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KIRBY-CHAMBERS, HEATHER MIRIAM

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A THERMODYNAMIC AND MECHANICAL ENGINEERING INVESTIGATION OF A RECIPROCATING JOULE CYCLE ENGINE

by

HEATHER MIRIAM KIRBY-CHAMBERS

A thesis submitted to the University of Plymouth
in partial fulfilment for the degree of

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Heather Miriam Kirby-Chambers

A Thermodynamic and Mechanical Engineering Investigation of a Reciprocating Joule Cycle Engine

ABSTRACT

Current automotive technology is driving towards developing cleaner and more efficient engines that still conform with the public's demand for power and performance. New technologies are also being developed which expand the public's knowledge and awareness of dual fuel, hybrid and even electric vehicles.

Gas Turbines, for use at an automobile level have been left largely unexplored (with the exception of Rover in the 1950's). The application of continuous combustion to a reciprocating engine results in an operating cycle identical to that of a Gas Turbine, but with a reciprocating compressor and expander. This eliminates the efficiency losses associated with scaling turbines down in size and also the considerable costs while maintaining the advantages of high thermal efficiency, cleaner combustion and hence good emissions characteristics, and allows a degree of operational flexibility.

This research undertakes to demonstrate that a reciprocating Gas Turbine engine operating with a Joule (Gas Turbine) Cycle has potential as an alternative form of motive power when applied to an automotive application.

A prototype engine has been developed and tested to obtain some basic data. A theoretical model of the cycle has been created on a spreadsheet to enable assessment of the performance of the engine under somewhat idealised conditions. Certain basic

assumptions have been made in the model and not all losses have been accounted for, but a clear indication of the engines potential has been gained from this exercise. If an engine could be produced that reflected the relatively high thermal efficiencies (about 40%) predicted by the model then it may prove suitable for automotive applications.

The culmination of this research however is the ability to state the potential of the Reciprocating Joule Cycle engine. Unfortunately the prototype engine was unable to sustain itself without external assistance, and even then was unable to operate for extended periods of time due to the combustion chamber. It was therefore felt that the engine would not be suitable for automotive applications due to the extensive development required. It may prove more applicable to static applications.

CONTENTS

TABLES & ILLUSTRATIONS

Illustrations

Nomenclature

A	Area (m^2)
Cp	Specific Heat Capacity at const. pressure (J/kgK)
Cv	Specific Heat Capacity at const. volume (J/kgK)
e	Regenerator effectiveness
E	Intercooler effectiveness
k	k factor
m_{flow}	Mass flow (kg/s)
P	Pressure (Pa)
Q	Heat Energy (J)
R	Gas constant ($JK^{-1}mol^{-1}$)
r_p	Pressure ratio
RPM	Speed (RPM)
Sp Wk	Specific work (kJ/kg)
T	Temperature (K)
v	Velocity (m^2/s)
V	Volume (m^3)
V_{flow}	Volume flowrate (m^3/s)
W	Work (J)

Greek Symbols

ρ	Density
η	Efficiency

Subscripts & Superscripts

1, 2, etc	Stage of cycle
act	Actual
ad	Adiabatic
c	Clearance
co	Compressor
ct	Cutoff ratio
exit	Exit
exp	Expander
i	intermediate
in	Inlet
ind	Inducted
k	No of Cylinders
limit	Limit
L	Loss
mech	Mechanical
n	Ratio of heat capacities
net	Net
opt	Optimum
s	Swept
th	Thermal
tot	Total
vol	Volumetric

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Many thanks to Ford (UK) for the donation of a 2.5 litre diesel engine and associated drawings, Lotus Engineering for access to their research information on valve timing equipment, and to Avesta Sheffield for the donation of valve seat material.

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Final thanks go to my husband Ptolemy, for always being there.

AUTHOR'S DECLARATION

At no time during the registration for the degree of Master of Philosopy has the author been registered for any other University award.

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Relevant technical seminars and conferences were attended to maintain insight into the solutions that the automotive industry would be developing to meet the demands of reduced emissions laid down by current and future legislation.

Conferences and Seminars attended:

International Seminar on: application of Powertrain and Fuel Technologies to meet Emissions Standards. Automobile Division & Combustion Engines Group of IMechE. 24-26 June 1996.

Lean Burn Combustion Engines. Combustion Engines Group of IMechE. 3-4 December 1996.

Future Engine and System Technologies: The Euro IV Challenge. Combustion Engines Group of IMechE in association with the Automobile Division. 3-4 December 1997.

International Conference on: Combustion Engines and Hybrid Vehicles. Automobile Division & Combustion Engines Group of IMechE. 28-30 April 1998.

Signed...H.M. Kirby-Chambers

Date.30..10..01......................

1.0 The Changing Automotive Climate

1.1 Preview

The amount of pollution that is affecting the environment is steadily increasing. In an attempt to curb this increase international conferences on pollution prevention are more regularly being held, one of the most recent being the International Pollution Prevention Summit held in Montreal, Canada, in October 2000. The event is actually a United Nations initiative and 2000 saw the sixth gathering to discuss cleaner production methods. Representatives from all over the world were invited to attend the meeting to share the latest methods that their country employ in various processes to reduce pollutant emission. As the economic situation of the countries involved in such a gathering is quite varied, thus steps that each nation implements reflects that variety. As the progress of pollution reduction in some countries will be inherently slow other more affluent regions take steps and apply more stringent measures to reduce the amount of pollutants they release. Hence Europe and the United States, amongst others, continue to legislate to limit the pollution products from various sources, ranging from power stations to automobiles.

Automotive manufacturers have taken steps to increase combustion cleanliness, improve fuel consumption, alter control methods and reduce rolling resistance. This has allowed them to meet legislation thus far, but the levels required are becoming ever more stringent. Development of technologies such as variable valve timing, transmission systems, catalysts, Exhaust Gas Recirculation (EGR) and reduction in overall vehicle weight are continuing to aid achievement of the standards set.

The introduction of this legislation has capped the level of 'Greenhouse Gases' that can be generated and released into the atmosphere. Much work still needs to be done however, to prevent the catastrophic climate changes that are beginning to happen (El Ninio, Acid Rain etc). It is interesting to note that much innovation to counter the pollution issues comes from Scandinavia, they have been suffering the effects of acid rain for many years. Companies such as Saab are continuing to develop systems that collect the exhaust products from the engine when it is initially started and the highest levels of pollutants are produced due to the engine and combustion system not being at the optimal temperature. These products are then recycled through the engine once it has reached its optimal conditions.

The research described in this thesis concerns the development of a Reciprocating Joule Cycle (RJC) engine, which has potential to reduce harmful gaseous emissions and also offers the possibility of reduced CO_2 emissions together with some other unique characteristics.

The aim of this project was to compare practical test data, obtained from a prototype engine, with theoretical data deduced from a simple computer model.

The objectives therefore were to:

1) Create a simple computer model of the reciprocating joule cycle engine utilising a spreadsheet,
2) Build a prototype reciprocating joule cycle engine and
3) Run the prototype engine and collect raw data from it.

1.2 Legislation

Steps were first taken by Europe in 1971 to control the emissions produced by the community's cars. To this date there have been 9 directives introduced, the next always more stringent than the last. 1998 saw the implementation of Euro III, the 10th of these directives, which came into force on 1st January 2000. During its preparation indications of emission levels of Euro IV were also given. Euro IV will come into action in 2005, the reason for revealing some indications of its contents was to act as an accelerant to technology. Forewarning manufacturers of what was to come, giving them an incentive to strive towards the goal of Euro IV before Euro III had been fully met. Manufacturers such as Vauxhall have dual-fuel cars available for sale to the public.

The United States took similar steps, with only California specifying a tighter schedule for the reduction in emissions seen in that state. The progress of the European standards can be seen by comparing the emissions levels that diesel engines are expected to meet. Fig 1 shows the emissions levels for diesel engines from Euro II, through Euro III and into Euro IV.

	EURO II	**EURO III**	**EURO IV**
	Passenger car 96EC Directive 94/12/EC	Passenger car 2000EC	Passenger car 2005EC (Targets)
Tailpipe – Gms/km	**IDI/DI**	**IDI/DI same levels**	**IDI/DI same levels**
CO	1.0	0.6	0.5
HC + NOx	0.7 (0.9)	0.56	0.3
HC	-	-	-
NOx	Not applicable	0.5	0.25
PM	0.08 (0.1)	0.05	0.025
Test Cycle	No Change	New: Delete 40 second idle	No Change
Additional Tests	None	Cold test added -7 °C	No Change
On-board diagnostics	None	Not mandated - optional	Added at 2005
In Use	80,000km	80,000km Campaign/recall risks	Possible 160,000km
Effective timing	January 1, 1997 (reg)	January 2000 (homol)	January 2005 (homol)

Fig. 1. Comparison of Diesel Emission Levels Across the Standards.[a)]

a) Data from Reference 1.

Some key additions to the regulations include [1].

- Modified test cycle – the deletion of the 40 second idle allowing sampling from engine crank.
- Addition of separate cold test at -7°C (effective from 2002)
- Standards for both combined HC + NOx and separate NOx.
- Addition of on-board diagnostics at 2005 for diesels.
- In-use emission testing.

The introduction of Euro III also removed the concession that High Speed Direct Injection (HSDI) engines benifitted from in Euro II. From 2000 they would also have to conform to the NOx emission levels. High Speed Direct Injection (HSDI) engines were allowed some leniancy to encourage development of the technology, its fuel economy benefits were seen as worthy of pursuit, and therefore assistance.

The Euro IV regulations were confirmed in 1999 and are already the next standard, Euro V, is being discussed in preparation for its implementation in 2008. Beyond this standard emission levels are destined to continue to reduce until the ultimate zero emission vehicle is produced, or technology falters and cannot be realistically pushed any further. The goal will be reached in time although, as ever, money for research and development is the limiting factor.

The enormity of the task is best illustrated graphically, as in Fig 2a and 2b. This shows how much has been achieved to date and how the boundaries are being pushed to their limits to reduce levels yet further as the standards progress.

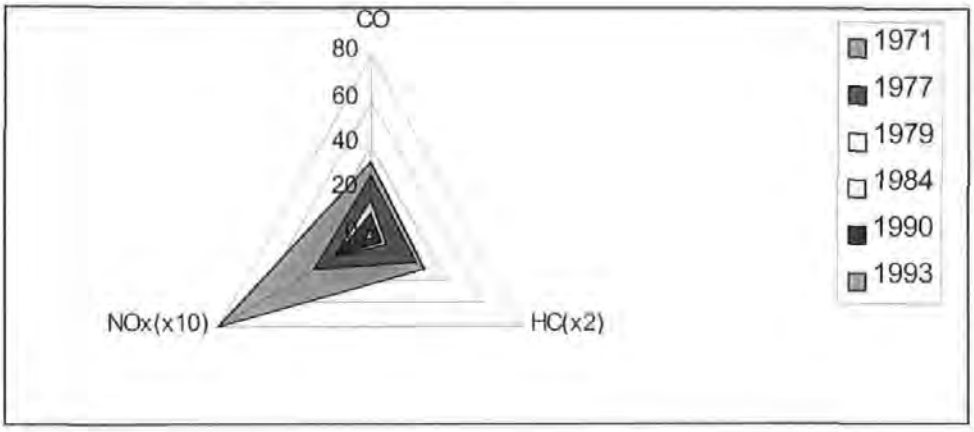


Figure 2a. European Emission Level Progress. [b)]

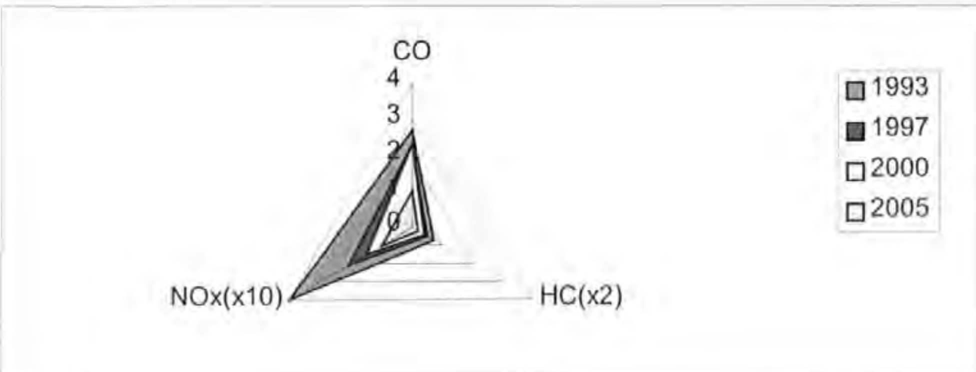


Figure 2b. Detail of Current European Emission Levels [b)]

b) Data from Reference 2.

Another influencing factor on the production of Euro IV regulations was a collaboration between the Automotive and Oil Industries. A large test programme was undertaken in cities across Europe to determine the performance of new engine technologies when combined with high quality fuels.

Analysis of the data has enable the European Commission to predict the impact on air quality under certain emission control measures, and also the associated costs [3].

The work of the Auto-Oil programme also highlighted the need for the two industries to continue to work closely together towards the achievement of the regulations. For example, catalysts for both diesel and gasoline engines require low sulphur content in the fuel to allow them to maintain efficiency, and a longer operational life. Ideally diesel (de-NOx) catalysts require a sulphur content of less than 50ppm to operate effectively. Low sulphur fuel also results in reduced particulate emissions.

The reduction of other fuel additives and/or properties can lead to the reduction of other emissions i.e. hydrocarbon emissions can be reduced by limiting the volatility of gasoline.

1.3 Current Automotive Technology

Assuming that the car will remain a preferred form of personal transport the ultimate aim has to be to manufacture a car that is sensitive to the environmental demands that are placed upon it: i.e. a car that produces negligible or zero emissions.

Recent advances include the introduction of the Direct Injection Gasoline engine, first marketed by Mitsubishi. Alongside the more conventional technology, some alternative power sources are gaining ground. Electric vehicles, hybrids, fuel cells and alternative fuelled Spark Ignition engines are stepping up to take their place in the arena. Natural gas powered buses can be seen in several European cities as well as Vancouver, Canada. BT and British Gas are also using many Natural Gas powered vehicles, and these may supersede the conventional fleets. In the United States, particularly California, electric vehicles are being tested to prove their driveability and range.

1.3.1 Direct Injection Gasoline Engines (GDI)

The DI gasoline engine is the latest development into which the major manufacturers have concentrated resources. The GDI produced lower CO_2 emissions, which is the main area of concern in the European legislation.

The concept is simple, similar to that of the DI Diesel engines. Instead of the fuel being injected into the cylinder via the inlet port, the nozzle injects directly into the combustion bowl of the piston. There are two types of GDI [4], the first operates on low injection pressure of the fuel, but it is air assisted. This has applications in the marine environment. The second utilizes a high pressure, single fluid system. Atomization occurs due to the high fuel system pressures of between 50-120bar (This is the type of GDI that Mitsubishi developed). Toyota have also followed this path and it can now be seen on the roads as the VVTI engine in their latest range of vehicles.

Advantages of the GDI include improved fuel economy, it can run under very lean conditions. This means that it can also operate under stratified conditions, this leads to reduced pumping work losses, and improved efficiency [2].

This type of engine will continue to have problems controlling the particulates and NO_x it produces until the fuel injection is better understood and more easily controlled. One suggestion to help with this problem is the inclusion of Exhaust Gas Recirculation (EGR).

1.3.2 High Speed Direct Injection (HSDI)

The HSDI is the newest form of Diesel engine and it has improved automotive diesels considerably, they are no longer the slow and unresponsive vehicles they once were. This engine basically increases the pressure and speed at which the fuel is injected into the cylinder, the effect of this is to increase the potential power output. This system also improves the fuel consumption, and provides impressive driving performances [5].

The system still works in conjunction with EGR and exhaust gas aftertreatment, so emissions may prove to be a problem area for the diesels. With this type of engine the products of reaction are highly dependent on the mixture formation and combustion. There is also a problem with the simultaneous reduction of particulates and NO_x. These areas mean that meeting EURO IV will be difficult but not impossible with adequate research and development.

Areas that still require work to aid in the clean up of the HSDI include combustion, control, cold start, oxicat and diesel fuel formulation.

1.3.3 Hybrids

Hybrids exist in two forms, parallel and series. A parallel hybrid consists of an internal combustion engine (ICE) and an electric traction system which both power the drive wheels directly. The Series hybrid provides all of the torque and power required to drive the vehicle from an electric traction drive. This drive system has a battery system, part of which is an Auxiliary Power Unit (APU). This APU is comprised of an ICE and generator, and is generally of a higher energy density than the electric drive system. This means that the vehicle can extend its range when utilizing this system [6].

Both types of hybrid vehicle are on test in various countries around the word. Schematic diagrams of the series and parallel hybrid engine types can be seen in Fig 3 and Fig 4. The concept is ideal for town driving, as when operating in electric mode little pollution is produced. When extended driving range is required the ICE comes into force. In the case of the series hybrid specifically, its range when in all electric mode is heavily dependent on the development of battery technology.

Some people see the introduction of the hybrid vehicle as a short term solution to the problem of automotive emissions. It is felt that they will aid in the reduction and control of smog problems in areas with extreme conditions until zero emission technology is available. However, the growth of this type of vehicle in the target areas may initially be slow despite the obvious benefits available to the environment, as potential customers will no doubt prove to be reluctant to move away from their more powerful conventional vehicles. In Christchurch, New Zealand, a hybrid bus service was introduced with the aim of helping to reduce the level of CO_2 pollution that the city experienced [7]. The introduction of the service has had a positive effect on the service area in Christchurch. As this is a vehicle designed with passenger comfort in mind it has encouraged the local population back onto the public transport service which has had the beneficial effect of reducing the number of cars that are being driven into the city for both business and recreational purposes. This return to the use of public transport has in turn had a positive effect on the economy of local businesses along the Shuttle's route, more people are frequenting these establishments. The tourist trade has also increased as word of this novel bus has spread, which generates yet more wealth for the city. This particular example of a hybrid system demonstrates that it is particularly well suited for the application. It is also a good test bed for this technology and others, such as the regenerative braking system that is used to recharge the 54 batteries which is the vehicles primary source of power.

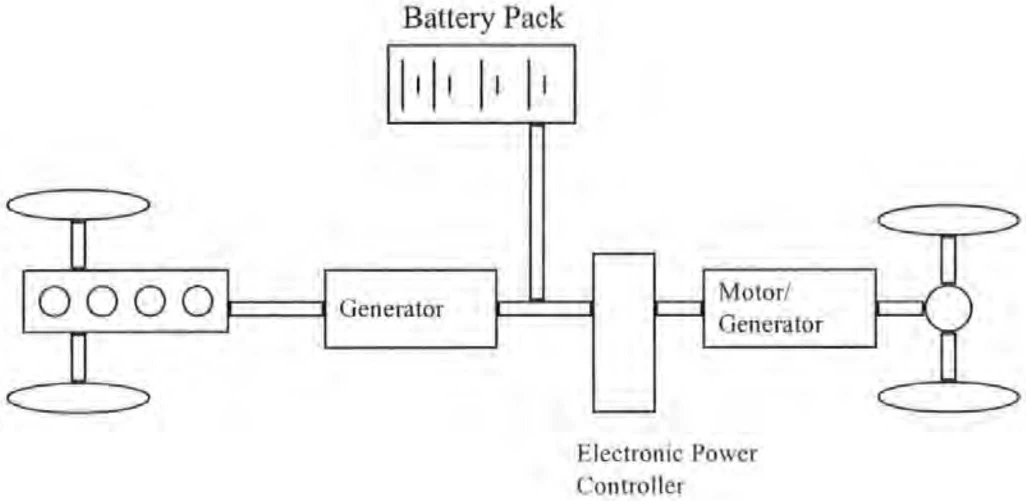


Figure 3. Series Hybrid Engine

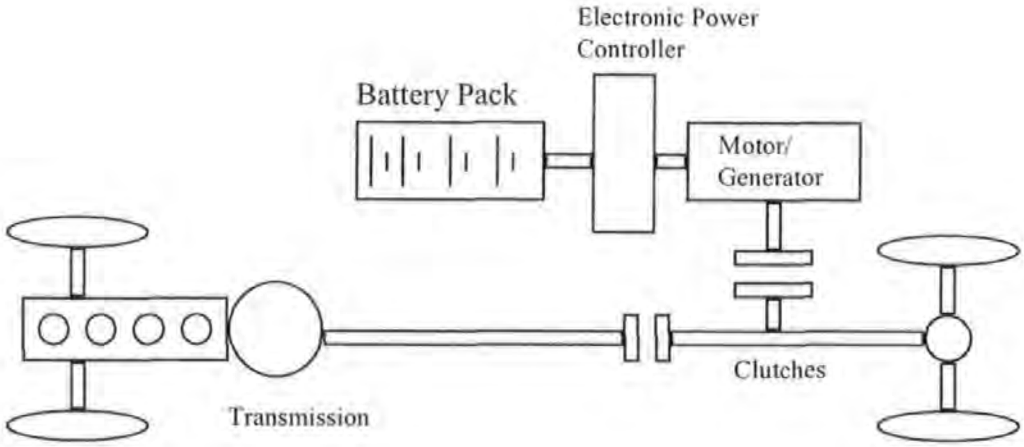


Figure 4. Parallel Hybrid Engine

1.3.4 Fuel Cells

The main type of fuel cell that is in operation today, for trial purposes, is the PEMFC (Proton Exchange Membrane Fuel Cell). In this particular cell Hydrogen gas and Oxygen, provided from air, combine in the presence of a platinum catalyst to form water. This reaction produces an electric current. The electrolytic process is illustrated in Fig 5. This technology is seen by many as the real way forward to zero emission vehicles. The only product from this type of fuel cell reaction is water. The primary fuel efficiency is currently up to 50% [8].

However, much development is still required before the fuel cell can be seen in mass production for automotive use, they are already widely used in the Space Programme. The size, weight and cost all need to be considerably reduced. Fuel storage, processing and onboard handling need further work, and the cell, energy store, electric drive and controls still need to be integrated into a cohesive system. The efficiency and power quotes for fuel cells varies depending on the type of cell and its application, however, they are reputed to have good part load characteristics as demonstrated in Reference 9. A cell producing 100kW AC nett electrical power has an efficiency of 37%.

Fuel Cells are however seen by many as the motive power source of the future. While it will take time to overcome the problems of hydrogen storage and a supply network, many companies are looking into intermediate methods of bringing the fuel cell into the automotive market. Current research is concentrating on methods of converting gasoline into hydrogen, which may not result in a clean extraction process.

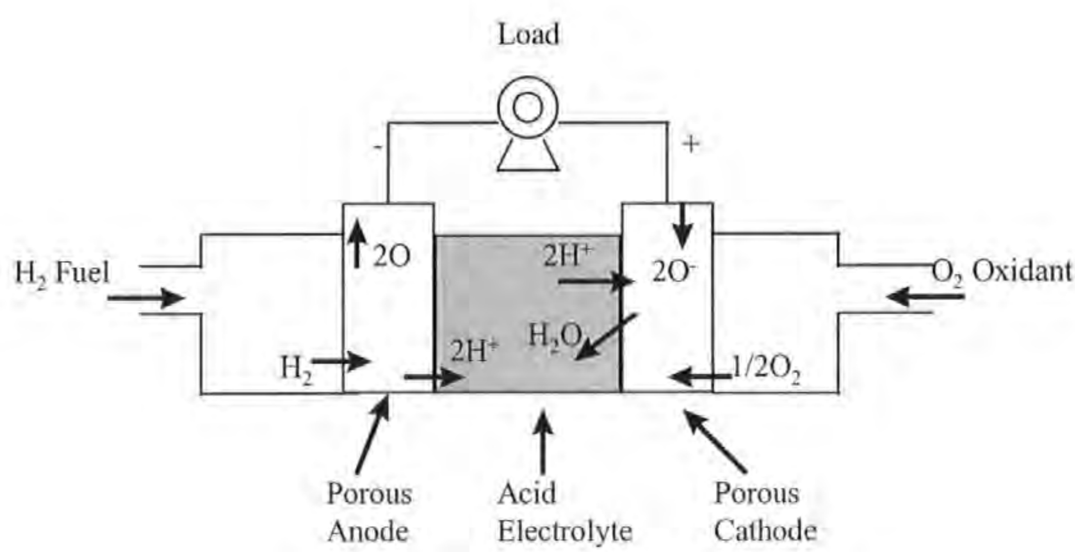


Figure 5. Proton Exchange Membrane Fuel Cell.

1.3.5 Gas Turbine

One area of technology that still appears to be overlooked for automotive application is the Gas Turbine.

When compared to gasoline engines the gas turbine weighs less, is more durable and more reliable. The gas turbine also has a higher thermal efficiency, for engines of comparable size a gas turbine with a 90% efficient regenerator will deliver a thermal efficiency of about 33% in comparison with roughly 27% for Spark Ignition (SI) engines [10]. The overall efficiency of the gas turbine does vary with the physical size of the engine, the decrease in this efficiency when the size is reduced can be compensated for by improving the efficiency of the constituent components of the engine, particularly the regenerator. When considering emissions it should be duly noted that gas turbines with no aftertreatment will produce lower emissions than SI and Compression Ignition (CI) engines with the best treatment. Gas turbines are also capable of burning a greater variety of fuels.

Limitations on the use of the gas turbine in automotive applications stem from regenerator losses, through leakage, and the maximum operating temperature attainable.

The maximum operational temperature of automotive gas turbines is currently in the region of 750°C. This restriction derives from the reduced size of this type of gas turbine. The turbine blades are smaller than in their counterparts in the aeronautical industry, which means that the cooling ducts cannot be incorporated, thus limiting the maximum operational temperature [10]. Another restriction on the production of highly efficient gas turbines on a small scale is the cost, which is driven higher by the need to utilize precision manufactured components to keep the desired performance.

1.4 Comments

These are just a sample of the developments that are taking place to meet the challenge of the emission problem. Other areas, aside from engine development are also working towards the same goal. These include exhaust gas aftertreatment – the goal of improving the catalysts light off time would bring large improvements to the emission results, fuel reformulation, transmission systems and vehicle weight. No matter how 'green' manufacturers make their new vehicle model, the perception is still that there is a compromise to be made between performance and environmental impact.

All of the alternative vehicles have to overcome the hurdle of current perception of what personal transport should be. (i.e. a push bike is a totally efficient, non pollution form of personal transport that is not widely acceptable)

Today's car owners demand and expect a lot from their vehicles. They are beginning to focus on the 'green' issues, but at the same they still expect the power and performance of their 'Golf GTI'. (Although Ford Special Vehicles have developed the Mustang Super Stallion Dual-Fuel car which runs on both gasoline and alcohol, producing 50 bhp more when running on alcohol). Electric vehicles are now appearing on the market and although they are praised for their emission qualities their speed, power, endurance and size are still no match for conventional technology. Alternative vehicles are beginning to meet the requirements and over time they will become more acceptable.

The current infrastructure for gasoline and diesel vehicles is far reaching, from fuelling stations to spares and repair, all easily available nationwide. For alternative fuels to make an impact they must have the same infrastructure to become viable, such as the current growth in Liquid Petrolium Gas (LPG) on forecourts. But fuelling is not the only issue. Vast sums of money are tied up in the conventional engine manufacturing market, and the supporting supply networks of companies reaches into every aspect of society. This is possibly the largest block of resistance to overcome. If automotive engine technology was to change many companies would suffer. Obviously concessions would need to be made and companies would need to diversify to sustain themselves during any transition to a radically different form of automotive engine.

1.5 The Reciprocating Joule Cycle Engine

This cycle can operate in one of two forms, reciprocating or rotational. It is the latter that has proved to be more successful over the last 100 years as the Gas Turbine undertakes a large variety of roles from power generation to airplane propulsion.

An operational Joule cycle engine differs from both gasoline and diesel engines in that each stage of the cycle takes place within separate, specific components. Air is drawn into a compressor where the temperature and pressure are raised. This high pressure air then moves into a combustion chamber where fuel is added, and combustion takes place at constant pressure. The high temperature products of combustion (approximately 1000°C) then move into an expander where it is expanded back down to atmospheric conditions, this essentially being the power stroke. Most operational Gas Turbines take advantage of the increased thermal efficiency that the incorporation of a regenerator brings, Fig. 6. The function of the regenerator is to take the waste heat out of the exhaust gases and use this surplus energy to pre-heat the gases which leave the compressor, before they enter the combustion chamber. This means that the temperature difference between the gases entering the combustor and the desired peak temperature is reduced, therefore less fuel is required to reach the target temperature.

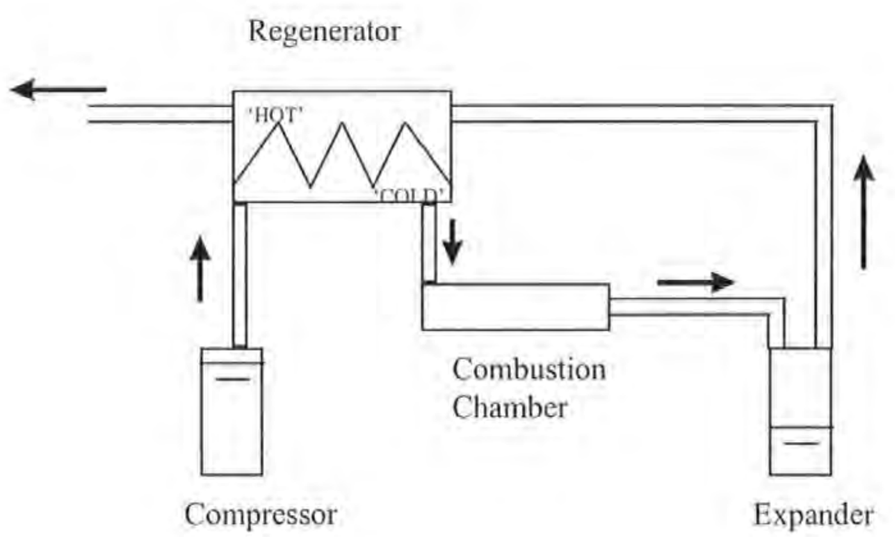


Fig. 6. Conceptual diagram of a reciprocating Joule Cycle Engine with Regenerator.

Utilizing reciprocating components for this type of small scale application takes advantage of the efficiencies, technology and marketability that are currently associated with the piston/cylinder arrangement. It is also widely known that as turbines are scaled down in size their efficiency drops considerably due to tip losses, and their manufacturing cost increases considerably. Another advantage that the RJC holds over current alternative engines is its relative simplicity, when compared to hybrids for example.

Factors that make the RJC favorable as a contender in the automotive field include its ability to deliver a comparable thermal efficiency to that of today's modern diesel engines of about 40%. Its emissions characteristics should be inherently clean, this being due to the continuous combustion process that takes place. Its configuration also means that multi-fuelling is easily suitable with very little, if any, alteration required. A three cylinder engine operates as a two stroke and will produce two power strokes in every complete cycle. Once the performance of the engine has been optimized developments can continue on the use of compressed air storage. The use of a store would eliminate the need for a starter motor. On start up a charge of air from the store would be passed through the combustion chamber thus powering the expander, which in turn will operate the compressor, and thus sustain the cycle. From an environmental view point this system would help combat pollution during city driving. In congestion or while waiting at traffic lights the engine would stop, rather than idle as is the current situation. This type of feature is now starting to appear in conventional cars in today's marketplace as the environmental awareness of the public is starting to impact upon the major car manufacturers. There is also the possibility of integrating regenerative braking into the system, this has been demonstrated on public buses, like the 'Shuttle' in New Zealand [7], as well as the possibility of removing the gearbox.

The removal of components such as the starter motor and gearbox would create extra space in the engine bay. This would be offset by the fact that to produce the same power output of a conventional engine, the RJC needs to be bigger than its counterpart.

The following chapters will look at the history of this type of engine, investigating the research that others have previously carried out and, where applicable, comparing their results to the outcomes of this project. An overview of the theory of the RJC engine will be covered and the details of the theoretical model will be explained. The process of preparing the donor engine for operation as an RJC engine will be detailed and explanations of some of the technical decisions will be given for some alterations. The methodology of the testing procedure will be stated followed by the results gained from any practical experiments carried out. The performance of the RJC engine will then be discussed and conclusions stated, with suggestions for areas of further research defined.

2.0 RJC Engine History

2.1 Introduction

The spark ignition and compression ignition engines have been the only form of motive power plant available to the automobile market for many years, their position unrivalled. The first engine to go into mass production was a 'V' configuration, designed by Daimler in 1889. Research into these two systems has continued and developments continue to be made in an attempt to improve them still further.

Ongoing legislation has laid down strict guidelines for the emissions of the SI and CI engines with which all newly constructed engines must comply. The aim was to produce Low Emission Vehicles (LEV's) by 2000. There were some authorities who were pushing even harder to obtain Ultra Low, or Zero, Emission Vehicles (ULEV's) in the same time period. This has added to the intensity of development by the larger corporations into these two engine types. Most effort has been focused on areas such as Variable Valve Timing, Fuel Injection Systems, Variable Intake/Inlet Systems and Low Heat Rejection. This work has a limited positive percentage effect on the efficiency and fuel consumption of the internal combustion engine. One such area of focus is on catalytic converters, such as the three-way converter, which deals with the problem after the fact and is effective under very specific operating conditions. Other methods include improving combustion efficiency, cleanliness of fuels and alternative fuels.

The percentage improvements that can be obtained from the internal combustion engine are finite, and there will come a point where no more can be done. Some people have looked into alternative technologies as possible methods of complying with the new regulations. Options explored, and still under development, include Two-Stoke cycles, Stratified Charge, Hybrid Engines and Gas Turbines to name but a few of the more common alternatives. Several of the latter ideas offer great potential but require substantial support for further research and development, as well as a positive reaction from manufacturers and customers.

2.2 RJC Engine

One such alternative that has received little attention is the Reciprocating Joule Cycle (RJC) Engine. The basic cycle is the same as the one employed in the gas turbine, the difference being that reciprocating parts are used rather than rotational. In its simplest form it consists of a reciprocating compressor linked to a reciprocating expander, via a combustion chamber. The Joule Cycle shows promising advantages over both the internal combustion and gas turbine engines, as modifications to the configuration can be easily made.

Since the late sixties work has been undertaken to show that the RJC is a suitable contender for the automotive market. To date, unfortunately, there has been little success.
A literature survey was carried out to identify any work of a similar nature to that which is proposed here. Extensive research showed that little work appeared to have been done to-date on the RJC Engine. However, a few engine designs that are similar were uncovered, none of these being of exactly the same configuration as the engine proposed by this thesis.

In total four engine designs were found to be suitable for comparison, three of them being based on the open cycle. Each will be discussed in turn.

2.3 'Britalus' Brayton Cycle Engine.

The Britalus Engine is based upon the simple form of the Joule (or Brayton) cycle, using a reciprocating compressor and expander linked via a combustion chamber. The design of this engine is rather unique, in that the compressor and expander utilise a three lobed cam in which six pistons are located. As the pistons rotate around the inside of the cam they are displaced within their cylinders, thus creating the required reciprocating motion, Fig 7 illustrates the concept.

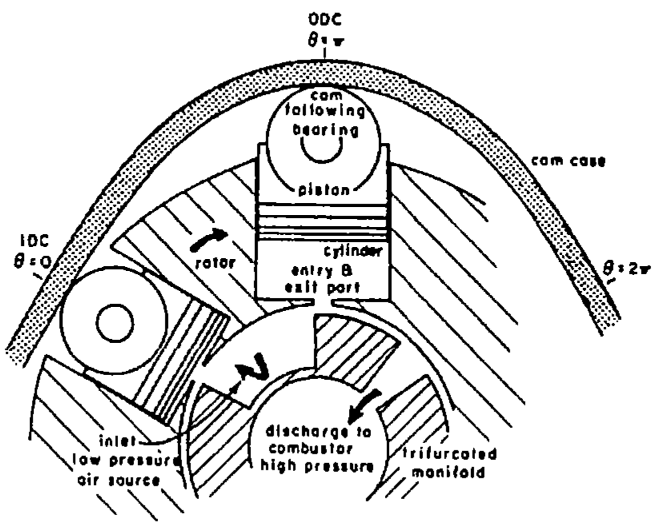


Fig.7 'Britalus' Joule Cycle Engine. [12]

Work started on the Britalus Engine in the 1970's and the idea, based on the rotating compressor and expander outlined above, was patented in 1982. The name of the engine originated from the home countries of the researchers that first worked on it, (i.e. Britain-Italy-United States) [11]. Most of the documented work on this engine has been completed by Reiner Decher, who has looked into various design and performance aspects in great detail, and these shall be discussed later in this section.

The engine as a whole entity does not appear to have been constructed and tested. The compressor on its own was, however, built and tested in the mid 1980's, modifications derived from the test results led to a second compressor being constructed for test. Indications were made that an expander was under construction [11]. Also in the mid 1980's, the Britalus gained an alternative name - PACE, Piston All-fuels Ceramic Engine [12], but only one paper was published with this name [13].

Decher's first paper [14] introduces the Britalus engine concept and continues to discuss the analysis of this engine under part load operation. From the outset this engine incorporated a compressor and expander consisting of a three lobed cam, as previously described. It was suggested that such a configuration could withstand high pressures and therefore any high performance engines of this design could be supercharged. This is the only type of cycle modification that is suggested throughout the course of Decher's work for improving the overall efficiency.

One advantage of such a compressor design is that each of the six pistons will complete three cycles per revolution, which would prove to be quite significant in the mass flow available to pass through the engine. The mass flow of the compressor [14], with no supercharging, is quoted as being:

$$m = 3.69\times10^{-4}\, NV^{cc} \qquad (1)$$

where $N = RPM/1000$

and V^{cc} = Displacement of one piston in cubic centimetres, however no indication has been given of the dimensions of the compressor.

A fundamental difference in the methods of analysing this cycle between Decher and the present author must be clarified. All of the work carried out by Decher was based upon a predetermined engine design with set dimensions which specify the volumes of gases that are involved. The RJC proposal, on the other hand, was considered more openly in terms of cylinder sizes. Certain operating parameters were set, i.e. the temperature of the gas at exit from the combustor, and the pressure at which the gas leaves the expander when the cycle is operating at a particular pressure ratio. This final condition must always be met by the computer model, therefore as certain operating parameters are altered the model would automatically compensate by adjusting the volume available in the expander.

Decher considered that the only feasible method of controlling the power output of the Britalus design, was by allowing the pressure ratio within the engine to vary. (Reference 15 looks at alternative methods of control to produce optimum part load operation). As more fuel enters the system the maximum temperature attainable will rise and because the geometry i.e. volume, of the compressor is fixed, the pressure within the system must also increase. This is an option in which there is little choice due to the design of the compressor and in particular the gas entry and exit ports - there is little possibility for flexibility of design. The RJC on the other hand has no fixed geometry which allows other possibilities to be considered for the control of the engine. The preferred first option for control is to maintain a fixed pressure ratio within the engine and to utilise valve timing to alter the amount of gas admitted to the cylinders, thus altering the mass flow and consequently the power output.

An advantage of the RJC proposal is highlighted when the expander is considered. Decher initially stated that the Britalus expander was designed to expand the combustion gases back to atmospheric pressure, and that losses must be accepted when higher pressure gases were not fully expanded. He did, however recognise that the development of a variable geometry expander would improve the performance of the engine by allowing full expansion under any load conditions and therefore decreasing the losses in specific work [15]. Regeneration was only suggested as a possibility for consideration by Tsongas and White [12] four years after the initial paper. Decher has however looked at the effects of over and under expansion on the Britalus [15] when it is operating at off design loads. The analysis shows that the expanders efficiency is not largely effected by over expansion, unless it is severe, and there is only cause for concern when it takes place at low pressure ratios.

Analysis of the air cycle by Decher [14] shows that for a given design pressure and expander inlet temperature ratio, the lower the pressure ratio of the engine, the higher the specific work. But this produces a narrow power output range, and vice versa for higher pressure ratios. The overall thermal efficiency is calculated from the difference between the compressor exit and the expander inlet temperatures. It is shown that as the power output is increased the component efficiencies fall, and even though this is compensated by a gain in the ideal cycle efficiency, the overall effect is that the thermal efficiency decreases. Work on the RJC has also shown this to be the case, but the reduction in overall thermal efficiency is only a matter of 2-3%.

Reference 15 concentrates on producing an optimum design of the engine that generates good performance at part load, and considers the effects of these design options. Three cycles are considered: I) no variable pressure ratio, II) constant density conditions at expander inlet and, III) a hybrid of the previous two. The effects of these conditions are studied by comparing the specific work and efficiency for variations in peak cycle temperature below the maximum value. Comparison of the results indicates that the hybrid cycle offers the best performance because it incorperates the advantageous elements of the other suggested cycles. These elements being: I) partial over compression if the efficiency loss is small, II) greater pressure variability in compressor through manifold rotation within the efficiency considerations and, III) to produce almost ideal performance in the mid range, the expander can be designed to suffer small efficiency loss at minimum and maximum power. This is demonstrated in the efficiency-specific work curve. Depending on the maximum cycle temperature, a peak efficiency of about 0.55 is obtained at specific works of about 0.6 and 1.3, these maximums occuring in roughly the centres of the load ranges. The conclusions drawn from this area of work are that if there is a large pressure ratio variability within the compressor there will be a significant variability in the specific work output of the cycle. The design can be optimized to compensate for losses experienced under certain conditions so that high efficiencies and specific work can be obtained for any load.

Decher's second paper [16] looks at the power scaling characteristics of the Britalus engine. The engine is considered for operation in the power output range of 10-100kW, and three areas are considered in detail; airflow capability, frictional losses and structural loads. Important parameters that effect the cycles overall efficiency were identified as being the components efficiencies and the maximum cycle temperature. Characteristics of the specific work and the cycle efficiency are shown for certain conditions. Pressure ratios that give the maximum possible specific work for equal component efficiencies highlights a pressure range between 10-20 and air cycle thermal efficiencies of about 50%. This paper also shows that if the maximum cycle temperature is raised the specific work and overall efficiency will also be improved.

Decher introduces a different Britalus engine design in 1986 [17]. The previous design was based around the pistons rotating inside a cam, or the cam revolving around the pistons. The new design still based on a circular concept, the pistons moving by means of a sun-gear arrangement. A schematic of the entire engine layout is also of some interest. The majority of the work covered in this reference is a summarisation of previous papers. It does, however, look at one particular aspect that is considered of prime importance to the performance of this engine, and that is the peak cycle temperature attainable. One example of the efficiency achievable is stated as being 40%, for a peak temperature of 2000K and component efficiencies of 90%. Under the same conditions the model for the RJC, for a basic cycle, concurs with the thermal efficiency predicted. The same limitations of materials suitability apply to both engine types, although the advances in ceramics technology are still under investigation.

Definite conclusions on the work of Decher are difficult to reach, as the parameters quoted between the papers vary. It can be said with some confidence, and judging by the comparison made in the last paragraph, that the RJC configuration proposed has the potential to produce a much more efficient engine.

Other aspects of the Britalus engine design have been discussed by Decher. These areas include further development of the expander, the use of variable speed drive and a look into the heat transfer characteristics. Unfortunately the author has been unable to obtain these to date.

Two other papers have been written relating to the Britatus engine. George Tsongas in association with Robert Jellesed in the first instance [11], and T. White in the second [12]. Both papers cover the development of a computer model for the easy analysis of the Britalus cycle. The complete model as described by Reference 12 is directed at easing the ability of the user to alter the design parameters of the engine, it shows the effects of alterations on the efficiency and specific work. As with the model for the RJC, it will not produce a combination of compressor and expander conditions to give an optimum performance. It is purely a tool for studying the effects of parameter alteration for the purpose of producing engine maps etc. Tsongas' model requires the user to choose a set of fluid properties and to input three operating parameters: the pressure ratio, the compressor rpm and the maximum cycle temperature. The performance is then calculated and displayed in graphic form for the user to analyse. In a similar manner the RJC model is manipulated by altering certain performance parameters, and the effects of these can be seen instantaneously on the graphs. Differences in the assumptions made in both of the models will obviously have some bearing on the results that would be obtained if the same parameters were keyed into each in turn. The RJC model assumes that the mass flow through the entire cycle is the same, no allowances have been made, as yet, for losses due to heat transfer, or piston speeds but some pressure loss has been incorporated. Mechanical efficiency has been accounted for in the compressor and expander individually, to allow for more realism. The model for the Britalus has made allowances for heat, mass, pressure and friction losses throughout the cycle. Also, in all cases of analysis, the compressor and expander are able to operate at different rpm's due to the use of a variable speed drive configuration being programmed in to the computer model. The Britalus model also considers that the specific gas properties of the working gas may be different before and after combustion. These differences were considered to be negligible by the present author.

Tsongas does make the point that the Britalus cycle could be improved in a number of ways and makes recommendations, including the use of intercooling and regeneration, as proposed for incorporation into the RJC.

2.4 Axial-Piston Rotary Engine

This is another example of a continuous combustion engine, which also compresses and expands the working gas in separate cylinders. The engine layout consists of seven or eight cylinders, which are connected to an angled back plate. The entire set-up rotates about the central axis, thus producing the necessary compression and expansion strokes. Gas is transferred through the system via inlet and exhaust ports, which makes the use of valves and cams etc. redundant. A clear schematic of the engine can be seen below in Fig 8.

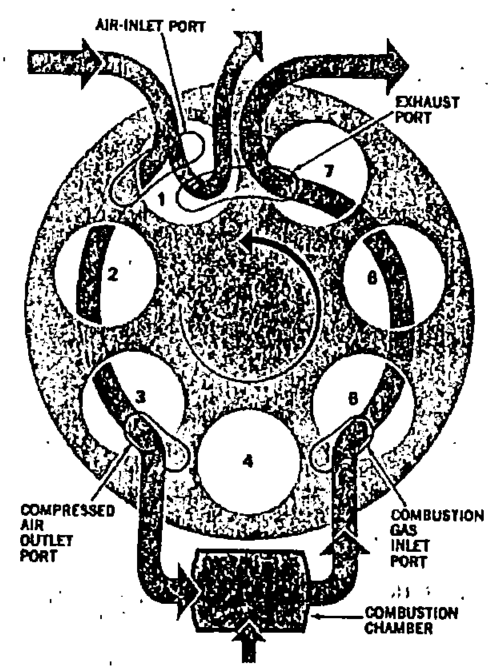


Fig. 8 Axial-Piston Rotary Engine [18]

Various prototypes of the engine have been built and tested, with either seven and eight cylinders. The work was carried out at the Institute for Automotive Studies in Aachen, West Germany. Results of the initial tests, carried out on the seven cylinder version, show some interesting points. Although at the early stages the engine only achieved a thermal efficiency of 16%, the areas for improving this value were clearly defined, and this is where further work was being carried out. One such area being the combustion chamber, which operated at temperatures of up to 2000°C. The structure consisted of a steel chamber which was water cooled. The heat losses experienced in the system could be as much as 40%, and if this could be reduced, it was predicted that the overall efficiency of the engine would at least double, making it comparable to conventional diesels. It also suggested that the gases entering the combustion chamber were pre-heated using a recuperator, so that less fuel would be required to reach a given burn temperature.

Although this type of engine could burn a wide variety of fuels and would produce very few polluting products, it can still only produce a thermal efficiency equivalent to that of conventional diesels. One major problem that is currently trying to be overcome is the fact that for its capacity, about 3 litres, it produced a low power output, only about 60hp [18].

2.5 Reciprocating Internal Combustion Engine with Constant Pressure Combustion

In one of the earliest references to a Reciprocating Joule Cycle, Warren and Bjerklie [19] looked at modifying existing technology to produce a competative alternative to the Internal Combustion Engine. The cycle was proposed as a 'reciprocating compressor and hot gas engine'. Combustion taking place continuously at constant pressure between the compressor and engine. This arrangement gave a higher compression ratio and maximum temperature, making the cycle relatively efficient. The investigating researchers did not see a need for regeneration.

This engine configuration was expected to have a 20-30% reduced fuel consumption, to almost eliminate exhaust pollutants, and to operate on a wide variety of fuels. It was proposed, and designed, around existing V8 configuration engines. Any even number of cyclinders and an in line style were also suggested as possibilities. One bank of the V acts as the compressor, the other the expander, with a water cooled combustion chamber linking the two across the top. The design of the engine, inlet and exhaust valves, the combustion chamber etc. are covered in some detail in Reference 19 but the basic configuration can be seen below in Fig 9.

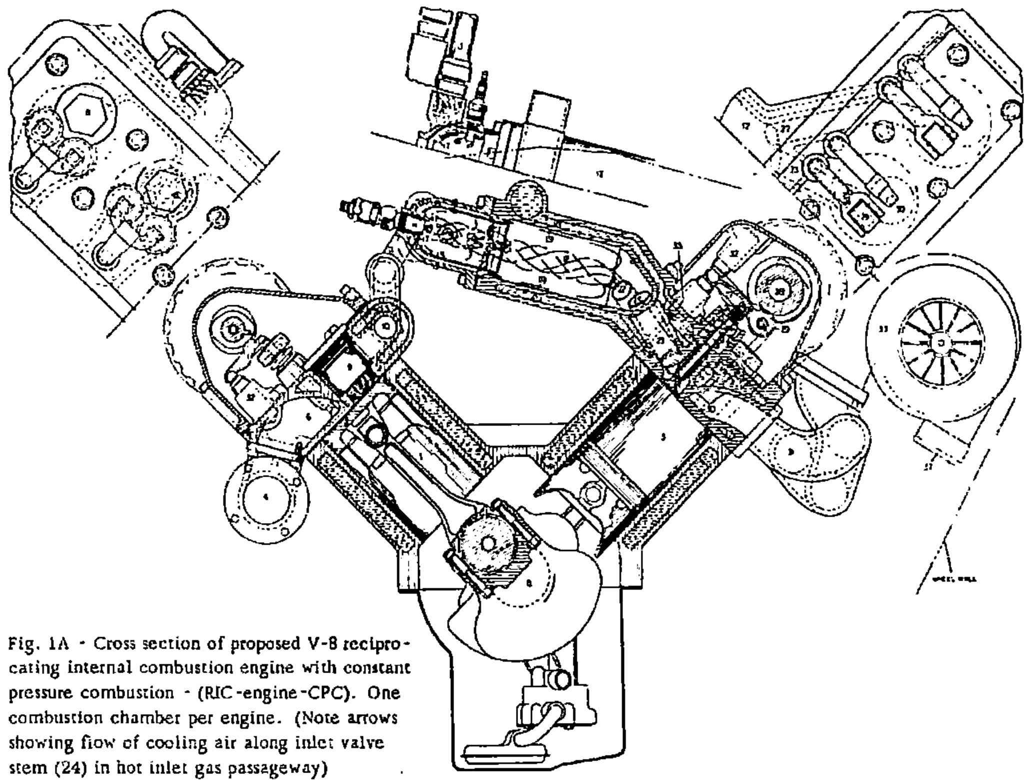


Fig. 1A - Cross section of proposed V-8 reciprocating internal combustion engine with constant pressure combustion - (RIC-engine-CPC). One combustion chamber per engine. (Note arrows showing flow of cooling air along inlet valve stem (24) in hot inlet gas passageway)

Fig. 9. 'V8' Configuration [19]

Computer analysis was made on the proposed engine, and the results compared with those obtained for an Otto cycle calculated from the same system. Extensive results were obtained for parameters such as torque v rpm, power required, bmep v rpm, all with and without supercharging. No mention, however, has been made of the engines overall thermal efficiency.

Another area of indepth research for this engine has been into its ability to reduce the amount of pollutants produced. It has been shown that unburned hydrocarbons are produced in excess by conventional engines when the combustable mixture is 'flame quenched' by the relatively cold cyclinder walls. This problem is eliminated in the proposed RJC engine, because combustion is completed in the combustion chamber before the quenching effect of the expansion cylinder walls can take effect. It is also proposed that combustion takes place in an excess of air, so that the amount of CO and HC present in the exhaust is reduced almost to zero. The amount of nitrous oxides (NO_x) produced should also be greatly reduced by the fact that combustion is taking place at a lower temperature and over a longer period of time than experienced in a conventional engine.

2.6 Valved, Hot-Gas Engine

This engine is a closed Joule cycle, consisting of a reciprocating expander and compressor, a heater, a cooler and a regenerative heat exchanger. The working gas being helium. A prototype was built and tested in the early 1970's [20], and from the test results obtained, improvements were sought after in a subsequent paper [21].

As documented in Reference 20, this was the first engine of this type to be built and tested. The cycle made use of a double acting piston. It was heated electrically - development of a hydrocarbon burner was anticipated, and the power was controlled by altering the pressure of the gas within the system. An alternative control method being to alter the expander swept volume, which would require the use of valve timing.

A simple computer analysis carried out on the cycle showed that the peak efficiency and work per cycle occurred at a pressure ratio of about 2:1, the efficiency being about 50%, and the work per cycle about 40%. The component efficiencies being at 90%, with the exception of the regenerative heat exchanger which was at 95%. When the prototype was tested the results were quite different to those anticipated. The overall efficiency of the engine was only 12%, some 30% less than the expected value. This drop in efficiency was put down to a serious gas leak in the system. Later work [21] showed that the problems were caused by heat transfer between the gas and the cylinder. Once solved the efficiency was expected to be back at around 50%.

2.7 Recent Investigations

Within the last 10 years, further work has been published on the use of the RJC, showing a marked progress towards creating a working engine. Patents have been awarded, and articles written about cars that can run on air. This literature adds weight to the case of the RJC.

The applications for two patents have been studied and there are similarities between the engines described [22, 23]. Key elements of each design are obviously the focus for these applications, which have no bearing on this investigation. Both designs utilise the basic cycle with only one of them including a regenerative heat exchanger, but both employ an opposed cylinder configuration. One incorporates an air bearing through which the pistons operate, which allows compensation for any differential expansion that takes place between the pistons and shells. Advantages to incorporating a well designed air bearing include the deletion of the need for lubricating oils, and the piston friction can be vastly decreased thus eliminating the need for the pistons and cylinders to be cooled. The bearing can also act as a seal to the pistons if the operational clearances are small enough, this being due to the fact that if a small amount of air is leaking out of the bearing then nothing can seep out of the cylinders.

A working prototype was built utilising the air bearing and a unique valving solution, its purpose being to prove the mechanics of the design and to aid with the finalisation of the air bearing. It was not however operated in anger, and was being considered for large scale power production.

In one of the documents, [23] it was stated that their particular engine would be capable of achieving efficiencies of 90%, assuming that the controls systems for the compressor, expander and regenerator could maintain the maximum possible efficiencies of the individual components when operating under a given set of conditions. However, it was also stated that practical engines would not run at this peak efficiency value, they would be more likely to operate at about 50% efficiency after size and cost considerations had been accounted for. The configuration allows the use of alternative fuels, such as methanol, and near zero emissions of hydrocarbons and monoxides are claimed. The thermal losses from the engine are also stated to be low, making an alternative application possible in large scale power production. The cost of manufacturing a reciprocating engine to this design is thought to be lower than that for a gas turbine, and the power output is claimed to be much more flexible.

An article published by the Daily Mail [24] in 1998 claimed that a car was running around Paris fuelled by nothing but air. The taxi that had the engine installed had a compressed air storage tank, which held the 'fuel'. This engine is unique when compared to the other Joule Cycle engines in that it operates with a compressor and two stages of expansion but has no combustor.

Air is drawn into the compressor via a filter. Once compressed the now hot air moves through the first expansion chamber. It is at this point that compressed air from a storage reservoir is added to the gas. The stored air expands on entry into the chamber and is heated by the compressed air from the cylinder. As this first expansion chamber is of a fixed volume, the pressure increases with the addition of the air from the compressed air reservoir. When this new air mixture is then expanded in the second chamber, work is produced which drives a shaft.

The vehicle was said to have a top speed of 68mph with a range of 124 miles or 10 hours running time. Although the claims for this engine are promising, no results of the test work in Paris have been seen.

2.8 Overview

Papers that have been written to date concerning a reciprocating Joule cycle engine all indicate that in theory, at least, the concept is more efficient than the current technology of the spark ignition engine, and comparable to that of the diesel. Most researchers have carried out at least a fundamental analysis of the cycle, with only Decher looking in more depth at the effects of part and full load operating conditions and heat losses, for example.

Of the work studied, few have been successful in building a complete prototype for test, those that were built [18, 20] performed well below expectations, due to inefficient combustion chambers and in-cylinder heat loss through heat transfer. These experiences were however useful in helping to develop the understanding of the operation of these designs and the problem areas were pinpointed for resolution. Unfortunately the result of any subsequent alterations to these engines is unknown.

All parties interested in the RJC set out to show that it has great potential in the transport industry, and in some cases the power production fields. Theoretical analysis throughout the work studied shows a minimum thermal efficiency of 33%, ranging up to 90% dependent upon many different parameters. When comparing basic Joule cycle thermal efficiency to basic Otto cycle thermal efficiency, the joule proves to be higher. However, on this basis alone the Joule cycle cannot be adopted for transportation purposes, there is some discrepancy as to the power output achievable with an RJC, particularly in relation to the physical size. One of the prototypes [18], which was a 3 litre engine, was only able to produce 60hp.

There is also no clear consensus throughout the documentation about the value of adding a regenerative heat exchanger. Some [20, 23] feel it is necessary to increase the thermal efficiency attainable by the engine, others consider that it is not necessary believing that no real gains would be made [19].

It is felt that the Reciprocating Joule Cycle engine does indeed hold a large amount of potential which is still waiting to be fulfilled. The efforts of previous researchers has validated, to a certain degree, the results that have been obtained from this project's theoretical analysis. The values may not be exactly the same but the trends follow the same patterns.

3.0 RJC Engine Theory

3.1 Combustion Engines

Internal combustion engines have been in use in one form or another since before the turn of the 20th century. The operational cycles of the various different types will not be detailed as they can be found in any standard text [25]. The cycle at the heart of this research will however be look briefly at.

3.2 Joule Cycle

Joule first proposed the cycle in the mid 1800's, it is commonly used today in the form of the Gas Turbine. This continuous combustion process is also known as the Brayton Cycle, after George Brayton who developed the first operational engine of this type in 1870.

As has already been mentioned this type of engine can take one of two forms; rotational or reciprocating. In either case the Joule cycle is different from conventional engines in that individual components are required to house each stage of the cycle. Figure 10 illustrates the Pressure/Volume relationship for a basic Joule cycle. Air induced into the engine is compressed in an isentropic process between 1-2. The addition of heat to the system at constant pressure is indicated between 2-3. Isentropic expansion then takes place between 3-4, providing the power to run the engine, and finally the remaining heat is removed from the system in a constant pressure process between 4-1.

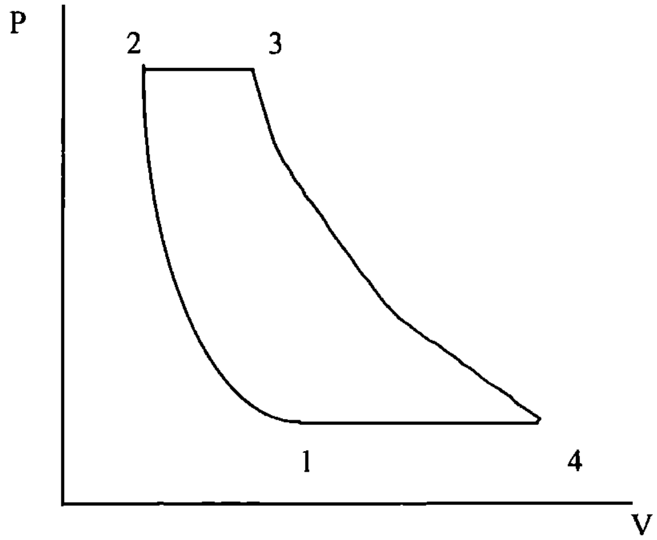


Fig.10. PV Diagram for Joule Cycle

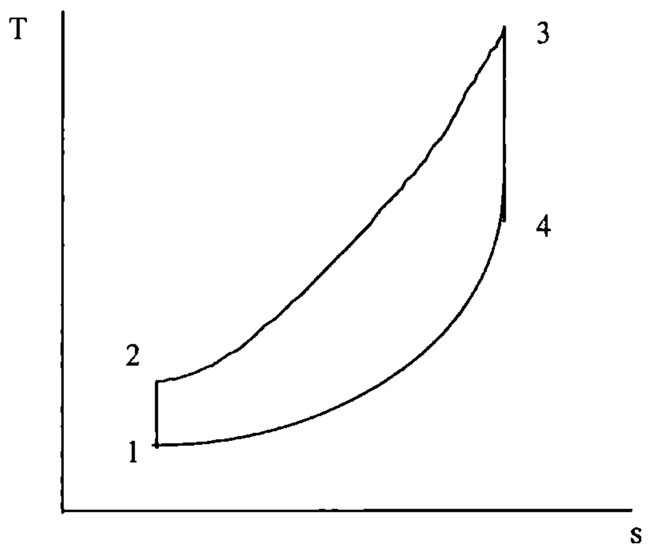


Fig. 11. Ts Diagram for Joule Cycle

The Ts diagram for the Joule cycle is shown above in Fig 11. A comparison to the Ts diagrams of the Otto and Diesel cycle shows a marked similarity to that of the Joule cycle. The difference here is that the lines 2-3 and 4-1 are constant pressure lines.

All processes of the Joule cycle are carried out in steady flow devices, so can be analysed as steady flow processes. The conservation of energy equation for a steady flow process can be given as follows for the Joule cycle when the potential and kinetic energies are neglected:

$$q - w = h_{exit} - h_{in} \tag{2}$$

Assuming constant specific heats at steady state conditions the thermal efficiency can be derived as follows;

$$q_{in} = q_{23} = h_3 - h_2 = C_p(T_3 - T_2) \tag{3}$$

$$q_{out} = q_{41} = h_4 - h_1 = C_p\,(T_4 - T_1) \tag{4}$$

$$\eta_{th.joule} = \frac{W_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_p(T_4 - T_1)}{1 - C_p(T_3 - T_2)} \tag{5}$$

so,

$$\eta_{th.joule} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)} \tag{6}$$

As processes 1-2, and 3-4 are isentropic and $P_2 = P_3$, $P_4 = P_1$,

so,

$$\frac{T_2}{T_1} = \frac{P_2}{P_1}^{(n-1)/n} = \frac{P_3}{P_4}^{(n-1)/n} = \frac{T_3}{T_4} \tag{7}$$

substitution gives:

$$\eta_{th.joule} = 1 - \frac{1}{r_p^{(n-1)/n}} \tag{8}$$

where r_p = pressure ratio, $\frac{P_2}{P_1}$

This shows that a Joule cycles efficiency is dependent upon the pressure ratio and specific heat ratio of the working fluid (if not air) as demonstrated below in Fig 12.

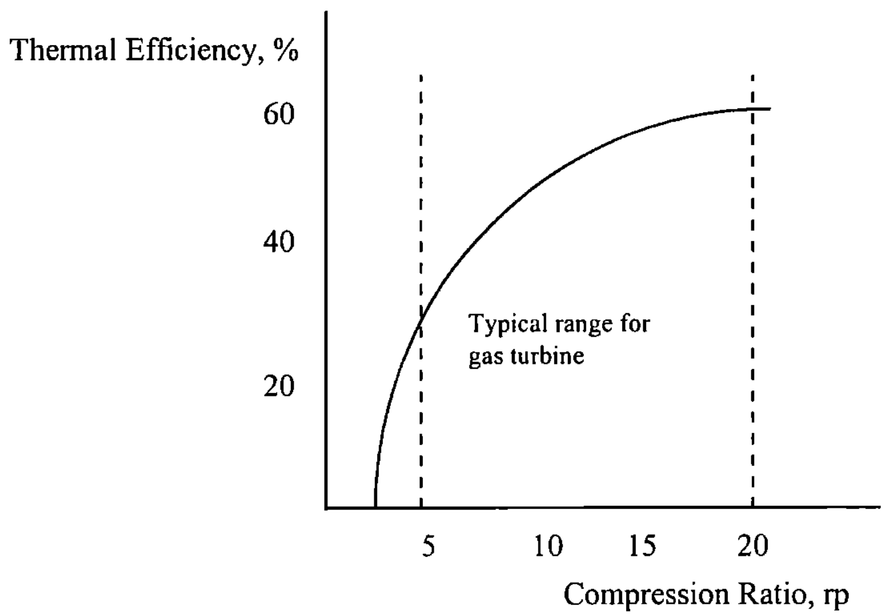


Fig.12. Thermal Efficiency v's Pressure Ratio for Joule Cycle.

The peak temperature (T_3) of the cycle occurs at the inlet to the expander. In gas turbines this temperature is limited by the maximum temperature that the turbine blades can withstand, this also limits the operational pressure ratio, and thus the thermal efficiency. Operational engines of this cycle are therefore defined by a fixed compressor inlet temperature, T_1, and fixed expander inlet temperature, T_3. The appropriate pressure ratio is then used to produce the optimum nett work output from the system.

The work achieved by the engine as a whole is a function of the work done by the compressor and expander. So initially the derivation of work for the individual components was considered.

Consider the compressor:

$$W_{co(net)} = \frac{m_{flow} \times C_p \times T_1 \times (r_p^{(n-1)/n} - 1)}{\eta_{mech}} \quad \text{J/cycle} \tag{9}$$

To gain specific work in kJ/Kg:

$$W_{co} = \frac{W_{co(net)}}{m_{flow}} \times \frac{1}{1000} \tag{10}$$

To find the work done by the expander all processes of the PV diagram are considered and then added to give the overall value. The expander can be considered to undertake either full or partial expansion of the gases that enter the cylinder. In this case partial expansion will be demonstrated. The PV diagram is as follows in Fig 13:

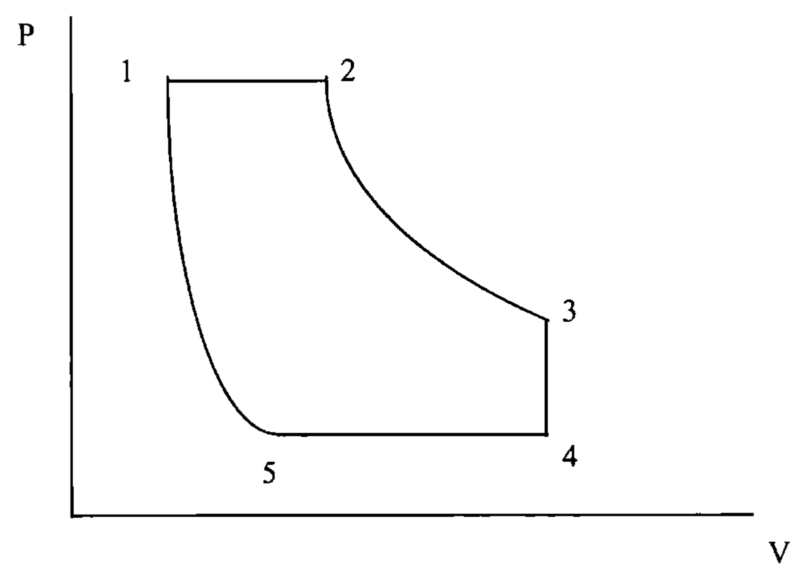


Fig. 13. PV Diagram illustrating partial expansion.

So,

$$W_{12} = P_1 \times (V_2 - V_c) \tag{11}$$

$$W_{23} = \frac{P_3 \times (V_s + V_c) - P_1 \times V_2}{1 - n} \qquad (12)$$

$$W_{45} = -P_{exit} \times ((V_s + V_c) - V_5) \qquad (13)$$

$$W_{51} = \frac{P_1 \times V_c - P_{exit} \times V_5}{1 - n} \qquad (14)$$

Where V_s = swept volume of the cylinder

and V_c = clearance volume of the cylinder

Net work is therefore:

$$W_{exp(net)} = (W_{12} + W_{23} + W_{45} + W_{51}) \times \eta_{mech} \quad \text{J/Cycle} \qquad (15)$$

As with the compressor, specific work is gained by:

$$W_{exp} = \frac{W_{exp(net)}}{m_{flow}} \times \frac{1}{1000} \quad \text{kJ/kg} \qquad (16)$$

The net specific work for the engine is therefore:

$$\text{Specific Work}_{(net)} = W_{exp} - W_{co} \quad \text{kJ/kg} \qquad (17)$$

3.3 Comments

The analysis undertaken for the Reciprocating Joule Cycle engine is very idealised. It looks at the processes that take place in each of the individual components and assumes that no losses occur as the fluid is transferred around the engine, i.e. the fluid will enter one components at the same conditions at which it left the previous one.

In reality however, this is not the case. As the fluid passes through the engine it meets various obstacles in the form of changing pipe diameters, new port shapes, valves and the fluid flow is diverted from linear to turbulent. All of these obstacles result in pressure and heat losses in the fluid. Energy is removed from the system irreversibly by these means, and they have not been accounted for in the analysis. The predicted theoretical results will therefore be higher than those obtained via experimental methods.

Other factors will also contribute to the theoretical results being different from those gained experimentally. Initially, the mechanical efficiencies of the components has been estimated, in reality this value may prove to be different. Steps can be taken however to more accurately calculate the actual mechanical efficiencies of the compressor and expander, these values can then be inserted into the theoretical analysis. Temperature losses have already been mentioned, but the combustion chamber is liable to be the largest source of such losses as it is not insulated, and this could have quite an impact on the performance of the engine. Even if the peak cycle temperature is achieved the amount of fuel input to the system would be larger than necessary for a fully insulated chamber, its efficiency is therefore expected to be low. Other inefficiencies may arise within the chamber if the fluid flow within it is not as expected. This can lead to the flame velocity being incorrect which can result in the flame being held within the wrong part of the chamber, which will in turn lead to a less than optimum fuel-air mixture.

4.0 Theoretical Computer Model

4.1 Introduction

As the prototype engine was to consist of essentially three separate components the best method for the analysis of the engine as a whole was seen to be to consider each component on an individual basis. Conceptually, the reader is following the gases as they flow through the engine, and are experiencing all pressures exerted. Analysis was kept at a simplistic level as an ideal cycle was assumed, hence a Microsoft Excel spreadsheet was created that calculated parameters at the various stages of the cycle. The spreadsheet was constructed piece by piece as the individual components were analyzed, with data from the first component analysis being required for calculations at the next stage.

As each of the components would be linked together via either internal ducting or external pipework it was felt that the effect of pressure losses through the system should be investigated. Again the analysis was kept at a simplistic level, looking at the effect of changing pipe diameters on the flow of gases through the engine. The impact of such losses could then be gauged.

Also built into the spreadsheet was the ability to analyze the system with the inclusion of intercoolers and regenerative heat exchangers. Although neither of these items were anticipated for inclusion in the prototype at this stage it was felt that they could be of benefit, and the effects of their inclusion in the system were worthy of note and comparison to the basic cycle.

The analysis of the engine was done over a range of compressor pressure ratios, so that the most efficient operational conditions could be found. It also allowed certain characteristics to present themselves immediately as conditions changed over the range of operational possibilities.

At each stage of the construction of the spreadsheet the data being created was checked to ensure that all of the calculations had been input correctly. To ensure that the correct information was there, under a specific set of parameters a base line piece of information was set so that certain values were known. The most effective method for achieving this was to set the pressure ratio of the compressor.

The diagram in Fig.14. illustrates the Ts diagram for the computer model of prototype engine, note that some processes have been included in the model that will not find inclusion into the prototype for some time. This was done for the purpose of ease of comparison. The compression process is modelled in two stages, the first taking place between 1-2, with intercooling decreasing the temperature of the gases from 2-3 with the final stage of compression between 3-4. Regenerative heat exchange from the exhaust gases is allowed for between 4-5. Combustion then occurs between 5-6, with the maximum cycle temperature being at 6 after the completion of combustion. Expansion is represented by 6-7 and as has already been mentioned regenerative heat exchange occurs and is shown on this part of the cycle between 7-8, from 8-1 the remainder of the heat from the exhaust gases is rejected to atmosphere.

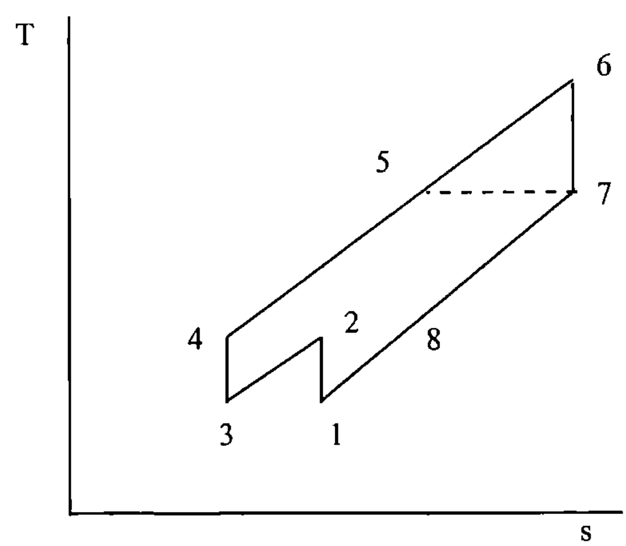


Fig.14. Ts Diagram for the Prototype Joule Cycle Engine

4.2 Compressor

Before any calculations could be carried out certain parameters of the compressor needed to be defined. Measurements were taken from the engine and this allowed the swept volume (V_s) to be stated and from this the clearance volume (V_c) was established. Air conditions on entering the compressor were also set at steady state conditions. A value for the RPM was also stated along with an estimate of the mechanical efficiency of the compressor, which was taken from standard compressor data.

By stating these parameters as fixed, and others throughout the analysis of the other components, the effects of certain changes could be monitored accurately when changing one parameter at a time. Once graphical information had been created, any subsequent changes to any of the base data could be seen instantaneously on the relevant graph, which proved to be an extremely useful tool.

4.2.1 Compressor Cycle Analysis

The compressor was to be single stage, the cycle consisting of air being drawn into the cylinder on the downward stroke, 4-1, with compression taking place on the subsequent upward movement of the piston, 1-2. At a predetermined position, 2, the exhaust valve would then open to release the now compressed gas into the engine, and it can be seen that this is a two stroke operation. From 3-4 the piston is commencing its stroke towards BDC. Both valves are closed and any gas remaining in the cylinder is expanded, at a pre-determined point, i.e. 4, the inlet valve opens and the induction part of the stroke begins again.

The PV diagram for the RJC compressor is as illustrated below in Fig 15:

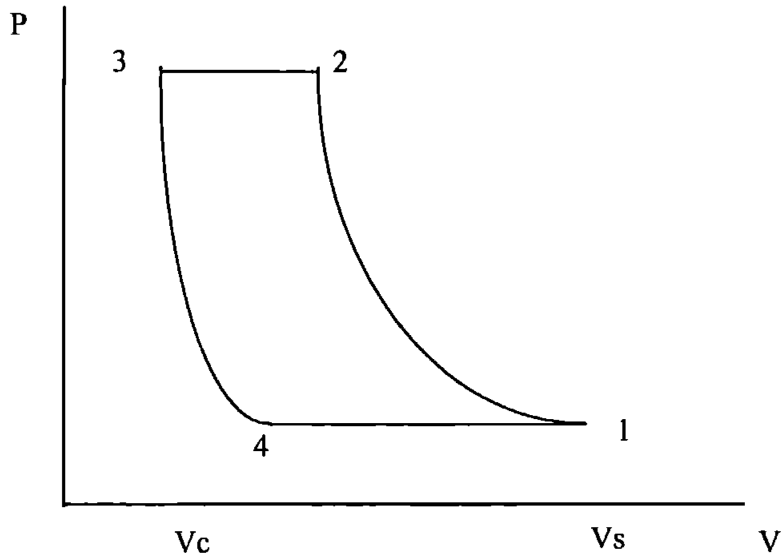


Fig.15. PV Diagram for RJC Compressor.

As has already been stated, the model makes allowance for the use of multi-stage compression. The relevant calculations will now be demonstrated in a full analysis of the compressor at a single pressure ratio, and the simulation of single stage compression will be explained. The spreadsheet, which is included on disk in the envelope inside the back cover, may be consulted for further information.

The data that was input as fixed information for calculations relating to the compressor is as follows, steady state conditions were assumed:

Pin = 101325Pa

Tin = 288K

n = 1.4

R = 287

RPM = 3000

ηmech = 0.85

Data relating to the capacity of the compressor was handled as follows:

$Vc/Vs = 0.015\ m^3$	fixed value (measured)
$Vs = 0.000624\ m^3$	fixed value (measured)
$Vc = 9.36 \times 10^{-6}\ m^3$	calculated
$Vs + Vc = 0.000633\ m^3$	calculated

Assuming a pressure ratio $r_p = 5$ for example.

In the event of multi-stage compression the optimum compression ratio between the two stages was found as follows:

$$r_{popt} = r_p^{1/k} \tag{18}$$

where k = number of stages i.e. 2

If there is only one stage then $r_{popt} = r_p$

The pressure gained at the intermediate stage of compression is defined as:

$$P_i = Pin \times r_{popt} \tag{19}$$

so:

$$P_i = 506625 \text{ Pa}$$

The volumetric efficiency is then found.

$$\eta_{vol} = 1 - V_c/V_s \, (r_{popt}^{\ 1/n} - 1) \tag{20}$$

$$\eta_{vol} = 0.967646$$

Mass flow:

$$m_{flow} = \frac{P_{in} \times V_s \times \eta_{vol}}{R \times T_{in}} \qquad (21)$$

$$m_{flow} = 0.00074 \text{ m}^3$$

Evident in the next equation are the effects of the inclusion of an intercooler. If two stage compression is taking place and ideal cooling is assumed then the temperature after the second stage of compression is assumed to be the same as it was after the first stage i.e. T_2 is equal to that of T_4. In reality however the intercooler is not 100% efficient so the temperature after cooling will be slightly higher than that of the inlet temperature, resulting in the final exhaust temperature after the second stage of compression being higher than that after the first.

The temperature of the gas at point T_2 on the PV diagram is calculated as follows, assuming an intercooler effectiveness of 100%:

$$T_2 = T_{in} \left(r_{popt}{}^{(n-1)/n}\right) \qquad (22)$$

$$T_2 = 456.14 \text{ K}$$

$$T_3 = T_2 - E \times (T_2 - T_{in}) \qquad (23)$$

$$T_3 = 288 \text{ K}$$

$$T_4 = T_3 \left(r_{popt}{}^{(n-1)/n}\right) \qquad (24)$$

$$T_4 = 456.14 \text{ K}$$

where E is the intercooler effectiveness.

The work done by the system is given as:

$$W_{co.net} = \frac{[m \times C_p \times T_1 \times (r_{popt}^{(n-1)/n} - 1)] + [m \times C_p \times T_3 \times (r_{popt}^{(n-1)/n} - 1)]}{\eta_{mech}} \quad \text{J/cycle} \quad (25)$$

$$W_{co.net} = 147.88 \text{ J/cycle}$$

In the case of single stage compression the second part of the expression equals zero.

Work done is therefore;

$$sp\ wk_{co} = \frac{W_{net}}{m} \times \frac{1}{1000} \quad (26)$$

$$sp\ wk_{co} = 199.78$$

4.3 Pressure Loss Calculation

For the calculation of the pressure drops through the pipe work it was assumed that the density of the air would remain constant. The cross sectional areas of the different pipe or cylinder sections were stated, and the effects calculated for different pressure ratios as with the remainder of the analysis.

Pipe area = 0.00049 m^2

Cylinder area = 0.00636 m^2

Combustion Chamber area = 0.01767 m^2

Tank area = 0.12566 m^2

Density air = 1.3 kg/m^3

For the analysis the system was considered in the following configuration shown in Fig 16:

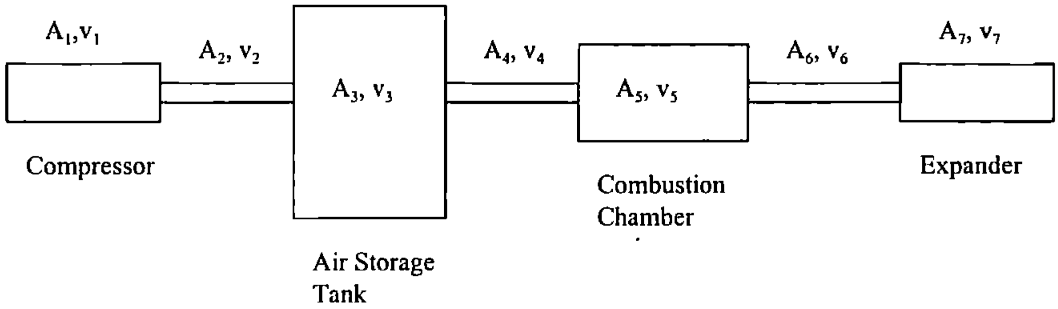


Fig. 16. Pressure Loss Considerations

When the fluid leaves the compressor and enters the pipe it is considered to enter a 'sudden contraction', it then experiences a 'sudden exit' as it enters the tank. As it leaves the tank and enters the next connecting pipe it undergoes a 'sudden entry' followed by a 'sudden enlargement' as it enters the combustion chamber. A 'sudden contraction' occurs on re-entry to the pipe followed by a 'sudden enlargement' as the gas finally enters the expander.

The cross sectional areas of the components are known and the velocities in each component can be deduced from the calculated mass flow at a particular pressure ratio.

So, for the sudden contraction form the compressor to the pipe:

$$P_L = \frac{k\rho v_2^2}{2} \qquad (27)$$

where $k = \left[1 - \frac{A_1}{A_2}\right]^2$

And $v_2 = \frac{m_{flow}}{\rho A_2}$

For the sudden exit from the pipe to the tank:

$$P_L = k\rho \frac{v_2^2}{2} \tag{28}$$

Where k = 1

At the sudden entry into the pipe:

$$P_L = k\rho \frac{v_4^2}{2} \tag{29}$$

where $k \cong 0.5$

The sudden enlargement as the air enters the combustion chamber:

$$P_L = k\rho \frac{v_4^2}{2} \tag{30}$$

where $k = \left[1 - \frac{A_4}{A_5}\right]^2$

Sudden contraction as gas re-enters the pipe:

$$P_L = k\rho \frac{v_6^2}{2} \tag{31}$$

where $k = \left[1 - \frac{A_5}{A_6}\right]^2$

And finally, the sudden enlargement as the gas enters the expander:

$$P_L = k\rho \frac{v_6^2}{2} \tag{32}$$

where $k = \left[1 - \frac{A_6}{A_7}\right]^2$

The pressure loss calculated as the gas enters the expander is multiplied by 2, as there are two expanders. The total pressure loss in the system is then calculated by the summation of all of the above results. This loss is then deducted from the ideal entry pressure to the expander, with the new value being represented by 'P1act' in the spreadsheet.

4.4 Combustion Chamber

The combustion chamber was included in the spreadsheet by the simple means of stating the maximum temperature of the gases leaving the chamber, and therefore entering the expander. By doing this the process was greatly simplified, but it did allow for the temperature sensitivity of the system to be gauged, i.e. simply by altering the maximum allowable temperature in the system the effect on the thermal efficiency could instantly be seen.

In reality the performance of the combustion chamber would have a great effect on the overall performance of the engine. Even if the initial design proved to have efficient combustion taking place, any heat losses through the chamber that were not accounted for could lead to large discrepancies between the practical results and the calculated ones. This was a shortcoming of the model which was understood, as long as the main characteristics of the system could be identified under perfect conditions any losses could be accounted for. It was also highly likely that a second prototype would be constructed in a different configuration, which would alter many of the loss characteristics in practice. These problems would be resolved when necessary, but for now a basic comparison between theory and practice was all that was required to serve the purpose of justifying the RJCE.

4.5 Expander Cycle

As with the compressor, certain basic parameters were fixed, these are listed later in the chapter. The expander would be working alternately from two cylinders, so the total expander volume characteristics were stated, which was simply double the value stated for the compressor. Efficiency was again stated along with an estimation of the attainable expander mechanical efficiency.

The cycle was then calculated step by step from the following PV diagram. The work done was calculated and from this the specific work done and the thermal efficiency and power output were calculated.

It was considered that full expansion of the gases may not take place due to restrictions of geometry, for this reason a partial expansion case was calculated and included in the model. The case for partial expansion gives worse results for work done, due to the area under the graph being reduced. Therefore if full expansion were to take place the results would be better than expected. The processes of expansion can be seen in Fig 17. The gases are induced into the expander during 1-2, the expansion process then takes place between 2-3. Process 3-4 drops the pressure of the gases to atmospheric without changing the volume of the gases. The expanded gases are then purged from the cylinder during 4-5. A small amount of the expanded gases that remain within the cylinder after the purge cycle are then re-compressed as the piston travels back towards TDC in process 5-1. This brings the cylinder back up to the pressure of the gases that are going to be drawn into the cylinder at the start of the next cycle. ? not needed

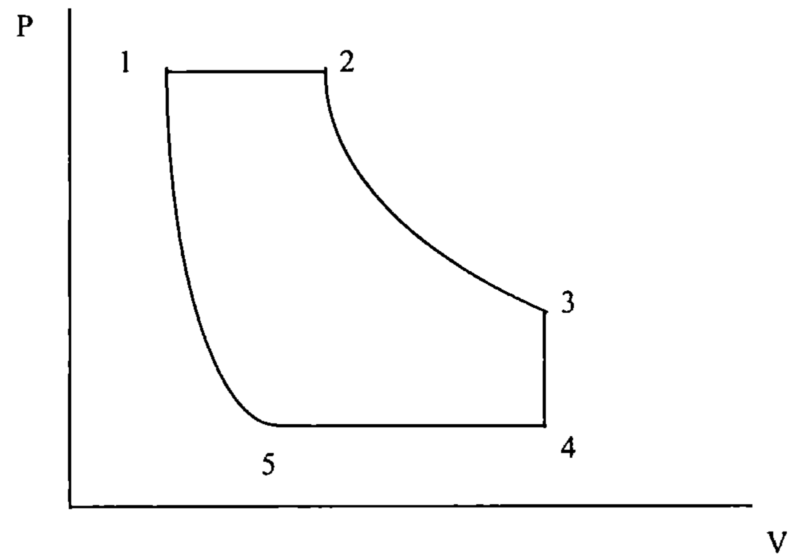


Fig 17. PV Diagram for the Expander

As with the progress of the compressor calculations, periodically the spreadsheet was checked with known data. This ensured that all entered calculations were correct, i.e. a pressure ratio of 1 was used so that the cycle would be doing no work therefore results of zero efficiency and power output were expected.

Data that was fixed for all of the expander calculations was as follows:

$Vc/Vs = 0.015\ m^3$

$Vc = 1.872 \times 10\text{-}5\ m^3$

$Vs = 0.0012480\ m^3$

$n = 1.4$

$\eta mech = 0.85$

$Pexit = 101325\ Pa$

$T_6 = 1273\ K$

If regeneration were to be seriously considered it would be calculated as follows:

$$Q_{46} = m_{flow} \times C_p \times (T_6 - T_4) \tag{33}$$

$$Q_{46} = 610.67$$

When considering the expander cycle the first place to start was with the calculation of the inlet pressure.

$$P_1 = P_{exit} \times r_p \tag{34}$$

$$P_1 = 506625$$

Pressure loss is then accounted for by:

$$P_1 - P_{LTOT} = P_{1act} \tag{35}$$

Hence P_{1act} becomes the inlet pressure for the expander.

The swept volume of the expander was inserted as a fixed value on the spreadsheet, all of the cylinders are of the same volume, therefore it is basically twice the value of $V_c + V_s$ for the compressor. At low pressure ratios the induced volume into the expander must be restricted to the volume of the cylinder that is practically available, it follows that the peak temperature of the cycle must also be lower than the target temperature. If the value of V_2, as described by the below equation:

$$V_2 = \frac{m_{flow} \times R \times T_6}{P_1} + V_c \tag{36}$$

$$V_2 = 0.000533 \text{ m}^3$$

is greater than $V_c + V_s$ then the equation for T_6 becomes:

$$T_{6limit} = \frac{T_6 \times (V_c + V_s)}{\frac{M_{flow} \times R \times T_6}{P_1} + V_c} \tag{37}$$

$$T_{6limit} = 1273 \text{ K}$$

So, the equation for V_2 must also be limited so that it is smaller than or equal to $V_c + V_s$.

$$V_3 = V_s + V_c \tag{38}$$

$$V_3 = 0.001267 \text{ m}^3$$

$$V_5 = V_c \text{ x } (P_1/P_{exit})^{(1/n)} \tag{39}$$

$$V_5 = 5.91 \text{ x } 10^{-5} \text{ m}^3$$

$$P_3 = P_1 \text{ x } (V_2/(V_s + V_c))^n \tag{40}$$

$$P_3 = 158564.3$$

The work done by the expander must be calculated in stages which are broken down as follows:

$$1\text{-}2 = P_1 \text{ x } (V_2 - V_c) \tag{41}$$

$$1\text{-}2 = 270.43$$

$$2\text{-}3 = \frac{(P_3 \text{ x } (V_s + V_c) - P_1 \text{ x } V_2)}{(1-n)} \tag{42}$$

$$2\text{-}3 = 197.64$$

$$4\text{-}5 = -P_{exit} \text{ x } ((V_s + V_c) - V_5) \tag{43}$$

$$4\text{-}5 = -122.36$$

$$5\text{-}1 = \frac{(P_1 \text{ x } V_c - P_{exit} \text{ x } V_5)}{(1-n)} \tag{44}$$

$$5\text{-}1 = -8.74$$

This gives a total net work:

$$W_{exp.net} = (W_{12} + W_{23} + W_{45} + W_{51}) \times \eta_{mech} \quad (45)$$

$$W_{exp.net} = 286.42$$

$$sp.wk_{exp} = \frac{W_{exp.net}}{m} \times \frac{1}{1000} \quad (46)$$

$$sp.wk_{exp} = 386.96$$

$$Sp\ Wk._{net} = sp\ wk_{exp} - sp\ wk_{co} \quad (47)$$

$$Sp\ Wk._{net} = 187.17$$

Considering the overall cycle of the engine, the temperature of the exhaust gases leaving the expander is calculated as follows:

$$T_7 = \frac{T_6}{(P_1/P_3)^{((n-1)/n)}} \quad (48)$$

$$T_7 = 913.5\ K$$

If regeneration takes place and the exhaust gas temperature is greater than that of the gases leaving the compressor, i.e. $T_7 > T_4$,

$$T_5 = e.\ (T_7 - T_4) + T_4 \quad (49)$$

$$T_5 = 456.14\ \text{K}$$

where e is the regenerator efficiency.

$$Q_{56} = m_{flow} \times C_p \times (T_6 - T_5) \tag{50}$$

$$Q_{56} = 610.67$$

$$Wk_{.net} = W_{exp.net} - W_{co.net} \tag{51}$$

$$Wk_{.net} = 138.54$$

$$\eta_{th} = Wk_{net} / Q_{56} \tag{52}$$

$$\eta_{th} = 0.226$$

$$\text{Power} = Wk_{net} \times \frac{RPM}{60000} \tag{53}$$

$$\text{Power} = 9.93\ \text{kW}$$

4.6 Analysis

Graphical data can be generated very easily from the spreadsheet results and can easily be manipulated to show the effects of altering specific parameters. This feature gives a great deal of flexibility that will prove to be extremely useful when analysing the data collected from the engine itself. This information allows provided a view of how the engine might react under certain conditions and will highlight any anomalous results collected from the engine, and hopefully this data will indicate the reasons behind the differences.

As mentioned previously, allowance was made in the spreadsheet for the inclusion of intercoolers and heat exchangers. Although this is a practical step that will not take place in this investigation, the effects of their inclusion in the system are still of interest. This allows a brief analysis of the affect of adding such components to see if the practical application would in fact be a benefit to the project at a later date.

As a result of this theoretical investigation, targets for the prototype engine performance could be set. A slight compromise between the thermal efficiency and power output was chosen, so the target specification looked as shown below in Fig 18:

Pressure ratio	10 : 1
Air flow	0.717 litres/cycle
Air : Fuel	1 : 60
Speed range	0 – 3000 rpm
Power Output	7.3 kW
Thermal Efficiency	28 %

Fig. 18. RJC Prototype Engine Target Specification

5.0 RJC Engine Prototype Design

5.1 Introduction

Three options were presented for the configuration of the first engine.

1) It could be built from individual components, i.e. separate compressor and expander. This would take advantage of the efficiencies of both of these individual components.
2) It could be built from a clean sheet, which would mean that ideal components and configuration could be incorporated.
3) It could be built from a donor engine which would be modified to suit its new purpose.

The first two options would be quite costly, and a clean sheet design would require a vast amount of time and resources to complete. It was decided that the best way forward for the initial prototype would be to modify an existing engine. Allowances could be made for the inclusion of an intercooler or regenerator at a later stage once the initial design had proven itself.

The first prototype was to be a simple RJC consisting of a compressor, combustion chamber and expander. The combustion chamber was to be specially designed and built and attached to the system external to the donor engine. The engine chosen for the conversion was a Ford FSD425, a 2.5 litre direct injection diesel engine with a straight 4 cylinder configuration. Being a diesel it was quite robust and would easily be able to cope with the workload of the RJC.

The main block of the engine was converted to house the compressor and expander, the combustion chamber being a separate unit. It was decided that cylinder No 1 would operate as the compressor and cylinder numbers 3 and 4 would become the expander. The latter two cylinders expand on alternate strokes. Cylinder No 2 would perform no function, but in the interest of retaining balance the piston was left in place but two piston rings were removed to reduce friction. It was understood that with this arrangement the supply of compressed air to the combustion chamber would fluctuate, however with the inclusion of a storage tank in the system the effects of this would be dampened, and the flow to the chamber would remain relatively continuous.

In preparation for the full conversion from diesel to RJC all four cylinders of the donor engine were honed and the bowl head pistons were replaced with flat topped pistons. This was done so that the clearance volume could be minimized therefore allowing the volumetric efficiency of each of the cylinders to be pushed as high as possible. For example, assuming a pressure ratio of 10:1 within the cylinder the volumetric efficiency with the bowl head pistons was 79%, the change to flat topped pistons increased the volumetric efficiency to 93%. The superfluous diesel equipment was removed, i.e. the injectors, and the ports were blocked by the simple insertion of a suitable diameter rods which were clamped in position.

Before the conversion began some thought went into the operation of the engine once complete. It would be running on Compressed Natural Gas and was by its very nature an experimental engine. This meant that the safe operation of the engine had to be considered and the appropriate Health and Safety checks carried out. A risk analysis was completed and safe working practices written with specific safety checks built into the normal operation of the engine. For example, to guard against the flame going out in the combustion chamber the spark used for ignition was operated by a timer circuit that created a spark two milliseconds, this period could be increased by adjusting a variable resistor. So if for whatever reason the flame did go out, within a few second there would be a spark to re-ignite the fuel-air mixture, thus not allowing any dangerous build up of fuel. The Safe System of Work including Risk Analysis can be seen in Appendix 1.

5.2 Compressor

5.2.1 Valve Timing Options

The compressor was required to deliver air at a designated pressure to the combustion chamber. The theoretical analysis of the cycle indicated that a compression ratio of 7:1 would develop a good efficiency. So the basic specification of the compressor was to produce air at nominally 7bar with every compression stroke.

To produce a compressor from the existing cylinder configuration the valve operation needed to be changed to ensure the desired result. Two options for the valving that were considered would have been applicable to both the compressor and expander, these being hydraulic and electromagnetic systems. The details of both of these systems can be seen in more detail in References 26 & 27. The advantage of the hydraulic system was that it could be programmed to simulate any cam profile, which would be advantageous for altering the valve timing quickly during testing. In discussion with Lotus Engineering Ltd, the designers of the system, highlighted its use as a research tool. It was not considered as a viable possibility for production cars due to its bulk on the engine, the instrumentation and the cost. All of these factors made this option infeasible for the initial prototype.

The electromagnetic system [27], briefly illustrated in Fig 19, appeared to offer a compact solution to the valve timing problem. The entire system had a simple design that offset the slightly more limited valve timing options. The system consisted of a metal disk that attached directly onto the valve stem, two electromagnets positioned above and below the disk and springs to assist with valve return. During operation the electromagnets were triggered in turn attracting the disk, thus opening or closing the valve.

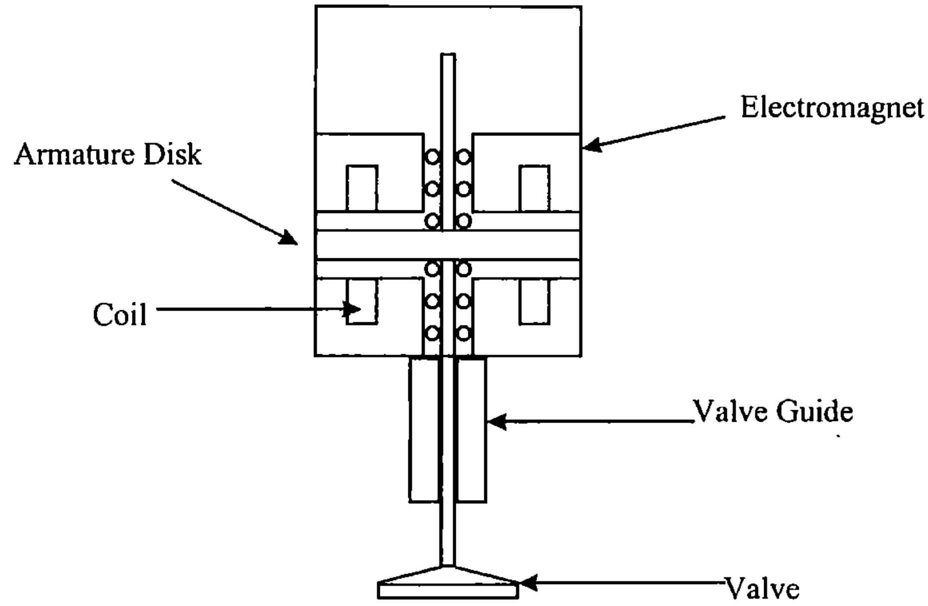


Fig. 19. Electromagnetic valve timing system.

Information on this system showed it to be very promising but in need of some further development. Further research uncovered the unreliability of the system. Monitoring of the valve position proved to be particularly difficult as the sensors monitoring the valve stem were not accurate, this lead to several clashes of the valve with piston. The material chosen for the electromagnets was also considered to be of a lower quality than actually required. So, after due consideration this system was also rejected.

The unfruitful search for alternative methods of valve timing brought the research back into the realms of conventional technology. At this point standard compressor valves were considered.

Reed valves are common in compressors, an example of one can be seen below in Fig 20. Their operation is simple as they rely on pressure difference to trigger their operation, rather than the use of cams. Reed valves are basically strips of metal that are restrained at one end. The thickness of the valve dictates the force required to open or close them, but they are flexible enough to perform the operation with ease.

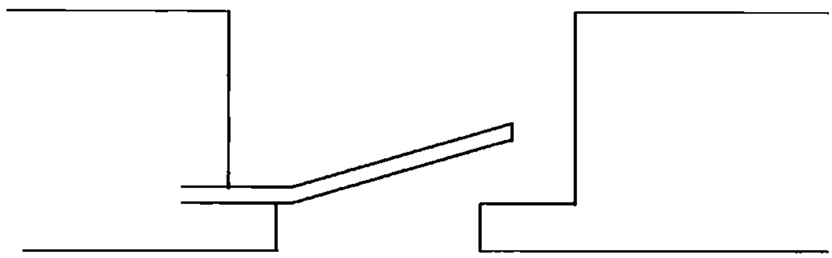
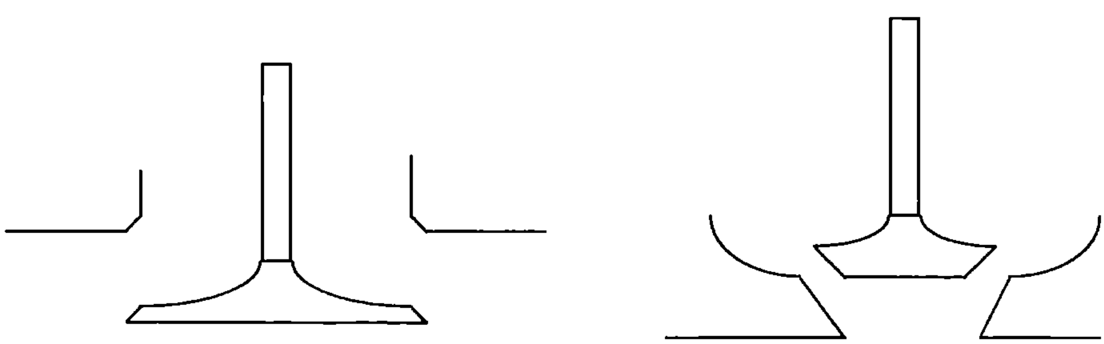


Fig. 20 Reed valve

Disk valves operate in a similar manner, relying on pressure difference to determine their position between open and closed. The use of this type of valve gear with the RJCs compressor was seen as a viable option as the technology is proven.

The initial solution was felt to be to convert the poppet valves to disk valves to achieve operation by pressure difference. To achieve this the rocker arm mechanism would be disguarded for both valves. The inlet valve could retain its conventional movement into the cylinder, the only modification being to replace the existing valve spring with a lighter one which enabled the valve to be opened on the induction stroke. The exhaust valve however required careful redesign, under standard operating conditions all of the valves within an ICE open into the cylinder. In this case the poppet valve was required to lift into the head as demonstrated below in Fig 21.



(a) (b)

Fig. 21 Proposed valve conversion, (a) conventional, (b) proposal.

This proposal to alter the operation of the exhaust valve required some work to be carried out on the valve and the head, and was considered to be the best route forward in creating the compressor.

5.2.2 Conversion of Exhaust Valve

Converting the exhaust valve to operate opposite to the conventional way required that the existing valve seat was removed and replaced. The new seat would have to be held securely in position, in such a way as to resist vibration and the pressure exerted by the valve seating itself. If the seat was to fall out during operation it would be disastrous for the valve and piston. After considering many options the shrink fit solution was decided upon.

The existing seats had been machined in situ and then induction hardened to give them the necessary properties to withstand the conditions under which they would be operating. Since these seats were integral to the head, the seat for the compressor exhaust would have to be machined out. This would prove to be a delicate operation to ensure that the water jacket would not be punctured. To aid in this procedure, copies of the original engine drawings were given to the project by Ford. This enabled assembly drawings to be accurately made.

The new seat was machined from high temperature 304 Stainless Steel. This material was chosen for its ability to withstand high operational temperatures and wear. It required heat treatment before being machined into the seat. The treatment involved heating the material to 1100°C for 8 hours, it was then allowed to air cool. The material was donated to the project by Avesta Sheffield. A Vickers hardness test was carried out on the material both before and after the treatment took place the hardness values being 235.5 Vickers before, and 168.8 Vickers after the treatment.

As was previously mentioned the method of installing the new seat into the head was to be by heat shrinking. The valve, now machined with a new profile to suit its new mode of operation, was placed into the exhaust port and it was pulled up hard against the base of the valve stem. Wearing the appropriate protective clothing, the new valve seat was immersed in liquid nitrogen and held there until the liquid ceased to bubble. At this point the valve was removed with tongs and placed into position, when released it slipped into the newly machined hole with relative ease. After an extremely short period of time the seat was tested and was found to be firmly held in position. The careful attention paid to the tolerancing of both the hole and the seat appeared to have been correct, the real test however was still to come.

After carrying out some initial test work on the compressor the head was removed for inspection and a slight modification to the newly inserted seat was necessary, as it had been pushed out of its seated position by the force of the valve acting upon it. This issue was resolved by machining tapped holes on either side of the seat and inserting cap head screws. A recess was machined for the heads of the screws, and their shoulders were positioned such that they held the seats in position.

With the seating problem resolved, the next step was to consider the placement of the valve spring, which would aid in returning the valve to its seat. The best position was on top of the head, around the valve stem in the traditional manner. In this design the function of the spring was to hold the valve down, rather than pull it up to its seat. To accomplish this either extension springs could be employed, or a compression spring which would push the valve. In either case a structure had to be made to support the spring. Another factor that was consided at this stage was the fact that the air above the valve, in the outlet ducting, would be at pressure. Therefore if the valve guide was not sealed large losses could be incurred up the stem as illustrated in Fig 22.

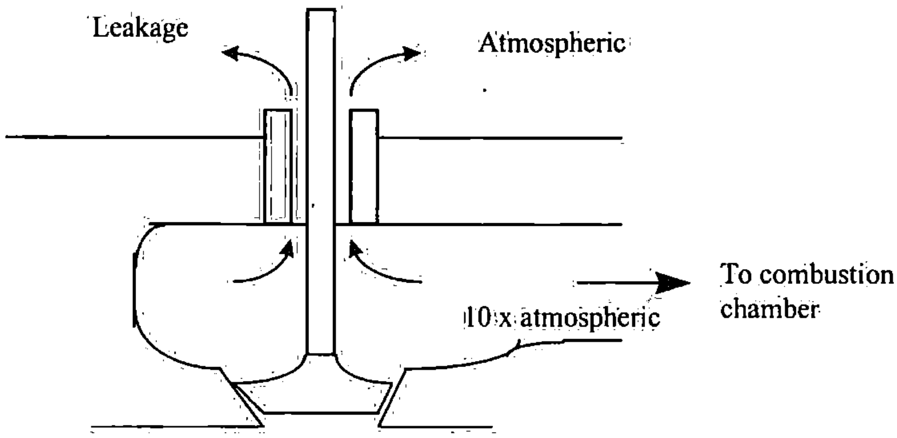


Fig. 22 Leakage

The solution to both the spring and the leakage problem was to attach a compression spring to the valve stem and place a cap over the top that would act as a seal, as illustrated below in Fig 23. A photograph of the system installed on the head of the engine cab be seen in Appendix 2.

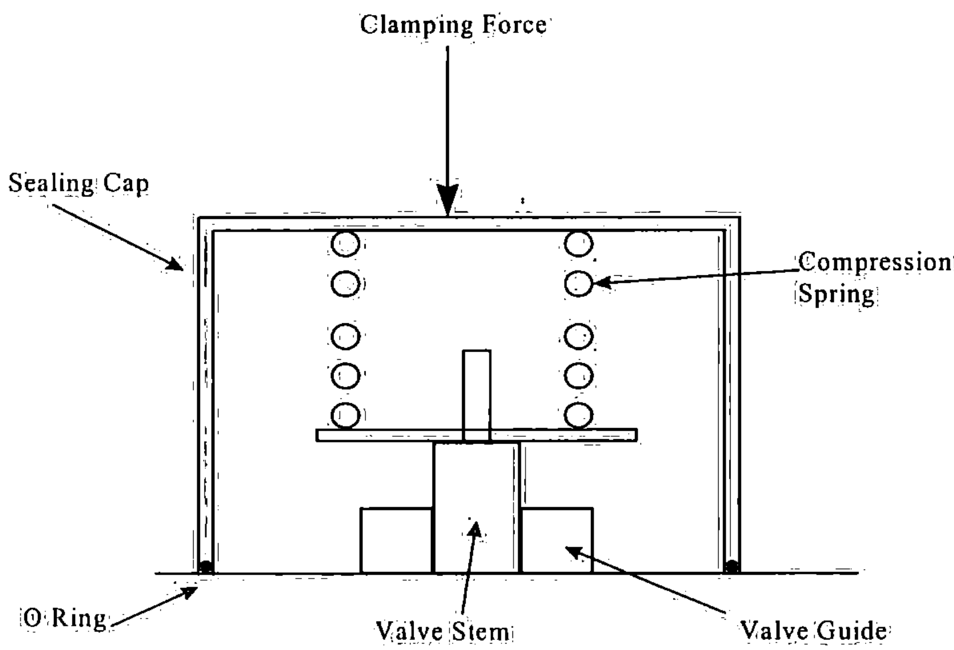


Fig 23. Exhaust valve solution.

Another opportunity for leakage to occur was through the old diesel injector ports, which ran directly into the cylinders. All of these were therefore blocked by placing a copper washer down the hole, this was then followed by a rod of suitable diameter. This was clamped into position using a rectangular bar that was bolted down into existing threaded holes in the head.

5.2.3 Compressor Spring Selection

As has already been stated, a light spring was used in conjunction with the inlet valve, which proved to be perfectly adequate for the purpose. The Exhaust valve spring did require some calculation, knowing the area of the valve and the pressure that would be applied, the force to hold the valve closed could be found. The space available for the spring was also known and from this information the spring rate was calculated, this is shown in detail in Appendix 3.

NOTE: This proposal was only partially successful. The principle worked perfectly well and the mechanical components were capable, but the amount of air induced into the cylinder using this method proved to be insufficient. So, the decision was made to fit a standard compressor disk valve onto cylinder No. 1. A valve was selected that fitted the diameter of the cylinder, and the head was cut in half to accommodate a new fixing structure for the new valve.

5.3 Expander

The expander of the prototype was to consist of two cylinders working alternately. This configuration would give two power strokes per revolution.

For successful operation of the expander the valving needed to be modified, as had been the case with the compressor. The inlet valves would be subject to high pressure on the back of them, but they would require precision timing to ensure that the most power was generated from the compressed combustion gases. To make the most of the pressure differences of the expander it was assumed that a slight modification of the seating arrangement of the compressor would work equally well in this application. The difference was that the inlet valves had their seating reversed so that they opened into the head, rather than the exhaust as was done in the compressor. This was done so that the pressure from the combustion process would aid in the closing and sealing of the inlet valves, which would still require mechanical operation for them to open. A photograph of the converted valve seats can be seen in Appendix 4 and a schematic demonstrates the configuration in Fig 24.

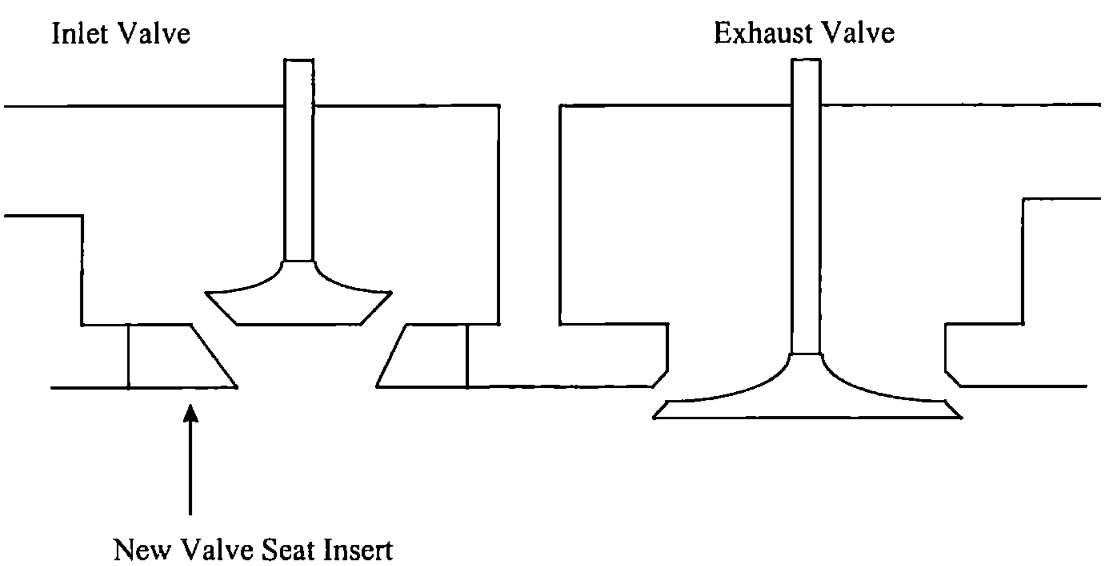


Fig. 24 Expander Valve Arrangement

As was mentioned with the compressor there were various leak paths that would need to be considered and minimised. In both functions these paths were the same, and the solutions included the seal between the new valve seats and the valves themselves. Leak tests were carried out and the amount of air leaking past the seat was found to be significant. It was greatly reduced after grinding the seats and valves with an appropriate paste.

The valves of the expander required mechanical operation, they could not rely on the pressure differences created within the system for their operation in the way that the compressor could. So, the valve gear for the expander also had to be redesigned.

5.3.1 Valve Gear

As the design of the valve gear was essentially starting from scratch for the expander it was decided that an overhead cam system would be utilised. Any thought of using the existing rocker system of the diesel was abandoned, it consisted of a cam shaft located within the crankcase, the lift being transferred to the valves by a push rod and rocker mechanism. By opting for an overhead cam (OHC) the system would be in a better position to facilitate the upward motion of the inlet valve into the head, and still maintain a conventional method of operating the exhaust valves.

As has already been discussed, the inlet valve to the expander was converted so that it would open into the head. This meant that the valve needed to be pulled upwards in order to allow gas to enter the cylinder. The exhaust valve would be operating as a conventional poppet valve, so it could be operated in the normal manner by an overhead cam. The system employed for the operation of the inlet valves involved the use of a 'stirrup' type mechanism, which consisted of a machined 'U' shaped piece of aluminum which was attached to the valve stem by means of the collets. The cam shaft ran through the centre of the 'U' and a cross piece was bolted on the top, this cross piece contained a roller which the cam profile ran against. This meant that when the cam for the inlet was vertically upwards the valve was opened. Appendix 3 illustrates the valve gear for the expander. Rollers were chosen as the running surface or follower, for both the inlet and the exhaust valves. This was achieved for the exhaust valves by machining a small block, that was attached to the valve stem by the use of the collets, which had the roller at its apex.

As the timing of the RJC was going to be unique a special cam shaft was made, rather than attempting to use an existing one. The profiles were calculated and the cams made to suit, the method of calculation will be explained later. Although the valve lift and duration of opening had been calculated, optimisation of the timing with relation to TDC was felt to be desirable to be altered as the experimentation program progressed. To achieve this a modular cam shaft was adopted. It consisted of a central shaft onto which the cam belt wheel bolted, spacers were then slotted onto the shaft between the main bearings and the cams themselves. Once all of the components were installed on the shaft the cams were aligned relevant to TDC and the whole shaft was then tightened into position, see Appendix 5. The cam shaft ran at the same speed as the crankshaft.

The whole of the exhaust valve timing system was designed around existing fixing holes in the head of the engine to minimise the amount of intrusive work into the head itself.

5.3.2 Valve Timing

A vital aspect of the valve gear is the cam profiles, which dictate the length of time that the valve will be open, and at which point during the cycle they should allow gas to enter or leave the cylinder. The calculation of the cam profiles for both the inlet and exhaust is in Appendix 6. As the computer model analysis of the engine was completed before the building of the engine began, it was a good tool in aiding with certain design aspects of the engine, such as the cam profiles. The model provided information about the engines performance under defined conditions, this was utilized to allow decisions to be made about the optimal pressure ratio that should be employed. This in turn provided information about volumes within the components during their cycles which was required for calculating the components like the cams for the expander. The springs used in conjunction with the new valve operating system also needed to be selected, this was done using the method shown in Appendix 3.

5.4 Combustion Chamber

The fuel used for the combustion chamber was compressed natural gas (CNG), the gas being injecting into an air stream so that good air/CNG mixing occurred, ignition would then take place, triggered by a spark plug. The system was based upon the same type of chamber that is used in gas turbines as the combustion process was continuous and this technology is proven.

The first prototype however more closely resembled a Bunsen burner type of arrangement, and this had a problem obtaining the correct fuel-air mixture when operational. As the CNG gas was introduced and the spark made ignition was almost explosive as the fuel was instantly burnt. It then took some time for the combustible air fuel ratio to build to the correct ratio again. This was obviously not the constant combustion solution desired and some redesign was done to produce a chamber that had a much closer resemblance to its counterpart in the gas turbine, and which produced the desired constant combustion.

A very basic chamber was constructed for laboratory testing away from the engine to assess the performance of the design, it is illustrated in Fig. 25. Air enters the main chamber normal to its main flow axis to induce turbulence through the main combustion tube. The majority of the air admitted to the main chamber flows around the outer edges of the inner tube, and then gradually enters through numerous holes along the side of the inner tube. This air is used to cool the combustion products on exit of the chamber. The remaining air enters the inner tube of the combustion chamber into the combustion zone through a split mixer cone. Air enters the cone through the slots, which are angled so that the air is forced to swirl and mix thoroughly with the injected fuel, which is injected into the cone at intervals along the leading inner edges of the cone. At the downstream end of the inner tube the outer jacket ceases to exist as all of the air has now been mixed into the combustion products and has assisted in cooling them. Ignition is achieved via a spark plug located close to the mouth of the cone.

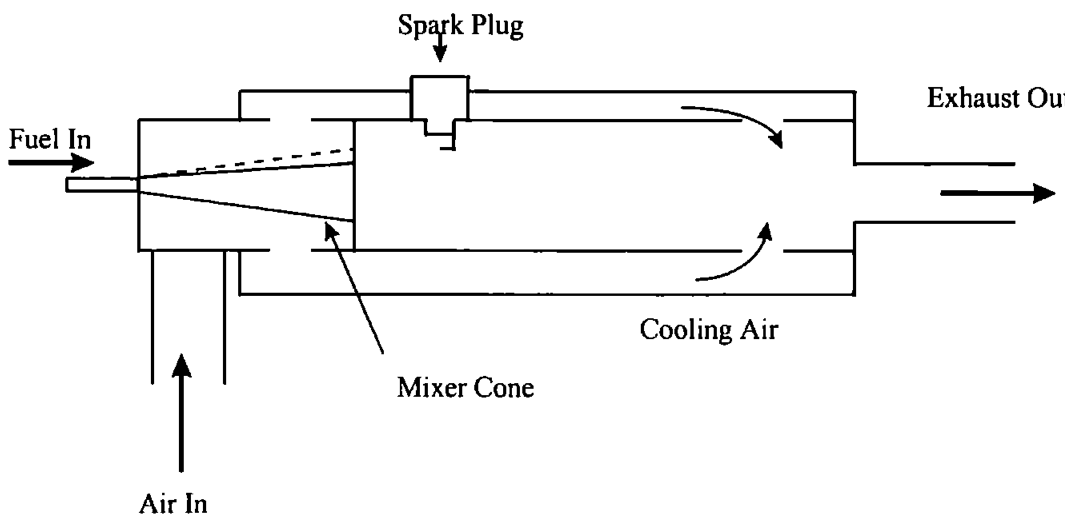


Fig. 25. Combustion Chamber

A very basic form of the above mentioned design was tested quite extensively at atmospheric pressure to ensure that the geometry of the cone would produce the desired flow characteristics. The most important criteria was the stability of the flame and how the flow of both air and fuel into the chamber would effect this. Secondary consideration was the temperature of the products of combustion. It was assumed that if the chamber performed well under atmospheric conditions it would operate in the same manner when at elevated pressure as long as the pressure ratio between the fuel and that of the air entering the chamber was maintained. The temperature range attainable with this type of design was gauged and found to be between 400- 860°C. Velocity and temperature measurements were taken which enabled the mass flow of the chamber to be calculated and compared to the data from the theoretical model, Appendix 7. Using this technique the geometry of the cone and baffle plate were modified until optimum flow was achieved through the chamber. As the testing progressed the method of ignition was modified until the best results were obtained and these were then incorporated into the final design. Simple calculations were also done to get a value for the air-fuel ratio in operation, Appendix 8.

The tests of this prototype proved to be successful. The chamber operated in the desired manner, the fuel mixed well with the incoming air and the required turbulence or swirl was created and could clearly be seen in the flame, as shown in the photograph in Appendix 9. At extremely low flows the flame did revert back to the interior of the cone, which was not desirable, but it was felt that the flows within the engine itself would be high enough to maintain the flame in its correct location. This design was finalized and the component manufactured. It was pressure tested to ensure there were no leaks from any of the welded joints, before being assembled to the engine.

For ease of ignition and also for the safety of the system a spark ignition timer circuit was built using a 555 timer chip. It was built so that the spark interval could be varied as required. After rigorous testing the circuit proved to have excellent performance and reliability. The circuit diagram can be seen in Appendix 10.

5.5 Further Work

After extensive laboratory testing the design of the combustion chamber had proven itself and a definitive chamber was manufactured. This was again tested independently before being assembled onto the engine. Manifolds were manufactured to link the compressor via the combustion chamber to the expander. An air storage tank was also built into the system directly after the compressor. This was done so that some pressure could be built up within the system, either from the compressor alone or from an external source directly connected to the tank, before the engine would commence operation. It also allowed some in situ testing of the individual components.

Once all of the components for the engine were ready some final preparation was needed to prepare the test bed. With all of the components assembled the fly-wheel of the engine was attached to a water dynomometer via a BMW drive shaft and specially made couplings. The engine was fitted with a starter motor, with temporary battery connections. The air storage tank had a connection to the external compressed air system installed so that it could be easily utilised whenever necessary. Finally the fuel line was installed. There was a critical installation that had to comply with safety standards to satisfy the health and safety aspect of the project. Installed in the line were a flowmeter, a pressure gauge and a shutoff valve that was in easy reach of the working area.

5.6 Evaluation

The final configuration of the engine to be tested consisted of a single cylinder compressor, with reed valves, a compressed air storage tank with mains air connection, a combustion chamber based on a gas turbine design, and a two cylinder expander with over head cam valve operation. Once constructed the prototype was by no means compact, but it looked robust enough to be able to cope with the testing work.

On the first run the air tank was filled from an external source, the compressor was disengaged so that the expander alone could be tested, and the engine was electrically started. The engine rotated under its own power until the compressed air supply in the tank ran out. Careful inspection of the engine after this first operation revealed that the valve operation system for the expander was not robust enough to cope with the motion and forces generated by the cams. The rods, which were part of the inlet valves lift system, were bent, the valve stems themselves bent and cracks appeared in the valve guides. The valves were straightened and the cracks in the guides filled, the valve gear was modified to make it more rigid and therefore more able to cope with the forces of operation.

During the initial test runs of the engine it became clear that it was necessary to fill the storage tank from the mains supply, and external air was required to get the engine started. The air tank gives an additional charge of air to the expander to assist in sustaining the engine's operation. It was concluded at this stage that the intake to the compressor was not large enough and not able to draw in sufficient air to compress and supply the expander. It must be borne in mind that the expander must achieve a high enough level of work to supply the compressor and produce an excess that drives the engine, if this does not happen the engine will not be able to sustain itself and will stop after a short period. This is what happened with the RJC if the additional mains supply was removed from the tank. At this stage the decision to split the head and fit a conventional compressor disk valve was taken.

Once the engine was running with its additional supply of air via the tank the ignition system was activated and the fuel admitted to the combustion chamber. It should be noted at this stage that the engine speed at no point exceeded 700 RPM. The chamber, after quite a short period of time, glowed red hot at the air input end, where the mixture cone is situated. This was an indication that the flame was not being held in the correct position, but instead it had reverted inside the cone. Although the system could operate in this manner it was not ideal, due to the excessive heating taking place at one area of the cone. Some post injection ignition could also be heard taking place towards the exit of the chamber, where a combustible mixture had become re-established further down the system and ignited. The conclusion drawn from this observation was that there was not sufficient air flow through the chamber to force the flame into its correct position which was at the mouth of the cone and extending into the main body of the chamber. This would be at least partially due to the speed of the engine. When designed, all calculations for the chamber were based on flow figures generated from an engine speed of 3000RPM, which was clearly not achievable with the current configuration. Under the circumstances it was felt that the chamber should only be operated for very short periods of time, this would prove to be a major limitation in the testing of the engine.

6.0 Methodology

6.1 Introduction

In order to assess the performance of the engine various sensors were positioned at different locations. Figure 26 illustrates the system and the location of these indicators.

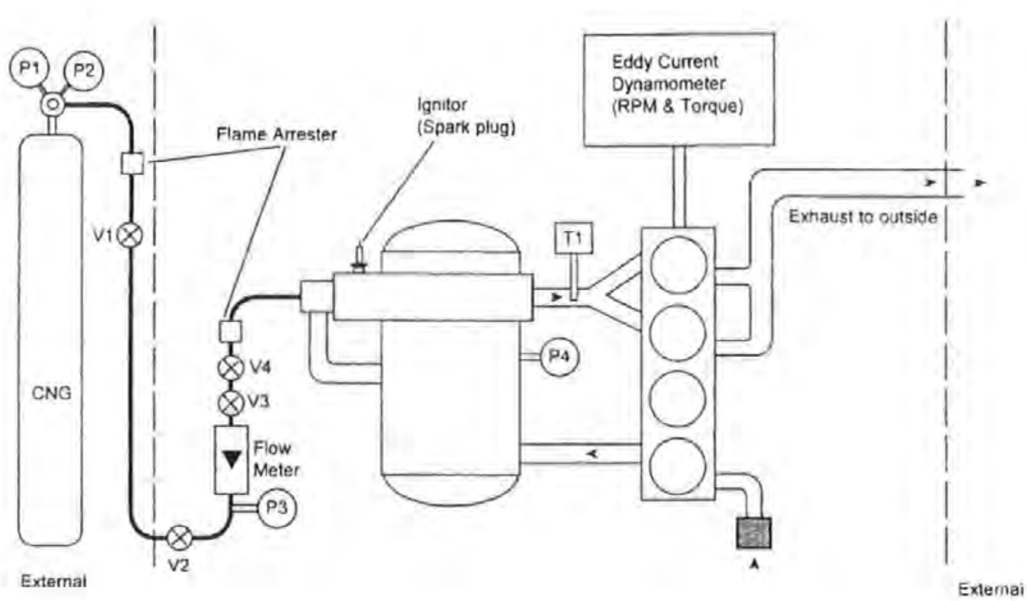


Fig. 26. RJC Test Rig and Sensor Location.

The key measured parameters being T1, the temperature of the inlet to the expander and P4, the operating pressure of the engine. Information of the engine speed and torque generated would be gathered from the Eddy Current Dynamometer.

The data gathered could then be input directly into the spreadsheet and compared to the ideal cycle. Any differences could then be analysed.

7.0 Results

7.1 Theoretical Results

As has been stated in Chapter 4, the theoretical model of the thermodynamic processes of the Joule cycle enabled the effects of different configurations to be gauged. By calculating the thermal efficiency for the engine as a basic cycle, then with the addition of intercooling and finally with both intercooling and regeneration it would enable the optimum configuration to be seen, with regard to thermal efficiency. Figure 27. illustrates the calculated thermal efficiencies with respect to the pressure ratio of the compressor.

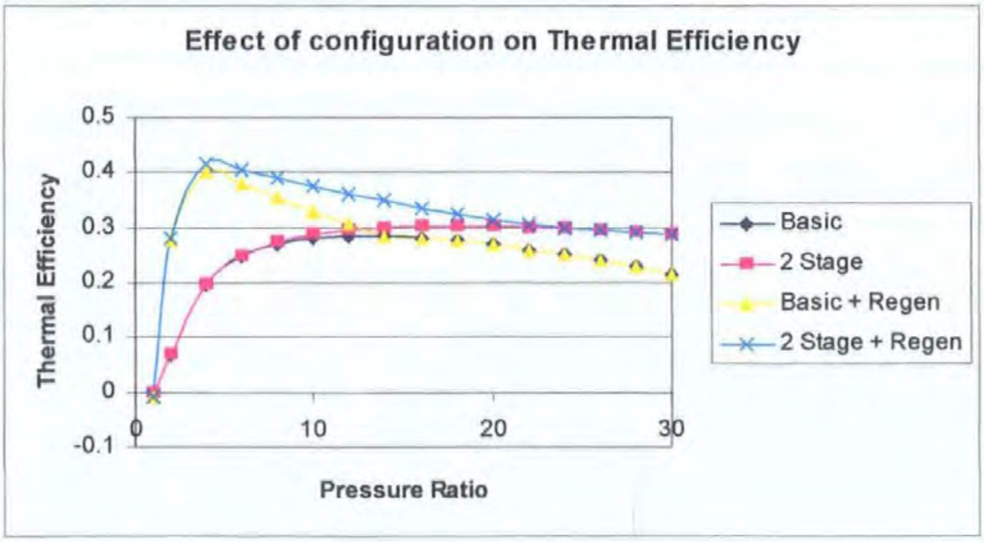


Fig. 27. Thermal efficiencies of RJC with different configurations.

Not only could the importance of configuration be gauged, but also key parameters such as peak temperature, T_6. The following Figs, 28 and 29, illustrate how altering the peak temperature effects the thermal efficiency and power output of the engine. Both of these graphs demonstrate the effect on a cycle that includes both intercooling and regeneration.

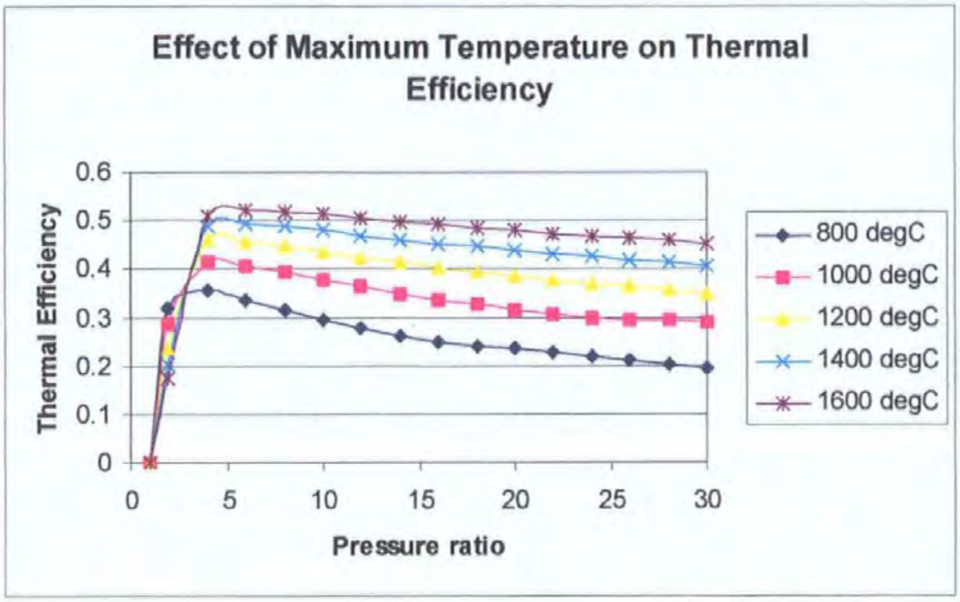


Fig. 28. Effect of Peak Temperature on Thermal Efficiency.

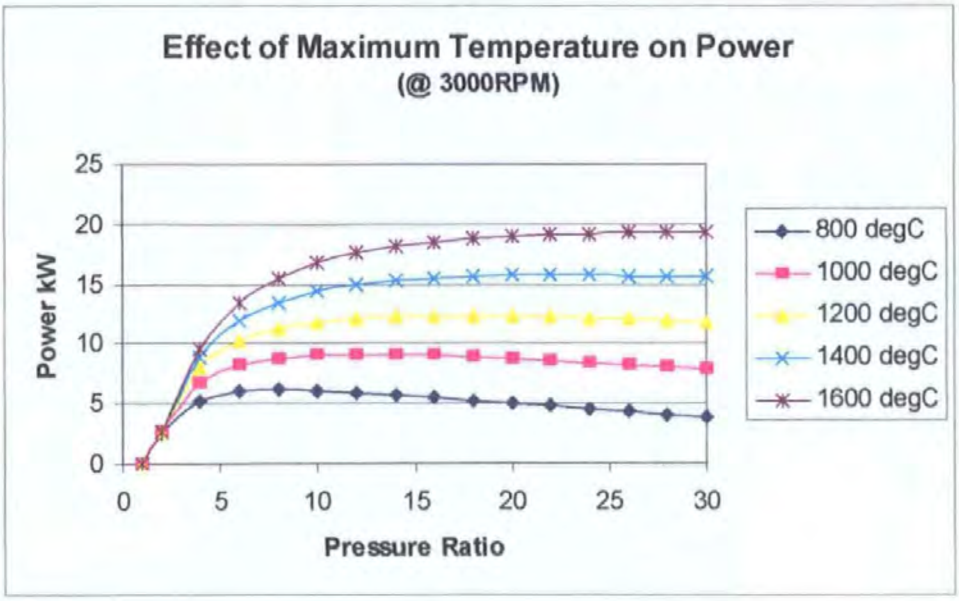


Fig. 29. Effect of Peak Temperature on Power.

Pressure losses in the system were also modelled, as described in Chapter 4. The resulting losses when compared to an ideal system were small. For example, consider a basic cycle with a pressure ratio of 8, the thermal efficiency with no losses is 27%, with losses it is 26.97%.

7.1.1 Comparison with Otto and Diesel

The thermal efficiency results from the model of the RJC are compared with those calculated for an Otto and Diesel cycle. In order to make a direct comparison between these engines a conversion needed to be made. The RJC operates with pressure ratio, both of the ICE's utilise volume compression ratio. The equation used was:

$$P_1V_1^n = P_2V_2^n \tag{54}$$

which gives:

$$rp = rv^n$$

The equations used for their thermal efficiencies are as follows:

Otto cycle

$$\eta_{th} = 1 - \frac{1}{rv^{n-1}} \tag{55}$$

Diesel cycle

$$\eta_{th} = 1 - \frac{1}{rv^{n-1}} \times \left[\frac{r_{ct}^{\,n} - 1}{n(r_{ct} - 1)}\right] \tag{56}$$

where r_{ct} = cutoff ratio

$$r_{ct} = \frac{V_3}{V_2}$$

On this basis the engines can be compared and the result is illustrated in Fig 30.

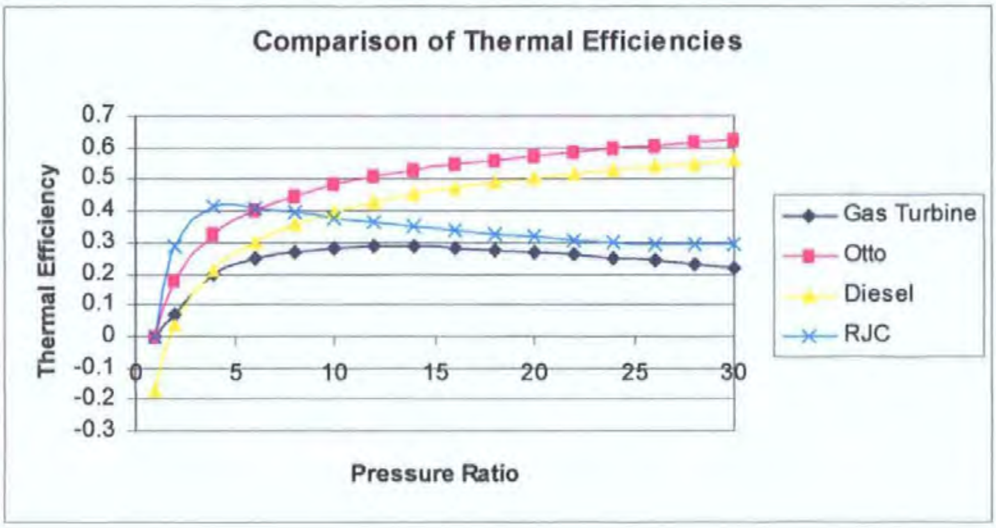


Fig. 30. Comparison of Thermal Efficiencies.

7.2 Practical Results

Starting the engine following the procedure documented in Appendix 1 proved to be successful.

The engine was operational for a short period of time before it became apparent that combustion within the chamber was not taking place in the manner expected. The chamber became cherry red from the heat generated in the region of the inlet, and occasional 'popping' sounds could be heard from nearer the downstream end. The popping sound indicated that combustion was not complete within the flame region. Unburned fuel was continuing to flow through the chamber along with the gaseous products of combustion. It is likely that at the locations where excess 'cooling' air was being introduced into the products of combustion the fuel/air mix was re-establishing itself and was then rapidly igniting, giving rise to the 'pop'.

During these short periods when the engine was operational it was able to reach speeds of up to 700 rpm. This information was put into the spreadsheet and the following effects were observed in Fig 31.

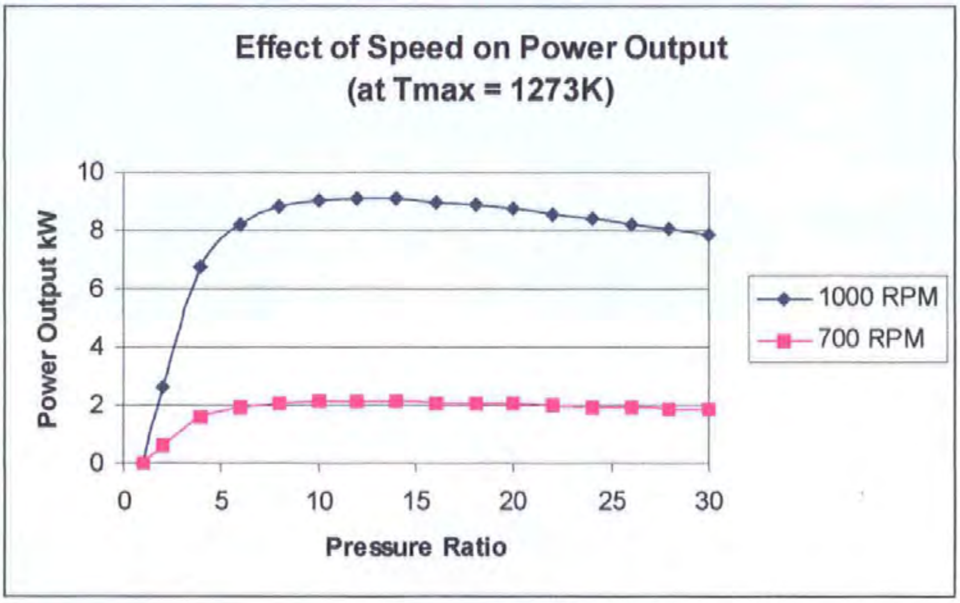


Fig. 31. Effect of Speed on Power Output.

The above graph demonstrates the effect on a two stage cycle with regeneration, the change in output will be greater for a basic cycle.

7.2.1 Combustion Chamber

As the engine was not operational for a significant period of time, no measurements of the temperature of the products of combustion could be made. When tested as a stand alone unit at atmospheric conditions the products of combustion were measured at temperatures up to 860°C (1133K).

The temperature of the products of combustion can be calculated theoretically and a comparison can be made to the actual temperature recorded. The theoretical value is obtained by calculating the adiabatic flame temperature at constant pressure as shown in Appendix 11. The calculation assumes complete combustion with no dissociation and that the only products of combustion are H_2O, CO_2 and N_2. The calculation in Appendix 11 demonstrated a stoichiometric air-fuel ratio, with the resultant adiabatic flame temperature, $T_{ad} = 2953.28K$.

A more accurate calculation of the adiabatic flame temperature that the chamber would theoretically achieve is illustrated in Appendix 12. By calculating the actual air-fuel ratio the process demonstrated in Appendix 11 can be reworked to give a more realistic temperature. With an air-fuel ratio of 70:1 the resultant temperature of the products of combustion would be considerably lower than that under stoichiometric conditions, as a large proportion of the air induced into the system will be introduced to the flame stream after combustion and will therefore have a cooling effect. This is seen to be the case, as the adiabatic flame temperature calculated in Appendix 12 is $T_{ad} = 1847.66K$, considerably lower than that under stoichiometric conditions.

There is however still a rather significant difference between the flame temperature calculated for the air-fuel ratio 70:1, and the temperature actually achieved, $\Delta T = 714.66K$.

8.0 DISCUSSION

8.1 Graphical data

Figure 26 demonstrates that the most efficient configuration for a Reciprocating Joule Cycle Engine will include multi-stage compression with intercooling and regeneration. This allows the cycle to maintain a relatively stable thermal efficiency from a low-pressure ratio of 5 up to about 30, in this range the efficiency drops from 40 to 30 %. This graph also demonstrates that the use of regeneration only provides real benefit at the lower pressure ratios, when a ratio of about 24 is achieved regeneration offers no benefit over a two stage cycle. This graph also clearly shows the influence of 2-stage compression. When it is included in the cycle it helps to maintain a more constant thermal efficiency after a pressure ratio of 10, for a basic 2 stage cycle the efficiency remains at about 30%.

The importance of the temperature of the products of combustion is demonstrated in Figs 27 & 28. For a pressure ratio of 10 an increase in the peak temperature from 1000 to 1600 °C increases the thermal efficiency from 38 to 51% (for a cycle with intercooling and regeneration). This trend is repeated when the power output of the cycle is considered with the same temperature rise, at 10:1 the power increases from 9 to 17kW for an increase of 600°C.

On first looking at the graph in Fig 29 it appeared that the Otto cycle proved to have the better thermal efficiency. One must consider however, that in practice the thermal efficiency of the gasoline engine is about 25% at operating compression ratios of about 10:1. The diesel in practice operates at 22:1 with an efficiency of about 35%. The graph shows that a conventional gas turbine is comparatively inefficient, however an RJC with intercooling and regenerative heat exchange offers reasonable efficiencies over the whole of its operating range.

8.2 Practical Data

As no numerical data could be obtained from the engine conclusions on its performance must be drawn from observations made. These observations may lead to some solid grounding in the best way to proceed with the research.

The observation of the maximum speed achieved by the engine was input to the spreadsheet and Fig. 30 in Chapter 7 illustrates the effect that this has on the power output of the engine. A comparison is made of the actual speed against the predicted speed for a peak cycle temperature of 1000°C. It is quite clear that this drop in speed has quite a detrimental effect on the performance of the engine. In reality of course, the peak cycle temperature while the engine was running would not have reached 1000°C. If the effect of reduced peak temperature on power output is briefly consider, at 700 rpm, for a basic cycle we can see from the graph in Fig.32 that even at a pressure ratio of 5:1 the power output would barely exceed 1kW.

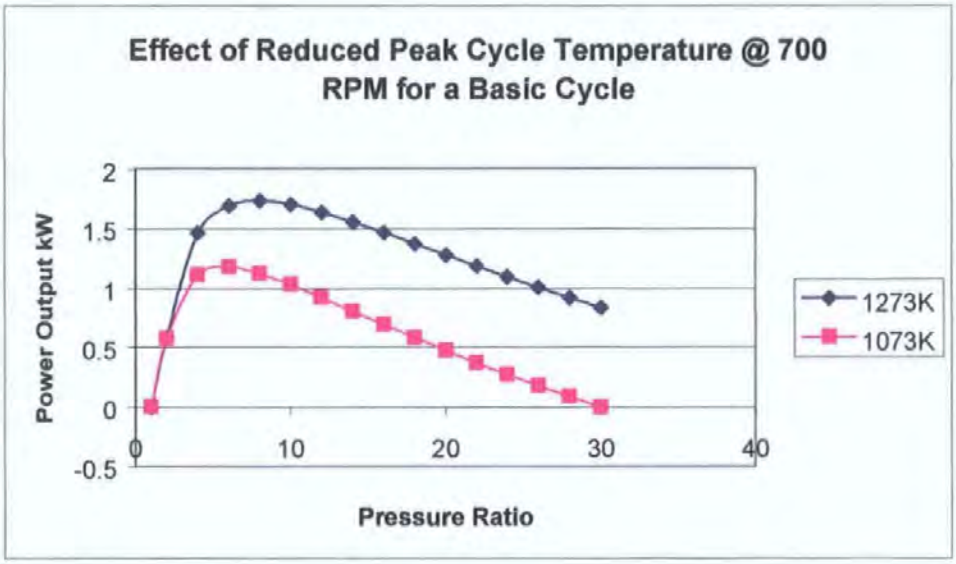


Fig. 32. Effect of Reduced Peak Cycle Temperature on Power Output for a Basic Cycle at 700 RPM.

When the engine first ran with an operational combustion chamber a reddening of the outer chamber at a particular location was considered indicative of the position of the flame. It readily became obvious that the flame was being held in the wrong part of the chamber. The chamber was designed to hold the flame from the face of the mixing cone, in reality the flame was being held within the mixing cone. Although the flame appeared to be relatively stable within this location the popping sounds that had been observed indicated that combustion was incomplete, and therefore inefficient. This inefficient combustion meant that the peak combustion temperature anticipated was not reached, which effected the maximum amount of work that the expander was capable of achieving. In practical terms this meant that the engine was not able to sustain its own operation, it required more energy to drive it than the engine could actually produce.

By the addition of compressed air, from an external line, into the storage tank the period of operation of the engine could be extended. Success with this method was limited due to continuing problems with the combustion process. In practice the run time of the engine was not exploited because it was felt that operating the combustion chamber in this manner was not conducive to the safety of the working environment. For this reason the engine was never run for a consistent period of time over which meaningful, quantitive results could be obtained.

The operation of the combustion chamber as an individual component can be assessed from the experimental work done with it under atmospheric conditions. The analysis done in Chapter 7 to find the Adiabatic Flame Temperature under both stoichiometric and more importantly, lean combustion conditions demonstrated some significant temperature differences. In practice the temperature of the products of combustion was approximately 440°C lower than the calculated value. This discrepancy is due to incomplete combustion, dissociation taking place and heat loss through radiation. If the operation of the chamber when installed into the engine is now considered. The drop in temperature actually achieved would be even greater as the flame was not even being held in the correct part of the chamber, and the possibility of incomplete combustion and the other losses was increased.

8.3 Comparison with other RJC Engines

When the analysis for this Reciprocating Joule Cycle Engine is considered against previous models it can be seen that with similar assumptions the thermal efficiency values can be achieved.

One of Dechers' models for the Britalus Engine quotes a thermal efficiency of 40% when the peak cycle temperature is considered as 2000K, and component efficiencies are 90%. Using the same assumptions for a basic cycle the RJC model predicts a similar value of thermal efficiency. The Valved Hot Gas Engine predicts 50% thermal efficiency with, again, 90% component efficiency assumed.

As with the RJC, whenever any of the previous models were put into practice in hardware, the results were vastly different to those theoretically predicted. All of the prototype engines suffered from technical difficulties, which resulted in a loss to the overall achievable thermal efficiency. Of those engines that published their performance results the Axial Piston Rotary Engine suffered from large losses from its combustion chamber, which reduced its thermal efficiency to 16%. The Valved Hot Gas Engine, a closed system, could only attain 12% thermal efficiency. These initial test runs, however dissapointing, proved to be an asset as the main problem areas for the engines were pinpointed. The same can be said of the RJC experience of initial run trials. The main areas of weakness highlighted from the exercise can now be corrected.

One apparent success story of this type of engine unfortunately offers little in the way of details. The configuration of this 'air' engine [24] is radically different to any of the other prototypes tested. If the reports about this particular engine are true, its success may be founded in the novel approach to this cycle. Without further details it is difficult to speculate.

8.4 Comparison with Today's Automotive Technology

Although the RJC holds the promise of thermal efficiencies better than the Spark Ignition and comparable to the Compression Ignition engines it has, as yet, been unable to deliver. When the growing options of todays' market place are considered it is obvious that the RJC will need to impress to gain a foothold in the market.

There is no doubt that conventional SI and CI engines can be improved upon to increase thermal efficiency, improve fuel economy and reduce emissions. Alternative engines are now emerging in the market place. Cars powered by hybrid engines and fuel cells are now commercially available, and their market share is set to grow. Although it is felt that the RJC could cope with the prospect of multi-fuelling it still has a long way to go in proving its efficiency, power and emissions capabilities. In development terms the RJC is decades behind the new engine types that are beginning to emerge. Although it offers potential advantages with multi-fuelling, an innate low emission characteristic, relatively high thermal efficiency and the possibility of including regenerative braking, extensive development is still required. This may mean that the RJC has missed on opportunity in the automotive field.

9.0 Conclusion

It is clear form the work previously described that the prototype engine is not operating in the manner in which it was hoped, in fact, it is currently unable to sustain itself for any reasonable length of time. This is in part due to the combustion chamber not operating under its optimum conditions and it is therefore not producing the required performance to sustain the engine. From the observations of the engine under running conditions it can be deduced that the velocity of air through the chamber is not sufficient to produce the necessary flame speed. This is evident from the colour change of the chamber at the air inlet end. As a consequence of this, incomplete combustion occurs. To remedy this situation the amount of air induced into the compressor could be increased, or a chamber that performs with lower flow rates could be designed and incorporated.

With a suitable combustion chamber capable of producing the necessary peak cycle temperatures the prototype has the potential for sustained running. It will however be equally vulnerable to the same losses experienced by its predecessors i.e. heat loss from the combustion chamber, component inefficiencies etc. It will also suffer from the additional problem of long connecting pipe runs. Previous engines have maintained a compact design and although pressure drop calculations for the RJC engine show there is little impact, it is felt that in reality the losses may prove significant.

For this reason, and the need for a relatively compact design in the automotive field, The RJC engine would need to be re-configured before considering presenting it for commercial use.

When the RJC engine is considered alongside its rivals in the automotive field it becomes apparent that it is never likely to enter service in this role. Too many resources have been dedicated to technologies such as the hybrid engine and fuel cells for any of the major manufacturers to divert their efforts towards this type of engine. Although it maintains the traditional piston/cylinder arrangement the RJC engine also incorporates some unfamiliar technology, namely the combustion chamber. Although the combustion chamber can ideally be based upon the 'cans' of gas turbine engines, the application of that principle would require adjustment, as this project can testify. Due to this aspect of the engine it would require substantial development to bring it to a level suitable for the market. For this reason, in combination with the fact that the final performance targets are constantly changing, it is felt that the RJC engine would be unsuitable for an automotive application. It may prove to have better potential within static, industrial power supply application.

10.0 Further Work

10.1 Current Prototype

For work to continue, and for a deeper understanding to be gained, the current experimental Joule Cycle engine must be developed still further to improve its stability and durability. When it has been developed into a more reliable platform further tests can be run and the amount of viable data obtained from the engine can be increased. This may lead to the improving the output of the engine by the inclusion of heat exchangers in the form of intercoolers or reheaters, whichever proves to be of greatest benefit.

Before that step is taken however the configuration of the engine could be optimized. As the current prototype was created from a donor engine there are many facets to the overall design that could be changed which may result in substantial benefits to the operation and performance of the engine. One obvious target for change would be the overall bulk and inherent mass of the engine. This opens the door to the opportunity of changing the configuration of the straight 'in line' as is current, to something that may prove to be beneficial in more ways than one. For example, if a horizontally opposed configuration was utilized not only would the weight and bulk issue be addressed, but with careful design of bearings the friction of the compressor and expander might be reduced.

10.1.1 Combustion Chamber

Another approach to improve the thermal efficiency obtained from the prototype would be to further develop the combustion system. The chambers that have been built and fitted to the engine have been crude in the extreme, but this was necessary to gain the relevant experience and understanding of the way in which this, the most complex component of the entire engine, would operate under these conditions.

Developing a combustion chamber that can operate over a wide range of flow conditions would be most beneficial, as the initial starting conditions tend to generate a low flow regime. However, as the engine builds momentum the flow increases and the chamber needs to be able to cope with the increasing turbulence being generated, and still be able to hold a steady flame in the correct region of the chamber. This may necessitate the need to develop a monitoring system for the chamber which monitors the combustion conditions and ensures that the correct amount of fuel is delivered into the system.

Another development of the chamber which could be considered is that of catalytic combustion. The combustion chamber that is currently mounted to the engine has been designed to allow the insertion of a catalyst. The burner and ignition systems would still be necessary as the catalyst needs to be brought up to temperature before it can initiate combustion within itself. Once at the correct temperature the ignition system can be extinguished and the naked flame within the chamber will also go out, leaving the combustion process to take place solely within the catalyst. This should make the chamber a safer system, and will also lead to even lower emissions from the engine.

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Appendix 1

University of Plymouth

Faculty of Technology

Department of Mechanical and Marine Engineering

Operation of a reciprocating Joule Cycle Experimental Engine

Safe System of Work including a Risk Analysis

M A Bell

BEng(Hons), MSc, PhD, FIMechE, Ceng

1. Description of the Reciprocating Joule Cycle (RJC) Engine

The RJC engine is essentially a reciprocating version of a gas turbine. Instead of a rotating bladed compressor and turbine, a reciprocating compressor and expander is used. The combustion chamber is essentially the same as for a gas turbine. The compressor is driven by the expander, with the excess energy available as motive power at the drive shaft.

The potential advantage of such an engine is extremely clean combustion combined with high thermal efficiency.

The diagram below, Fig 32, shows the essential features of the experimental engine.

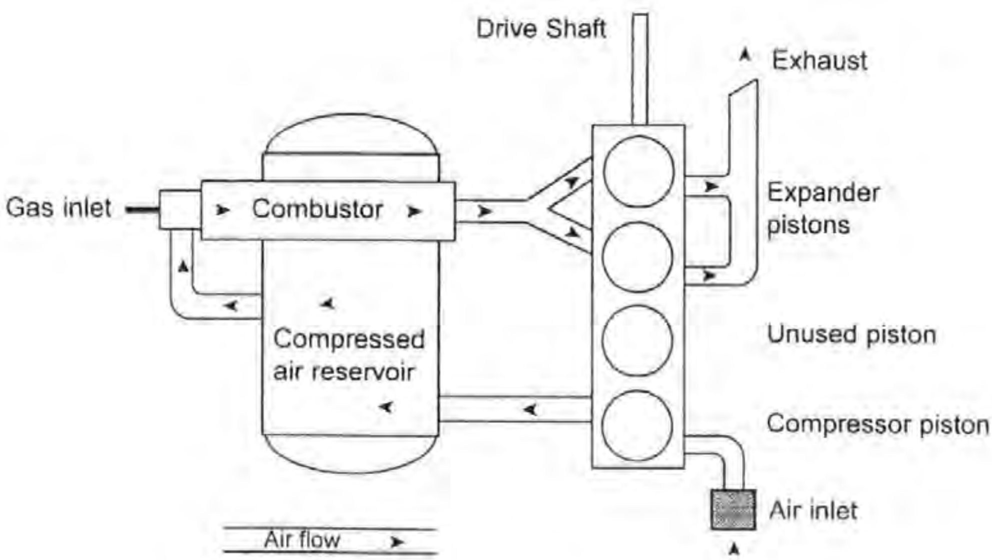


Fig. 33. Four cylinder Diesel engine converted to RJC Engine – main features

2. Operation of the RJC Engine

The engine operates in a very similar manner to a gas turbine.

Starting would normally be by motoring the engine using a starter motor. The compressor and expander rotate together setting up an airflow in the combustion chamber. An ignitor is turned ON and fuel is supplied. Fuel ignition should be virtually instantaneous. If ignition fails to occur within a few seconds the ignitor is turned OFF and a purge cycle commenced to eradicate excess fuel, and the starting sequence is repeated. If ignition repeatedly fails to occur an investigation id required.

For the purposes of experiment starting will be carried out differently from above. The engine will be motored using compressed air rather than using a starter motor. The compressor will be 'unloaded' by means of holding the inlet valve open using an unload lever.

Once ignition has occurred and the engine accelerates to a steady running speed, the compressor will be brought into operation and the compressed air supply disconnected. In this way the initial operation of the engine can be ascertained on a step-by-step basis rather than relying on all components operating together. Once all components of the engine have been ascertained as operating correctly the above starting procedure will be attempted and if it proves satisfactory will be adopted as the normal means of starting the engine.

Engine power output is primarily controlled by fuel flow which in turn determines the expander inlet air temperature. Engine speed is determined by the fuel flow and dynamometer setting.

2.1 Engine Operating Pressure

Because this experimental engine is designed with fixed expander inlet valve timing the operating pressure of the engine is intimately tied to its operating temperature. At a given engine speed, as the expander air inlet temperature increases (by increasing the fuel flow) the density of the inlet gases tends to decrease and the expander therefor needs less air mass flow. The compressor continues to supply air at the same mass flow rate, so the pressure will tend to rise (increasing air density) until a balance is achieved between the air mass flow supplied by the compressor and that needed by the expander. Engine operating pressure is thus self limiting and controlled by the expander inlet temperature.

2.2 Engine Speed

If the engine speed increases the air mass flow through the engine as a whole will increase. If the fuel flow remains the same the expander inlet temperature will tend to fall, and the engine will tend to produce less power, thus reducing engine speed. Engine speed is therefore stable and self limiting controlled by fuel flow and dynamometer setting.

2.3 Engine Operating Temperature

The highest engine operating temperature (apart from localised flame temperatures in the combustion chamber) is the expander air inlet temperature. This temperature is controlled by the air-fuel ratio (AFR) of the engine. The delivery end of the combustion chamber, the delivery manifold, the expander inlet valves, and to an extent the expander piston face and upper cylinder are subject to this temperature. All other parts of the engine operate at comparatively low temperatures.

There is the need to monitor this temperature to prevent thermal damage to the engine, and to guard against failure of the combustion chamber and expander inlet manifold from the effects of pressure and excessive temperature.

As indicated above, pressure is intimately tied to temperature. The control of this engine is therefore essentially achieved by controlling the expander inlet temperature. Provision has been made to monitor this temperature and accurately regulate and measure fuel flow.

3. Engine Test Rig

A diagram of the test rig is shown below in Fig 33 with all of the controls and instruments essential to safety shown. Other measurements may be taken for experimental purposes but these are not necessarily indicated.

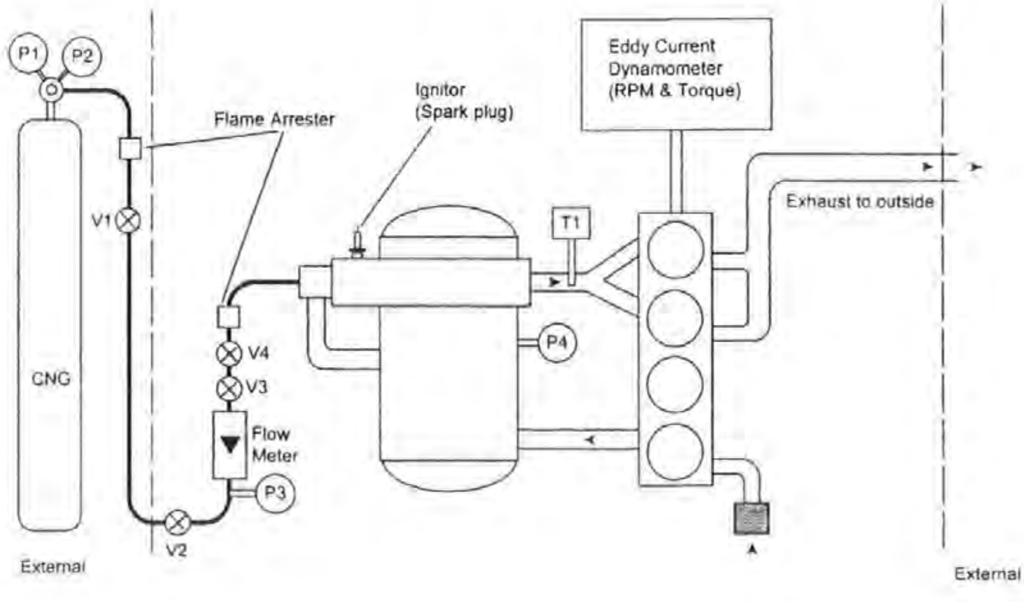


Fig. 34. RJC Engine – Test Rig

Key:

P1 = CNG bottle pressure

P2 = CNG regulated supply pressure

P3 = Line supply pressure

P4 = Engine operating pressure

T1 = Expander Inlet Temperature thermocouple

V1 = Shut-off valve (lever operated)

V2 = Shut-off valve (lever operated)

V3 = Regulating valve

V4 = Shut-off valve (lever operated)

The key control is valve V3, and the key monitored parameter is T1.

4. Risk analysis

The main safety issue is that of preventing uncontrolled fire or explosion with the use of natural gas as the fuel.

(a) Fire

Fire could occur in the event of leakage of gas from the supply pipe to the surroundings or because of a ruptured gas supply pipe. The means of arrest of such a fire are the lever operated shut-off valves V1, V2 and V4. Any of these valves can be closed in the event of fire. If valves V2 and V4 are not accessible because of a fire, valve V1 remains accessible at all times. Mr. Tim Partridge (Second Investigator) is trained in fire fighting. Fire extinguishers will be ready to hand in the event of their being needed.

(b) Explosion

Explosion could occur if a combustible mixture within the system ignited either intentionally or unintentionally. The main risk is that a combustible mixture could accumulate in the compressed air reservoir.

The risk is made negligible by the operating procedure. Both pre and post operation, the compressed air cylinder is thoroughly purged with air. In normal operation the air flow prevents any possibility of unburned fuel accumulating within the compressed air cylinder.

If ignition is delayed, there is the possibility of 'explosion' within the combustion chamber and inlet manifold. This, however, could be regarded as a normal part of the start up process. If a stoichiometric mixture ignited in the combustion chamber and inlet manifold it could be expected that the resulting products could reach maximum adiabatic flame temperature estimated to be around 2300°C. Assuming that this gas could freely expand back into the compressed air reservoir, and because the volume ratio of the compressed air reservoir to that of the combustion chamber is ~6:1, the maximum pressure ratio that can be expected is of the order of 2.1. i.e. if the engine was operating at a typical start up pressure of 4 bar gauge, the maximum expected pressure under these circumstances would be 9.5 bar gauge. i.e. within the system's safe operating limit (150 psi gauge or 10.3 bar gauge). In practice it would be unlikely that the whole volume of the combustion chamber would be filled with a stoichiometric mixture.

(c) Assurance of ignition/combustion

In common with the gas turbine generally, the combustion chamber does not have visual means of ascertaining ignition or continued combustion. Instead, the thermocouple (T1), continuously indicates whether hot gas, produced by combustion, is present. On ignition temperature T1 rises very rapidly. If it fails to do so within a short time fuel is cut off and a complete purge and start up procedure is repeated. If combustion ceases, temperature

T1 drops very rapidly, and under these circumstances the fuel supply would immediately be cut off.

Continuous careful monitoring of temperature T1 is essential for safe operation of the system.

(d) Other risks

Mechanical failure due to overspeed is possible but unlikely to be hazardous. As mentioned above, the engine speed is self limiting, and will be less than 3000RPM. The engine is capable of speeds in excess of twice this figure without hazard. Any mechanical failure is likely to bring the engine to a stop, hence minimizing the hazard.

Noise is no worse than the operation of other IC engines in the laboratory. Ear defenders are available.

Toxic gases (exhaust products). Unlike a typical petrol engine CO emissions are expected to be zero or negligible. However, in any event, exhaust is directed to the external environment.

5. Safety & Emergency Procedures

Basic safety and emergency procedures have been devised and are attached to this schedule as Appendix A. They reflect the description and analysis given above.

Also attached is a completed Occupational Health and Safety Centre ***Risk Assessment*** form.

6. Responsible Person

The responsibility for all matters of health and safety within the Department rests with the Head of Department, Dr John Chudley. For the purposes of these experimental procedures it is suggested that the responsible person in the first instance should be the Chief Investigator, Dr Murray Bell.

7. Further Enquiries

Any further enquiries should be directed to the author of this document, Dr Murray Bell, tel. 01752 232656.

M A Bell
16/2/99

Appendix A

University of Plymouth

Faculty of Technology

Department of Mechanical and Marine Engineering

OPERATING PROCEDURE FOR THE RECIPROCATING JOULE CYCLE ENGINE

Location: BRUNEL W8

SAFETY PRECAUTIONS

The engine is situated in a separate bay in the Energy laboratory in Brunel W8. This enables that area to be isolated from the remainder of the laboratory via a metal roller shutter door and allows the engine to be operated without disturbance from, or to, the main laboratory area.

The fuel for the engine is compressed natural gas (CNG), and this has been installed such that the main supply bottle is outside of the building in accordance with CORGI regulations. The CNG supply pie is of galvanized steel with screw fittings designed to supply the gas with pressure up to 10 bar gauge. The gas supply system embodies two regulators to control pressure, a manual shut-off valve in the working area next to the engine, a pressure relief valve, a non-return valve and a shut-off valve outside of the working area.

The compressed air reservoir is pressure tested annually. The system will be operated to a maximum pressure of 8 bar gauge. Its maximum design operating pressure is 10 bar gauge (150 psi). A copy of the test certificate can be obtained from Mr Brian Lord, Chief Technician.

Only authorised personnel will be permitted to operate the engine. These persons are Dr M A Bell, Mrs H Kirby-Chambers, Mr T Partridge, Mr R Cox (Technician). Warning signs situated in the area will inform other visitors and personnel of the caution required and the possible hazard of operating the engine without authority.

The combustion chamber has been already tested at atmospheric pressure and at a pressure of 3 bar absolute. Ignition and steady combustion at these conditions has been confirmed. The combustor has been designed to operate at a maximum pressure of 10 bar gauge and proof tested (cold) to twice this pressure.

STARTING and OPERATING PROCEDURE

1. Ensure all gas supply valves are OFF.

2. Check all electrical connections, i.e. thermocouples, starter motor, ignition and ensure correct operation.

3. Purge the system for a minimum of 2 minutes by motoring the engine using the starter motor.

4. Stop the purge and turn the engine manually until both expander inlet valves are closed.

5. Pressurize the compressed air reservoir to a maximum of 80 psi gauge.

6. Unload the compressor using the decompresion lever.

7. Ensure the CNG is set to the correct supply pressure and flow rate.

8. Start the engine using the starter motor and allow it to rotate steadily using the compressed air supply.

9. Switch on ignition.

10. Turn on the CNG supply.

11. Monitor the expander inlet temperature indicator. If the temperature does not begin to rise rapidly within 5 seconds shut off the gas supply and recommence the start up procedure form '3' above. If ignition does not occur after three attempt investigate the cause.

12. Continue to monitor the expander inlet temperature and regulate it to endure a steady engine speed. Do not allow the temperature to exceed 700°C.

13. Bring the compressor into operation by deactivating the decompression lever.

14. Continue to monitor the temperature and speed until steady operating conditions are achieved.

15. Remove the compressed air supply while continuing to monitor the engine.

16. Regulate the engine speed and load by means of the dynomometer setting and CNG supply.

17. On completion of the engine run, shut of the CNG supply using the work area shut off valve.

18. Unload the compressor.

19. Supply compressed air and motor the engine for a further 2 minutes to purge all gases from the engine.

20. Remove the compressed air supply and allow the engine to come to a stop.

21. Bleed any residual compressed air from the compressed air reservoir and switch off electrical supplies.

IN CASE OF

EMERGENCY

SWITCH OFF THE

GAS SUPPLY AT

THE MANUAL

SHUT OFF VALVE

OCCUPATIONAL HEALTH & SAFETY CENTRE

RISK ASSESSMENT

~~Division~~/Faculty/~~Centre~~ TECHNOLOGY	Record No
Department DUME	Site of Activity BRUNEL W8
Name HEATHER KIRBY-CHAMBERS	Date 21/9/98
Signature H Kirby-Chambers	Review Date

Activity — Define work eg experiment, practical or project — PROJECT / EXPERIMENTAL

Administration, general clerical, use of Display Screen Equipment, photocopying and typing.

TESTING PROTOTYPE ENGINE

WHEN CARRYING OUT AN ASSESSMENT PLEASE REFER TO THE HSE GUIDANCE (5 STEPS TO RISK ASSESSMENT)

HAZARD

Look only for hazards which you could reasonably expect to result in significant harm under the conditions in your workplace. Use the following examples as a guide:-

* Slipping/tripping hazards (eg poorly maintained floors or stairs)
* Chemicals (eg Battery Acid)
* Fire (eg from flammable materials) ✓
* Pressure systems (eg steam boilers) ✓
* Electricity (eg poor wiring)
* Fume (eg welding)
* Noise
* Work at height (eg from mezzanine floors)
* Moving parts of machinery (eg blades)
* Ejection of material (eg from plastic moulding)
* Vehicles (eg fork-left trucks)
* Dust (eg from grinding)
* Manual Handling
* Poor lighting
* Low temperature

LIST HAZARDS HERE:

* Computer workstation.
* Faulty or wet electrical equipment.
* Walkway obstructions (trailing leads, stools, bags).
* Injury due to lifting heavy objects.
* Wet or slippery floors.
* Fire.
* Risk of burns from heat sources.
* Incorrect use of equipment.
* Overcrowding.

* POSSIBLE GAS LEAK.
* UNCONTROLLED DETONATION.

If other hazards are identified please refer to specific Risk Assessments eg Manual Handling/DSE Users.

WHO MIGHT BE HARMED?

There is no need to list individuals by name. Think about groups of people doing similar work or who may be affected, eg

- Office staff
- Maintenance personnel
- Contractors
- People sharing your workplace
- Operators
- Cleaners
- Members of the public
- Students
- Staff with disabilities
- Visitors
- Inexperienced staff
- Lone workers

PAY PARTICULAR ATTENTION TO:- THEY MAY BE MORE VULNERABLE

List groups of people who are especially at risk from the significant hazards which you have identified:

- University staff
- Inexperienced staff
- Students
- Visitors

IS THE RISK ADEQUATELY CONTROLLED?

Have you already taken precautions against the risks from the hazards you listed? For example, have you provided:-

- Adequate information, instruction or training?
- Adequate systems or procedures?

Do the precautions:-

- Meet the standards set by a legal requirement?
- Comply with a recognised industry standard?
- Represent good practice?
- Reduce risk as far as reasonably practicable?

If so, then the risks are adequately controlled, but you need to indicate the precautions you have in place. You may refer to procedures, manuals, company rules etc giving this information.

List existing controls here or note where the information may be found:

All reasonable precautions against the risks from the hazards listed have been undertaken.
Refer to the Office Regulations for more details

WHAT FURTHER ACTION IS NECESSARY TO CONTROL THE RISK?

What more could you reasonably do for those risks which you found were not adequately controlled? You will need to give priority to those risks which affect large numbers of people and/or could result in serious harm. Apply the principles below when taking further action, if possible in the following order:-

- Remove the risk completely
- Try a less risky option
- Prevent access to the hazard (eg by guarding)
- Organise work to reduce exposure to the hazard
- Issue personal protective equipment
- Provide welfare facilities (eg washing facilities for removal of contamination and first aid)
- Further training

List the risks which are not adequately controlled and the action you will take where it is reasonably practicable to do more. You are entitled to take cost into account, unless the risk is high:

Appendix 2

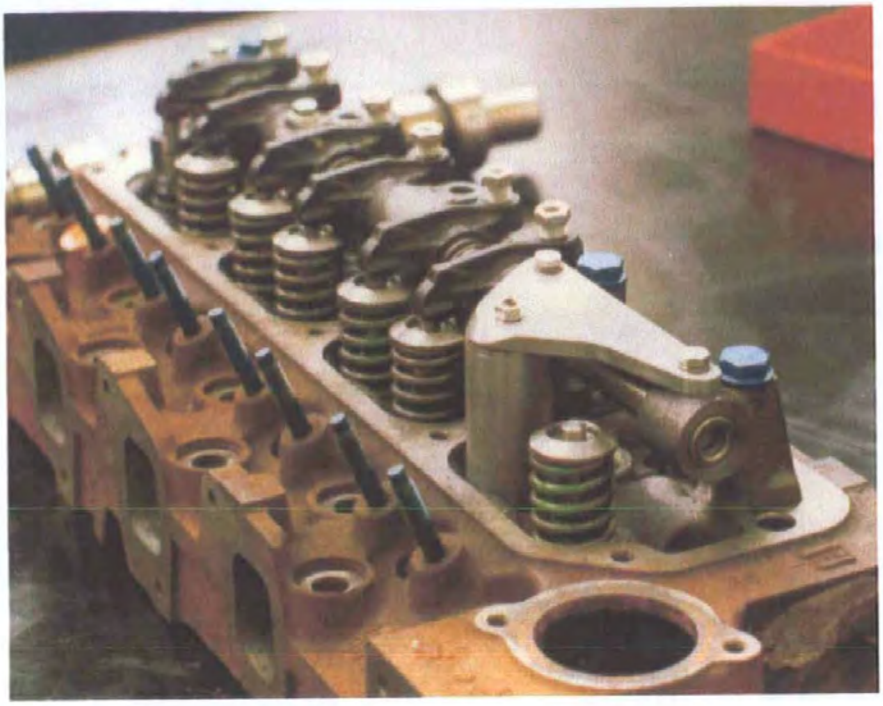


Plate 1. Compressor Valve Modification.

Appendix 3

Spring Selection

Another important aspect of the valve gear was the choice of springs to be used on the inlet and exhaust. The first step was to obtain an initial estimation of the forces required to close the valves during operation.

The springs active on a conventional valve system are quite substantial in order to withstand the large forces that are subjected on them by the action of the cams. The requirements for the RJC valve gear would prove to be no exception. The only limiting factor on the spring to be used was the physical space available. In order to select springs suitable for the task the forces acting on the valves had to be ascertained. The main function of the springs was to ensure that the roller remained in contact with the cam at all times, the wrong type could lead to the roller losing this contact thus causing the valve to close later than desired.

Calculations were carried out considering the forces during the 30° period of cam closing.

The closing action of the valve was considered to be linear as illustrated below in Fig 34.

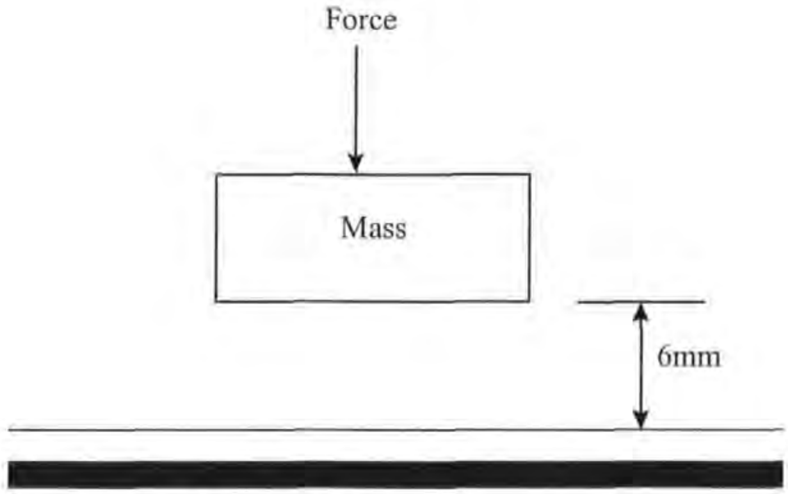


Fig. 35. Block Illustration of Forces on Valve.

Assume speed = 2000RPM

Time = 30°

$$\text{Time} = \frac{\text{Distance}}{\text{Speed}} \tag{57}$$

$$\text{Time} = \frac{30/360 \times 2\pi}{(2000 \times 2\pi)/60}$$

$$t = 0.0025\text{s}$$

Using the equation of motion;

$$S = ut + \tfrac{1}{2} at^2 \tag{58}$$

u = 0

$$a = \frac{2s}{t^2}$$

$$a = \frac{2 \times 0.006}{6.25 \times 10^{-6}}$$

$$a = 1920 m/s^2$$

The mass of the inlet valve and its gear is 0.33Kg, and 0.167Kg for the exhaust.

Therefore the force applied to the valves can be evaluated from $F = ma$.

Inlet $\quad F = 631.68N$

Exhaust $\quad F = 320.64N$

Two springs were chosen to be used on the inlet valve timing system, this meant that the force required by the springs only needed to be half of the value expressed above.

These forces give a starting point for the spring selection, along with the dimensional restraints of the valve gear. Due to the itterative nature of the process a spreadsheet was created. The calculations for the spring selection will be followed in detail to illustrate the process.

Consider the exhaust valve @ 2000rpm.

Outside diameter OD = 30mm
Free Length FL = 36mm
Force F = 340N @ 6mm deflection

Some assumptions need to be made, these can be estimated from the appropriate table found in any suitable text book on spring design.

Assuming the spring material is music wire

Modulus of Elasticity G = 81000Mpa

Safe Stress S = 550Mpa

We now need to find the wire diameter, d, actual stress ,s, and number of coils, N.

$$d = \left[\frac{2.55FD}{S} \right]^{1/3} \tag{59}$$

$$d = 3.5mm$$

Where D = mean wire diameter.

$$s = \frac{FD}{0.393d^3} \tag{60}$$

$$s = 569N/mm$$

This allows the number of coils required to be calculated from:

$$N = \frac{GdF}{\pi SD^2} \tag{61}$$

Appendix 4

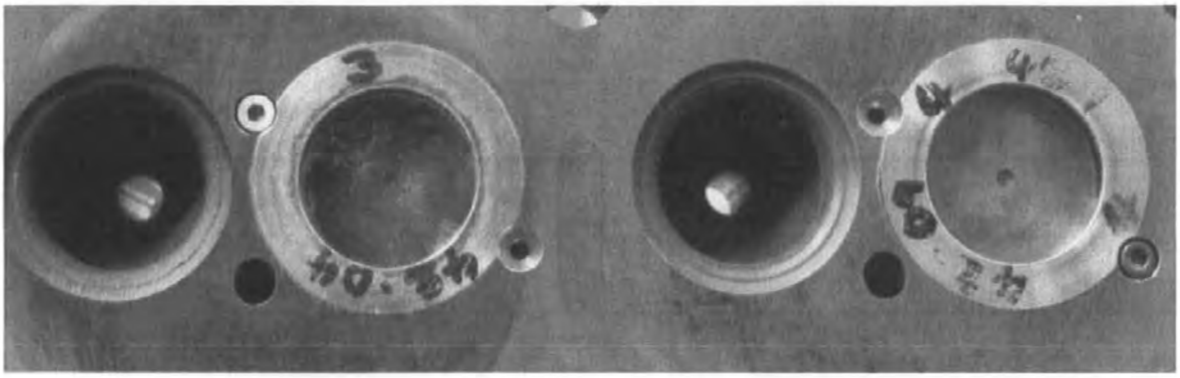


Plate 2. Expander Inlet Valve Seat Inserts.

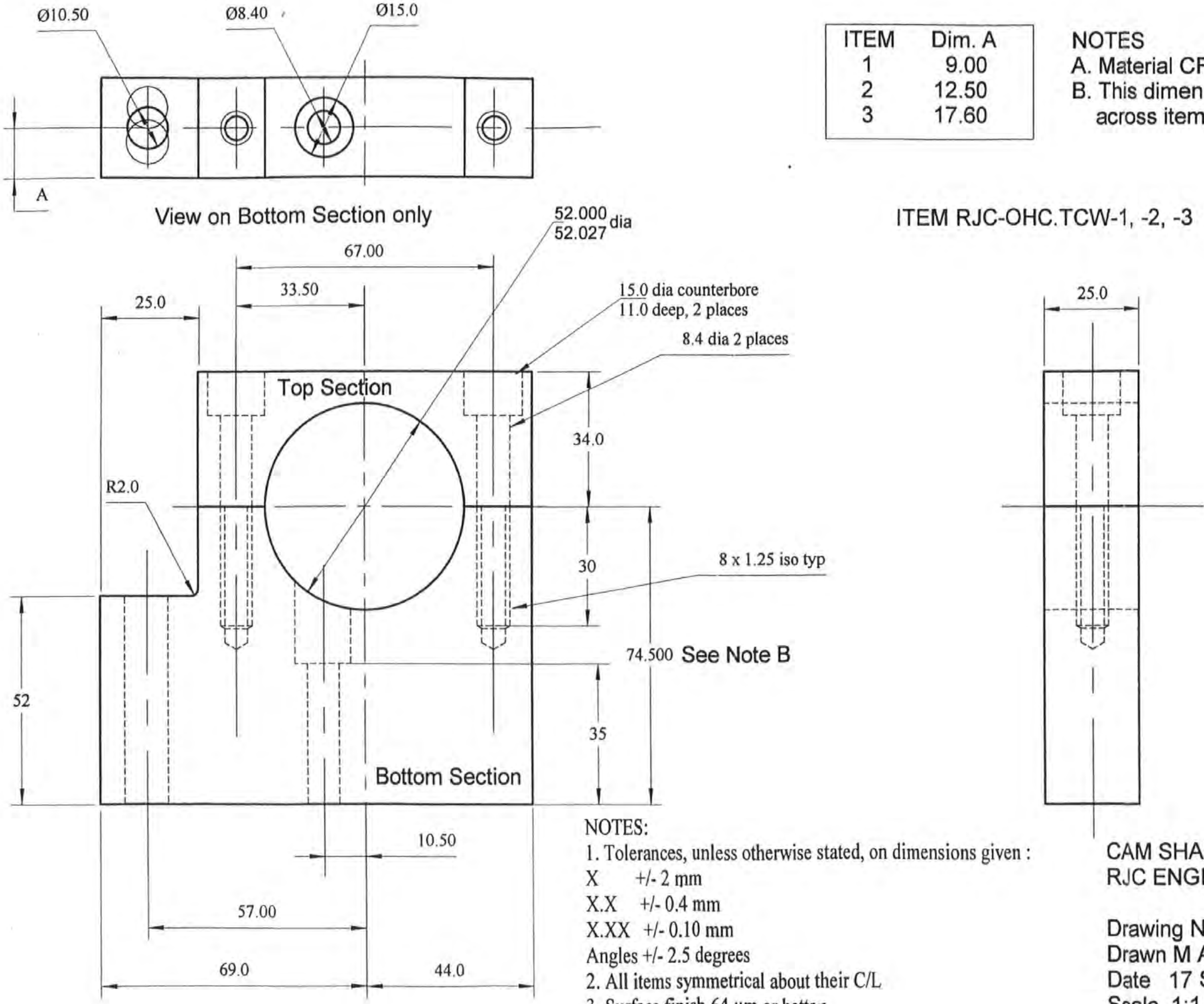
Appendix 5

The following 5 drawings illustrate the expander valve timing mechanism.

ITEM	Dim. A
1	9.00
2	12.50
3	17.60

NOTES
A. Material CRS
B. This dimension to be within +/- 0.025 across items 1 to 3

ITEM RJC-OHC.TCW-1, -2, -3 1 OFF EACH



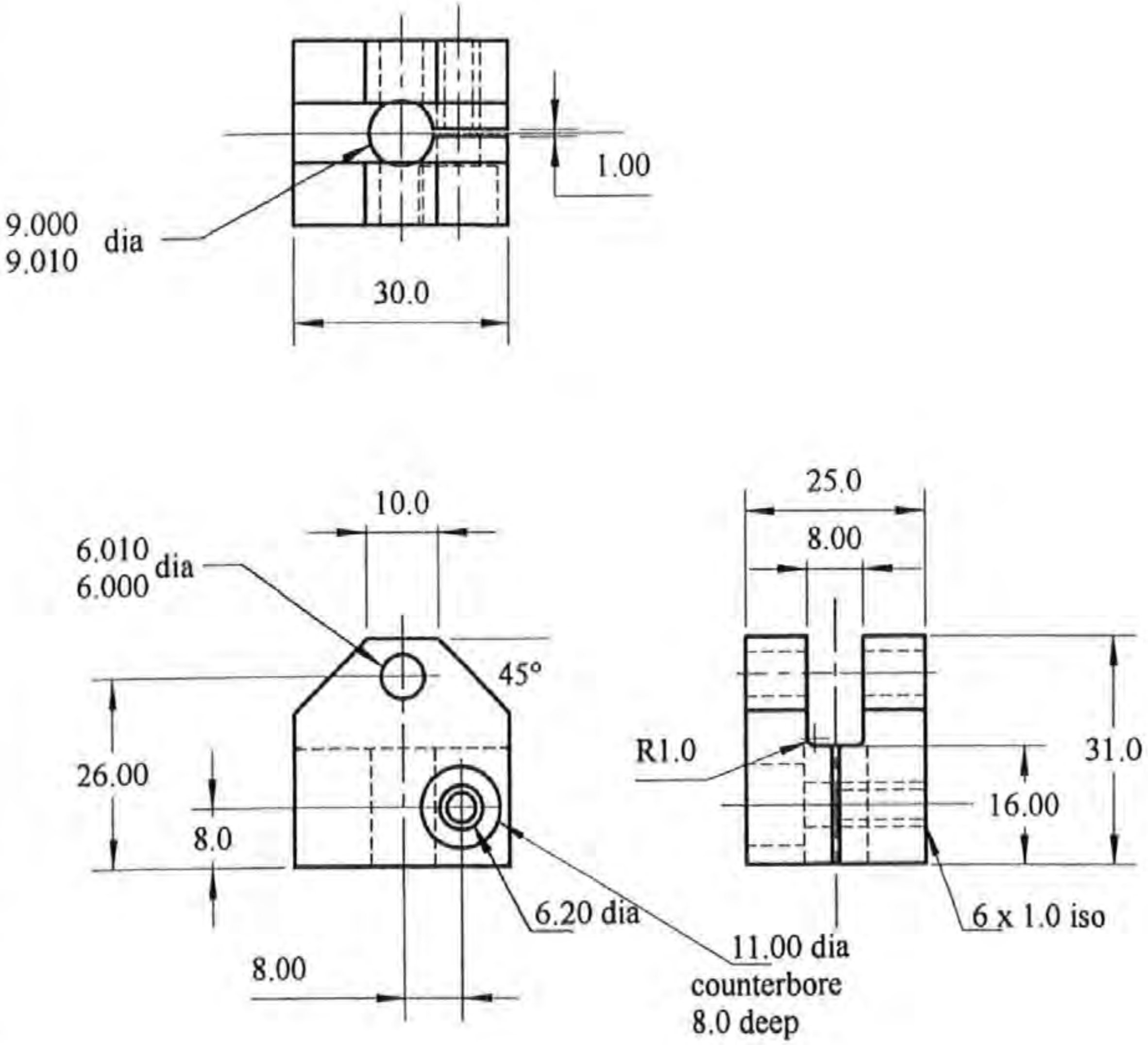
NOTES:
1. Tolerances, unless otherwise stated, on dimensions given :
X +/- 2 mm
X.X +/- 0.4 mm
X.XX +/- 0.10 mm
Angles +/- 2.5 degrees
2. All items symmetrical about their C/L
3. Surface finish 64 µm or better

CAM SHAFT BEARING HOUSINGS
RJC ENGINE

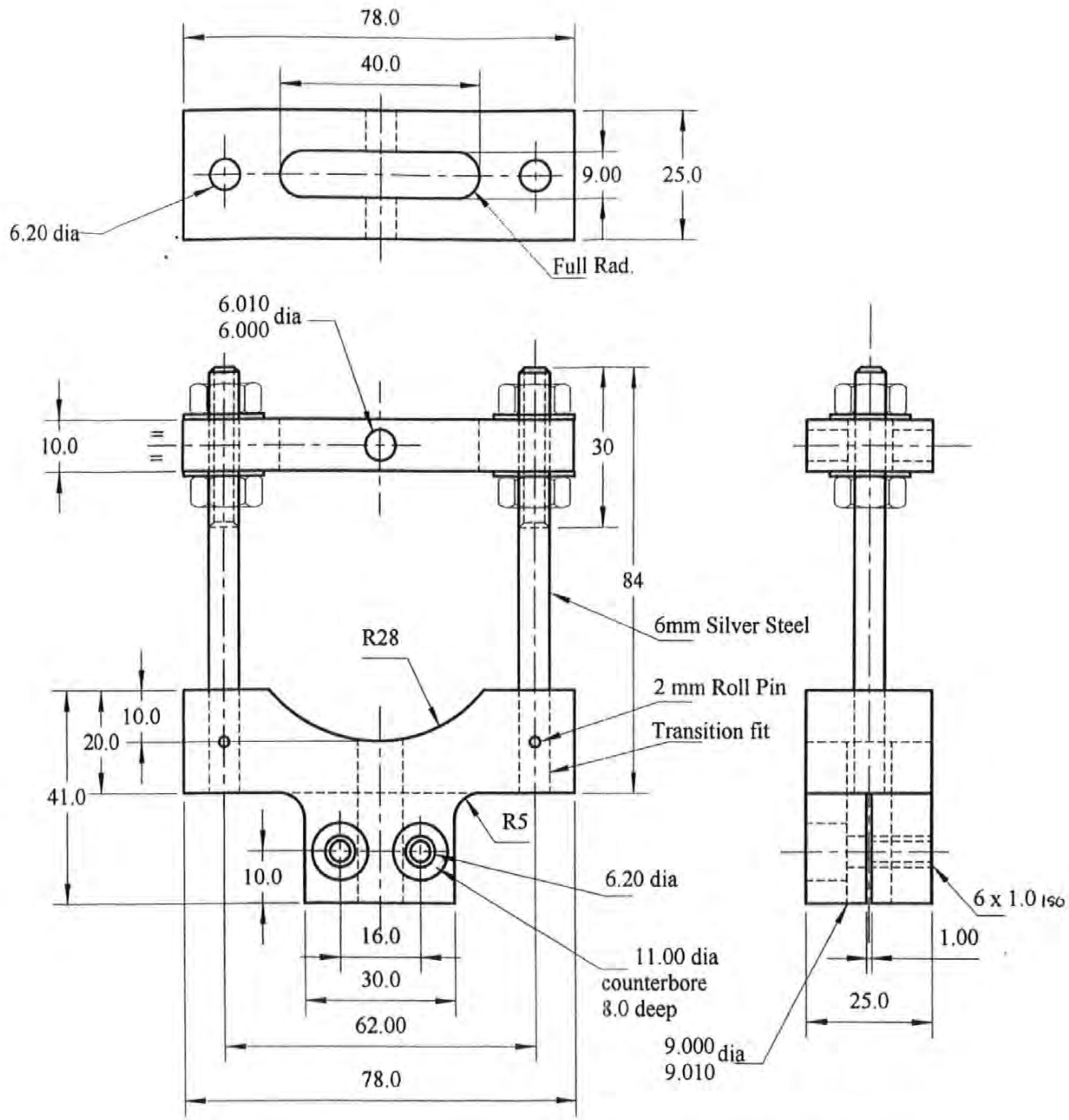
Drawing No. RJC-OHC2.TCW
Drawn M A Bell
Date 17 Sept., 1997
Scale 1:1

ITEMS RJC-OHC0.TCW-1 & -2, 2-OFF REQUIRED

MATERIAL CAST ALUMINIUM ALLOY UTS 300 MN/m² min



ITEM RJC-0HC0.TCW-1



ITEM RJC-0HC0.TCW-2

NOTES:

1. Tolerances, unless otherwise stated, on dimensions given :

X +/- 2 mm

X.X +/- 0.4 mm

X.XX +/- 0.10 mm

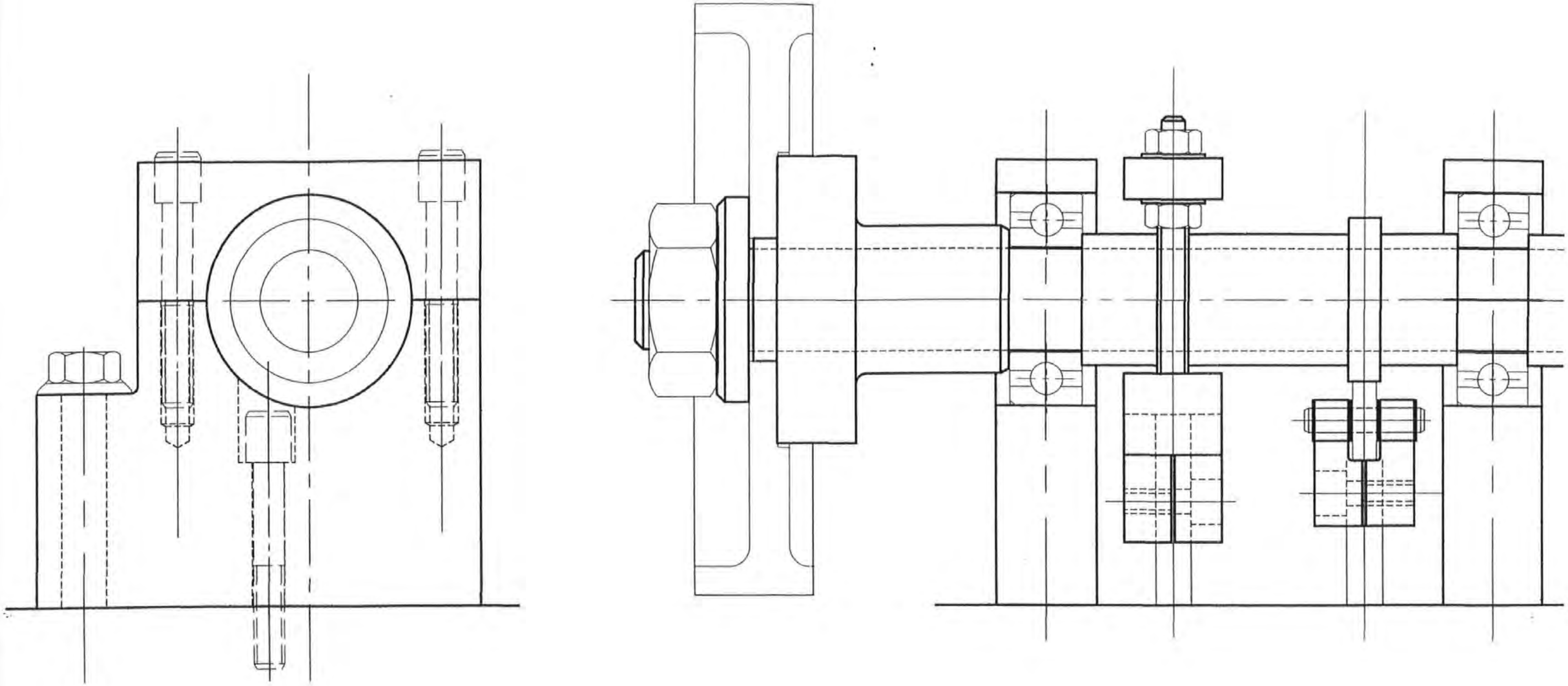
Angles +/- 2.5 degrees

2. All items symmetrical about their C/L

3. Surface finish 64 μm or better

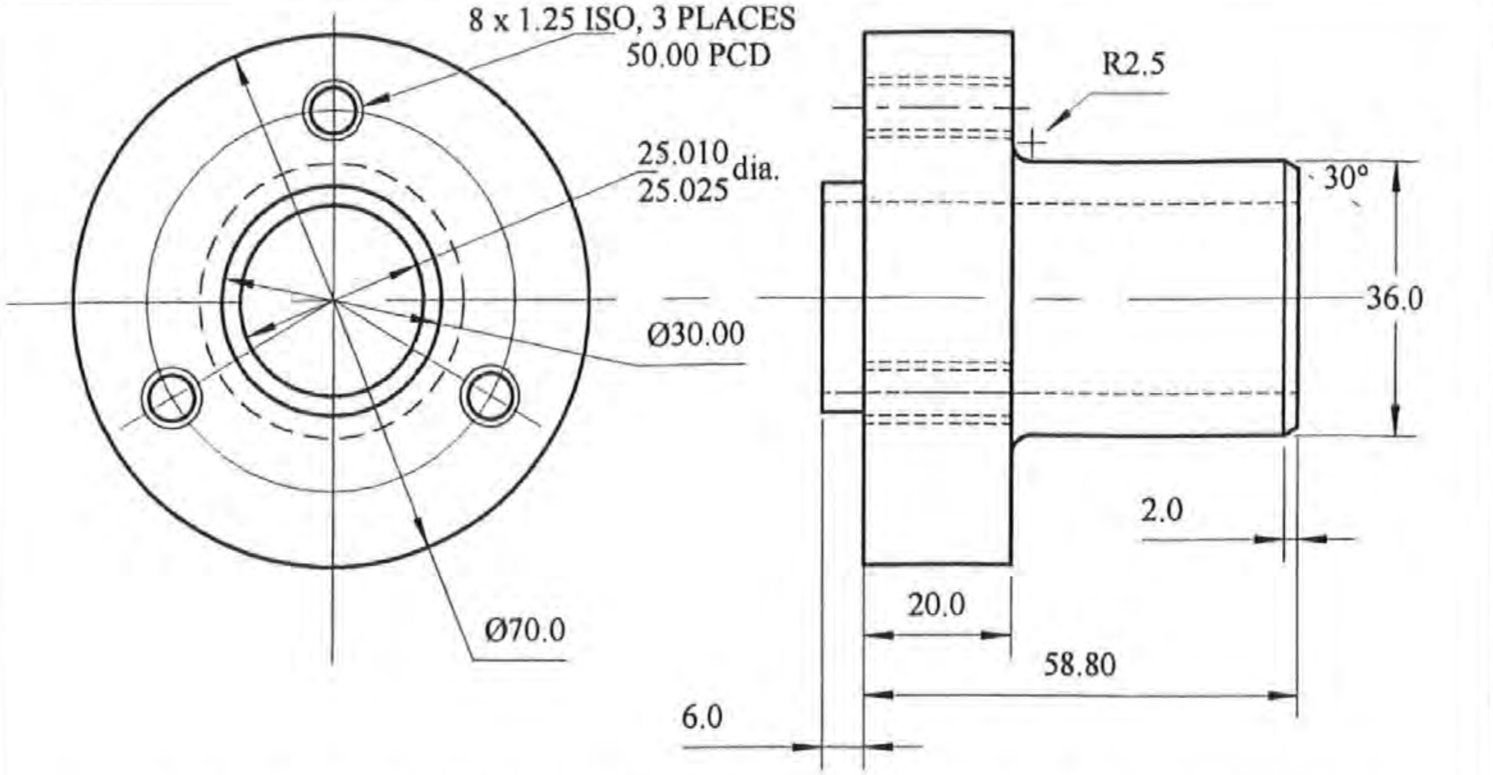
CAM FOLLOWER HOUSINGS
- RJC Engine

Drawing No. RJC-OHC0.TCW
Drawn: M A Bell
Date: 30 Sep., 1997
Scale 1:1

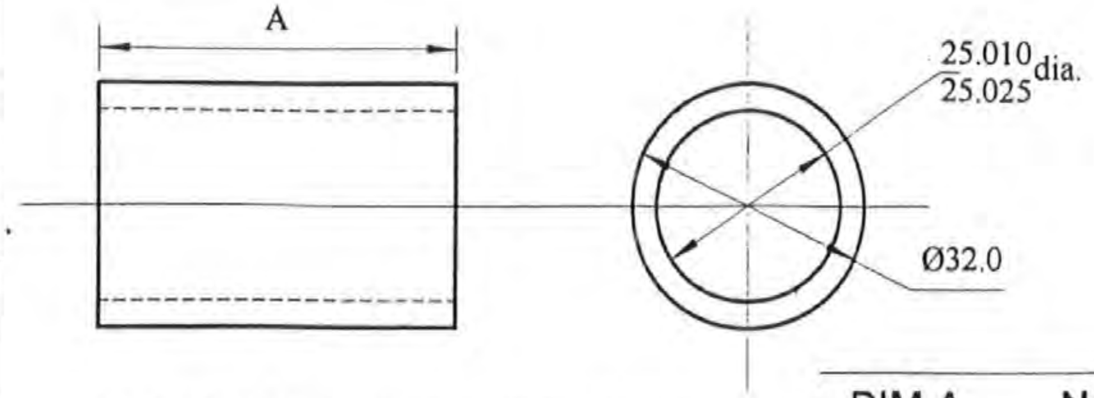


OVERHEAD CAM ASSEMBLY -
RJC Engine

Drawing No. RJC-OHC1.TCW
Drawn: M A Bell
Date: 30 Sept., 1997
Scale 1:1



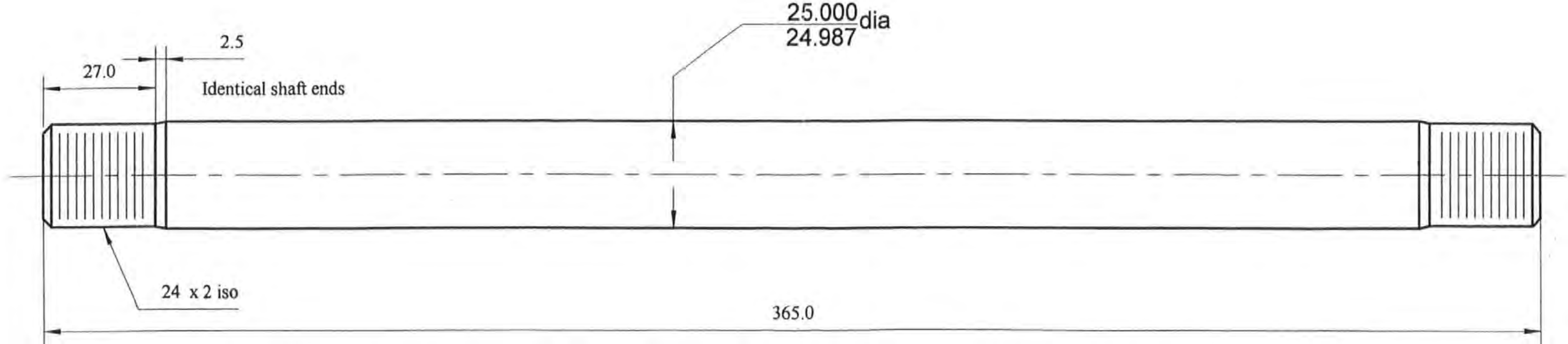
DRIVE PULLEY FLANGE : Material - Mild Steel, 1 OFF ITEM RJC-OHC3.TCW-2



SPACERS : Material - Mild Steel

ITEM RJC-OHC3.TCW-3-'A'

DIM A	No. off
40.39	2
19.17	4



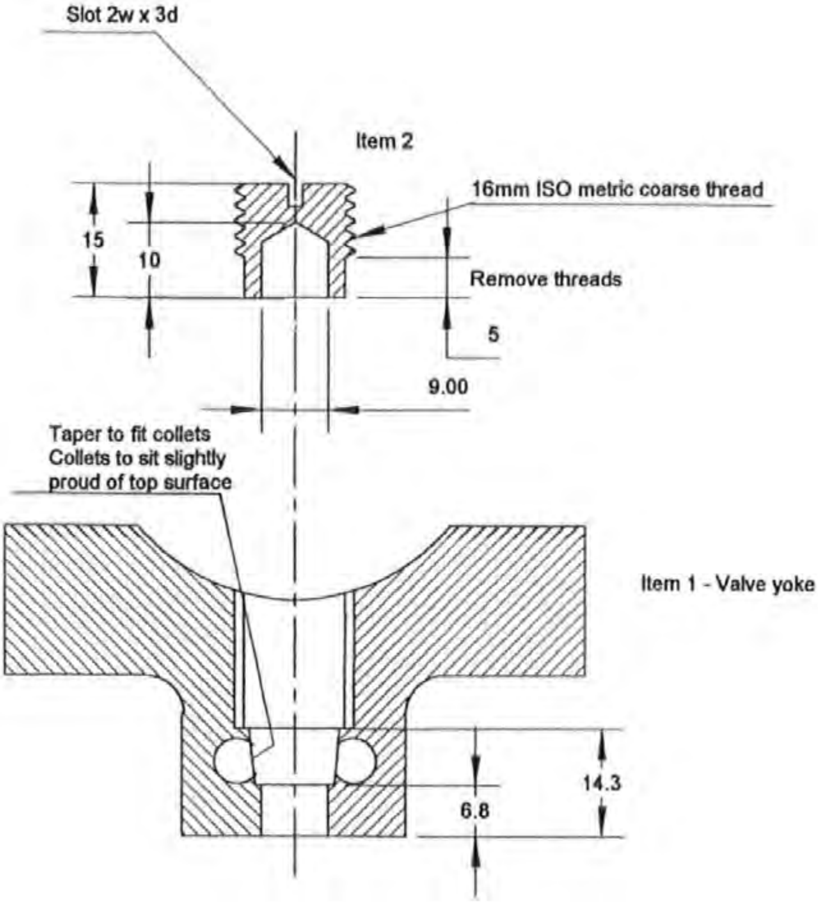
SHAFT : Material - EN6 or equiv.

ITEM RJC-OHC3.TCW-1

NOTES:
1. Tolerances, unless otherwise stated, on dimensions given :
X +/- 2 mm
X.X +/- 0.4 mm
X.XX +/- 0.10 mm
Angles +/- 2.5 degrees
2. All items symmetrical about their C/L
3. Surface finish 64 µm or bett

CAM DRIVE SHAFT - RJC ENGINE

Drawing No. RJC-OHC3.TCW
Drawn M A Bell
Date 29 Sept., 1997
Scale 1:1



Notes:

1. Material Item 2 - MS

Collet taper in Inlet Valve
Cam Follower Block
Scale: Full size
Drawn: M A Bell
Date: 5/8/98

Appendix 6

Expander Cam Profile Calculations

Before the profile of the cams could be worked out, the movements of the piston inside the cylinder needed to be more fully understood, particularly its relationship to the crank angle. The clearance of the piston from TDC can be found from:

$$Sk = r\,[\,1 + (l/r) - \cos\varphi - ((l/r)^2 - \sin^2\varphi)^{1/2}] \qquad (62)$$

where r = Crank Radius, mm

and l = Con rod length, mm

The crank angle can be calculated from:

$$\varphi = 2\pi nt \qquad (63)$$

Where the angle is in radians,

n = engine speed

and t = time, s.

The average piston velocity can be found from:

$$vm = 2ns \qquad (64)$$

where s = stroke, mm.

For a known geometry within the cylinder the piston clearance can be calculated, a conversion table can then be employed to give the conversion to the crank angle [28].

Using this method the required opening and closing times of the valves were calculated and from this the profile was calculated. Parameters such as the lift of the valves were predetermined by the geometry of the mechanism. The inlet valve was set up to open at TDC and close 53° After Top Dead Centre. The exhaust valve was due to open at BDC and close 24° Before Top Dead Centre. The total lift by the cams was set at 8mm and the opening and closing processes for each cam occurred over a 30° period, with the fully open or closed position being half way through this motion.

Appendix 7

Combustion Chamber Flow Calculations

Below in Fig 41 is an illustration of the apparatus used to measure the flow of gases through the combustion chamber, the engineering drawing of the chamber can be seen on the next page.

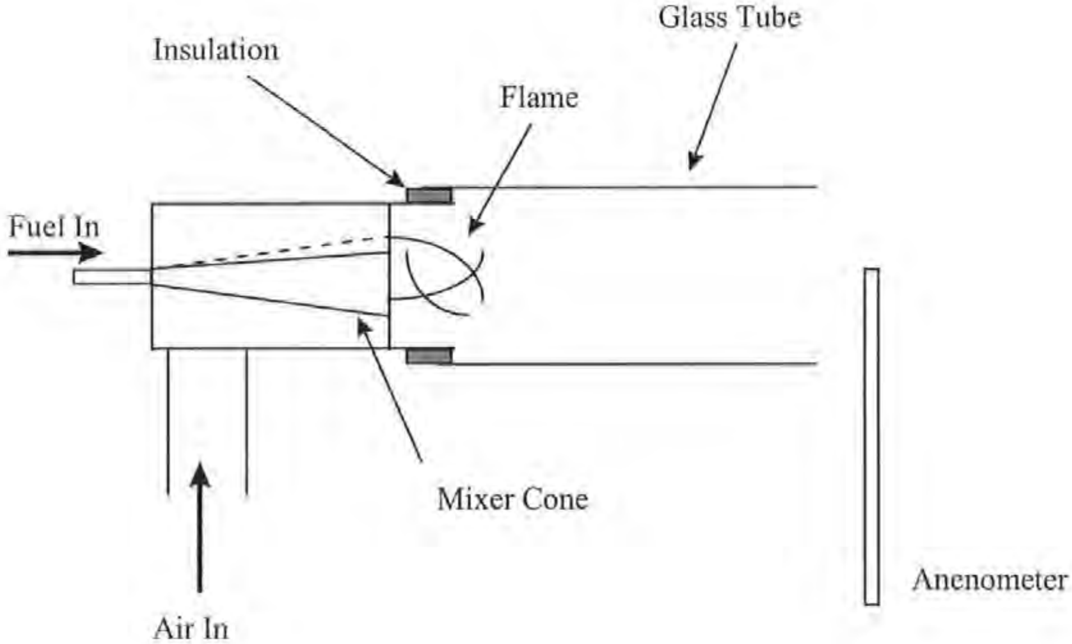


Fig. 41. Flow Measurement through Combustion Chamber.

Once the rig was set up the air and gas flow was turned on and the mixture ignited from the open end of the glass tube. The gas and air were then adjusted until a blue flame was produced that was held in front of the face section of the cone. Once satisfied that the flame had stabilized the fuel was turned off at the bottle, the air was left to run through the chamber to bring the temperature of the apparatus back down to room temperature. Once the rig had cooled sufficiently velocity readings were taken.

Baffle plates were changed to gauge the effect of changing the gap between the cone segments and changing the slot sizes that ring the cone.

Using a baffle plate with slots 2mm wide the following results were found:

Velocity (air and gas) = 0.1 - 0.2 m/s

From the model, air mass flow = 0.000733 Kg/cycle = 0.0366 Kg/s (air)

now,

$$m_{flow} = \rho A v \tag{65}$$

and,

$$\rho = \frac{P}{RT} \tag{66}$$

$$\rho = \frac{101325}{283 \times 293}$$

$$\rho = 1.2219$$

so,

$$m_{flow} = 1.2219 \times \frac{\pi(0.090^2)}{4} \times 0.2$$

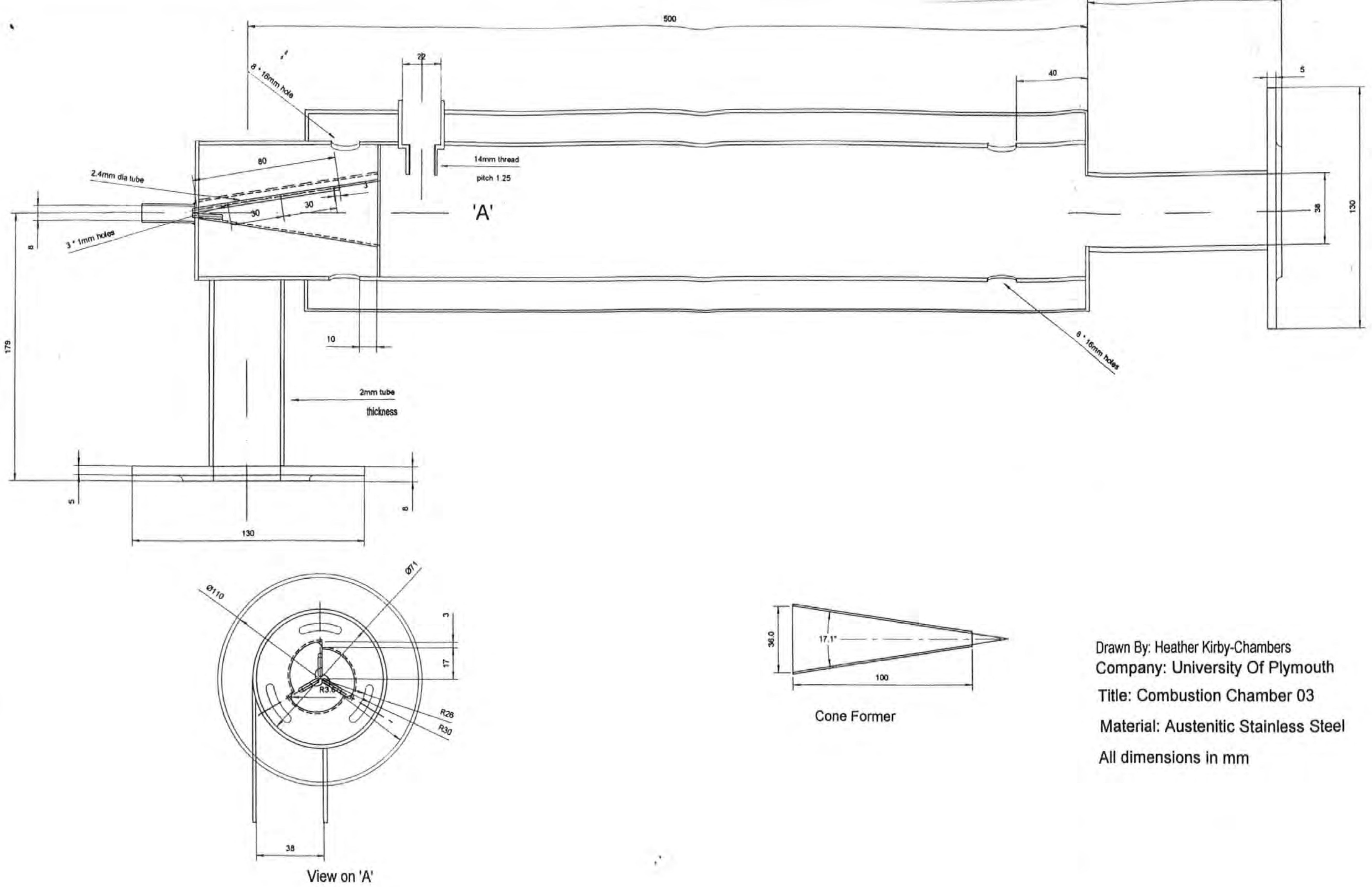
$$m_{flow} = 1.55 \times 10^{-3} \text{ Kg/s}$$

When using a baffle with 3mm slots the velocity was found to be:

Velocity = 1.15 m/s

Thus giving:

$$m_{flow} = 9.032 \times 10^{-3} \text{ Kg/s}$$



Drawn By: Heather Kirby-Chambers

Company: University Of Plymouth

Title: Combustion Chamber 03

Material: Austenitic Stainless Steel

All dimensions in mm

Appendix 8

Combustion Chamber Air-Fuel Ratio Calculations

Assuming that complete combustion takes place:

$$(0.8CH_4 + 0.2C_2H_6) + y_s(0.21O_2 + 0.79N_2) \rightarrow aCO_2 + bH_2O + cN_2 \qquad (67)$$

Now consider the equation one side at a time, and the unknown quantities can be found:

	LHS	RHS	Solving gives:
C	0.8 + 0.2 x 2	a	a = 1.2
H	4 x 0.8 + 6 x 0.2	2b	b = 2.2
O	y_s x 2 x 0.21	2a + b	y_s = 10.95
N	y_s x 2 x 0.79	2c	c = 8.65

$(y_s)_{mol} = 10.95$ molecules molar AFR

$$(y_s)_{mass} = (y_s)_{mol} \frac{\text{molar mass of air}}{\text{molar mass of fuel}} \qquad (68)$$

$$(y_s)_{mass} = 10.95 \times \frac{(0.21 \times 32 + 0.79 \times 28)}{(0.8 \times 16 + 0.2 \times 30)}$$

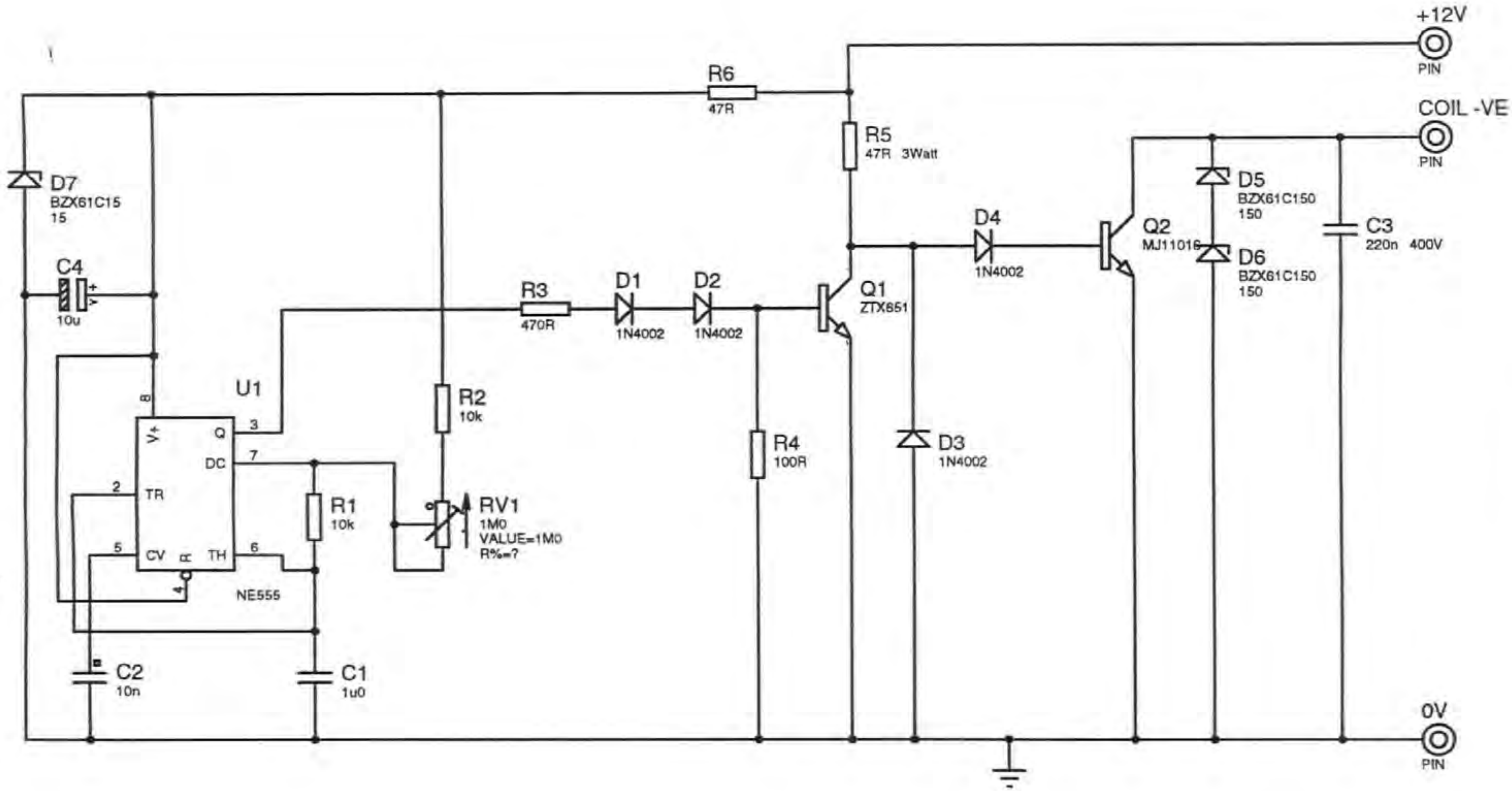
$$(y_s)_{mass} = 10.95 \times \frac{28.8}{18.8}$$

$(y_s)_{mass} = 16.79$ AFR by mass

Appendix 9

Plate 3. Swirl of Flame

Appendix 10



+12V
PIN
COIL -VE
PIN
0V
PIN
R6
47R
R5
47R 3Watt
D7
BZX61C15
15
C4
10u
U1
NE555
R3
470R
D1
1N4002
D2
1N4002
Q1
ZTX851
D4
1N4002
Q2
MJ11016
D5
BZX61C150
150
D6
BZX61C150
150
C3
220n 400V
R2
10k
R4
100R
D3
1N4002
R1
10k
RV1
1M0
VALUE=1M0
R%=?
C2
10n
C1
1u0

Appendix 11

Constant Pressure Adiabatic Flame Temperature

During a constant pressure process the flame temperature reached when a fuel is completely burned adiabatically, is known as the Adiabatic Flame Temperature. In reality the flame temperature reached is lower than this value due to dissociation, excess air, incomplete combustion and radiation. Under the conditions described for complete combustion the absolute enthalpy of the reactants must equal that of the products in their final condition, so:

$$H_{reac}(T, P) = H_{prod}(T_{ad}, P) \tag{69}$$

Consider a stoiciometric compressed natural gas-air mixture. The adiabatic flame temperature can be calculated assuming that combustion is complete and the products of the reaction are CO_2, H_2O and N_2 only.

The equation for the reaction is the same as used to calculate the stoichiometric air-fuel ratio in Appendix 6:

$$(0.8CH_4 + 0.2C_2H_6) + 10.95(0.21O_2 + 0.79N_2) \rightarrow 1.2CO_2 + 2.2H_2O + 8.65N_2 \tag{70}$$

Firstly consider the enthalpies of formation for:

Reactants

	No of Moles	H_f° (KJ/kmol)	Total
CH_4	0.8	-74.831	-59.864
C_2H_6	0.2	-84.667	-16.933
O_2	2.29	0	0
N_2	8.65	0	0
			-76.797

Products:

	No of Moles	H_f° (KJ/kmol)	Total
CO_2	1.2	-393.546	-472.255
H_2O	2.2	-241.845	-532.059
N_2	8.65	0	0
			-1004.314

Now consider the Sensible Enthalpy of both reactants and products at the target temperature of 1200K.

Reactants:

	No of Moles	Hs (KJ/kmol)	Total
CH_4	0.8	0	0
C_2H_6	0.2	0	0
O_2	2.29	29.775	68.185
N_2	8.65	28.118	243.221
			311.406

At this point the enthalpy equation is to be considered

$$\Sigma(\Delta H_f)_R = \Sigma(\Delta H_f)_P \qquad (71)$$

giving Equation 72 below

$$\Sigma(\Delta H_f^{\circ})_R + \Sigma(\Delta H_f)_R = \Sigma(\Delta H_f^{\circ})_P + \Sigma(\Delta H_f)_P \qquad (72)$$

$$(-76.799) + (311.406) = (-1004.314) + \Sigma(\Delta H_f)_P$$

$$\Sigma(\Delta H_f)_P = 1239.314 \text{ KJ/kmol}$$

The adiabatic flame temperature for the reaction can be calculated from this value of sensible enthalpy for the products. The method being to start from this value and work backwards through the processes illustrated in the tables above. Estimates for the flame temperature must be made, and the associated total enthalpy found. A further calculation is then required to find the actual adiabatic flame temperature.

		2700K		3000K	
	No of Mole	Hf	Total	Hf	Total
CO_2	1.2	134.284	161.141	152.891	183.469
H_2O	2.2	109.979	241.954	126.563	278.438
N_2	8.65	81.652	702.289	92.730	802.114
			1105.384		1264.021

With these two values of enthalpy at particular temperatures the adiabatic flame temperature can be calculated. In choosing the two temperatures in this process we are trying to obtain total enthalpy values that are on either side of the one calculated in Equ.72. This means that the final calculation gives a more accurate value of the adiabatic temperature.

$$\frac{T_{ad} - T_L}{T_U - T_L} = \frac{\sum(\Delta H_f)_P - H_{fL}}{H_{fU} - H_{fL}} \qquad (73)$$

$$\frac{T_{ad} - 2700}{300} = \frac{1239.314 - 1105.384}{1264.021 - 1105.384}$$

$$T_{ad} = \left[\frac{133.93}{158.637} \times 100\right] + 2700$$

$$T_{ad} = 2953.28K$$

This is the adiabatic flame temperature of a stoiciometric compressed natural gas-air mixture undergoing complete combustion, and assuming that no dissociation takes place. This method can also be used to calculate the temperature when there is an excess of air involved in the reaction.

Appendix 12

Calculate actual AFR in the experimental chamber.

Using:

$$m_{flowfuel} = \rho A v \tag{74}$$

$$m_{flowfuel} = \frac{137 \times 10^5}{283 \times 293} \times \frac{\pi}{4}(0.005^2) \times 0.04$$

$$m_{flowfuel} = 1.29 \times 10^{-4} \text{ kg/s}$$

It is known from Appendix 5 that the mass flow of air through the chamber was 9.32×10^{-3} kg/s. Using the stoichiometric air-fuel ratio calculated in Appendix 6 the actual air-fuel ratio in the combustion chamber can be calculated.

$$AFR = \frac{m_{total}}{m_{flowfuel}} \tag{75}$$

$$AFR = 70.016$$

Knowing the air-fuel ratio that was present in the chamber during the practical testing at atmospheric conditions the theoretical adiabatic flame temperature can be calculated and compared to that achieved in practice. Using the same methodology to calculate the adiabatic flame temperature as demonstrated in Appendix 11 the following can be reached:

$$T_{ad} = 1847.659\text{K } (1574.659°\text{C})$$

The difference between the theoretical and practical results will be due to numerous factors including incomplete combustion, dissociation and heat loss through radiation.