Sasol Advanced Fuels Laboratory

UNIVERSITY OF CAPE TOWN

Determination of the effectiveness of a Hot Tube igniter for initiating HCCI combustion

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Abstract

Background

Homogeneous Charge Compression Ignition (HCCI) is a new internal combustion system that promises high efficiency and dramatically reduced nitrous oxide (NO_x) and particulate matter (PM) emissions when compared to current spark ignition (SI) and compression ignition (CI) engine technologies. In its simplest form, HCCI can be described as lean autoignition of a homogeneous fuel/air mixture that occurs without a flame front. HCCI can in theory be achieved using almost any fuel, provided that it evaporates readily and has a short enough ignition delay that it can be made to autoignite under the conditions typically found in an IC engine.

Basically HCCI incorporates the best features of a SI (petrol) and CI (diesel) engine. Like in a SI engine, the fuel and air in the cylinder is allowed to be well mixed before the onset of combustion which promotes cleaner burning (Low PM) and like in a CI engine the engine is operated overall fuel-lean and therefore has no throttling losses and near zero NO_x emissions. The mixture is also compression ignited in the same way as in a CI engine. This causes combustion to occur simultaneously throughout the combustion chamber and thus no flame front is present.

Motivation

Controlling combustion phasing is one of the biggest challenges with HCCI. The purpose of the study was to determine the effectiveness of a Hot Tube igniter for initiating HCCI combustion. The igniter consists of a capped tube that protrudes from the combustion chamber and is heated from the outside. By controlling the temperature of the tube, it is possible that some control can be gained over the start of combustion.

Sasol sponsored this research project and their interest in HCCI lies on the fuels side where HCCI testing capabilities are a prerequisite for any fuels research that the company wants done in this field. Upgrading an existing single-cylinder research engine with a fuel injection system and comprehensive instrumentation, data acquisition and control hardware provides this capability.

Results

After spending a lot of effort on upgrading the engine, HCCI running was achieved with great success from the start.

A series of tests were done using high and low octane fuels and stable HCCI operation was achieved with all fuels tested. The fuels used were n-Heptane (low octane) and Methanol (high octane, low surface ignition temperature).

The Hot Tube igniter was tested concurrently for temperatures up to 400°C and was found to be ineffective in initiating HCCI combustion. This was a surprising result because with Methanol, which is known to have a low hot surface ignition temperature, the Hot Tube had no effect on combustion. The modelling suggested that autoignition should be initiated in the Hot Tube. The Hot Tube did, however, lower that rate of pressure rise in the cylinder.

The experimental results for normal running (without the Hot Tube) correlated well with combustion modelling that was done prior to the test program.

Conclusion

The Hot Tube igniter proved ineffective in initiating HCCI combustion for the relatively low tube temperatures that were investigated.

The work done during this project provides the Sasol Advanced Fuels Laboratory with a tool with which further HCCI research can be done.

Recommendations

Refinements to the instrumentation and data acquisition system are strongly recommended.

If a way can be found to reliably heat the Hot Tube to higher temperatures than the ones used in this study, more tests can be done to confirm the findings.

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List of Acronyms and Abbreviations

ATDC: After Top Dead Center

CAD: Crank Angle Degrees

CA 10: Crank Angle at which 10% of total heat has been released

CA 50: Crank Angle at which 50% of total heat has been released

CI: Compression Ignition

CO: Carbon Monoxide

CO_2: Carbon Dioxide

COV: Coefficient of Variation

CR: Compression Ratio

DAQ: Data Acquisition

DC: Direct Current

DI: Direct Injection

EGR: Exhaust Gas Recirculation

EVO: Exhaust Valve Opening

HC: Hydrocarbon

HCCI: Homogeneous Charge Compression Ignition

HT: Hot Tube

HTT: Hot Tube Temperature

IMEP: Indicated Mean Effective Pressure

IVC: Inlet Valve Closure

λ: Air/Fuel equivalence ratio

NO_x: Nitrous Oxides

NTC: Negative Temperature Coefficient

PC: Personal Computer

PFI: Port Fuel Injection

PM: Particulate Matter

PRF: Primary Reference Fuel

ROHR: Rate of Heat Release

RPM: Revolutions per Minute

SAFL: Sasol Advanced Fuels Laboratory

SI: Spark Ignition

τ: Ignition Delay

THR: Total Heat Released

VVA: Variable Valve Actuation

1 Introduction

1.1 HCCI Combustion

The continued pressure placed on engine and fuel manufacturers to reduce internal combustion (IC) engine emissions has lead to a global drive to develop new cleaner engines and combustion systems. The possibility of a looming global fuel crisis has also focussed manufacturers' attention on improving engine efficiency and reducing fuel consumption.

Homogeneous Charge Compression Ignition (HCCI) has been identified as being a promising new combustion system that could provide solutions to the challenges mentioned above. HCCI combines the advantages of both spark ignition (SI) and compression ignition (CI) in a single system that offers both dramatically reduced emissions and improved efficiency. As with any emerging technology, there are a number of challenges that will have to be overcome before the system can be commercialised. Continued research is being done worldwide to bring HCCI into the market.

In principle HCCI combustion is the controlled autoignition of a lean homogeneous fuel/air mixture. Premixed fuel and air gets compressed during the compression stroke and the pressure and temperature rise causes the mixture to autoignite. Combustion is purely controlled by the chemical kinetics of the fuel and thus there is no direct control over the start of combustion. This lack of control over combustion phasing is one of the biggest challenges facing HCCI development.

1.2 Hot Tube ignition

This study investigates the use of a Hot Tube igniter to initiate HCCI combustion. Hot Tubes were originally used on SI engines before the electric spark plug ignition was developed. The igniter consists of a capped tube that protrudes from the combustion chamber and is heated from the outside. During engine running some portion of the mixture gets compressed in the tube exposing it to the hot inner surface. Heat transfer from the tube to the mixture can cause a sufficient rise in the temperature to make the mixture in the tube autoignite before the rest of the mixture in the combustion chamber. The pressure and temperature rise due to autoignition can then "nudge" the bulk of the mixture into autoigniting. By controlling the temperature of the tube, it is possible that some control can be gained over the start of combustion.

1.3 Objectives

- For the purpose of this study it is necessary to upgrade and modernise an existing experimental engine setup. A Ricardo E6 single-cylinder experimental engine dating from 1968 is available as test engine but needs a lot of work to get it up to modern engine test standards.
- By upgrading the engine, create a tool with which further HCCI research can be done.
- A Hot Tube igniter needs to be designed and fitted to the engine
- A range of tests with and without the Hot Tube needs to be done with different fuels to capture engine data for analysis and comparison.
- Through analysis of the data captured during the tests, the effectiveness of the Hot Tube needs to be determined.

1.4 Thesis Outline

This document is structured in such a way that the scientific method used for the investigation can be followed.

Chapter 2 is a review of the relevant literature. It starts by giving a broad overview of HCCI in general and this is followed by a discussion on the proposed use of a Hot Tube igniter.

Chapter 3 discusses the experimental design and the computer modelling that was done to determine experimental operating conditions.

Chapter 4 covers the experimental setup. The test apparatus and procedures are discussed.

Chapter 5 gives the results of the engine testing and analyses done on the engine data. The results are also interpreted and discussed throughout.

Chapter 6 gives the conclusions of the study and recommendations for further work.

2 Literature Review

2.1 What is HCCI?

Homogeneous Charge Compression Ignition is an internal combustion process that promises high efficiency and dramatically reduced nitrous oxide (NO_x) emissions when compared to current spark ignition and compression ignition engine technologies. In its simplest form, HCCI can be described as lean autoignition of a homogeneous fuel/air mixture that occurs without a flame front [Zhao F. et al., 2003(a)]. HCCI can in theory be achieved using almost any fuel, provided that it evaporates readily and has a short enough ignition delay that it can be made to autoignite under the conditions typically found in an IC engine.

Basically HCCI incorporates the best features of an SI and CI engine. Like in a SI engine, the fuel and air in the cylinder is allowed to be well mixed before the onset of combustion which promotes cleaner burning and like in a CI engine it is operated overall fuel-lean and can therefore benefit from a higher thermal efficiency and no throttling losses. The mixture is ignited by compressing it and thereby raising the temperature and pressure of the mixture in the same way as in a CI engine. This causes combustion to occur simultaneously throughout the combustion chamber and thus no flame front is present. The absence of a flame front also means that the flammability limits associated with SI combustion is irrelevant for HCCI running.

The reduction in emissions is due to the fact that: 1) the peak burned gas temperatures are much lower than in the flame front of a SI engine or the diffusion flame combustion in a CI engine. This causes much lower NO_x emissions; and 2) there is no fuel-rich diffusion burning taking place. Particulate matter (PM) emissions are therefore reduced to near-zero levels.

HCCI engines potentially offer efficiency levels comparable to those of diesel engines while engine-out NO_x emissions can be as low as tailpipe NO_x emissions of a conventional SI engine with aftertreatment.

Despite all the advantages mentioned above, HCCI poses its own unique set of challenges:

- High hydrocarbon (HC) and carbon monoxide (CO) emissions
- Narrow operating range in terms of load and speed
- Difficulty in controlling combustion phasing
- Low specific power output

As is the case with all homogeneous combustion systems, a significant amount of fuel is stored in in-cylinder crevices and escapes combustion [Zhao F. et al., 2003(b)]. In a normal SI engine, burned gas temperatures are high enough to consume this unburned fuel during the power (expansion) stroke. In an HCCI engine this is not the case with burned gas temperatures being much lower due to lean operation and therefore HC emissions are high compared to conventional SI engines. The burned gas temperatures are also too low to complete the CO to CO_2 oxidation reaction during combustion and CO emissions increases.

HCCI engines need to be operated fuel-lean to limit the rates of pressure rise due to the near simultaneous combustion of all of the in-cylinder mixture. This limits the load range over which the engine can be operated and thus the maximum power output for a given engine displacement will always be lower than a SI or CI engine.

Since a homogeneous mixture is compression ignited, there is no direct control over the start of combustion (combustion phasing). The timing of the autoignition event is controlled purely by the pressure and temperature history of the charge in the cylinder. This means that the mixture conditions at inlet valve closure (IVC) and the subsequent development thereof needs to be precisely controlled. This is easy enough at steady state conditions but becomes especially difficult during transient engine operation.

HCCI engines can be used wherever an IC engine is used today – from automotive to heavy duty truck engines and every size class of transportation engine. HCCI is particularly well suited to stationary applications where operation is limited to narrower load and speed ranges, because of the engine's characteristic narrower band of operation. HCCI combustion, its challenges and other aspects of this research project will be discussed in greater depth in the remainder of this chapter.

2.2 Why HCCI?

The driving force behind all the current HCCI research around the world is the ever increasing pressure to reduce vehicle emissions and improve fuel economy. This inevitably means that engine manufacturers have to find new and innovative ways to get their engines to run cleaner and more efficiently.

As mentioned in the previous section, HCCI offers significant emission reductions and this makes it a very attractive combustion system to achieve future emissions targets. The regulated exhaust emissions from vehicles are the following [Schaberg, 2005]:

- Total hydrocarbons (THC – Europe) and non-methane hydrocarbons (NMHC – USA)
- Carbon monoxide (CO)
- Nitrogen oxides (NO_x)
- Particulate Matter (PM) (diesel only)

HCCI combustion principles can be applied to both SI and CI engines and the results are equally impressive. In SI engines, NO_x emissions can be reduced to near-zero level without any aftertreatment due to the lean autoignition causing lower peak gas temperatures during combustion which practically eliminates the formation of NO_x.

CI engines have the characteristic that one can either reduce PM or NO_x, but not both at the same time and as one is reduced, the other increases. It is referred to as the PM – NO_x trade-off and is illustrated qualitatively in Fig. 2-1. In Fig. 2-2 the attainable HCCI operating range is compared to the current CI range. Pollutant formation is not the focus of this review but a detailed discussion of it can be found in Heywood (1988a).

Due to its premixed nature, PM emissions are virtually eliminated under HCCI conditions in a diesel engine, while NO_x emissions are also reduced dramatically due to lower peak gas temperatures in the cylinder.

Although HC and CO emissions are higher for SI and CI engines running in HCCI mode, the potential benefits of dramatically reduced NO_x and PM outweigh these disadvantages and work continues to find ways to eliminate these problems.

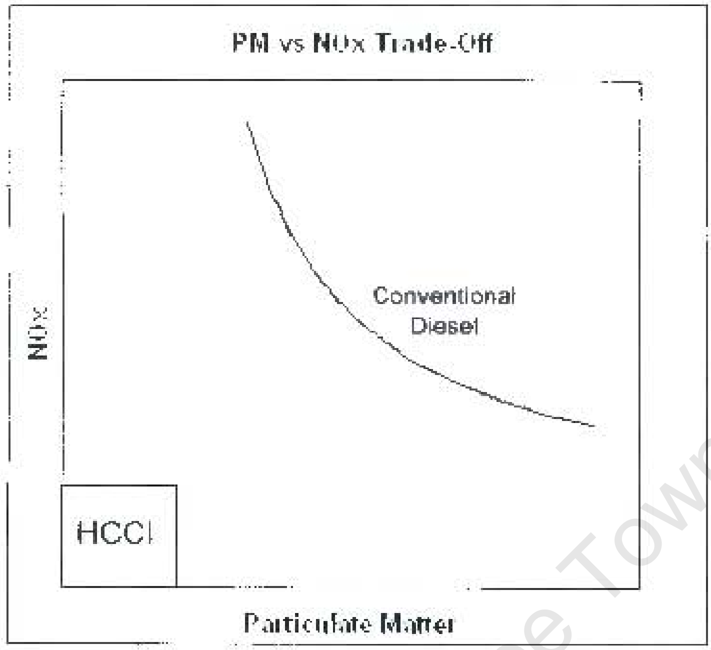

Figure 2-1 The PM vs NOx trade-off for diesel engines.

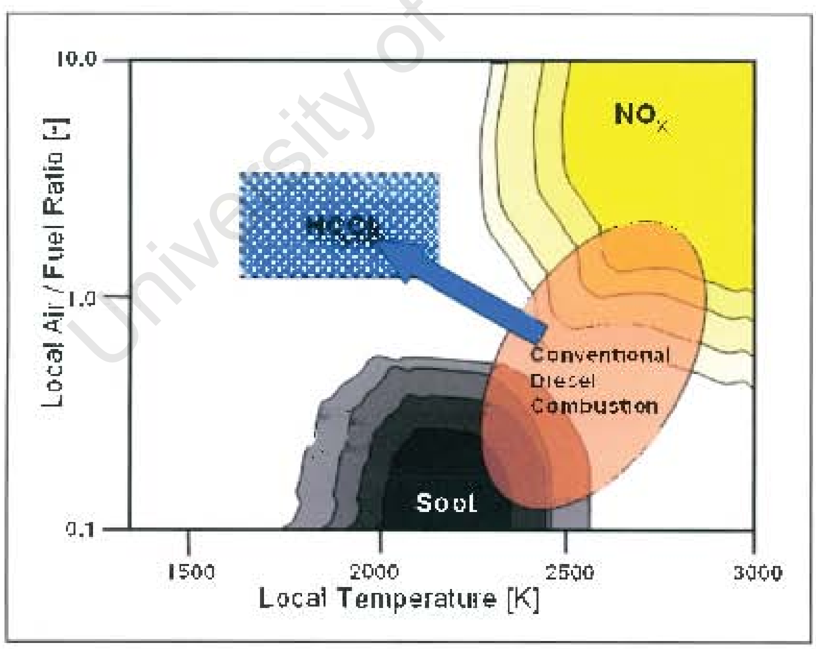

Figure 2-2 Comparison between load ranges for HCCI and CI running. Adopted from Velji (2005).

HCCI also offers the potential to reduce fuel consumption by 15 – 20 %, especially at low load conditions in an SI engine. This is possible because the engine can be operated unthrottled and therefore pumping losses are eliminated. Reductions in fuel consumption are also made possible by improved thermodynamic efficiency due to lean operation. Fuel economy and CO_2 emissions are linked to a large extent and since there has been a stronger focus on CO_2 emissions of late, HCCI might be part of the solution to achieving some of the CO_2 reduction goals [Schaberg, 2005].

2.3 Challenges with HCCI

As stated previously, HCCI combustion poses a unique set of challenges that have to be overcome before it can be commercialised on a large scale. These challenges include expanding the limited operating range, finding ways to accurately control combustion phasing and getting the engine to run smoothly during transient operation. Each of these challenges will be discussed below.

2.3.1 Control of combustion phasing

Combustion phasing is controlled by the chemical kinetics of the fuel meaning that the pressure, temperature and concentration history of the fuel/air mixture in the cylinder determines when autoignition will occur. Also, because autoignition is time dependent, engine speed will also influence combustion phasing. Therefore the parameters that influence the mixture history, need to be controlled.

The key parameters that can be utilised for HCCI combustion phasing control can be summarised as follows:

- Inlet conditions (temperature and pressure)
- Fuel type, concentration and additives
- Compression ratio
- Amount of EGR or residuals trapped in the cylinder
- Glow plugs
- Water injection

To achieve steady HCCI operation one requires a critical amount of thermal energy to be present in the cylinder at IVC. The lower the load and the higher the speed, the more energy is required. For SI-like HCCI some form of energy input is usually needed to assist in getting the fuel to autoignite since SI fuels are generally very resistant to autoignition. This energy input can be in the form of inlet air heating or EGR.

The effect of intake air temperature has been studied widely in the literature and has proved to be very effective in initiating autoignition. A higher intake temperature advances autoignition and increases heat release rates. Due to the fact that autoignition is very sensitive to changes in temperature, the controllable range when using intake heating is fairly limited. Compounding this problem is the fact that when engine speed and load changes, autoignition timing will also vary, unless the intake temperature can be changed to compensate for the change. Temperature compensation is very slow on a cycle to cycle basis and it is therefore not an effective means to control autoignition [Zhao F. et al., 2003(c)].

EGR promotes HCCI combustion by heating the intake charge. EGR makes use of the thermal energy in the exhaust gases to heat the fresh intake charge. This can be achieved by either routing some of the exhaust gases back into the intake (external EGR) or by trapping some exhaust gas in the cylinder (internal EGR) through variable valve actuation (VVA). A further advantage of the use of EGR is the dilution effect which can be used to control heat release rate, due to its effect on chemical reaction rates. This increases the ignition delay and lowers heat release rate and peak cylinder pressures [Ishibashi Y. et al., 1996].

Using the compression ratio as a means to control HCCI has also been investigated [Christenson, 1999]. A higher compression ratio will increase the mixture temperature during compression and thus advance the autoignition timing. A higher compression ratio lowers the optimum intake temperature which results in higher intake density and higher output [Hiraya, 2002]. It also improves the thermal efficiency of the engine due to larger expansion ratios. High compression ratios can however lead to knock problems at higher engine loads especially when low octane fuels are used. A variable compression ratio could solve this problem, but the control response of the systems tried to date are once again too slow for control during rapid transient operation [Zhao F. et al., 2003(d)].

Fuel selection is obviously a very important factor for HCCI engine development. In theory, an HCCI engine can run on any liquid or gaseous fuel that can be mixed with air prior to autoignition [Aceves et al., 2003]. Volatility and autoignition characteristics have been identified as important fuel parameters and the choice of fuel will greatly influence engine operating parameters such as compression ratio, intake temperature and EGR required. Fuels with single-stage ignition[1] are less sensitive to load and speed changes and thus allow HCCI operation over a wider range of engine operating conditions [Zhao F. et al., 2003(e)]. HCCI fuel requirements will be discussed in greater depth in section 2.6.

Forced induction or boosting can be used to increase the engine's IMEP by increasing intake charge density and extend the engine's operational range of air-fuel ratio for HCCI operation [Zhao F. et al., 2003(f)]. Boosting leads to high cylinder pressures and this could limit its application. Studies done by Christensen et al. (1998) and Olsson et al. (2001) showed the potential benefits of boosting to be higher achievable IMEP and efficiency at high loads, but drawbacks included lower efficiency at low loads and increased turbocharger pumping losses due to lower exhaust temperatures.

The use of glowplugs for HCCI control has been investigated. Problems with response time, limited heating capacity and complicated control were identified as being key issues for their application especially during rapid transients [Zhao F. et al., 2003(g)].

The use of water injection to control phasing and heat release has been studied lately. Christensen et al. (1999) showed that water injection could be used to control ignition timing, but that it caused an increase in the already high emissions of HC and

[1] Single stage ignition refers to the way certain fuels react when autoigniting. The heat release associated with combustion occurs in a single stage once a certain level of pressure and temperature is reached. This is opposed to two-stage ignition where oxidation happens in two stages – partial oxidation followed by a distinct time delay before oxidation goes to completion.

CO. NO_x emissions were even lower than for normal HCCI combustion. Water consumed was in the order of the fuel flow [Zhao F. et al., 2003(h)]. Kaneko et al (2002) investigated the effect of water injection on diesel HCCI combustion. They found that it had a significant effect in suppressing the heat release rate and that the low NO_x, smokeless combustion range could be extended to higher loads.

2.3.2 Expansion of Operating Range

Currently the load and speed range for which acceptable HCCI combustion can be achieved is in the order of 0.5 – 4 Bar BMEP and 600 – 4000 rpm [Zhao et al,2002], although Duffy et al. (2005) reported a BMEP of over 10 Bar achieved in tests done at Caterpillar. Fig. 2-3 shows a comparison between the load and speed ranges achievable with HCCI, SI and CI. The figure was adopted from a presentation by Xu (2005) but it seems to indicate that neither CI nor SI engines can be operated at low speed and high load. It is known that heavy duty engines are operated in this region. However, the point made that HCCI engines can not be operated at the same levels of speed and load as SI and CI, is valid.

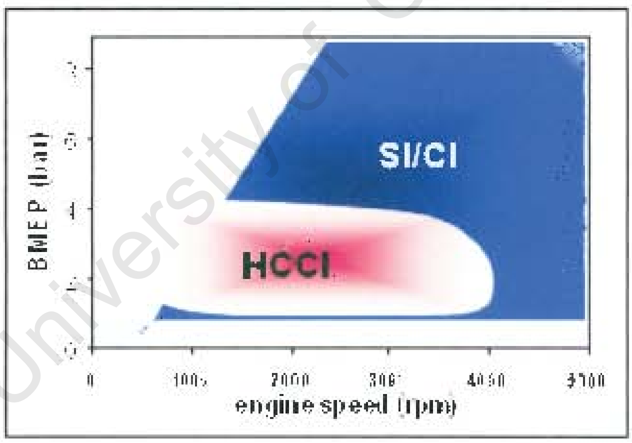

Figure 2-3 Comparison between load and speed ranges for HCCI and SI running. Adopted from Xu (2005)

Expanding the operating range of HCCI engines is as important as finding ways to accurately control combustion phasing. High load operation is limited by excessively high rates of heat release and high engine noise levels. Thus, HCCI operation is limited to part-load conditions and this reduces the benefits of this technology.

Various techniques to widen the operating range have been investigated. These include different fuel injection strategies, the use of in-cylinder residuals or EGR, water injection, fuel additives and boosting. The pros and cons of these were discussed in section 2.1.1. Until now no single technique has been found that can be applied to mass production.

Supercharging has been named as a promising way to expand the operating range, but questions surrounding engine structure limits and combustion noise remain unanswered [Zhao F. et al., 2003(i)].

High load diesel-HCCI shows very small NO_x emission benefits over normal CI running and the efficiency advantage of gasoline-HCCI is almost lost at high load due to the essentially unthrottled operation of an SI engine at these loads. Emissions from a gasoline-HCCI engine compared to a stoichiometric SI engine with three-way aftertreatment, does not show a great advantage for HCCI.

HCCI operation at near idle loads and at cold start is also still problematic. At low loads there is not sufficient thermal energy available in the residuals or EGR to trigger autoignition. Also, HC and CO emissions increase at low loads due to reduced combustion efficiency. For cold start heat loss to the combustion chamber walls are very high and therefore HCCI operation becomes a problem.

Taking all of the abovementioned into account, it seems inevitable that for HCCI to be commercialised, it will have to be run as a dual-mode engine switching between HCCI and SI or HCCI and CI to maintain good combustion. The challenge will remain to expand the HCCI range to maximise the benefits of this technology.

2.3.3 Transient Operation

Under normal driving conditions an engine will only run at steady state conditions (ie. fixed load and speed) for limited periods of time. Thus, for HCCI engines to become a viable power source for automotive applications, they will need to be able to handle transient operation very well.

HCCI combustion is very sensitive to changes in operating conditions, especially any changes that affect intake charge temperature. Many of the measures used to control autoignition in the engine are not suited to transient operation, since the speed with which they can be controlled simply is not fast enough to keep up with rapid transient operation. Intake air heating, variable compression ratio mechanisms and glow plugs for instance can not react fast enough during rapid transients to maintain good combustion. Many of the issues regarding transient operation and mode-switching is currently being researched and good progress is being made in finding ways to ensure smooth operation [Xu, 2005].

2.4 Gasoline HCCI Combustion

Most of the current research in the HCCI field concentrates either on gasoline-like HCCI or diesel-like HCCI. In this section aspects specific to gasoline-HCCI will be discussed as the work done during this study will concentrate on gasoline-like operation.

The biggest challenge with HCCI combustion is to find a way to use existing mechanical measures like different fuel injection strategies and VVA [Zhao F. et al., 2003(j)] to control combustion phasing and duration. For gasoline-HCCI a thermal energy input is required to promote combustion due to the resistance of gasolines to autoignition. While raising the compression ratio and intake temperature promotes combustion, it is not a very practical approach to controlling combustion. For a dual-mode gasoline HCCI engine the compression ratio is limited to around 12:1 to still allow acceptable full load operation without severe knocking and there are limits to the amount of intake heating that can be done using heat available in the exhaust or engine coolant. The most efficient way to promote autoignition is the use of high levels of EGR and various strategies exist to implement it in the engine.

To get the full efficiency benefit of HCCI, the engine will have to be operated where a normal SI engine is highly throttled. A number of aspects concerning gasoline-HCCI will be discussed below.

2.4.1 Dilution Strategies

The use of EGR has been found to be the most practical approach to achieve good HCCI combustion in gasoline engines.

EGR can either be implemented by routing some of the exhaust gases back into the intake (external EGR) or by trapping exhaust gases in the cylinder through VVA strategies. A disadvantage of external EGR is heat loss that occurs along the transfer route, while VVA increases mechanical and control complexity.

EGR is very effective at light loads, but falls short at idle and near-idle loads where there is not enough thermal energy available in the exhaust to heat the intake charge sufficiently and at high loads there is too little dilution to moderate the heat release.

2.4.2 Compression Ratio

Raising the compression ratio is one of the most obvious ways to promote autoignition, but the need to integrate gasoline-HCCI into a dual-mode SI engine limits it to around 12:1. Increasing compression ratio also creates problems with combustion noise and the higher in-cylinder gas temperatures cause higher levels of chemical reactivity, advanced ignition timing and much higher heat release rates. Studies performed by Najt et al. (1983) and Thring (1989) both concluded that lower compression ratios suited gasoline-HCCI better.

Christensen et al. (1998) showed the potential of using a variable compression ratio, but such systems are complex and difficult to implement.

The best compromise seems to be to select the highest compression ratio for which full-load SI running is possible. Other ways will have to found to promote autoignition at low loads.

2.4.3 Fuel Injection

For fuel delivery one has the choice of two systems – port fuel injection (PFI) or direct injection (DI). PFI gives a long mixing time and therefore a very homogeneous mixture. DI offers the potential advantage for combustion phasing control through changes in injection timing. The charge stratification that can be obtained with DI can influence heat release significantly [Marriot et al. 2002].

2.4.4 Autoignition Control

Gaining direct control over autoignition is the aim of much of the current HCCI research. It would solve many of the inherent difficulties with HCCI combustion.

Marriot et al. (2002) used a secondary direct injection to trigger combustion but this increased NO_x emissions. With late injection timings the combustion was more stratified and smoke emissions increased.

The effect of a spark seemed to yield conflicting results. Ishibashi et al. (1996) found it to be effective in a two-stroke engine, but Ma et al. (2001) and Law et al. (2000) found it to have no effect in a four stroke engine.

2.4.5 Emissions

In HCCI mode, low peak burned gas temperatures result in dramatically reduced NOx emissions, while the same low temperatures cause increased HC and CO emissions. At low loads HC emissions increase due to reduced combustion efficiency and due to the lower temperatures, CO can not be oxidised to CO_2. HC and CO emissions can be as much as 50% higher than in normal SI engines.

Higher EGR levels causes the emissions to be reingested into the engine and it can be burned during following cycles. The high EGR levels also cause reduced exhaust temperatures at lower loads which could cause problems with catalyst functioning [Zhao F. et al., 2003(k)].

2.5 Hot Tube Ignition

The main focus of this study is to determine the effectiveness of a Hot Tube igniter in initiating HCCI combustion. It is proposed that similar to their initial use as igniters in early SI engines, the Hot Tube could be used to initiate autoignition in an HCCI engine.

Hot Tube ignition was used on many of the very early IC engines. Designers like Wright, Kjelsberg and Day [Little, 2003] used it on different types of engines, but Daimler and Maybach gave this type of ignition a definite boost and consolidated it as the state of the art when they used it on their first ever gasoline driven engine in 1883.

They also used this ignition on their "Grandfather Clock" engine which had an output of 0.75 kW at 600rpm with a displacement of 0.246 litres [Little, 2003]. From this time Hot Tube ignition was the preferred ignition system used by most designers.

The hot tubes of that era all consisted of a capped tube with the open end connected to the combustion chamber. Heat was supplied to the outside of the tube, usually with a Bunsen-type burner. Most designs featured another tube surrounding the first to isolate the system against heat loss and asbestos was often used to line the outer tube. Ignition was achieved when the fuel/air mixture gets forced into the tube at the end of compression stroke and the fuel ignited off the hot inner surface of the tube. The flame then propagated to the rest of the combustion chamber. Fig. 2-4 shows a typical Hot Tube design.

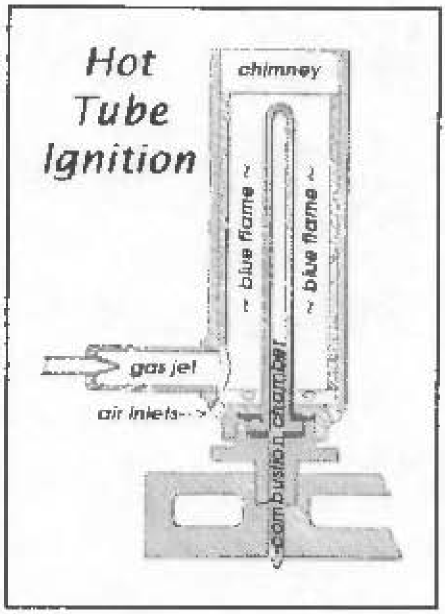

Figure 2-4 An early Hot Tube design

The popularity of the Hot Tube was due to the fact that, by changing the effective length of the tube, the ignition timing could be set. Longer lengths advanced ignition and shorter lengths retarded it. Hot Tube durability was, however, a serious problem due to the poor quality of materials available at the time. The earliest iron tubes had to be replaced almost on a daily basis [Little, 2003]. Thicker iron tubes and later ceramic versions improved durability dramatically, but it was still fairly poor compared to what is achievable today.

Hot Tube ignition was still used until around 1910 when Robert Bosch introduced its electrical ignition using a spark plug similar to what is used currently. This electrical system improved ignition dramatically and the use of Hot Tubes quickly went out of fashion.

The subsequent use of heated surfaces in the combustion chamber were mostly limited to glow plugs used in CI engines to assist combustion during cold starting. The use of glow plugs to initiate or assist HCCI combustion has been investigated, but its operation proved to be problematic [Zhao F. et al., 2003(I)].

In this study, the use of an electrically heated Hot Tube is proposed as a possible means to control HCCI combustion. When running an engine in HCCI mode, fuel and air is mixed homogeneously and compressed to such levels that it autoignites. The rationale behind using a Hot Tube is that it is possible that by exposing some portion of the mixture to elevated temperatures in the Hot Tube, one can initiate HCCI combustion at that point. This primary autoignition in the Hot Tube can then cause a sufficient rise in pressure and temperature to "nudge" the rest of the mixture in the combustion chamber into autoigniting. In this way control can be gained over combustion phasing. This study will test this hypothesis.

The advantage of using a Hot Tube over other approaches found in literature is that a small portion of the mixture is forced into the tube and is exposed to a relatively large heated area. Heat can then be transferred to the mixture to heat it. One of the problems with glow plugs is that they have a relatively small surface area and with the fluid motion in the cylinder during compression, the heat transferred to the mixture is spread to the whole mixture, which limits the heating effect.

2.6 The role of the fuel

Considering the nature of HCCI combustion it is obvious that the choice of fuel is vital when designing such an engine. Engine hardware and control strategies have to be tailored around making the fuel autoignite at the right moment during every engine cycle to maximise the fuel economy and emissions benefit of this combustion system.

The autoignition timing of a fuel depends on the temperature, pressure and concentration history of the charge during compression [Zhao F. et al., 2003(m)]. All the proposed HCCI control strategies employ different dilution, valve timing, fuel injection and charge heating strategies to control this mixture history and get the fuel to autoignite at the right time.

It is accepted that current HCCI operation is limited to part load. High loads cause excessive rates of pressure rise and engine noise becomes unacceptably high, while at near idle loads peak gas temperatures are too low to sustain combustion and CO and HC emissions increase dramatically [Zhao F. et al., 2003(n)]. For these reasons, an HCCI engine for automotive use will inevitably be a multi-mode engine, switching between HCCI and SI or HCCI and CI running depending on load. For such engines the fuel requirements are extreme. One would need a gasoline that autoignites readily while in HCCI mode (fuel must have a low octane rating) and when in SI mode the fuel will need to have a high octane rating to avoid knocking. For diesel the opposite is true. In HCCI mode a low cetane fuel is ideal, while at full load a high cetane fuel with short ignition delay is preferred.

Taking the above into account it is possible that HCCI engines might have to be dual-fuelled, with one fuel for HCCI running and another for SI or CI full load operation. This is a concern because of the cost penalty of having two fuel systems and the increased complexity of controlling transient operation.

The use of fuel additives can be a possible solution to the problem of needing different fuel characteristics for different operating modes. On board blending of ignition promoters or inhibitors according to engine requirements can be used to tailor the fuel to specific running conditions.

A large amount of research has been done to determine the effects different fuels have on HCCI combustion. Especially the primary reference fuels (PRF's) n-heptane and iso-octane have been studied extensively. These fuels are particularly attractive because of their relative simplicity compared to gasoline and diesel and the fact that their behaviour can be modelled accurately using existing chemical kinetic mechanisms. These results can then be compared to actual experiments and have been useful for finding the effect of fuel octane rating on HCCI combustion [Zhao F. et al., 2003(o)].

3 Experimental Design

This chapter discusses aspects of the experimental design such as the combustion and ignition delay modelling and how these were used to determine the engine operating conditions needed for the different experiments.

3.1 Combustion Modelling

Computer modelling is used extensively in research to predict the behaviour of systems and substances under hypothetical conditions and therefore it can dramatically reduce the need for costly experimental work. For this exact reason, an engine and ignition delay model was developed to predict the behaviour of different fuels in an engine under HCCI conditions. The model predictions give a very good idea of what can be expected in the engine and at what conditions it needs to be run at to obtain a certain response. The models used will be discussed below.

3.1.1 Engine model

Even though there is an ever growing understanding of physics and chemistry, this understanding is still inadequate at a fundamental level to develop complete engine models incorporating the mechanics, thermodynamics and fluid dynamics into one model. For this reason, all modelling focuses only on certain limited aspects of an engine and its working.

The engine model used for the combustion modelling was originally developed to model an SI engine and was adapted for HCCI modelling. The model was developed in Microsoft Excel® using recognised techniques found in literature [Heywood, 1988]. It is a zero-dimensional thermodynamic model that predicts engine and fuel behaviour from IVC to EVO, by applying the laws of conservation of mass and energy. Combustion phasing is predicted using the ignition delay model discussed below.

The Hot Tube igniter is included in the model and is treated as a separate volume to the combustion chamber where heat is added to the charge. Pressure is assumed to be equal in the combustion chamber and the Hot Tube volume and the effect of different tube temperatures can easily be modelled.

3.1.2 Ignition delay model

An ignition delay model is incorporated in the engine model to predict combustion phasing. Ignition delay is the time it takes for a fuel/air mixture to autoignite under specific pressure and temperature conditions. Thus, ignition delay is a critical parameter for HCCI combustion since the combustion event and phasing is purely controlled by the ignition delay chemistry.

In the ignition delay model used, the predicted combustion phasing is based on a three exponential equation description of ignition delay as proposed by Yates et al. (2004) and Viljoen et al. (2005).

The three exponential equation description of ignition delay is a vast improvement over the single exponential equation description proposed by Douad and Eyzat (1978) as it can account for the so-called Negative Temperature Coefficient (NTC) behaviour of some fuels. The three exponential equation model involves two regimes; a two-stage low temperature regime and a single stage high temperature regime. The ignition delay for each of the different stages are given by a single Arrhenius equation:

$$\tau_i = A_i p^{-n_i} e^{\frac{B_i}{T}}$$

The two stages of the low temperature regime are sequential and are therefore expressed as the sum of the individual ignition delays, τ_1 and τ_2, while the high temperature regime presents a third competing reaction pathway and an ignition delay value of τ_3. The description of ignition delay for the full temperature and pressure spectrum is then given by the inverse sum of the two options:

$$\tau_{overall} = \left\{ (\tau_1 + \tau_2)^{-1} + (\tau_3)^{-1} \right\}^{-1}$$

This model for calculating ignition delay was originally developed for predicting knock in SI engines which are operated at around stoichiometric air/fuel ratios. HCCI engines are operated much leaner than this and therefore an air/fuel ratio correction factor was incorporated in the model. It was adopted from the unpublished work done by Londleni (2005). The correction is applied to $\tau_{overall}$ and is in the following form:

$$\tau_{corr,overall} = \tau_{overall} \lambda^{0.775}$$

This corrected overall ignition delay value is then used to calculate the combustion phasing by applying the conservation of ignition delay model of the form:

$$1 = \int_{to}^{tf} \frac{dt}{\tau_{corr,overall}}$$

The ignition delay value is calculated at every timestep in the model and at the timestep at which the integral reaches a value of 1, autoignition occurs.

Figures 3-1 and 3-2 show the ignition delay maps for the full temperature and pressure spectrum for n-Heptane and iso-Octane respectively. These maps illustrate visually how ignition delay is affected by pressure and temperature.

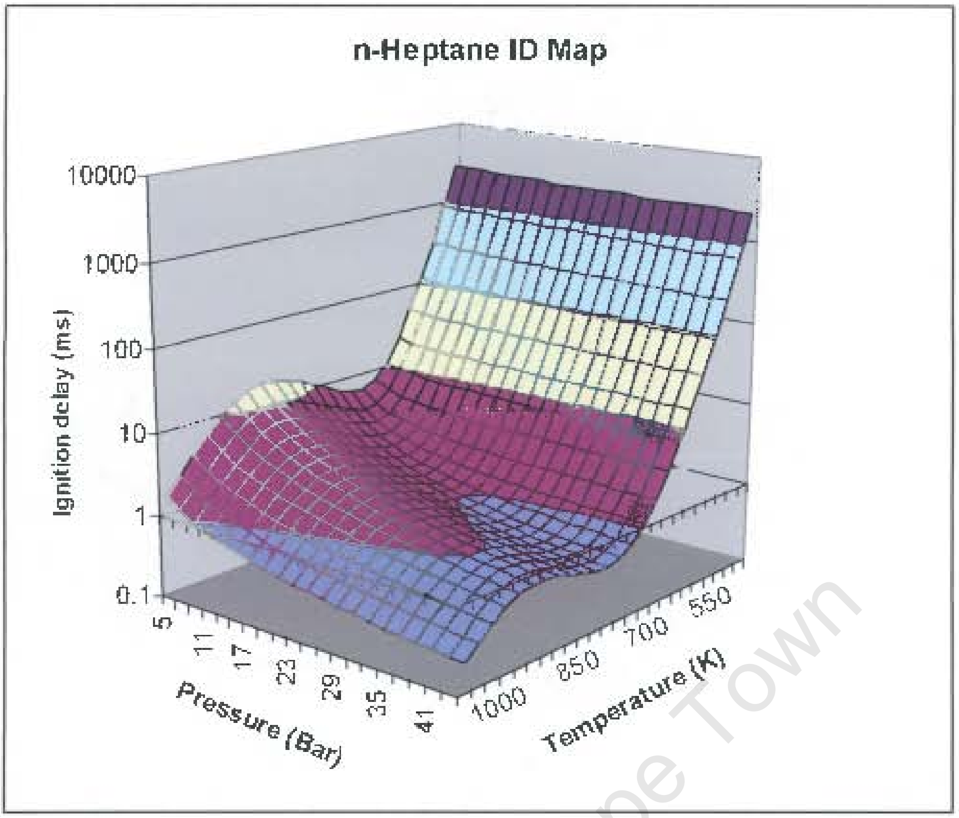

Figure 3-1 n-Heptane ignition delay map

These maps show the difference in ignition delay of iso-Octane (octane number = 100) and n-Heptane (octane number = 0). It is clear that iso-Octane with its higher octane rating exhibits a higher resistance to autoignition. Also visible on both maps is the so-called Negative Temperature Coefficient (NTC) region where ignition delay becomes longer the higher the temperature. This phenomenon is characteristic of parafinic fuels and is visible as a two-stage heat release on cylinder pressure traces. An example of such a pressure trace is shown in Fig. 3-3 where it is compared to a motored pressure trace to highlight the shape of the pressure development.

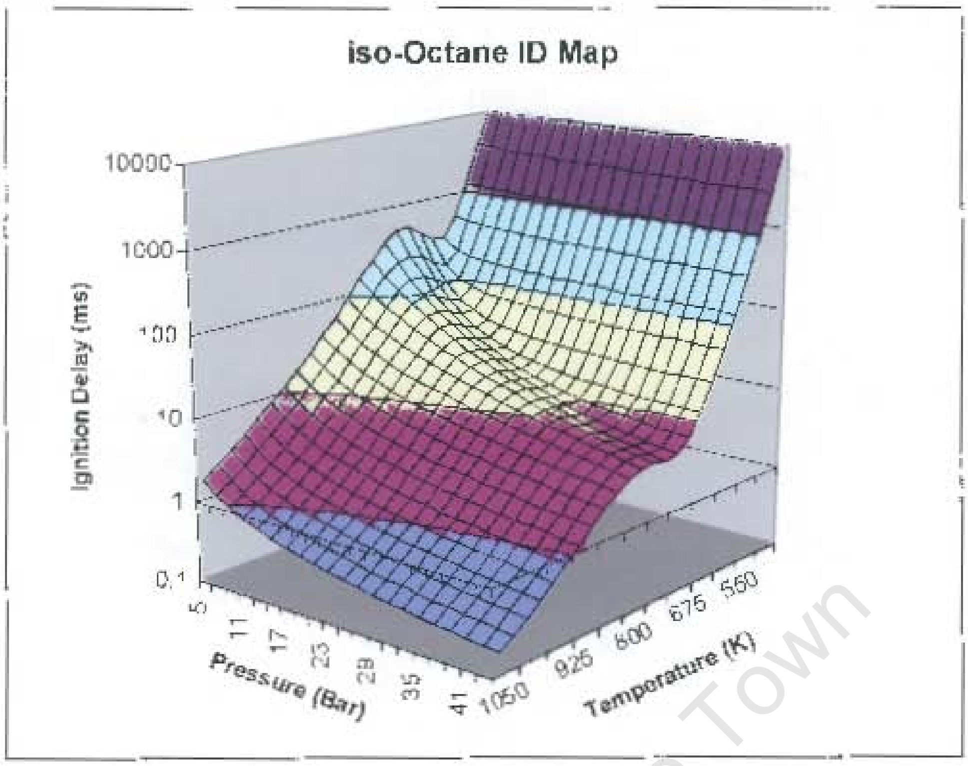

Figure 3-2 iso-Octane ignition delay map

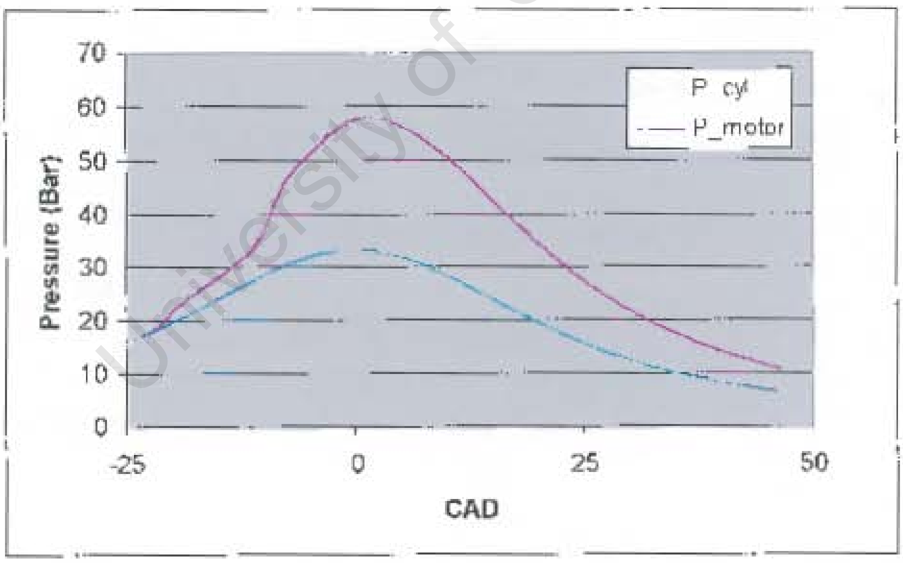

Figure 3-3 Graph showing typical two-stage HCCI ignition of n-Heptane

3.1.3 Modelling Results

Table 3-1 gives the predicted combustion phasing for n-Heptane. The prediction is for an inlet air temperature of 340 K and an engine speed of 1000 rpm. These values and the compression ratios indicated in table 3-1 were chosen after preliminary engine testing showed it to be where stable HCCI running was achievable.

Table 3-1 Predicted combustion phasing for n-Heptane

		HT*	Normal*
lambda	**CR**	**Phasing**	**Phasing**
2	7.65	4°	12°
2.5	8.2	2°	6°
3	9.2	-2°	0°
3.5	11.8	-6°	-6°
4	14	-6°	-6°

*HT = Running with Hot Tube, Normal = Running without Hot Tube

4 Experimental Setup

In this chapter the experimental apparatus, test procedures and operating criteria are discussed. One of the main objectives of the project was to develop a tool with which further HCCI research can be done. Most of the work discussed in this chapter went towards achieving this objective.

The design and implementation of the Hot Tube igniter is also discussed.

4.1 Apparatus

In the following sections various aspects of the setting up of the engine and instrumentation is discussed.

4.1.1 System requirements

To establish a tool for fundamental fuels research there are a number of requirements that need to be met. Ideally, the engine setup that is used must be as flexible as possible in changing various operating parameters to affect overall engine operating conditions. Parameters like engine speed and load, inlet air temperature and pressure (density), the amount of fuel supplied to the engine, valve timing, EGR, spark ignition timing etc. need to be easily variable. Also of great importance is the ability to capture engine data and store it for post-processing.

More specifically for HCCI combustion testing the following requirements were set:

- The engine used must be flexible enough to easily change compression ratio, inlet air temperature and amount of fuel supplied to the engine.
- It must have a wide enough speed range to cover as much as possible of the known HCCI operating range.
- The engine has to be instrumented to such an extent that condition monitoring, control and data acquisition (DAQ) can be done.

- The engine must be safe to operate. Covers and safety guards need to be fitted to reduce the possibility of injuries resulting from contact with moving parts.

4.1.2 Suitability of Ricardo E6 Engine

It was decided to use a single-cylinder Ricardo E6 experimental engine. Being an experimental engine it is ideally suited to the purpose of HCCI research. The following characteristics made the engine especially attractive:

- The engine is fitted with a DC dynamometer to motor or load the engine.
- The engine is fitted with a laminar flow element for air flow measurement and a heating element for inlet air heating.
- The engine is very robust.
- The compression ratio can be varied between 4:1 and 20:1 while the engine is running by simply turning a handle on the side of the engine.
- The engine has a usable speed range of 600 – 3000 rpm.
- The cylinder head has two spark plug holes, making it very easy to fit a pressure transducer and the Hot Tube igniter simultaneously.

Looking at the characteristics mentioned above, it is clear that the Ricardo E6 offered a good platform from which to start. Modifications and additions to the existing engine is discussed in the following section.

4.1.3 Modifications to Ricardo E6 Engine

The University of Cape Town acquired the E6 engine in 1968 and since then IC engine technology has developed dramatically. Extensive modernisation was needed to upgrade the engine to present standards.

In its original form the E6 was fitted with a single barrel downdraught carburettor, a magneto-type ignition system and analogue temperature and pressure gauges. Fuel and air flow measurements were performed using a pipette and manometer respectively, while engine speed and load control was handled by a very clumsy manually operated unit that controlled the dynamometer. Although these systems were not necessarily inaccurate, much more elegant electronic solutions exist today for engine control and condition monitoring. It was therefore decided to make the following modifications and additions to the engine:

- Disable the carburettor and design and fit an electronic fuel injection system for more accurate fuel delivery. See Appendix A.
- Replace the manual dynamometer controller with an electronically controlled DC drive that offers seamless switching between motoring and loading conditions. See Appendix A.
- Replace analogue temperature and pressure gauges with thermocouples and pressure transducers, the output of which can be fed into a computer for control and monitoring purposes.
- Fit a crank angle encoder to the engine for use with the DAQ system.
- Fit an engine coolant heater to control the coolant temperature.
- Install feedback control on both the inlet air and engine coolant heaters.
- Fit a lambda sensor.
- Replace the analogue spring balance used for torque measurement with a loadcell.
- Build a mobile cabinet to house the different control units and a computer from which data acquisition can be done.
- Fit the necessary safety guards to ensure that engine is safe to operate.
- Design and fit a new adjustable ignition distributor to replace the existing magneto-type ignition system for alternative SI operation.

All of the above was done and figures 4-1 – 4-4 show some of the results of these changes. The modification and preparation of the engine was a very time-consuming operation and took a significant amount of time to finalise.

Figure 4-1 The fuel injection system hardware.

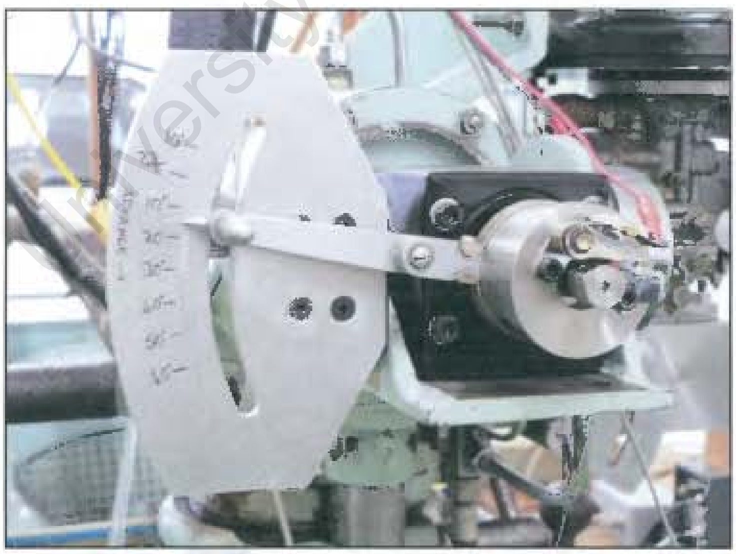

Figure 4-2 The ignition distributor that replaced the old magneto ignition.

Figure 4-3 The crank angle encoder fitted to the engine.

Figure 4-4 The cabinet housing the instrument read-outs and the DAQ computer.

4.1.4 Hot Tube igniter

The Hot Tube igniter was proposed as a possible means to initiate HCCI combustion by exposing some portion of the fuel and air mixture in the combustion chamber to the hot inner surface of the tube at the end of the compression stroke. Such an igniter needed to be designed for this purpose.

The design requirements for the Hot Tube was as follows:

- The igniter must be of simple and robust design.
- The igniter must fit onto the engine without modifications to the engine itself.
- Since the tube will form part of the combustion chamber its volume will affect overall combustion chamber volume and thus the compression ratio. Therefore it was decided that it must not limit the maximum compression ratio of the engine to below 16:1 (CR_{max} of the standard engine is 20:1).
- The tube must be electrically heated and automatically controlled.

With these criteria in mind work went ahead to design the Hot Tube. It seemed logical to design the tube to fit into one of the existing 14mm spark plug holes. The tube length and inside diameter determines its volume and these were chosen to meet the maximum compression ratio criteria. The tube was machined from stainless steel. It was chosen as material for its corrosion resistance, ease of machining and availability. Heating would be done through a 300W coiled electric heater wound around the outside of the tube. The maximum temperature that the tube can be heated to is 400°C and is limited by the heating element. Figures 4-5 and 4-6 show the design of the tube and the tube installed on the engine respectively.

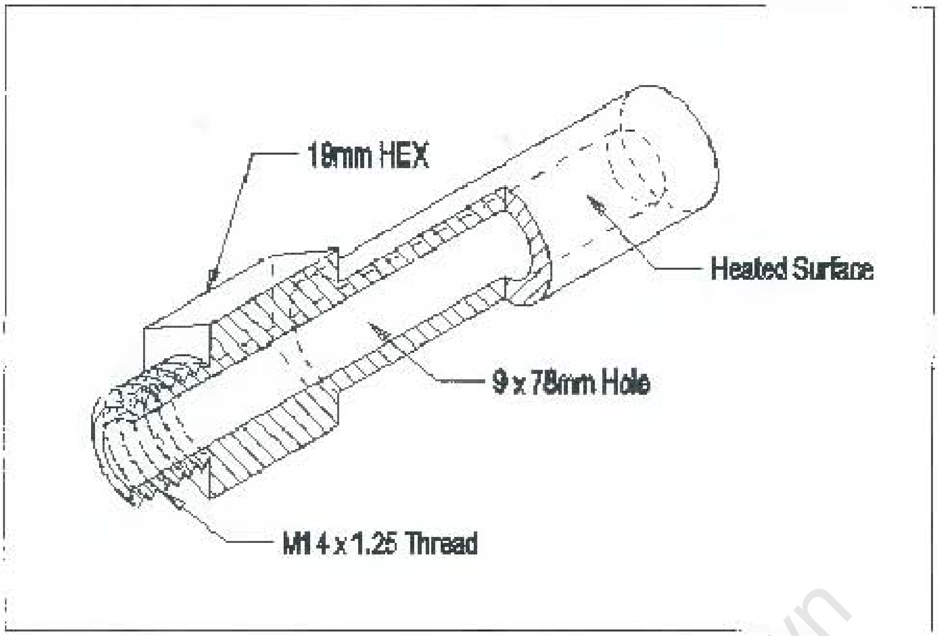

Figure 4-5 Design drawing of the Hot Tube igniter.

Figure 4-6 The Hot Tube igniter installed on the engine.

4.2 Experimental Procedure

With the engine and Hot Tube igniter prepared, suitable test conditions and an experimental procedure had to be decided on for assessing the effectiveness of the Hot Tube. These are discussed in the following two sections.

4.2.1 Test conditions

As this was the first time ever that the Ricardo E6 was run in HCCI mode, it was necessary to first determine the achievability of HCCI combustion with this engine. Using the combustion model discussed in the previous chapter, a set of operating conditions was determined for the preliminary HCCI experiments. HCCI combustion was easily achieved under these conditions, proving both that the model was fairly accurate and that the engine could easily be run in HCCI mode.

Since the operating conditions used for the preliminary tests fell within the known HCCI range and stable operation was achieved, it was decided to run the experiments for this study at the same conditions. The operating conditions used are given in Table 4-1.

Table 4-1 The operating conditions used during the HCCI experiments.

Engine Speed (rpm)	1000
Inlet Air Temp (K)	340
Engine Coolant Temp (°C)	60
Hot Tube Temp (°C)	varied 150 - 400
Lambda	varied 2 - 4
Compression Ratio	varied

4.2.2 Test procedure

To determine the effectiveness of the Hot Tube igniter, a series of experiments would be done where the minimum compression ratio needed for complete combustion at different λ-values and Hot Tube temperatures would be determined. The theory is that the hotter the tube becomes, the more heat will be transferred to the mixture inside it which would raise the mixture temperature and shorten the ignition delay. The shortened ignition delay should then promote combustion. The minimum compression ratios needed at each λ-value for Hot Tube and normal running can then be compared to see the effect of the Hot Tube igniter.

Fig. 4-7 shows a flowchart of the experimental procedure.

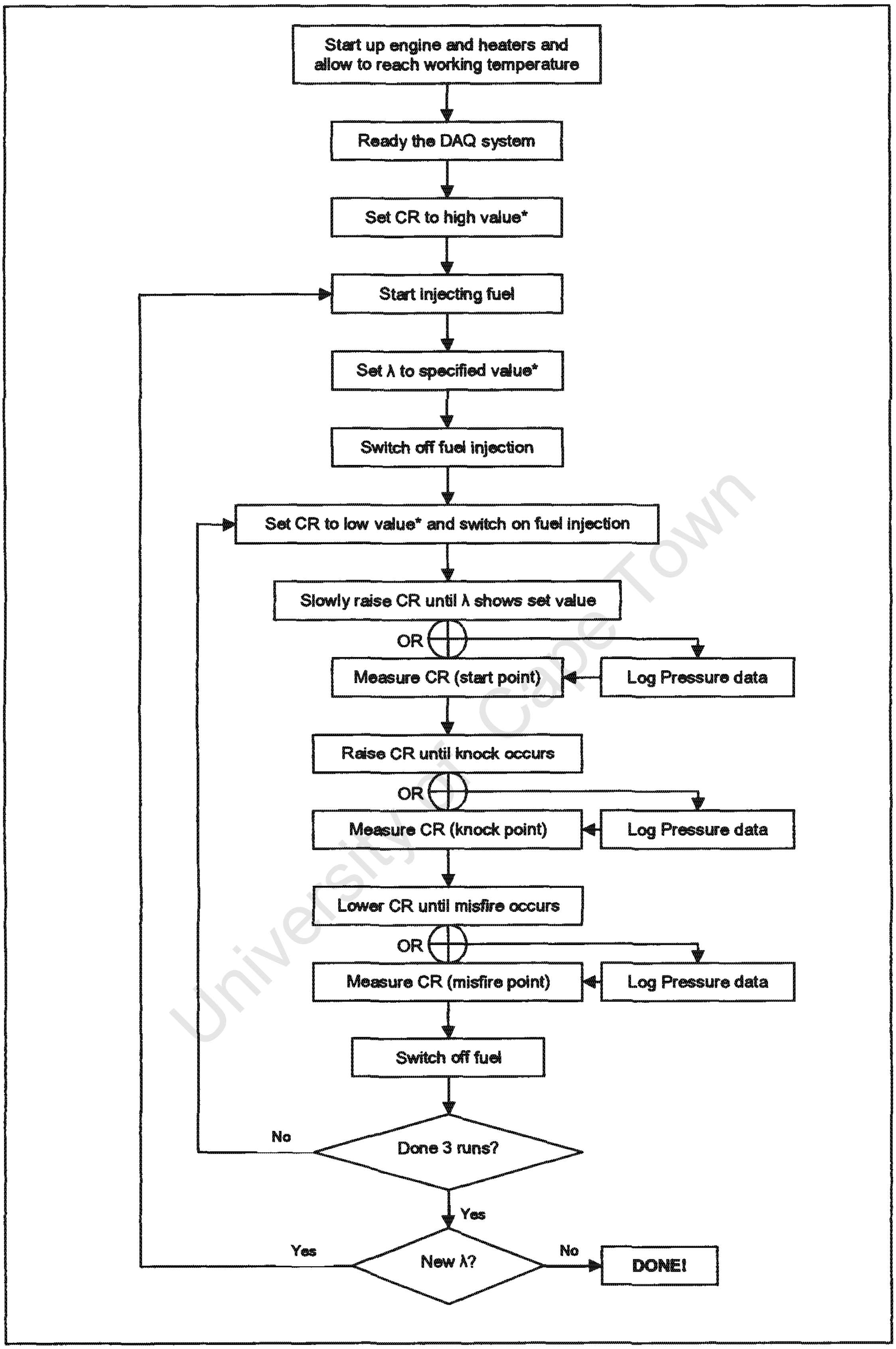

Figure 4-7 A Flowchart of the experimental procedure used. (*Elaborated in text)

For each test the engine would be motored and run until all the operating parameters set out in Table 4-1 reached steady state. Due to the large thermal inertia of the system this takes 10 -15 minutes. The DAQ system would also be readied. Once the conditions reached steady state the experiments could be started.

With the Hot Tube installed the minimum compression ratios needed at a certain λ-value was determined for a range of Hot Tube temperatures. This meant that for each λ-value from λ = 2 to λ = 4 (in 0.5 steps) the minimum compression ratios needed for complete combustion were measured at different Hot Tube temperatures (150°C – 400°C in 50° steps). Three runs were done at each temperature point resulting in 90 experiments (5 λ-values * 6 Temperature points * 3 runs at each point). After each change in Hot Tube temperature the system is left to reach steady state before measurements are done.

Lambda (λ) was set to the specified value by adjusting the fuelling and checking the lambda sensor readout at a reasonably high compression ratio where one could be sure that complete HCCI combustion is occurring based on the combustion model. λ is calculated based on airflow. Then the fuelling is switched off and the compression ratio is decreased to well below where any combustion would occur.

The procedure for the experiments without the Hot Tube is essentially the same as the one described above, but instead of having to do 90 experiments, only 15 is necessary – 5 λ-values * 3 runs at each point.

The exact criteria for the different conditions – starting, knocking and misfiring – are as follows:

- **Starting** – The "start" point would have been reached when, while raising the compression ratio, the indicated λ-value gets to within 0.1 of the set λ-value.
- **Knocking** – The "knock" point would have been reached when, while raising the compression ratio, the typical pressure oscillations associated with this phenomenon reaches a magnitude of 0.2 Bar.
- **Misfiring** – The "misfire" point would have been reached when, while lowering the compression ratio, the indicated λ-value increases to a value 0.1 greater than the set λ-value.

4.2.3 Data Acquisition and Post Processing

The only meaningful way to compare Hot Tube and normal running was to analyse the in-cylinder pressure development. A pressure transducer was fitted to the engine for this purpose and its output was captured using a PC-based data acquisition system. The stored data was then processed and analysed using a spreadsheet application such as Microsoft Excel®.

The DAQ system was developed using National Instruments® hardware and software. A 1 MHz DAQ card was used in conjunction with LabView® software to create an application that can store cylinder pressure data in a format that can be exported to MS Excel®.

Inherent to HCCI combustion is very low cycle-to-cycle variability. Due to this, smaller amounts of data can be analysed and taken to be representative of a certain set of operating conditions. To verify this, pressure data from a large number of consecutive engine cycles were analysed and the coefficient of variation (COV) of IMEP and maximum cylinder pressure was determined from it. The COV of IMEP was 2.7% and COV of maximum pressure was 1.6%. The low variability made it possible to capture data of very few engine cycles and then select any single cycle to base the analysis on.

Fig. 4-8 is a graph showing an average pressure trace and ones for +1 and -1 standard deviation.

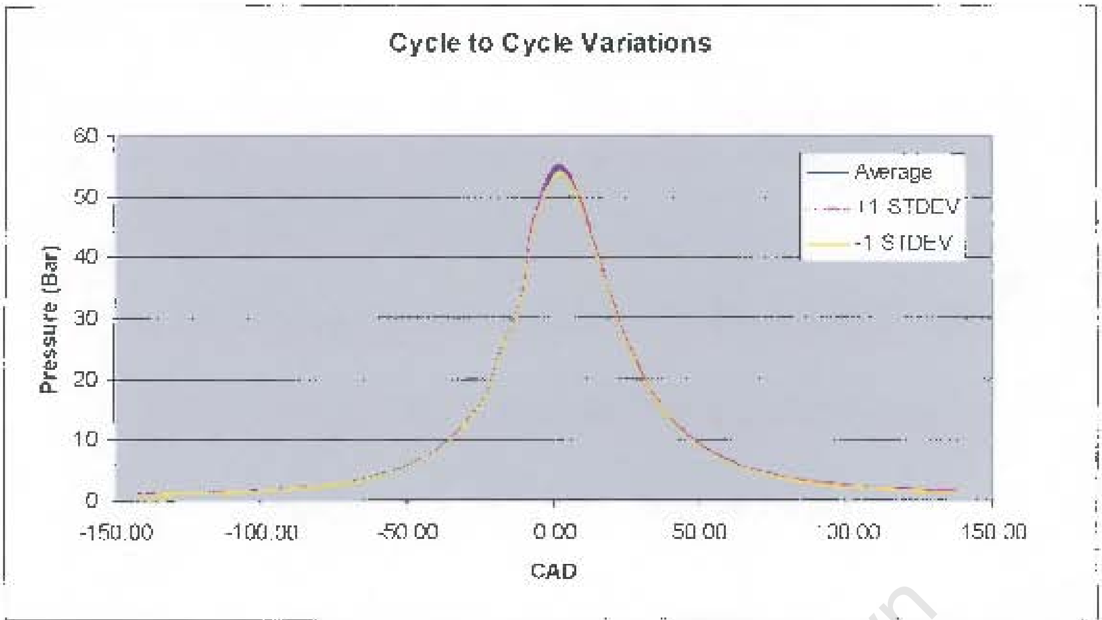

Figure 4-8 Graph showing low HCCI combustion variability.

5 Experimental Results and Discussion

The objective of the experimental work was to determine the effectiveness of the Hot Tube igniter when used as an initiator for HCCI combustion. The tests were done with two different fuels, N-Heptane (PRF 0) and Methanol. N-Heptane was chosen because it autoignites under very mild engine operating conditions and thus it should be easy to achieve steady HCCI running. Methanol was chosen because it is known that although it has a high octane number, it has a low hot surface ignition temperature [Hagen, 1977 and Downs, 1950] which could make it a suitable fuel to use in conjunction with the Hot Tube igniter.

The results are presented graphically or in table form with a discussion under each heading.

5.1 Hot tube test results

The first set of tests were conducted with the hot tube fitted to the engine and using N-Heptane and Methanol as fuels. Tests were done at five different lambda values, each time keeping lambda constant and adjusting the Hot Tube temperature (HTT) to determine the effect this had on the compression ratio needed to sustain the different running conditions i.e. starting, knocking and misfiring. A detailed discussion of the engine operating conditions and the exact test procedure that was followed can be found in Chapter 4.

5.1.1 N-Heptane

Tables 5-1 – 5-5 show the results of tests done at the five different lambda values using N-Heptane. The values shown are averages of three runs at each operating point.

Table 5-1 Compression ratios needed at $\lambda = 2.0$

Lambda 2.0			
HTT*	**Start**	**Knock**	**Misfire**
150	8.17	8.17	7.64
200	7.97	7.97	7.57
250	7.96	7.98	7.50
300	7.94	7.98	7.47
350	8.09	8.12	7.54
400	8.04	8.08	7.47

*HTT = Hot Tube Temperature.

Table 5-2 Compression ratios needed at $\lambda = 2.5$

Lambda 2.5			
HTT	**Start**	**Knock**	**Misfire**
150	8.78	9.46	8.46
200	8.73	9.38	8.43
250	8.71	9.19	8.40
300	8.68	9.01	8.42
350	8.76	9.35	8.45
400	8.73	9.32	8.45

Table 5-3 Compression ratios needed at $\lambda = 3.0$

Lambda 3.0			
HTT	**Start**	**Knock**	**Misfire**
150	10.14	11.44	-
200	10.06	11.44	-
250	9.94	11.20	-
300	9.73	11.24	-
350	9.95	11.44	-
400	10.04	11.20	-

Table 5-4 Compression ratios needed at $\lambda = 3.5$

Lambda 3.5			
HTT	**Start**	**Knock**	**Misfire**
150	10.98	-	-
200	10.92	-	-
250	10.89	-	-
300	10.34	-	-
350	10.53	-	-
400	11.00	-	-

Table 5-5 Compression ratios needed at $\lambda = 4.0$

Lambda 4.0			
HTT	**Start**	**Knock**	**Misfire**
150	11.86	-	-
200	11.55	-	-
250	11.87	-	-
300	11.53	-	-
350	11.47	-	-
400	11.50	-	-

The reason for there not being any data listed for the knocking and misfiring conditions for $\lambda = 3$, 3.5 and 4 is that accurate measurements could not be made at these conditions. At $\lambda = 2$ and $\lambda = 2.5$ there was a very definite compression ratio at which the main heat release started or stopped occurring, while at $\lambda = 3$, $\lambda = 3.5$ and $\lambda = 4$ there was a gradual increase or decrease in heat release as one increased or decreased the compression ratio. In other words, at $\lambda = 2$ and $\lambda = 2.5$ the engine was either running or not, while at $\lambda = 3$, $\lambda = 3.5$ and $\lambda = 4$ partial combustion would take place for a range of compression ratios smaller than the ones listed as the starting compression ratio. A possible explanation for this can be that at $\lambda = 2$ and $\lambda = 2.5$ the fuel concentration in the cylinder becomes too high to allow only partial combustion of the mixture. Once the main heat release starts occurring at some point in the combustion chamber the bulk of the mixture is "nudged" into autoigniting due to the high concentration and the rise in pressure and temperature in the cylinder.

The presence of the Hot Tube in the combustion chamber gave it acoustic characteristics which made measuring knocking very difficult as it introduced combustion chamber resonances which caused pressure oscillations that interfered with the identification of knocking at λ = 3 - 4. These pressure oscillations will be discussed in greater depth later in this chapter.

From these results it can be seen that for a set air/fuel ratio (lambda value) the compression ratio needed to sustain the different engine running conditions show no dependence on the Hot Tube temperature.

Table 5-6 and Fig. 5-1 below shows the average compression ratios (calculated from the results given above) needed to sustain the three operating conditions.

Table 5-6 Averaged compression ratios

Lambda	Start	Knock	Misfire
2	8.03	8.05	7.53
2.5	8.73	9.28	8.43
3	9.98	11.33	-
3.5	10.78	-	-
4	11.63	-	-

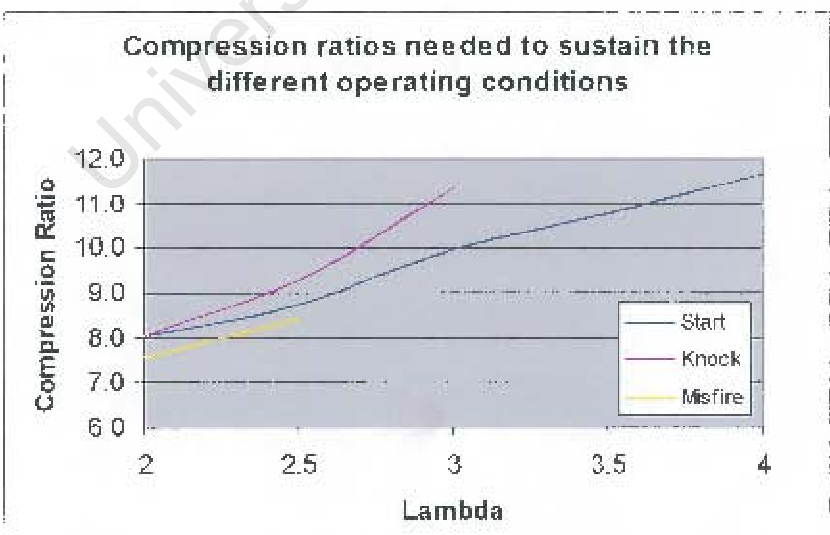

Figure 5-1 Compression ratios needed to sustain the different operating conditions

As was expected the compression ratio needed increases as lambda increases due to the higher pressures and temperatures needed to achieve autoignition of the leaner mixtures.

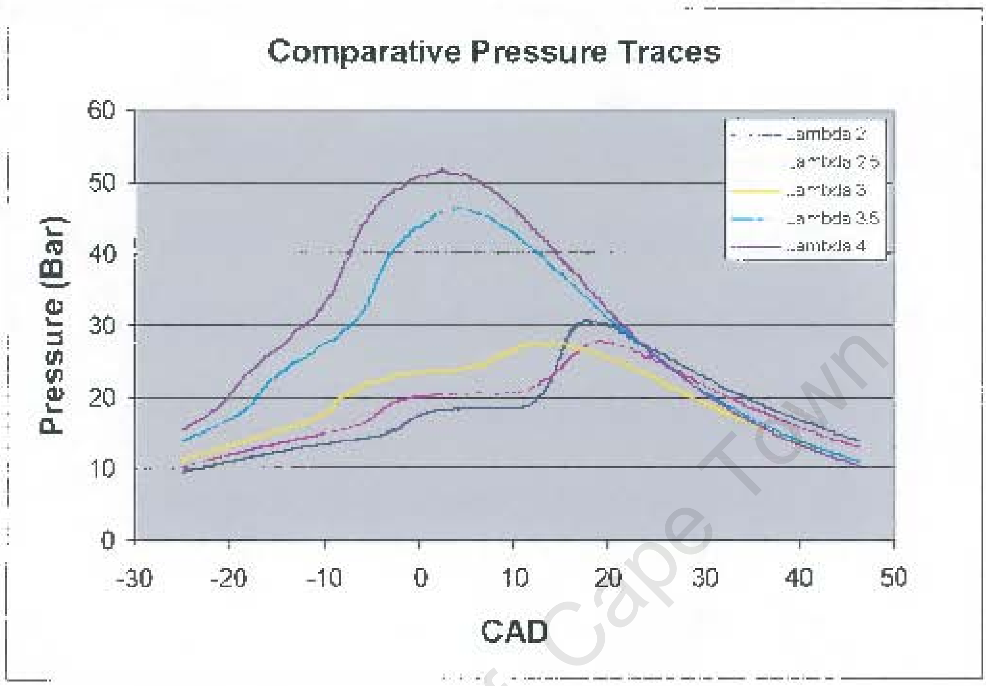

Figure 5-2 Comparative pressure traces for different λ-values.

From Fig. 5-2 above it can be seen that N-Heptane shows very distinct two-stage ignition which is a characteristic of paraffinic fuels. These pressure traces were not taken at the same compression ratio and that explains the shift in phasing as λ increases.

5.1.2 Methanol

After finishing the N-Heptane tests, it was attempted to run the engine on methanol. Although it has a high octane number, methanol is also known to have a low surface ignition temperature [Taylor, 1977] which could make it suitable for use in conjunction with the Hot Tube.

Every attempt was made to get the engine to run in HCCI mode using the identical parameters set out for the N-Heptane test, but to no avail. Thereafter the following steps were taken to try and make the engine run:

- Raising the compression ratio to the maximum value of 16.6:1
- Increasing the inlet air temperature from 340 K to 390 K
- Decreasing λ from 2 to ~1.25
- Lowering the engine speed from 1000 rpm to 600 rpm

All of the above changes could still not get the engine to run. It seems then that the Hot Tube was also not effective in initiating HCCI combustion with methanol.

5.2 Comparison between Hot Tube and Normal Running

After running the engine with the Hot Tube fitted, the tests were repeated for the same conditions but this time without the Hot Tube. As was the case when testing with the Hot Tube, λ was kept constant and the compression ratio changed to determine the compression ratio's needed to sustain the three different engine running conditions.

5.2.1 N-Heptane

N-Heptane was used for the first set of tests and the results are presented below.

5.2.1.1 Required compression ratio determination

Table 5-7 shows the compression ratios needed to sustain the different engine running conditions for the five λ-values tested. The values are averages of three runs at each operating point

Table 5-7 Compression ratios needed for different engine running conditions

Lambda	Start	Knock	Misfire
2	7.57	7.57	7.24
2.5	8.32	8.32	8.02
3	9.05	9.39	8.79
3.5	9.96	12.19	-
4	11.13	-	-

Now Table 5-8 shows a comparison between the compression ratios needed for Hot Tube and normal running.

Table 5-8 Comparison between Hot Tube and normal running

Lambda	Start		Knock		Misfire	
	Normal	HT	Normal	HT	Normal	HT
2	7.57	8.03	7.57	8.05	7.24	7.53
2.5	8.32	8.73	8.32	9.28	8.02	8.43
3	9.05	9.98	9.39	11.33	8.79	-
3.5	9.96	10.78	12.19	-	-	-
4	11.13	11.63	-	-	-	-

It is interesting to note the difference in the compression ratios needed for the different running conditions when comparing Hot Tube and normal running.

The fact that the compression ratio needed for starting was consistently higher for the Hot Tube can be explained in the following way. Poor scavenging of exhaust gases in the Hot Tube during exhaust stroke causes dilution of the fresh charge that gets compressed into the Hot Tube during the next cycle. This dilution causes an increase in the ignition delay and to now obtain complete combustion of all the charge in the combustion chamber, the compression ratio needs to be higher to get the diluted portion of the mixture in the Hot Tube to autoignite.

This is an unexpected result as the combustion model suggested that the heat transfer from the wall of the Hot Tube to the compressed gas would heat the gas sufficiently to make it autoignite before the bulk gas in the main part of the combustion chamber. The effect of exhaust residuals were not taken into account explicitly in the model but the work done by Londleni (2005) suggested that the level of dilution expected in the Hot Tube would not influence the ignition delay significantly. If poor scavenging is the reason for the poor performance of the Hot Tube, the dilution clearly has a much greater influence than expected.

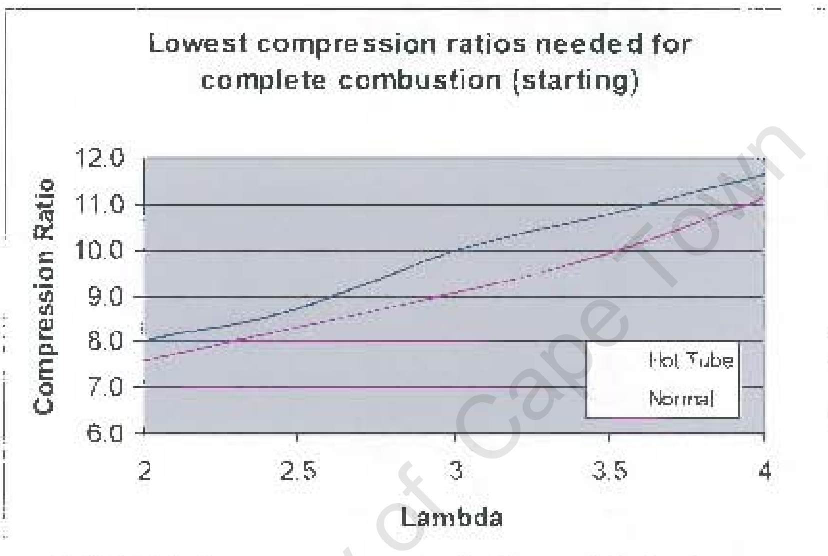

Figure 5-3 Comparison of compression ratios needed for complete combustion.

Not much can be said about the difference in compression ratio at knocking conditions because the measurements taken with the Hot Tube fitted were much less accurate than those for normal running without the Hot Tube. This is due to combustion chamber resonance effects as mentioned earlier. Much lower levels of pressure oscillations (knock) were detectable under normal running conditions and therefore all the compression ratios measured are lower.

5.2.1.2 Comparison of heat release

As discussed under "Hot Tube ignition" in Chapter 2 the use of the Hot Tube was not only intended to initiate HCCI combustion, but to also lower the rate of heat release

(ROHR) by prolonging combustion. By analysing the in-cylinder pressure data, the rates of heat release and total heat released (THR) can be calculated. A comparison between the heat release rates will therefore shed further light on the effectiveness of the Hot Tube. The method for calculating ROHR and THR is discussed in Appendix B.

Figures 5-4 – 5-8 below show a comparison of the heat release rates between Hot Tube and normal running. The graphs show the rates of heat release for the different λ-values where for each λ-value the compression ratio was the same for Hot Tube and normal running.

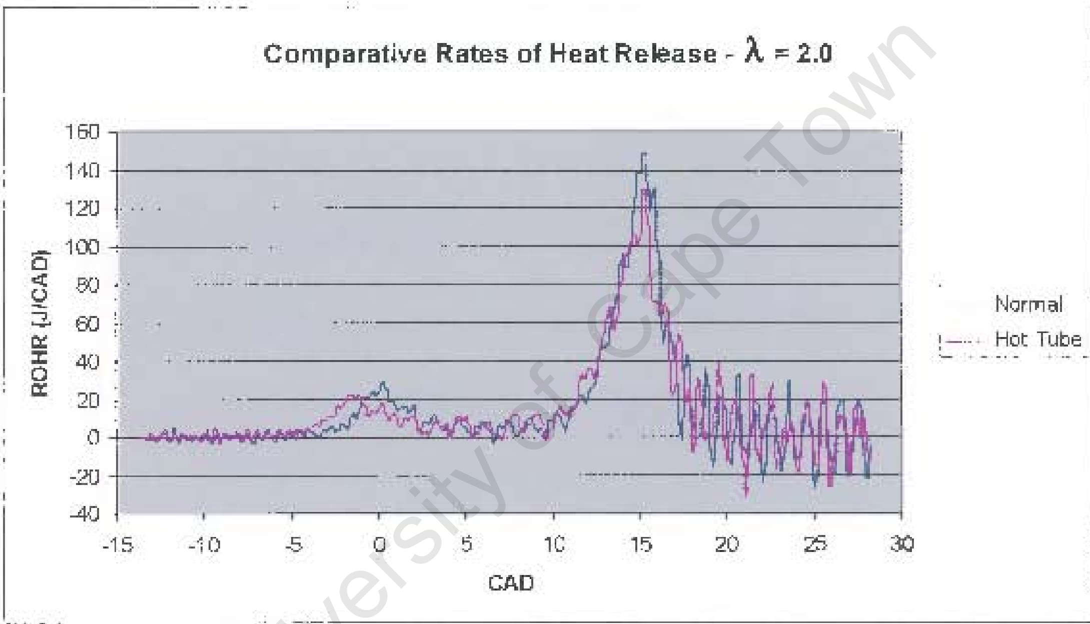

Figure 5-4 Comparison of rates of heat release at λ = 2

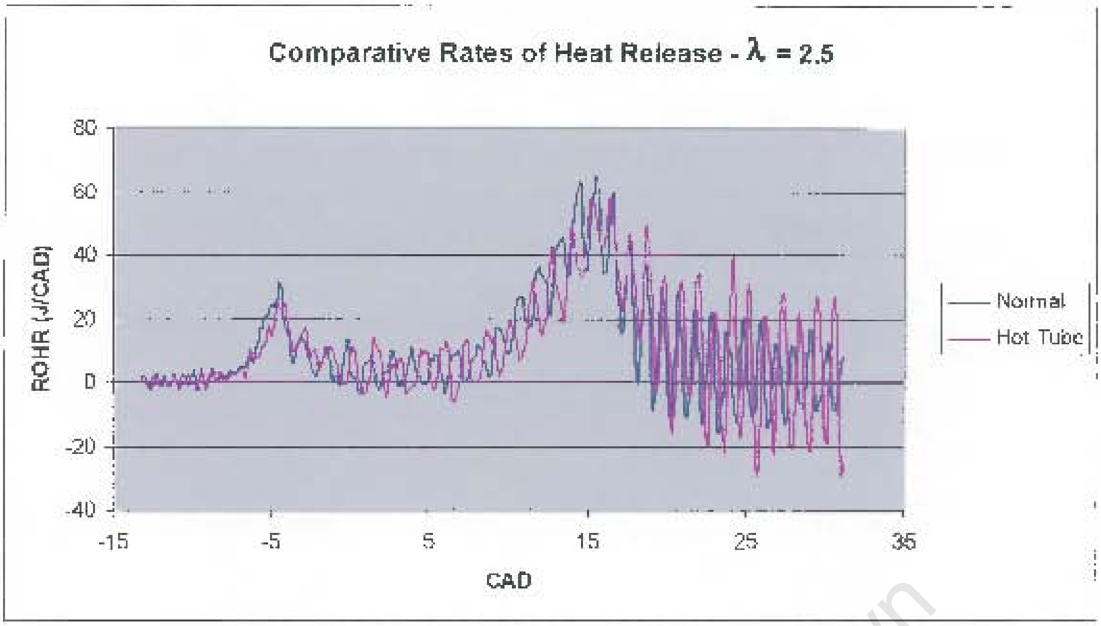

Figure 5-5 Comparison of rates of heat release at λ = 2.5

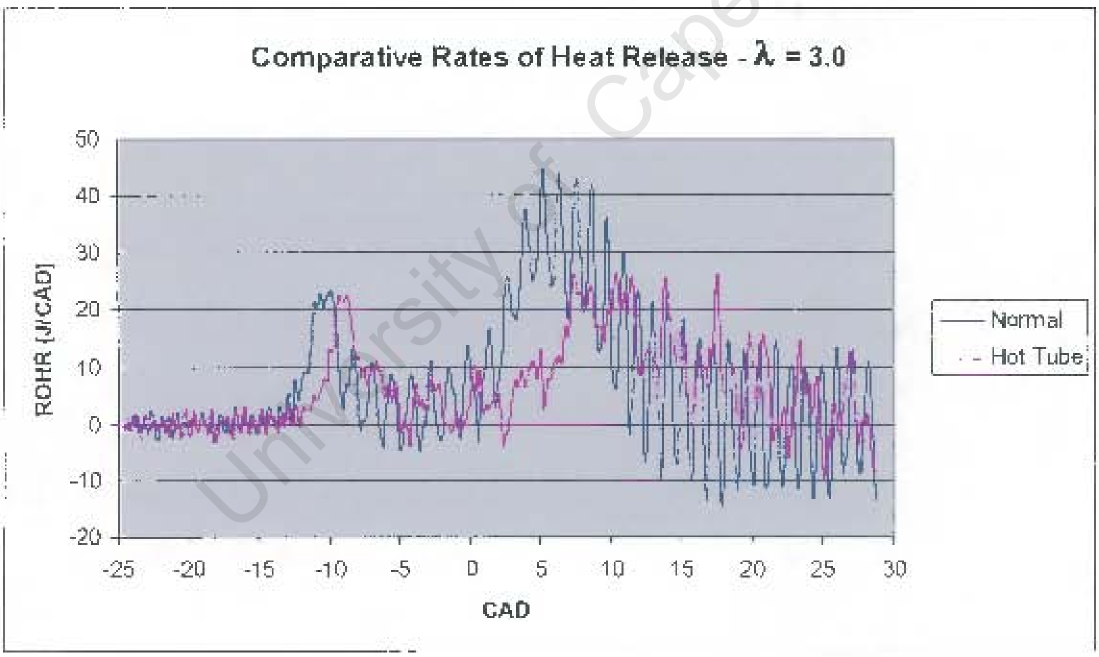

Figure 5-6 Comparison of rates of heat release at λ = 3

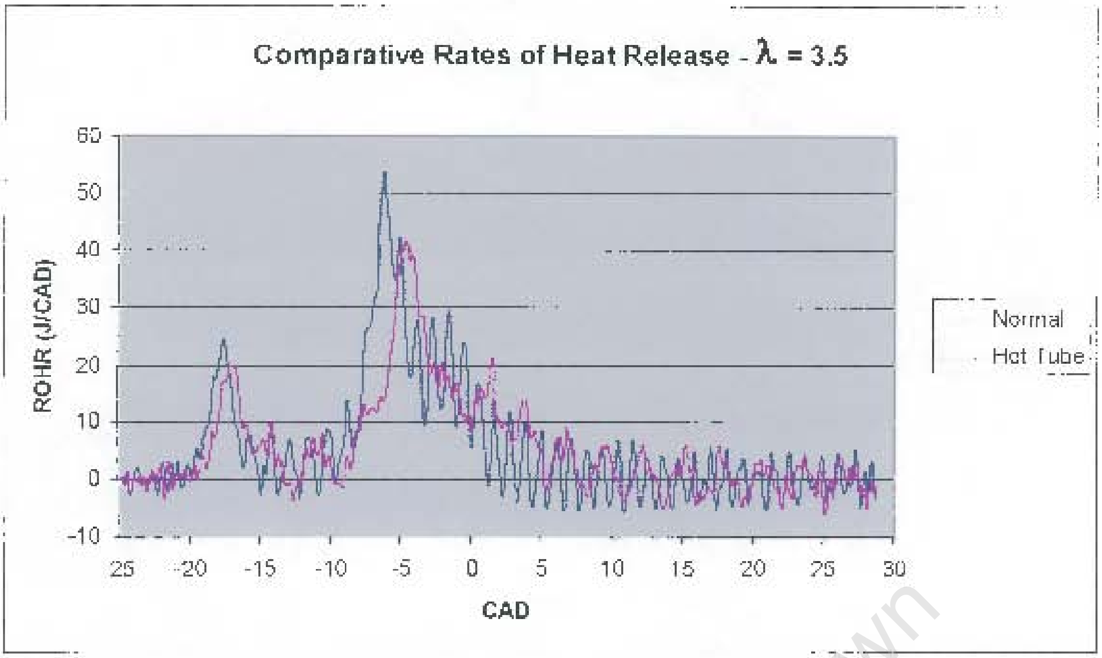

Figure 5-7 Comparison of rates of heat release at λ = 3.5

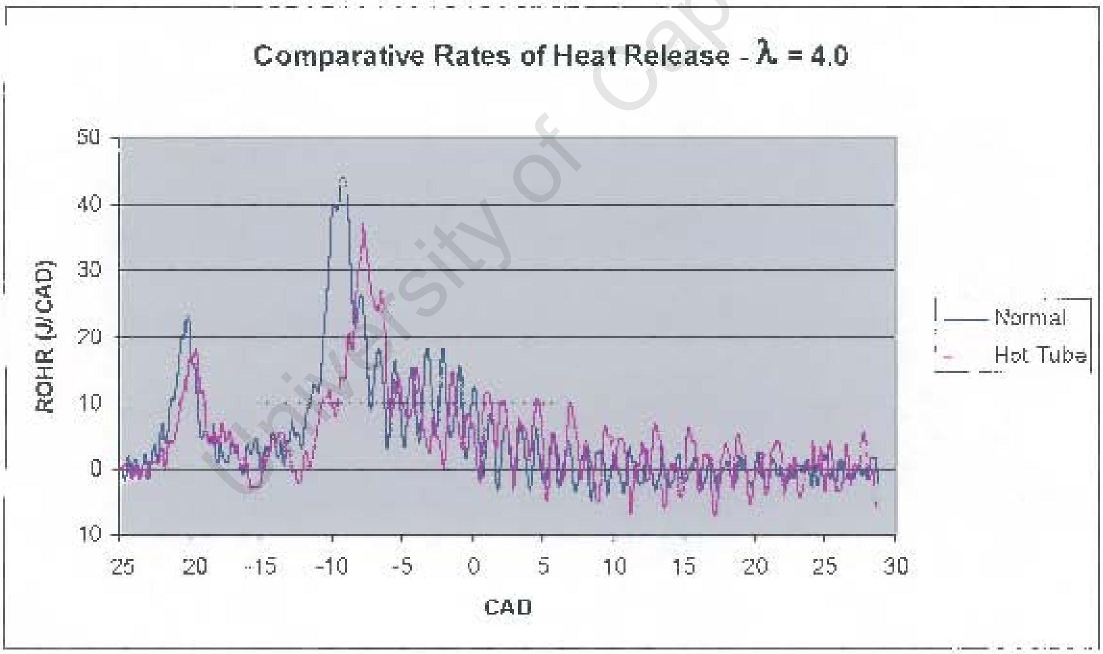

Figure 5-8 Comparison of rates of heat release at λ = 4

From Fig. 5-4 – 5-8 a clear trend is visible where the heat release for Hot Tube running occurs later and the rate is lower than under normal running conditions. Again, this can only be explained by the fact that there are more exhaust residuals present under Hot Tube running conditions compared to normal running. Diluting the mixture delays heat release and also lowers the rate of heat release.

This is also an unexpected result. It was thought that the Hot Tube would advance autoignition but judging by these results it is clearly not the case.

The use of the Hot Tube causes the combustion chamber to have an irregular shape and it seems that this geometrical effect overrides any of the other expected effects. As was shown in section 5.1.1 the Hot Tube temperature has no effect on engine operation and this strengthens the argument that its thermal effect is not nearly as great as was expected. It has to be said that the Hot Tube temperature could only be controlled to values up to 400°C. It was limited by the heating element used. Further work could include finding a way to heat the Hot Tube to higher temperatures and seeing whether this has any effect.

5.2.1.3 Comparison of total heat released

Figures 5-9 – 5-13 show a comparison of the total heat released for the different λ-values.

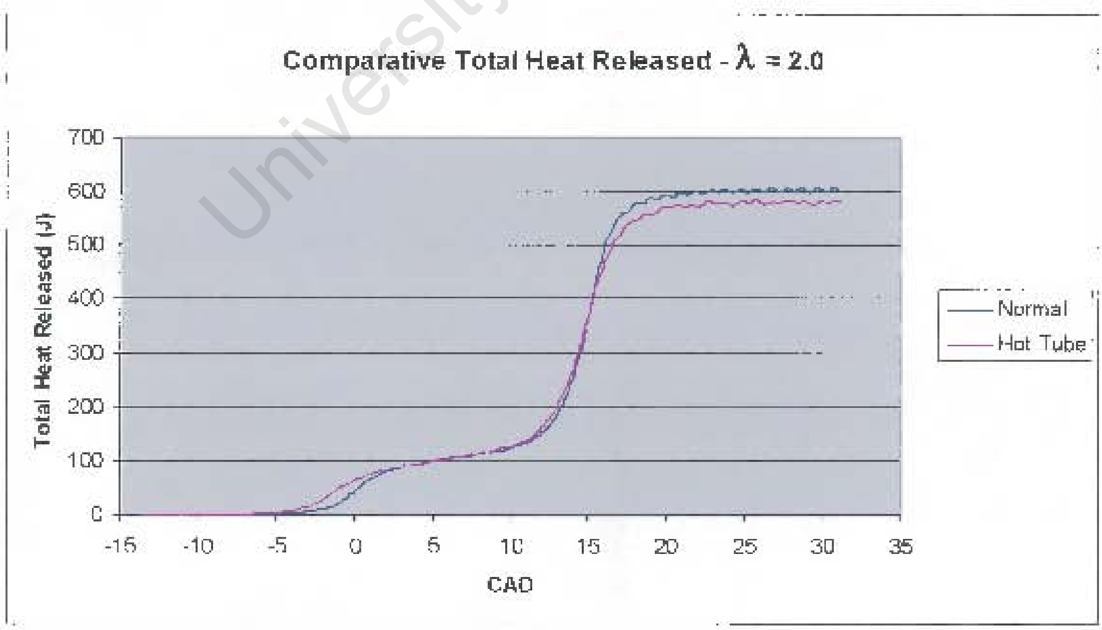

Figure 5-9 Comparison of total heat released – λ = 2

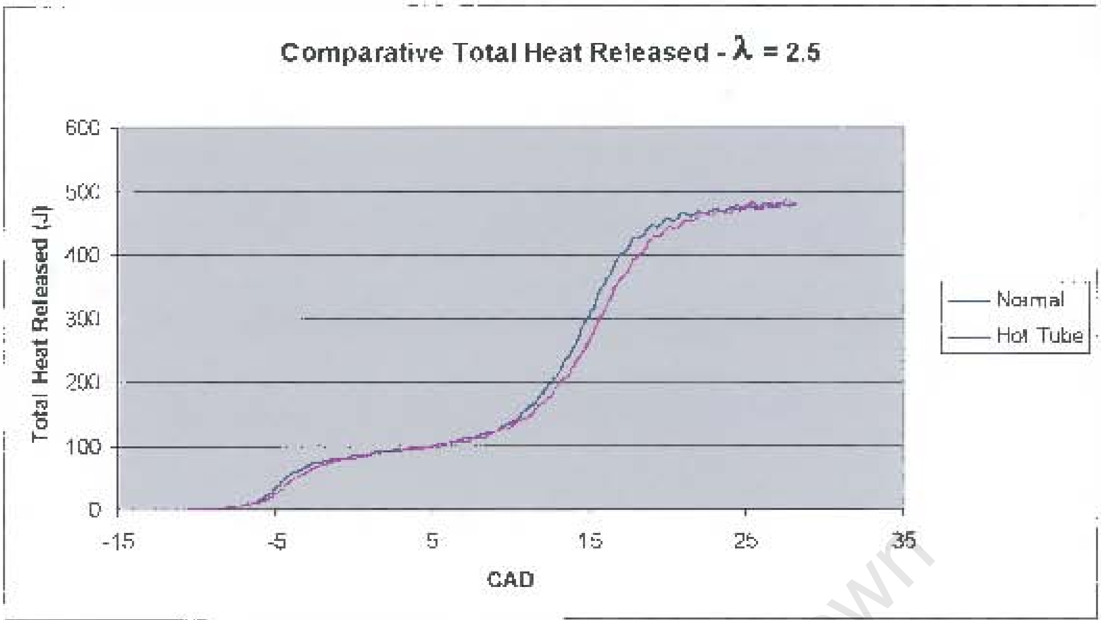

Figure 5-10 Comparison of total heat released – λ = 2.5

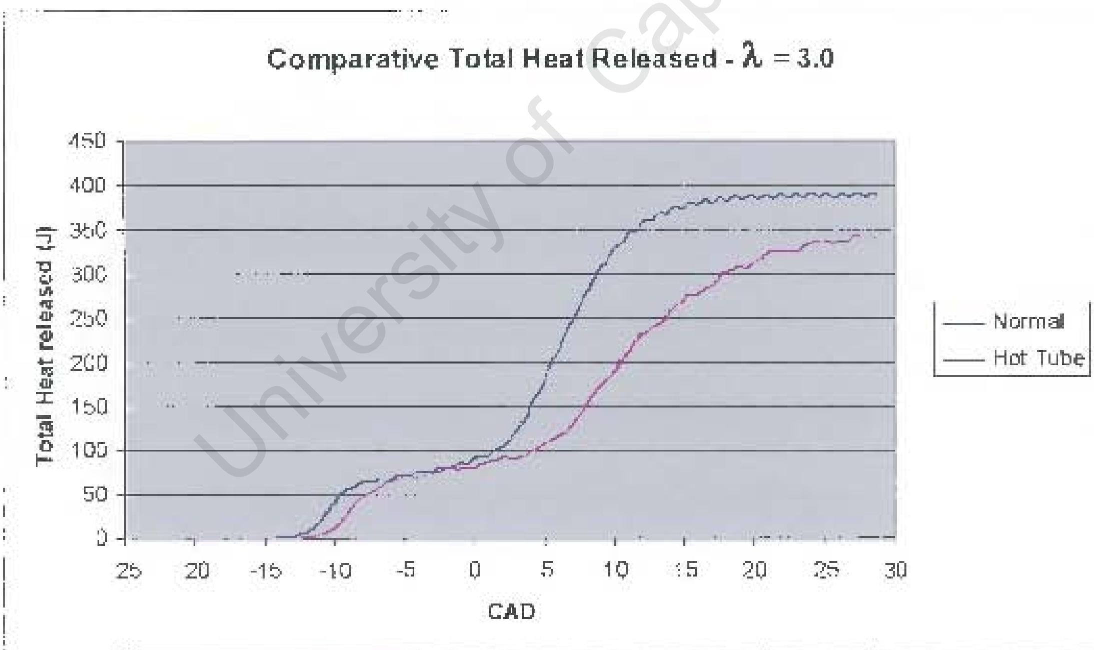

Figure 5-11 Comparison of total heat released – λ = 3

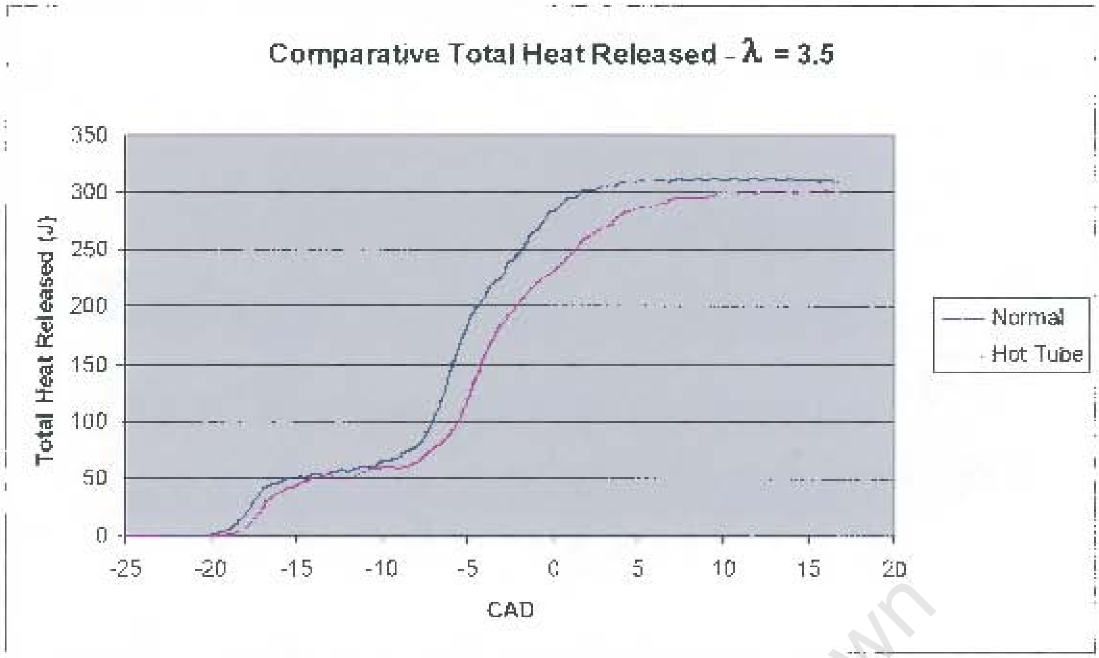

Figure 5-12 Comparison of total heat released – λ = 3.5

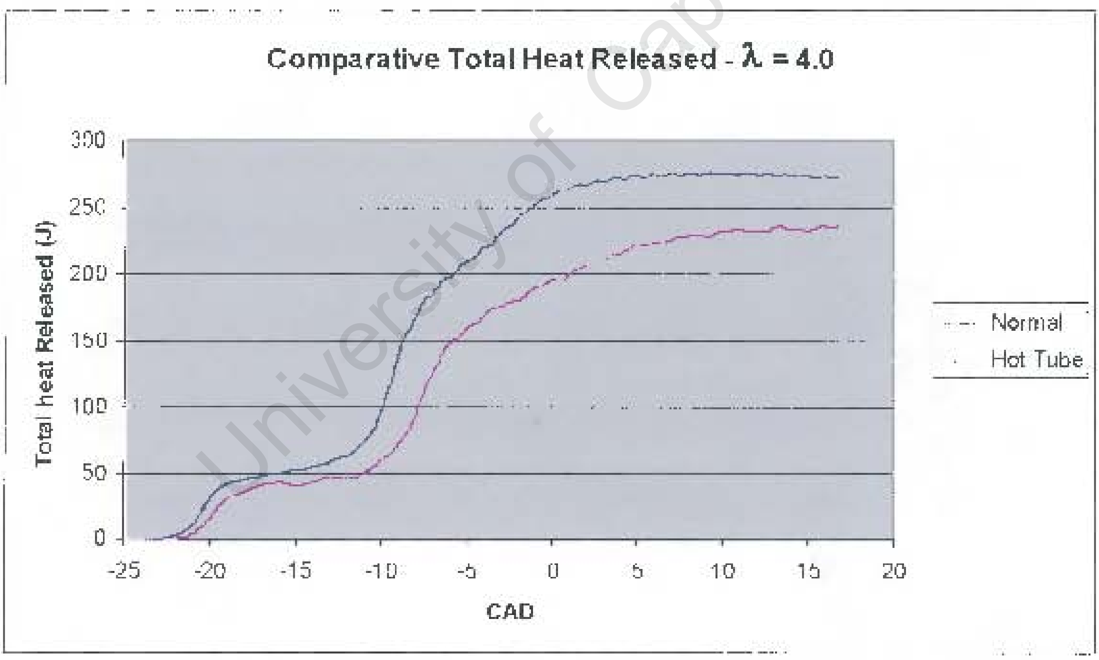

Figure 5-13 Comparison of total heat released – λ = 4

Figures 5-9 – 5-13 show that for each case total heat released is lower when using the Hot Tube. This can be attributed to the increased amount of residual gases present during Hot Tube running. Due to the residuals less fuel and air can enter the cylinder than during normal running. Less fuel in the cylinder means less energy and this reflects clearly on the graphs. At $\lambda = 3$ and $\lambda = 4$ the difference is substantial and can not only be due to the presence of residuals. It is suspected that inaccurate fuelling caused these discrepancies. The method used to obtain the data, from which these graphs were derived, was to first run the engine with the Hot Tube fitted and then remove the Hot Tube and repeat the runs at the different λ-values. The fuelling is adjusted manually on the engine and the air/fuel ratio is checked using a lambda sensor. This is a rather crude method and therefore it is impossible to be sure that the exact same amount of fuel was injected for the two runs at each λ-value.

To confirm the results shown above another test was run. The result is shown in Fig. 5-14.

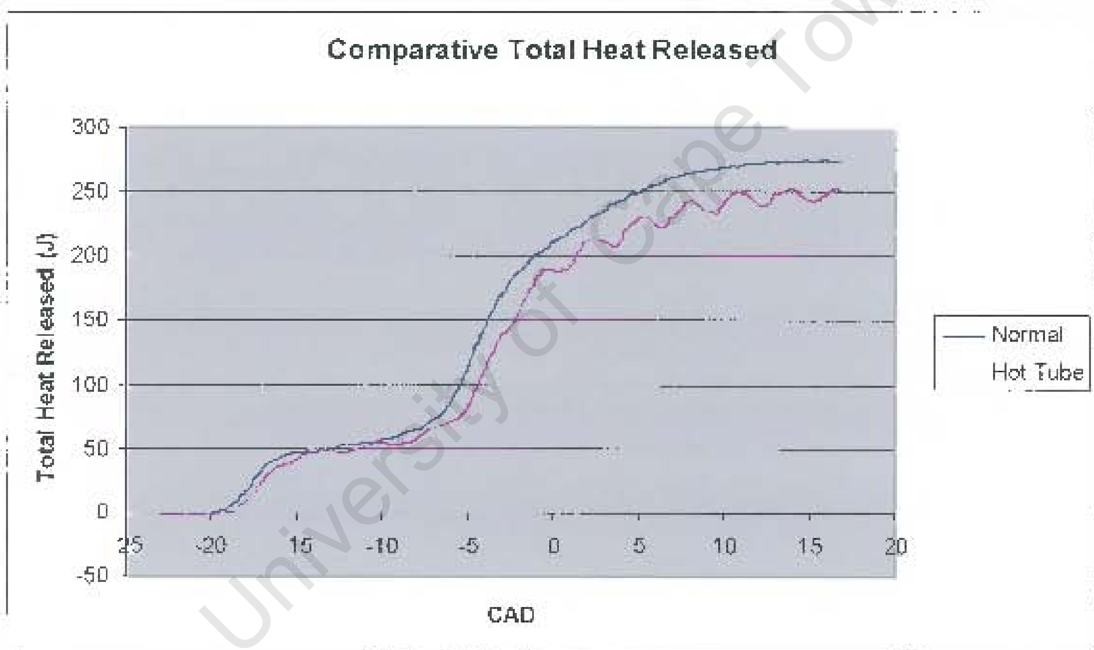

Figure 5-14 Comparison of total heat released with fuelling kept constant

This time the Hot Tube was in place for the first run and then removed for the second run. The fuelling was not adjusted at all between the two runs and thus the result can be considered repeatable. This proved beyond doubt that total heat released was lower when using the Hot Tube.

In all cases, the calculated heat release results obtained correlate well with the theoretical amount of heat that should be released for complete combustion based on fuel mass flow and the calorific value of the fuel.

5.2.1.4 Correlation with combustion model

Having obtained all the experimental results shown and discussed above, it can now be compared to the results predicted by the combustion model. Table 5-9 gives the predicted combustion phasing for Hot Tube and normal running and compares it to the actual results obtained from the engine. The actual results were obtained by noting the start of the main heat release on the ROHR graphs (Figs. 5-4 – 5-8).

Table 5-9 Comparison of predicted and actual combustion phasing (Unit °ATDC)

		HT		Normal	
Lambda	**CR**	**Predicted**	**Actual**	**Predicted**	**Actual**
2	7.6	4°	9°	12°	11°
2.5	8.2	2°	6°	6°	5°
3	9.2	-2°	2°	-2°	-2°
3.5	11.8	-6°	-11°	-6°	-13°
4	14.0	-6°	-13°	-6°	-15°

The predicted results correlate fairly well with the actual results obtained especially for normal running. The predicted results for Hot Tube running do not correlate with actual results as well as with normal running. The reasons for this were discussed in 5.2.1.2 and 5.2.1.3. For the least it can be said that the results are directionally consistent with a reasonable spread.

5.2.1.5 Summary

Table 5-10. gives some further results of the analysis of cylinder pressure data from the tests done with n-Heptane. In the table CA 10 and CA 50 are the Crank Angles at which 10% and 50% of the total heat had been released, respectively.

For each condition CA 50 corresponds well with the position at which the maximum rate of heat release happened. This is a trend that is in accordance with literature

[Kalghatgi et al, 2003]. It is also interesting to note that the maximum rate of pressure rise is consistently lower for Hot Tube running. The pressure rise is caused by the combustion of the fuel and the ROHR determines the rate of pressure rise. It was shown previously that the ROHR was lower for Hot Tube running and it follows then that the rate of pressure rise will be lower for the same reasons.

Table 5-10 Further results of the n-Heptane tests

LAMBDA	Test Condition	Max dP/dt (bar/CAD)	THR (J)	CA 10	CA 50	Max ROHR (J/CAD)	Position Max ROHR (J/CAD)	Max P (Bar)	Position Max P (CAD)
2.0	Normal	4.6	608	0.8	14.5	148.8	15.3	32.2	17.9
	Hot Tube	3.7	585	-0.2	14.3	129.4	15.3	30.8	17.9
2.5	Normal	1.6	486	-4.4	13.7	65.1	15.6	28.8	18.0
	Hot Tube	1.5	490	-3.7	14.6	58.7	15.2	27.9	19.1
3.0	Normal	1.6	390	-10.1	5.3	44.7	5.2	34.1	11.1
	Hot Tube	1.4	347	-8.8	9.1	26.4	7.4	27.5	11.8
3.5	Normal	3.8	313	-17.4	-5.7	53.8	-6.2	49.2	2.0
	Hot Tube	3.3	303	-16.6	-4.1	41.7	-4.6	46.4	4.3
4.0	Normal	4.6	275	-20.2	-9.0	43.9	-9.2	58.0	1.6
	Hot Tube	3.5	237	-19.5	-7.2	36.8	-7.7	51.7	2.5

5.2.2 Methanol

As was mentioned in 5.1.2 the engine could not be run on methanol in HCCI mode with the Hot Tube in place. Removing the Hot Tube made it possible to raise the CR to a higher level than with the Hot Tube in place. This enabled HCCI running and the pressure trace, rate of heat release and total heat released are shown in fig. 5-15 – 5-17.

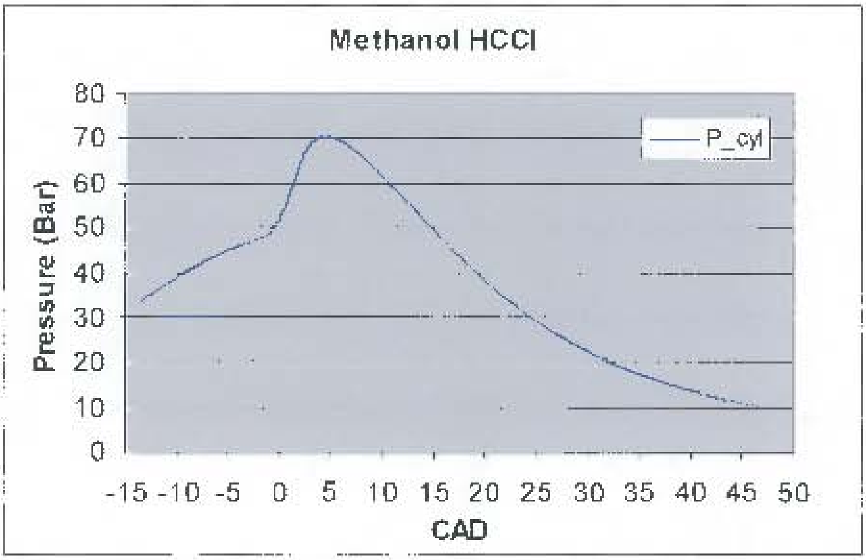

Figure 5-15 Methanol HCCI pressure trace

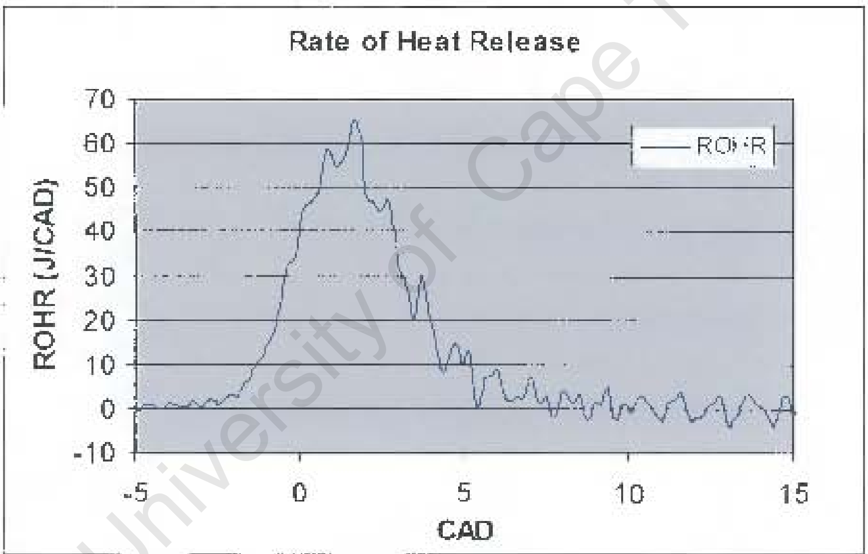

Figure 5-16 Rate of heat release

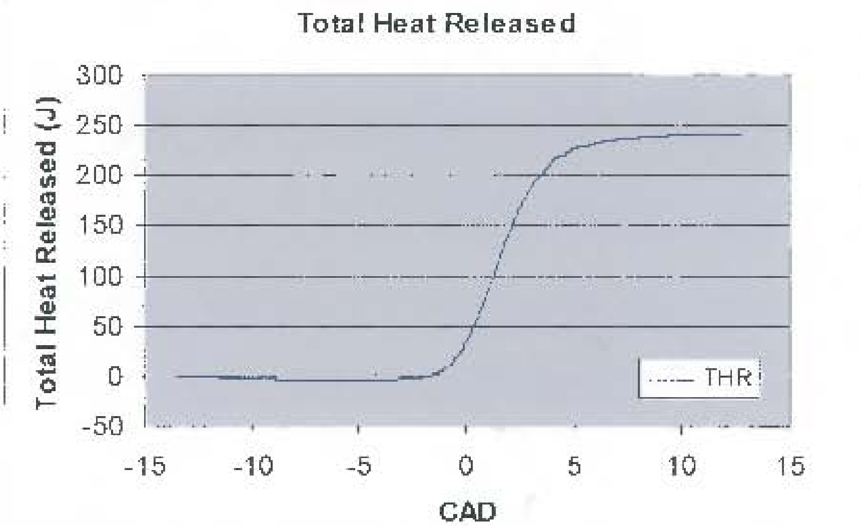

Figure 5-17 Total heat released

It is worth noting the single stage heat release of methanol compared to the two-stage heat release experienced with n-Heptane. This is typical of non-paraffinic fuels.

HCCI combustion was achieved at the following engine operating conditions shown in Table 5-11:

Table 5-11 Engine operating conditions for HCCI using methanol

Lambda	4
T_inlet	~390 K
RPM	1000
CR	19.55

No direct comparison can be made between Hot Tube and normal running with methanol since HCCI was only achieved without the Hot Tube, but only because the CR could be raised to 19.55:1 while the maximum CR that could be used with the Hot Tube was only 16.6:1.

5.3 Pressure Oscillations

In-cylinder pressure oscillations were visible in several of the experiments in which the Hot Tube was used. It is thought that the presence of the Hot Tube gives the combustion chamber certain acoustic characteristics that cause these oscillations. These oscillations are not to be confused with oscillations caused by knocking combustion for which the frequencies are much higher. Fig. 5-18 shows a typical example of these oscillations in a test done with n-Heptane at λ = 4. Three consecutive traces are shown.

Further investigation into possible in-cylinder acoustic vibrations showed that the characteristic frequencies experienced in the experiments do not correlate with any known acoustic vibration modes. The frequencies are much lower and Table 5-12 compares the measured frequencies to the possible resonant frequencies given by C.S. Draper's acoustic theory (1938).

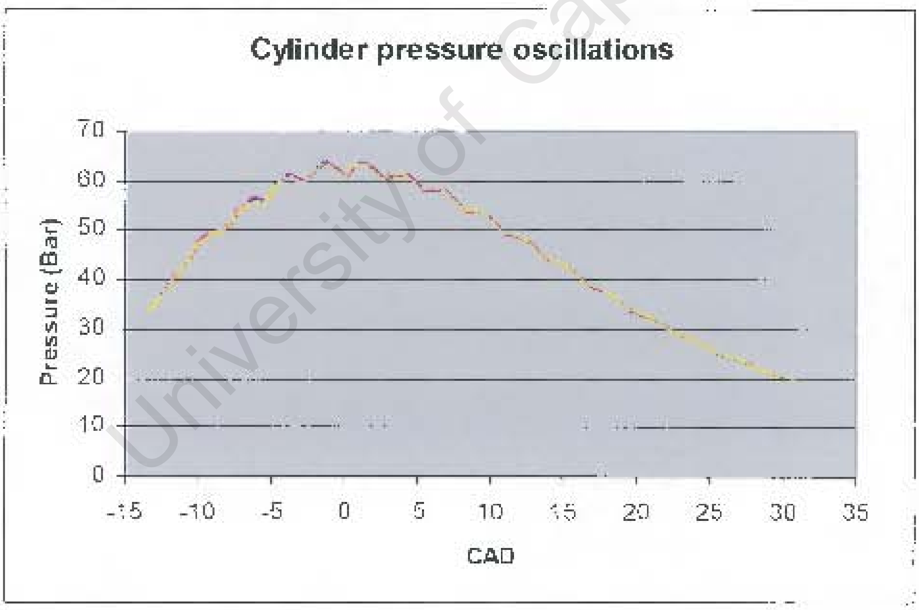

Figure 5-18 Cylinder pressure traces showing pressure oscillations

Table 5-12 Comparison of measured and theoretical resonant frequencies

Mode	Frequency (Hz)
Measured Frequency	±2200
First Circumferential	4616
Second Circumferential	7649
First Radial	10532
Third Circumferential	9602

This is a surprising result and the only possible explanation that can be given for these oscillations are that the presence of the Hot Tube changes the combustion chamber's acoustic properties significantly and therefore Draper's theory for calculating resonant frequencies does not apply. Since acoustic analysis is not the focus of this study, it will be left at that and be recommended as a point for further investigation.

Pressure oscillations were also present during tests without the Hot Tube, but their frequencies correlate with those found under knocking conditions in SI engines and the ones calculated from Draper's theory. Studies by Eng (2002) and Vressner et al. (2003) on pressure oscillations during HCCI combustion came to a similar conclusion and thus this phenomenon can be attributed to known acoustic behaviour under rapid pressure rise rates.

6 Conclusions and Recommendations

6.1 Effectiveness of the Hot Tube igniter

- The Hot Tube proved to be ineffective in initiating or advancing HCCI combustion at all the conditions tested. It was expected that the Hot Tube would advance or promote HCCI combustion, but in all cases it retarded combustion. It is believed that poor scavenging of the Hot Tube causes dilution of the mixture at the end of compression and this had a more pronounced effect on ignition delay than was expected.
- The temperature of the Hot Tube was too low to induce hot surface ignition of the mixture. The maximum temperature the tube could be heated to was 400°C and this was limited by the heating element used. Therefore the only way that the tube could influence combustion was by transferring heat to the mixture to shorten the ignition delay.
- The Hot Tube was effective in lowering the rate of pressure rise in all the cases tested. This could be a useful finding as it shows that by introducing an irregularity into the combustion chamber, combustion duration is increased.
- The Ricardo E6 proved to be a suitable engine for HCCI research and the improvements made to the existing engine setup makes it possible to use the engine for further HCCI research and testing.

6.2 Recommendations for further work and system improvements

Recommendations for further work include the following:

- Experiment with the use of higher Hot Tube temperatures to determine whether it has any effect on combustion.
- Although it may be technically challenging, experiment with irregularly shaped combustion chambers to see if it can reduce the rate of pressure rise during combustion.

Recommendations for system improvements include the following:

- Currently the amount of fuel that is injected into the engine is determined by a lambda sensor in the exhaust and adjusted manually by turning a dial on the control unit. It is recommended that a more reliable and repeatable way is found to measure fuel flow. All the instrumentation is in place, it is merely a matter of sorting out the LabView application to read inputs and calculate values correctly.
- Related to the first point above is another software issue. As was mentioned, the engine is fitted with all the necessary instrumentation to enable online condition monitoring. A great deal of effort was spent to try and set up a LabView-based application to read and display various operating parameters, but a lack of computer programming expertise and some electronic bugs got the better of the author. It is therefore recommended that an expert be appointed to sort out these problems and set up a reliable and user friendly control and condition monitoring software application.

7 References

Aceves S.M. et al. (2003). Fuel and Additive Characterisation for HCCI Combustion. SAE Technical Paper, No. 2003-01-1814.

Christensen M. et al. (1998). Supercharged Homogeneous Charge Compression Ignition. SAE Technical Paper, No. 980787.

Christensen M. et al. (1999). Demonstrating the multi fuel capability of a HCCI engine with variable compression ratio. SAE Technical Paper, No. 1999-01-3679.

Christensen M. et al. (1999). Homogeneous Charge Compression Ignition with water injection. SAE Technical Paper, No. 1999-01-0182.

Draper C.S. (1938). Pressure Waves Accompanying Detonation in the Internal Combustion engine. Journal of Aeronautical Sciences, Vol. 5, No. 6. p219-226.

Douaud A.M. et al. (1978). Four Octane Number Method for Predicting the Anti-knock Behaviour of Fuels and Engines. SAE Technical Paper, No. 780080.

Downs D. et al. (1951). An Experimental Investigation into Pre-ignition in the Spark-ignition Engine. Proc. Inst. Of Mech. Engineers, p125.

Duffy K. (2005). Heavy Duty HCCI Development [presentation]. SAE HCCI Symposium, Lund.

Eng J.A. et al. (2002). Characterisation of Pressure Waves in HCCI Combustion. SAE Technical Paper, No. 2002-01-2859.

Hagen D.L. (1977). Methanol as a Fuel: A Review with Bibliography. SAE Technical Paper, No. 770792.

Heywood J.B. (1988). Internal Combustion Engine Fundamentals, International Ed. McGraw Hill, London.

a) p567-667
b) p748-822

Hiraya K. et al. (2002). A study on the gasoline fuelled compression ignition engine – a trail of operation region expansion. SAE Technical Paper, No. 2002-01-0416.

Ishibashi Y. et al. (1996). Improving the exhaust emissions of two-stroke engines by applying the Activated Radical Combustion. SAE Technical Paper, No. 960742.

Kalghatgi G. et al. (2003). A Method of Defining Ignition Quality of Fuels in HCCI Engines. SAE Technical Paper, No. 2003-01-1816.

Kaneko N. et al. (2002). Expansion of the operating range with in-cylinder water injection in a premixed charge compression ignition engine. SAE Technical Paper, No. 2002-01-1743.

Law D. et al. (2000). Controlled Combustion in an IC-engine with fully variable valve train. SAE Technical Paper, No. 2000-01-0251.

Little T.A. (2003). Test Apparatus to Facilitate HCCI Combustion testing using Hot Tube ignition. Degree Dissertation. University of Cape Town, South Africa.

Londleni S. (2005). Personal Communication.

Ma T. (2001). Method of operating an IC engine. International Patent Application, WO 01/11233 A1.

Marriot C. et al. (2002). Experimental investigation of direct injection gasoline for premixed compression ignited combustion phasing control. SAE Technical Paper, No. 2002-02-0418.

Najt P.M. et al. (1983). Compression-ignited Homogeneous Charge Combustion. SAE Technical Paper, No. 830264.

Olsson J. et al. (2001). A turbo charged dual fuel HCCI engine. SAE Technical Paper, No. 2001-01-1896.

Schaberg P. (2005). MEC594Z Cleaner Engines, Cleaner Fuels – Course Notes. University of Cape Town.

Taylor C.F. (1977). The Internal-Combustion Engine in Theory and Practice, Volume 2: Combustion, Fuels, Materials, Design. MIT Press, Massachusetts. p153.

Thring R.H. (1989). Homogeneous Charge Compression Ignition (HCCI) engines. SAE Technical Paper, No. 892068.

Velji A. (2005). An Experimental Study to Assess the Potential of HCCI Combustion with Various Fuels for Light Duty Diesel [presentation]. SAE HCCI Symposium, Lund.

Viljoen C.L. et al. (2005). An Investigation of the Ignition Delay Character of Different Fuel Components and an assessment of Various Autoignition Modelling Approaches. SAE Technical Paper, No. SAEFL-100.

Vressner A. et al. (2003). Pressure Oscillations During Rapid HCCI Combustion. SAE Technical Paper, No. 2003-01-3217.

Xu H. (2005). The UK Foresight Vehicle Research on Extending HCCI Engine Operating Boundaries [presentation]. SAE HCCI Symposium, Lund.

Yates A.D.B. et al. (2004). Understanding the Relation between Cetane Number and Combustion Bomb Ignition Delay Measurements. SAE Technical Paper, No. 2004-01-2017.

Zhao H. et al. (2002). Performance and analysis of a four-stroke multi-cylinder gasoline engine with CAI combustion. SAE Technical Paper, No. 2002-01-0420.

Zhao F., Asmus T.W., Assanis D.N., Dec J.E., Eng J.A. and Najt P.M. (2003). Homogeneous Charge Compression Ignition (HCCI) Engines – Key Research and Development Issues. Warrendale, Society of Automotive Engineers.

a) p3
b) p3
c) p328
d) p328
e) p334
f) p331
g) p336
h) p330
i) p337
j) p8
k) p338
l) p336
m) p235
n) p235
o) p236

Appendices

Appendix A – Modifications to the Ricardo E6 Engine

Fuel Injection System

In its original form the Ricardo E6 was fitted with a carburettor for fuel supply to the engine. It was fed from a fuel tank mounted on an ancient wooden cabinet. A pipette was used for fuel flow measurement and was also mounted on the original cabinet. Figure A-1 shows the original cabinet.

Figure A-1 The original cabinet housing the fuel tank and fuel flow measurement pipet.

For this project the fitment of a fuel injection system was a necessity. It was decided to design and build the system in-house and to keep it as simple as possible. For this reason all the system hardware used was chosen to be readily available. The fuel pump, pressure regulator and injector are all standard Bosch units while the two fuel filters (high and low pressure) are GUD units acquired from a local motor spares dealer.

The electronic control board was designed to take a trigger input from the crank angle encoder and use a pulse width modulator to control the duration of the injection pulses based on an input from a manually operated pot. Figure A-2 shows an early version of the board used and Figure A3 shows the final fuel system

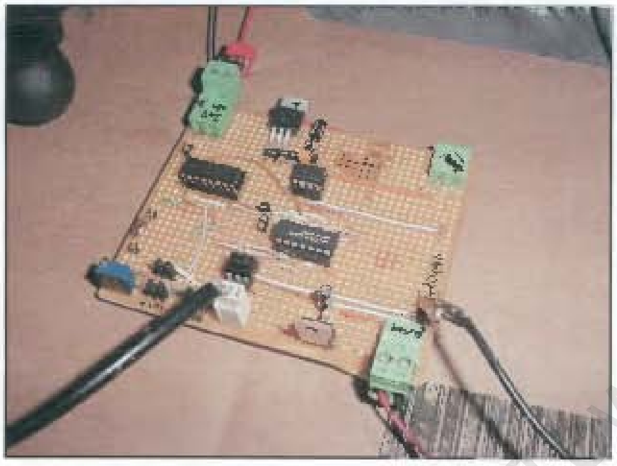

Figure A-2 Early version of the injection control board.

Figure A-3 The fuel injection system.

The fuel injector was calibrated to determine the range of pulse width outputs needed and the calibration curve is shown in Figure A-4.

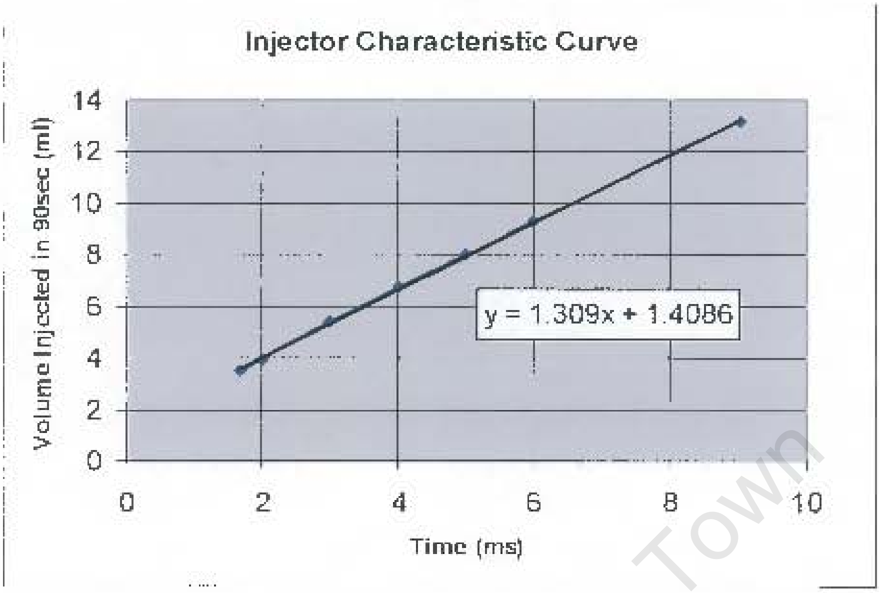

Figure A-4 Injector calibration curve.

The engine is fitted with a laminar flow element for air flow measurement and this was also calibrated for the purpose of being able to determine air/fuel ratio. The calibration curve is shown in Figure A-5.

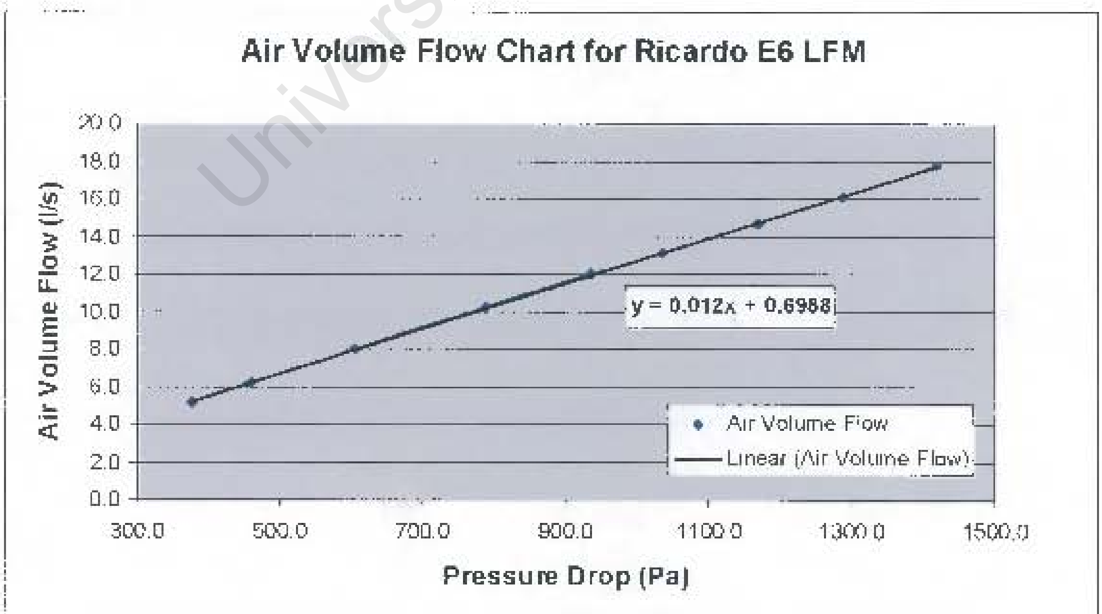

Figure A-5 Calibration curve for the laminar flow element

Dynamometer Controller

The dynamometer controller that was used on the original E6 engine was a very clumsy manually operated unit that did not support intermittent engine running very well. Switching between motoring and loading conditions was difficult and it was impossible to keep engine speed constant during intermittent engine running. The dynamometer used is essentially a DC motor and it was decided to fit a new electronic DC drive to control the electric motor. With this unit the speed can be kept constant and the drive then automatically switches between motoring and loading conditions to maintain the set speed. Figure A-6 and A-7 show the old controller and new DC drive respectively.

Figure A-6 The old dynamometer controller.

Figure A-7 The new DC Drive used for dynamometer control.

The following pages are excerpts from the instruction manual for the DC Drive.

DC400
Thyristor power converter

for DC drive systems

20 to 1000 A

9 to 522 kW

INSTRUCTION MANUAL

2 System overview of DC400

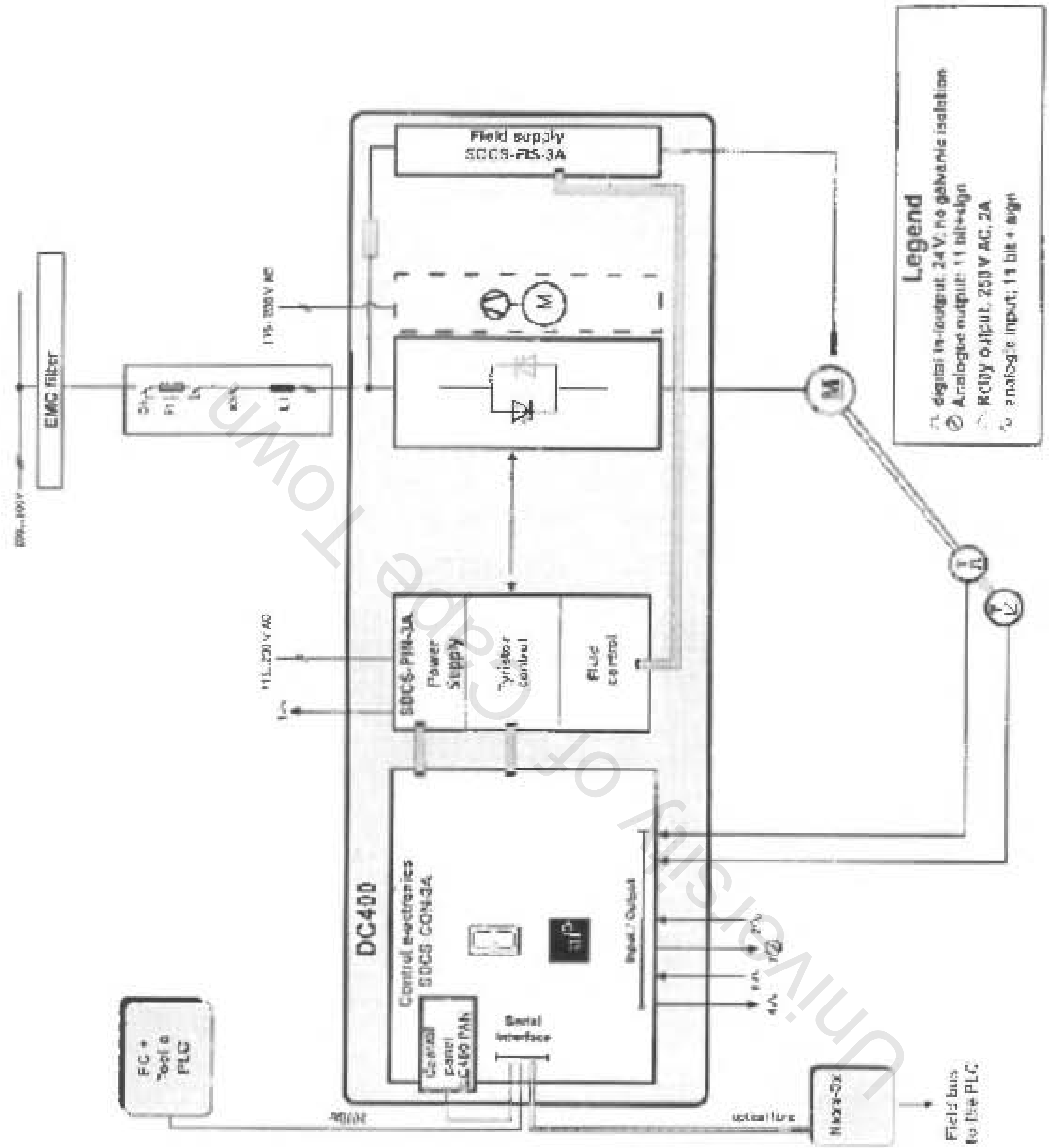

Fig. 2-1: System overview of DC400

SAFTRONICS
www.saftronics.com

II-1

2.1 Environmental conditions

System overview of DC400

Mains supply - power part

Voltage, 3-phase:	230 to 500 V in acc. with IEC 38
Voltage deviation	±10% permanent
Rated frequency:	50 Hz or 60 Hz
Static frequency deviation:	50 Hz ±2 %; 60 Hz ±2 %
Dynamic: frequency range:	50 Hz: ±5 Hz; 60 Hz: ± 5 Hz
df/dt:	17 % / s

Mains supply - Electronics supply

Voltage, 1-phase:	115 to 230 V in acc. with IEC 38
Voltage deviation:	-15% / +10%
Frequency range:	45 Hz to 65 Hz

Degree of protection

Power converter module:	IP 00

Paint finish

Power converter module, cover:	RAL 9002 light-grey
housing:	RAL 7012 dark-grey

Environmental limit values

Permissible ambient temp. with rated current I_{DC}:	40 to 105°F (+5 to +40°C)
Ambient temp., power conv. module:	105 to 120°F (+40°C to 55°C; s.Fig. 2.1/2
Alteration in the ambient temp.:	< 0.5°C / minute
Storage temperature:	-20 to 130°F (-40 to +55°C)
Transport temperature:	-20 to 158°F (-40 to +70°C)
Relative humidity:	5 to 95%, no condensation
Pollution degree:	Grade 2

Site elevation:

< 1000 m above M.S.L.:	100% without current reduction
> 1000 m above M.S.L.:	with current reduct., s. Fig. 2.1/1

Vibration converter module: 0,5 g, 5 Hz to 55 Hz

Noises: (1 m distance)	Size	as module
	A1	55 dBA
	A2	55 dBA
	A3	60 dBA
	A4	66...70 dBA, dependent on fan

Current reduction to (%) for armature circuit and field supply

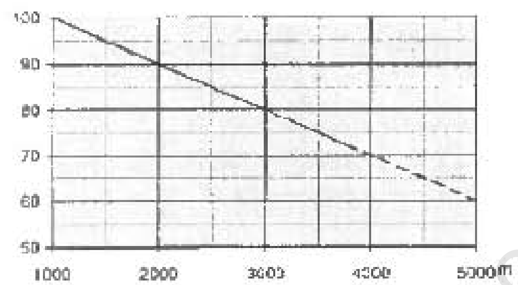

Fig. 2.1/1: Effect of the site elevation above sea level on the power converter's load capacity.

Current reduction to (%) for armature circuit and field supply

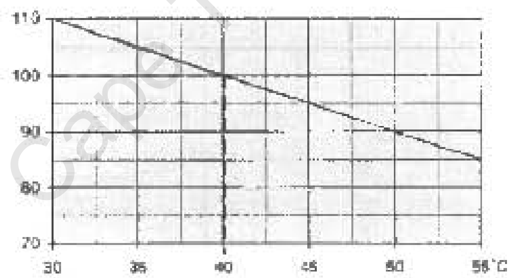

Fig. 2.1/2: Effect of the ambient temperature on the converter module load capacity.

Compliance with standards

The power converter modules and cubicles are designed for industrial applications. Within the EU, the components satisfy the requirements European guidelines, shown in the table below.

European Union Directive	Manufacturer's Assurance	Harmonized Standards Converter module
Machinery Directive 89/392/EEC 93/44/EEC	Declaration of Incorporation	EN 60204-1 [IEC 204-1]
Low Voltage Directive 73/23/EEC 93/68/EEC	Declaration of Conformity	EN 60146-1-1 [IEC 146-1-1] EN 50178 [IEC --] see additional IEC 664
EMC Directive 89/336/EEC 93/68/EEC	Declaration of Conformity Provided that all installation instructions concerning cable selection, cabling and EMC filters or dedicated transformer are followed.	EN 61800-3 ① [IEC 1800-3] where limits are under consideration EN 50081-2 / EN 50082-2 has been supplied the accordance with 3ADW 000 032 installation in accordance with EMC The Technical Construction File to which this Declaration relates has been assessed by Report and Certificate from ABB EMC Certification AB being the Competent Body according to the EMC Directive.

Standards in North America

In North America the system components satisfy the requirements as listed in the table below.

Safety for Power conversion Equipment ≤ 600 V	Standard for modules UL508 C
Industrial control Equipment: Industrial products ≤ 600 V	CSA C 22.2 No. 1105

Please note:

applies for power converter modules only.

II 2

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6 Operating Instructions

General
This manual is designed to help those responsible for planning, installing, start-up and servicing the thyristor power converter.
These people should possess:

- basic knowledge of physics and electrical engineering, electrical wiring principles, components and symbols used in electrical engineering, and
- basic experience with DC drives and products.

> **CAUTION!**
> To avoid unintentional operating states, or to shut the unit down in case of any imminent danger according to the standards in the safety instructions it is not sufficient to merely shut down the drive via signals 'RUN', drive 'OFF' or 'Emergency Stop' respectively 'control panel' or 'PC tool'.

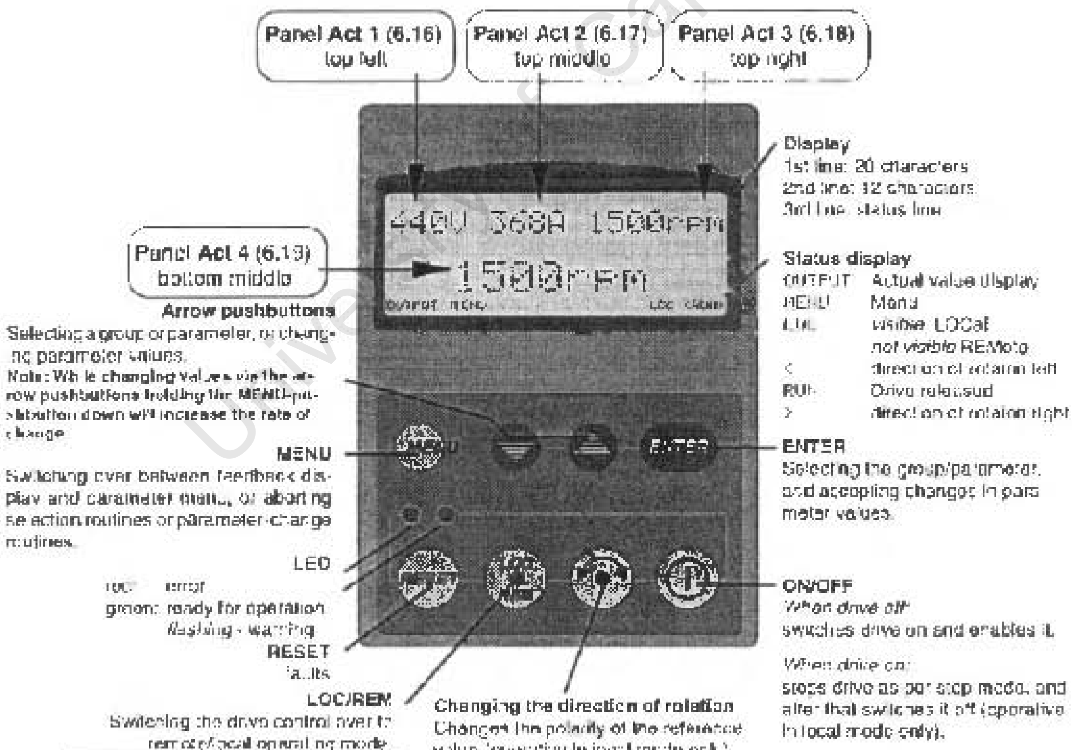
Operating panel DC400 PAN

The Control and Display Panel is used for parameter setting, for feedback value measuring and for drive control with series DC400 thyristor power converters.

Panel link
The DC400 PAN is connected to the drive via a serial interface and is removable under power.

Initialization
After switch on electronics supply the panel shows actual values immediately.

OUPUT display
The panel display can show up to four actual values. Three values at the first line and one at the second line. For individual display it is possible to arrange these in any order via Parameter **Panel Act 1...4**.



> In LOCal Mode terminals are disabled except **Emc Stop** (UI3) and **Reset** (UI5).
> In LOCal Mode the converter works **always** speed controlled.

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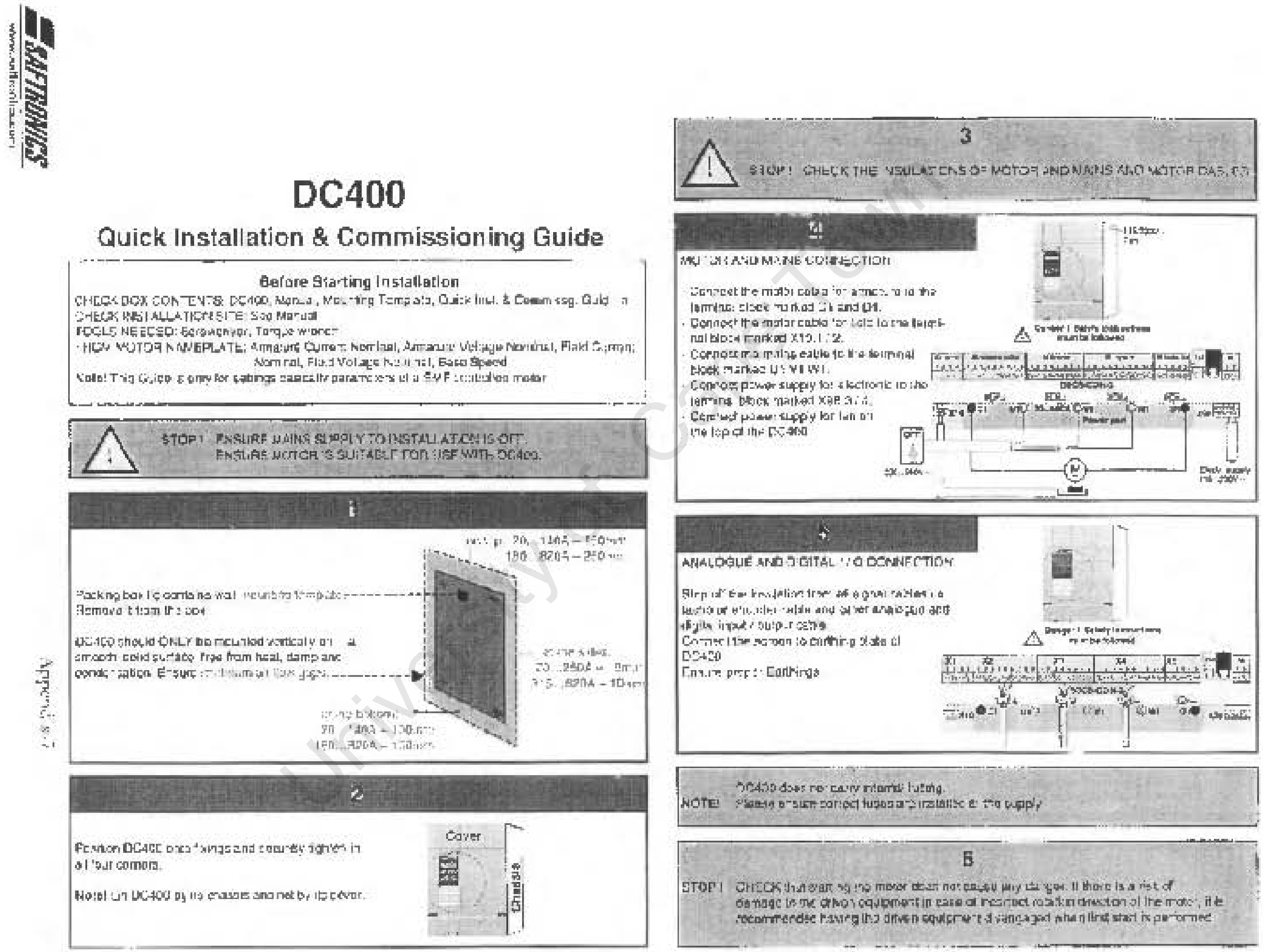
VI-1



DC400

Quick Installation & Commissioning Guide

Before Starting Installation

CHECK BOX CONTENTS: DC400, Manual, Mounting Template, Quick Inst. & Commissg. Guide
CHECK INSTALLATION SITE: See Manual
TOOLS NEEDED: Screwdriver, Torque wrench
NOTE MOTOR NAMEPLATE: Armature Current Nominal, Armature Voltage Nominal, Field Current Nominal, Field Voltage Nominal, Base Speed
Note! This Guide is only for setting basically parameters of a SMF controlled motor

STOP! ENSURE MAINS SUPPLY TO INSTALLATION IS OFF.
ENSURE MOTOR IS SUITABLE FOR USE WITH DC400.

1

Packing box lid contains wall mounting template. Remove it from the box.

DC400 should ONLY be mounted vertically on a smooth solid surface free from heat, damp and condensation. Ensure minimum air gaps.

2

Position DC400 onto fixings and securely tighten in all four corners.

Note! Lift DC400 by its chassis and not by its cover.

3

STOP! CHECK THE INSULATIONS OF MOTOR AND MAINS AND MOTOR CABLES

4

MOTOR AND MAINS CONNECTION

- Connect the motor cable for armature to the terminal block marked C1 and D1.
- Connect the motor cable for field to the terminal block marked X15.1/.2.
- Connect the mains cable to the terminal block marked U1 V1 W1.
- Connect power supply for electronic to the terminal block marked X96.3/.4.
- Connect power supply for fan on the top of the DC400

5

ANALOGUE AND DIGITAL I/O CONNECTION

Strip off the insulation from all signal cables i.e. tacho or encoder cable and other analogue and digital input / output cable.
Connect the screen to earthing plate of DC400
Ensure proper Earthings

NOTE! DC400 does not carry internal fusing.
Please ensure correct fuses are installed at the supply

6

STOP! CHECK that starting the motor does not cause any danger. If there is a risk of damage to the driven equipment in case of incorrect rotation direction of the motor, it is recommended having the driven equipment disengaged when first start is performed.

Application macro Standard

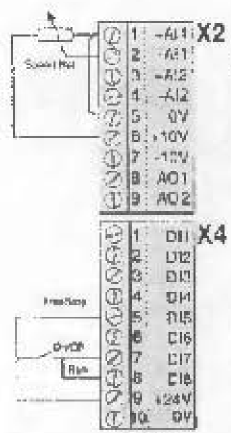
This example of connection bases on application macro Standard and EMF speed feedback

Note! The drive may start when mains is switched on.

For analogue speed reference, connect potentiometer (2-10 kOhms) to terminal X2:1,2,5,6.

Switch on mains

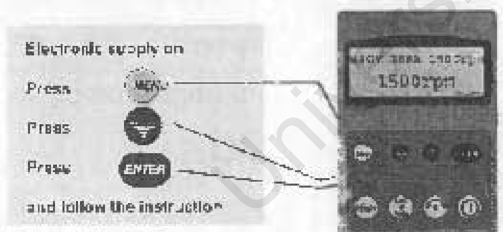
To start the drive activate digital inputs DI7 / DI8



Note! For further information on I/O settings, refer to DC400 Manual, Chapter application macros and / or parameter list.

Note! There are several ways for commissioning the DC400.
- parameter programming by panel
- guided commissioning by panel via panel wizard
- parameter programming by PC Tool *Drive Window Light*
- guided commissioning via PC commissioning wizard (as a part of DriveLink)

This guide describes parameter programming by panel.
For guided commissioning by panel via panel wizard start the panel wizard as follows:



Continue with point 9

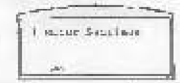
8 Parameter programming by panel

Display shows the OUTPUT mode

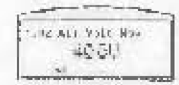
For application macro Standard the following DC400 parameter must be set:

1.01 - Armature Current Nominal
1.02 - Armature Voltage Nominal
1.03 - Field Current Nominal
1.04 - Field Voltage Nominal
1.05 - Base Speed
7.01 - Language

Instructions for settings the parameters:
- Press MENU to enter the MENU. MENU flag becomes visible



- Press ENTER to select the Motor Settings group
- Select the parameter with UP and DOWN buttons



- Press ENTER to get the parameter set mode
- Alter the value by using UP and DOWN buttons
- Store the modified value by pressing ENTER
- After setting all parameters press MENU button twice to resume OUTPUT display.

9

Motor is now ready to run.

Drive controlled by digital inputs.	Close On/Off switch to turn on motor.
Drive controlled by panel.	Set the control mode to local by pressing the LOC/REM button. Press START/STOP button to turn on motor

10

Note: Before increasing motor speed, check that the motor is running in desired direction.

To set the reference by analogue input:
- To increase or decrease the speed reference turn the potentiometer.
- To stop motor open On/Off switch.

To set the reference by panel:
- To increase the reference press UP
  To decrease the Reference press DOWN
- To stop motor press START/STOP button.

Note: Always disconnect mains supply before working on DC400 or motor.

Appendix B – Heat Release Calculation Method

Using in-cylinder pressure history to analyse combustion is a recognised technique found in literature [Heywood, 1988]. For this study it was used as a means to compare Hot Tube and normal running of the engine by calculating the rates of heat release and total heat released as a function of crank angle degrees.

The equation used for calculating the combustion energy (heat) release rate was as follows and the full derivation can be found in Heywood (1988) pp. 385-387:

$$\partial Q_{ch} = \left(\frac{c_v}{R}\right)Vdp + \left(\frac{c_v}{R} + 1\right)pdV + (h' - u + c_v T)dm_{cr} + \partial Q_{ht}$$

The first two terms on the right together represent the sensible energy change and work transfer to the piston, while the third and fourth represent the energy change due to crevice effects and the heat transfer to the walls respectively. The first term δQ_{ch} and the crevice and heat release terms on the right are often combined and are then termed *net heat release*. This net heat release rate was used during analysis of data during this study

The heat release rate is calculated at every data point captured and are displayed in the various graphs in Chapter 5. Total heat released is calculated by using the heat release rate to determine the actual amount of heat released for each data point and summing these values over the whole combustion cycle.