The Development of a Prototype External Heat Engine Based on the Ericsson Cycle

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2000

Thesis submitted to the Department of Mechanical Engineering, University of Cape Town in fulfillment for the Degree of Master Of Science in Mechanical Engineering

Synopsis

The aim of this thesis was to develop a prototype external heat engine based on the Ericsson cycle, as an alternative to the internal combustion engine, to be used as a small-scale power source for rural Africa. Subsequently test and evaluate its viability and potential to fulfil the requirements of such an application.

Despite the wide range of possible prime movers, it appears there is still a need for a simple, low-tech, low-output power plant for developing countries. This created an opportunity to re-visit the origins of basic engine design in order to seek an alternative solution to the modern internal combustion engine.

The hot air or external heat engine developed in the 1800's provides an attractive alternative as it has a number of advantages over the modern internal combustion engine. A hot air engine is a cyclical heat engine that uses an external heat source, heat exchangers, pistons and a gaseous working fluid contained within the engine to convert heat to mechanical work by volumetric expansion. The project looked at old and new engines in an attempt to capture the best of both.

Two experimental engines were constructed during the course of this project, the first engine was built to provide insight into the functioning of an unconventional external heat engine and to test the validity of theoretical predictions made using a thermodynamic computer model. This engine was designed to function off a cycle consisting of a polytropic compression, a polytropic expansion with heat addition and a constant volume heat rejection process, achieved using a two-stroke principal to exchange the hot exhaust gas with cold recharge gas.

Based on experience gained from this model, the second generation engine was designed to circumvent the problems experienced with the first engine. It functioned off a near Ericsson cycle, with the compression and expansion truncated for practical purposes and valve control being achieved with solenoid valves controlled by a computer. A thermodynamic computer model similar to the one used for the first engine was employed to optimise the design of this engine.

Experimental investigations were carried out with the Ericsson engine to examine how closely the actual cycle resembled that predicted by the thermodynamic model and to determine engine performance. The power and mean effective pressure produced by the engine were determined and compared with friction data. Hence the potential of this engine to meet the criteria necessary to function as a small-scale rural power source was judged and resultant conclusions as to the engines feasibility were drawn.

The actual pressure–volume diagrams obtained closely conformed to the theoretical expectations for the cycle and the truncated Ericsson cycle functioned sufficiently well. However, the friction in the system was too high a percentage of the total engine output and therefore the engine was unable to operate unaided.

Although the hot air engine has the potential to provide cheap power efficiently, in practice these engines need to be highly pressurised and run at temperatures close to their material limit in order to obtain useful work from them. Therefore, although with the use of low friction seals and high pressurisation the engine could potentially produce the 5kW design target, due to the complexity these efforts would add to the engine it is recommended that other options be explored for rural power generation in Africa.

Declaration

I declare that this thesis is essentially my own work and that it has not been submitted for a degree at any other university.

Signed by candidate

J. Hussey

December 2000

Acknowledgements

The author would like to acknowledge the following for their contributions made towards this project:

My supervisor Dr. A Yates, for his guidance and advice, his practical experience was invaluable. His support and encouragement throughout the duration of this project was sincerely appreciated.

The staff of the Mechanical Engineering workshop, for their endless advice and patience, and for all the work they did in making both engines a reality.

Eskom, for funding this project and thus providing the means by which to carry out this thesis.

The Energy Research Institute, for the opportunity to do this project, for their support and enthusiasm throughout the year.

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1. Introduction

The continued trend for conventional engine designs to become increasingly high-tech has exposed the lack of infrastructure and service support in third-world countries. The high price of fuel and spares for these engines only serves to exacerbate the situation. As a result, these engines are subjected to harsh operating conditions, a lack of adequate maintenance and poor fuel quality, all of which drastically reduces the engine's life. This poses the question as to whether such engines are appropriate for developing countries.

On a global level, the increasingly high cost and dwindling resources of fossil fuels, combined with current environmental concerns, is forcing the world energy market to consider alternatives to the internal combustion engine in the future. However, the majority of this research is aimed at high-tech solutions and for the transport industry specifically. These endeavors are largely inappropriate for small-scale power generation in rural Africa, were this sort of technology is far beyond the reach of the average person.

From a technological point of view, it could be said that the level of development in rural Africa is not far removed from that of mid-nineteenth century Europe. Hence, it may be instructive to look back at the solutions that were available at that time and consider the type of prime movers that were built and operated within the framework of the prevailing skills and educational background.

Many of the engines in the 1800's were thermodynamically well conceived, but their development was impeded by machining limitations and inferior materials by today's standards. Hence many only achieved a fraction of the performance of which they were capable. Modern technology and machining processes now provides the potential to implement these designs more effectively. [1]

The steam engine was the principal prime mover of the 19th century, but the poor materials available for construction of the boilers left them prone to explosions with disastrous effects for those operating them. They also required a plentiful supply of water, which is a precious commodity in rural Africa.

The hot air engine, which was also conceived and developed during the 1800's, provides an attractive solution to these problems. It is an external combustion engine and as such relies on heat to be transferred from an external source, so does not require the stringent fuel specifications necessary for internal combustion engines. Its ability to operate from any heat source, its long life and durability combined with low maintenance and running costs make it a worthy contestant for a small-scale power source.

William Beale, who was instrumental in the development of several recent hot air engines, stated that hot air engines "are so simple, yet effective, they are an excellent choice for power generation in developing countries." [2]

These engines have the potential to provide power and efficiency superior to diesel engines. Even better is their potential for lower life cycle cost, less weight, and almost no emissions. The engines are quieter, smoother and much cleaner than conventional internal combustion engines mainly due to their continuous external combustion. [3]

As a power system (converting heat to work), hot air engines have unique advantages: [3]

- **Multi-fuel capability:** Hot air engines can use any form of heat supplied from an external source.
- **Quiet operation:** There are no periodic explosions in external combustion engines, they have the potential to operate at very low noise levels.
- **High Efficiency:** Many of the feasible cycles have the potential to achieve the maximum possible Carnot cycle efficiency.
- **Low internal wear and lubrication consumption:** The combustion products in a hot air engine are not in contact with the moving parts. Therefore, there is no contamination of the lubricating oil as in a diesel engine.
- **Flat 'part-load' characteristics:** Hot air engines have flat part-load characteristics, that is, the efficiency remains more or less constant over wide load changes. They also have a flat torque-speed curve, as the cylinder torque is less variable than that of an internal combustion engine because there is one cycle per 360 degrees of crankshaft revolution.
- **Flexibility in design:** The few essential elements of a hot air engine allow for a wide variety of mechanical configurations. The pressure ratio is lower and not as quick changing as in an internal combustion engine, so bearing loads are lower and components can be lighter.

Shortly after World War II, fresh attention was brought to the hot air engine, with specific attention being given to the Stirling cycle. Advances in thermodynamics, a greater understanding of the underlying principals and considerable development in the materials available for the construction of such an engine aided its revival. However, most of the efforts in this field of research were aimed at technical applications like vehicles, artificial hearts and space programmes. Very little was achieved in designing a competitive engine for rural power generation. [5]

Despite the wide range of possible prime movers, it appears there is still a need for a simple, cheap and reliable power source for developing countries. The internal combustion engine, in its current guise no longer fits the criteria and the cost of alternatives such as solar panels puts them beyond the reach of most third world communities. These factors, combined with the aforementioned advantages of the hot air engine, led to the concept for this thesis. To investigate the possibility of using an external combustion hot air engine for power generation in rural African communities.

In order to determine the most appropriate cycles for such an application, it was instructive to start the research by exploring the history of the hot air engine.

1.1 Background

Powering engines by the variations in a volume of air as it altered temperature was first envisioned by Henry Wood in his patent of 1759. His vision was to pump heated air into a large cylinder, cool the air and let atmospheric pressure do the work on the inward stroke of the piston. The first to build a working model of Wood's proposal was Sir George Cayley in 1807. Further technological advancements by the Stirling brothers in 1816 earned them a place in history as the 'inventors' of the hot air or 'Stirling' engine [4]. Figure 1.1 shows Stirling's first engine built in 1816.

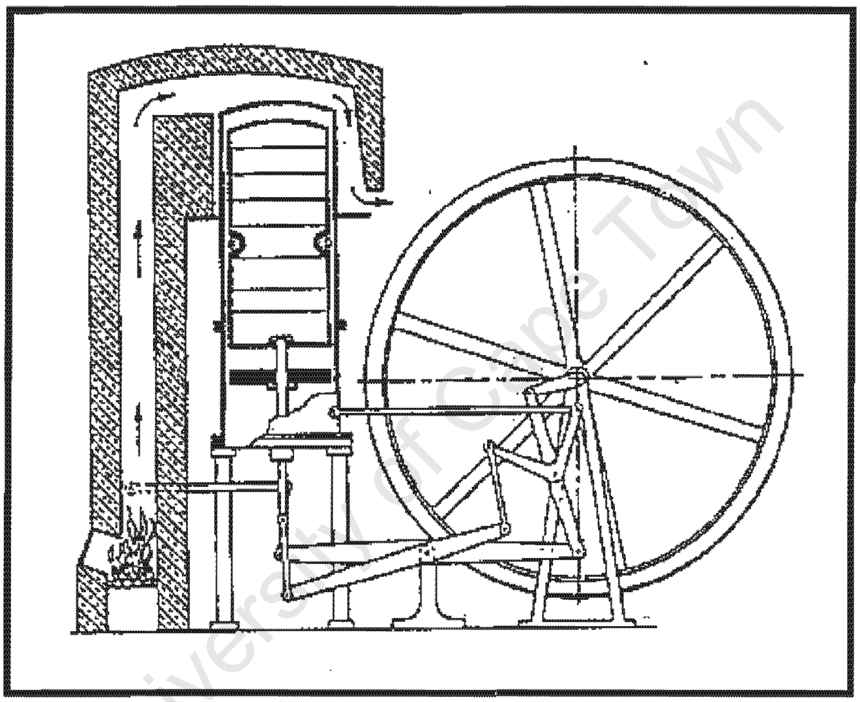

Figure 1. 1: Stirling's Engine of 1816

(Source: http://sesusa.hypermart.net/hist.ory.1816.htm)

The idea of using air as the working fluid was appealing because steam engines, the principal alternative, were prone to explosions due to the lack of strength in the materials available to construct boilers. Hot air engines would not explode because the energy stored in the working cylinder was insignificant in comparison to the energy stored in a heated boiler. The machine would simply stop when the heater section failed from thermal stress or imperfections in the material.

Throughout the 1800's a variety of hot air engines came to the forefront in an effort challenge the steam engine as the prevailing power unit. Their safety record made them more attractive despite their low power output, which limited their use to small industries and workshops.

Well known names such as Stirling (1815), Ericsson (1833), Joule (1852), Brayton (1867), Otto (1867) and Diesel (1897) produced a number of engines with varying degrees of success. Ericsson in particular was an extraordinarily inventive man, producing a range of successful engines that embodied a wide range of different thermodynamic cycles. An example of one of his most successful designs is shown in figure 1.2.

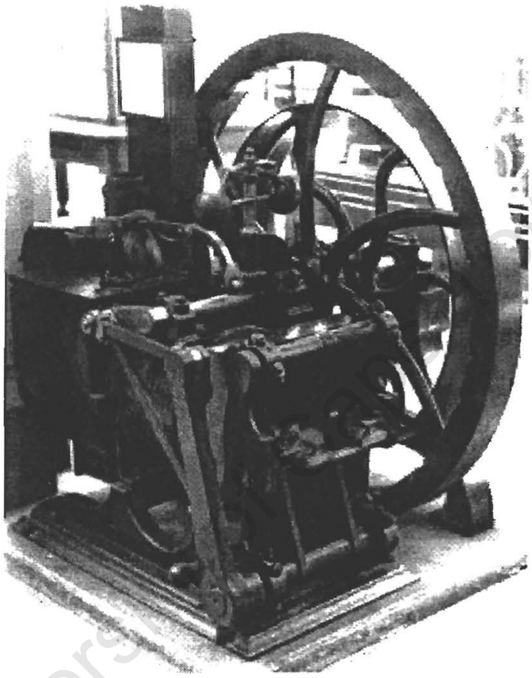

Figure 1. 2: Ericsson's engine of 1860

(Source: http://www.argonet.co.uk/users/bobsier/pio4B.html)

Towards the end of the nineteenth century, it became apparent that the temperature limitations of the external heat engine could be avoided by placing the combustion process inside the engine, giving birth to the internal combustion engine. With this development the maximum cycle temperature and hence the thermodynamic efficiency and specific power of engines could be significantly increased. External combustion engine development was therefore abandoned. Thus only the Otto, Diesel and Brayton/Joule cycles survived to become the modern day petrol, diesel and gas-turbine engines. The move away from external combustion however did not come without a penalty in the form of exhaust noise, increased exhaust emissions and stringent fuel specifications.

In 1937 Philips reconsidered the hot air engine as a power source for a small electric generator for radio sets. 1938 saw the first hot-air engine constructed at Philips. Shaft power was 16 Watts at 1000 rpm [3]. Up until 1979 extensive research was done at Philips, using different working gasses (hydrogen and helium) exotic heat sources (isotope heaters and metal combustors) and different drive mechanisms (swash plates and the Rhombic drive).
A number of comprehensive Computer programs were developed for use in engine design and optimisation. Tests were conducted with various engines including a 250kW Stirling engine in a bus, and a 127kW version for vehicular use in conjunction with Ford. This research culminated in the current state of the art Stirling engine, exemplified by a four cylinder, double-acting, engine capable of 25kW at 40% efficiency [3].

1.2 Aims and Objectives

The aim of this project was to investigate the feasibility of producing a simple prime mover that has the capability of running on any combustible fuel primarily for use as a power source in rural Africa. Specifically, the Ericsson engine designs were to be revisited as the source material for the project.

The thesis objectives were as follows:

- To determine the most appropriate cycle to run an external combustion engine for use as a power source in rural Africa.
- To develop a computer model of this thermodynamic cycle.
- To build an external combustion engine based on this cycle.
- To compare the test results with results predicted from the computer model.
- To evaluate the performance of this engine as a rural power source.

1.3 Project Outline

A number of theoretical cycles were considered. The efficiencies and Indicated Mean Effective Pressures (IMEPs) for each cycle were compared for the same temperature ratios. From these, the most suitable cycle could be determined and a prototype engine was built based on this cycle.

A thermodynamic model of the engine was constructed using an Excel spreadsheet, and a large number of configurations were tried and analysed for various pressures, temperatures and compression ratios. From this the optimum configuration was determined.

The experimental engine was built and tests were carried out to determine its performance compared to what was predicted. Further investigations into the fundamental principals of the engine were conducted, and the potential of the engine was assessed. The flaws with the concept were evaluated and decisions were made on the cycle and layout for the second-generation engine.

A similar thermodynamic computer model was used to investigate the new engine concept, determine the likely efficiency and IMEP and to decide on variables such as operating pressure and compression ratio. This second engine concept was designed and built and tests were carried out in order to investigate it's viability.

Finally, the performance and potential of this engine to meet its criteria were judged and resultant conclusions drawn.

2. Literature Review:

2.1 Hot Air Engines

In a hot air engine heat is applied externally, in order to raise the temperature of a working medium to provide the required degree of volumetric expansion to produce motive force. [6]

Hot air or external combustion engines can be split into two general categories. [5]

Stirling Engines:

The flow of the working gas is controlled by volume variations. Stirling engines use a closed cycle, i.e. the same working fluid is alternately heated and cooled during each cycle. These volumetric changes are brought about by displacing a fixed amount of gas to and from a hot and a cold zone in an enclosed cylinder. An important feature of the Stirling engine is the regenerator. This thermodynamic 'sponge' stores energy as the gas passes from the heater to the cooler and releases it as the gas flows back through the regenerator to the heater.

Ericsson Engines:

The flow of the working gas is controlled by valves. The Ericsson engine makes use of an open cycle, that is a fresh working medium is induced each cycle. The engine generally has a compression and an expansion space, the flow between these two being controlled by the opening and closing of valves. The Ericsson engine does not necessarily include a regenerator or cooler. Without the use of these heat exchangers, economisers and other waste heat recovery systems are necessary to improve the efficiency of these engines.

Very little literature was found to be available on the subject of Ericsson engines. The majority pertaining to Stirling engines. However, this literature is still applicable to the design of an Ericsson type engine, because the thermodynamic principals are essentially the same. It is therefore intuitive to look at Stirling engines in order to determine the design variables.

These engines can be subdivided according to the layout of the working cylinders. [7]

The Alpha type:

This type of engine consists of two pistons interconnected by a series of heat exchangers. This layout has the potential for the smallest dead volume. If the pistons were horizontally opposed in the same cylinder, the dead volume could theoretically be zero [7]. The friction associated with this engine is high however, because of the sealing rings required on both pistons.

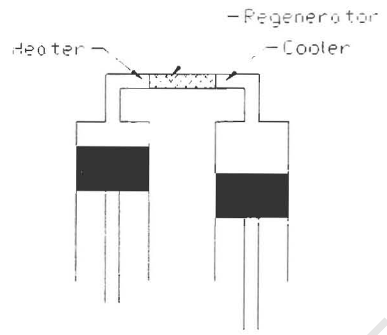

Figure 2. 1: α Type Stirling Engine

The Beta type:

This engine layout has a piston and displacer in one cylinder. Heating and cooling is done in different regions of the cylinder. The dead volume and the mechanical friction associated with this layout can be very small, but the requirement for the piston and displacer to be in the same cylinder reduces the flexibility of the design [7].

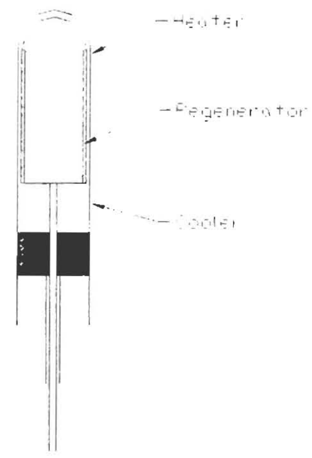

Figure 2. 2: β Type Stirling Engine

The Gamma type:

This design consists of a piston and displacer in separate cylinders. Heating and cooling are done in the cylinder containing the displacer. The mechanical friction associated with this design is potentially small and this also gives the greatest flexibility in terms of layout [7]. However the dead volume can become large because of the connecting passage.

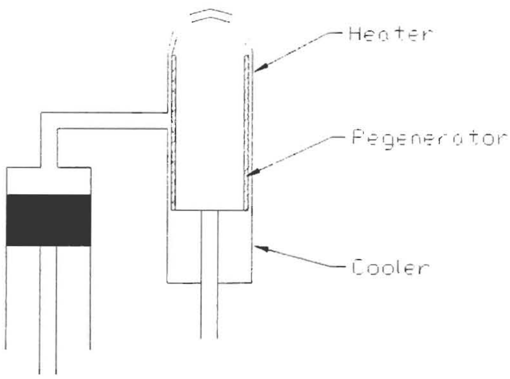

Figure 2. 3: γ Type Stirling Engine

2.1.1 Fundamental Elements of a Hot Air Engine

Regenerator:

This engine component is used to store heat energy during part of the cycle and to return it to the working gas during another part. The inclusion of the regenerator results in a substantial increase in engine efficiency, and reduces the loading on the heater and cooler. According to experiments carried out by Phillips, a well designed regenerator matrix can save up to 95% of the heat input by the heater [6].

It is usually composed of a matrix of fine wires or porous metal, but can be simply an enlarged metal surface.

Pistons and Displacers:

A piston is defined as having a large pressure and a small temperature difference between its top and bottom [5]. It's function is to utilise the expansion or contraction of the working fluid to produce motion. A Buffer space is often used on the underside of the power piston, it is pressurised to the mean or minimum cycle pressure to reduce loads on the mechanism [8].

A displacer is defined as having the same pressure top and bottom but a high temperature difference. Thus it does no work on the gas it only shifts the gas from one region to another. The length of the displacer should be about 3 times its diameter for effective isolation of the hot and cold zones or if the displacer is to act as the regenerator. [6] [5]

Both must necessarily be connected to some kind of mechanism arrangement to create the gas flow necessary for the cycle. The assumption of sinusoidal piston motion for a crank linkage is only very approximate and as such, every engine should be analysed in regard to its driving mechanism [5].

Due to the fact that the flow in a Stirling engine is controlled by volume variations, the piston and displacer (or the two pistons as in α type) must as such have different motions. The most simple is a ninety degree phase difference between piston and displacer. This however, produces only a rough approximation of the ideal cycle, therefore a number of other more complex mechanisms have been developed in an attempt to better reproduce the ideal cycle.

With an Ericsson type engine however, there is no requirement for the two pistons (or piston and displacer) to move relative to each other, because flow of the working fluid is controlled by the valves. Therefore, the resultant mechanism can be considerably simpler.

Working Gas:

Many types of working gas can be used e.g. Argon, Helium, Hydrogen or Carbon Dioxide. Selection is based upon cost, efficiency and required performance. Air is cheap and easily replaced if leaks occur, but because its heat transfer characteristics are inferior to that of other gases it can't be used for high-speed engines of high specific output [5].

Gasses with low molecular weights significantly reduce aerodynamic drag. Hydrogen and helium also have better heat transfer properties, but they pose difficult sealing problems, as most materials are porous to hydrogen [9]. Furthermore hydrogen embattlement of metals can occur rapidly.

Heater and Cooler:

Heating of the gas must occur during part of the cycle and with a closed cycle, cooling must occur in another. This is achieved by the presence of a continuously operative heater and cooler, which provide the hot and cold-sink temperatures for the engine. The heat supplied to the engine can take any form. For developing countries, this opens up endless possibilities in the form of biomass, coal combustion or solar power.

For adequate performance, a temperature ratio of at least three is recommended [6], this is best achieved using water cooling for the closed cycle engine. If fuel combustion is used for the heater, the process takes place outside the engine and occurs continuously at atmospheric pressure. A continuous combustion process is easier to control than an intermittent one and therefore complete burning of the fuel can be readily achieved, significantly reducing emission levels. [3]

Walker states that finned heat exchangers are best suited for small engines of less than 1kW, where simplicity and compact design are important. For larger engines, tubular heat exchangers are favoured [5].

Seals:

Sealing is by far the most difficult recurring problem in hot air engines. Hot air engines need seals to contain the working fluid and prevent the ingress of lubricant into the cylinder. The original engines of the1800's used only air and were barely pressurised, therefore sealing wasn't a problem, but the use of light gases such as hydrogen, or a highly pressured engine poses difficult fluid sealing problems. These problems are compounded by the fact that seals behave in a different manner from day to day, and seals that appear identical can behave in very different manners [5].

Seals are required to: [3]

- Effectively contain the highly pressurised or low molecular weight gases.
- Work without lubrication.
- Sustain a long life with a low maintenance requirement.
- Be readily replaced by relatively unskilled personnel.

Walker recommends the use of a lip seal fabricated from Rulon, a compacted PTFE material. However, Walker also states that these types of seals, together with the roll sock seal are unsuitable for production engines, as their leakage characteristics are not reproducible and they are expensive and difficult to manufacture. [5]

2.1.2 Performance Characteristics of Hot Air Engines

There is generally an overly sanguine view of the capabilities of hot air engines and there have been many proposals to convert internal combustion engines, using low-pressure air and a small furnace.

As a simple indicator, W. Beale proposed that the power output of many Stirling engines was directly proportional to the mean cycle pressure and conformed approximately to the following equation: [5]

$$P = 0.015p \cdot f \cdot V_0$$

Where: P = Engine power (Watts)

P = Mean cycle pressure (Bar)

f = engine speed (Hertz)

V_c = Displacement of power piston (cm^3)

Walker cites the following example to illustrate the point. Using a twin cylinder engine of 50mm bore and stroke, with air at a mean pressure of 2 bar running at 1200rpm will only produce about 75 watts. The mechanical friction of the engine will probably be greater than this and hence it will not work. Since this is very close to the type of engine envisaged for the project, the above illustration is illuminating.

Furthermore, it can be seen that hot air engine performance only approaches that of internal combustion engines (10-15 bar imep) when highly pressurised (50-100bar). Pressurising of the working gas results in a significant increase in performance due to the increase in the pressure excursion. Table 2.1 highlights this point.

Table 2. 1: Effect of Pressurising a Hot Air Engine

Pressure Ratio	Minimum Pressure Bar	Maximum Pressure Bar	Pressure Excursion Bar	Mean Pressure Bar
2	1	2	1	1.5
2	100	200	100	150

The theoretical efficiency of all reversible cycle heat engines is the same, and is given by the following equation: [10]

$$\eta_t = 1 - \frac{T_c}{T_h}$$

Where: T_h and T_c are the hot- and cold-sink temperatures.

The efficiency of all real engines must always fall below this value.

The best designed engines may achieve 60% of this maximum value, but this can easily drop to below 30%. If no high temperature materials are used, 600 degrees may be the maximum temperature attainable, thus reducing the efficiency even more. Hence, the selection of proper materials is key to the success of any external heat engine.

2.1.3 Imperfections

Hot air engines having a small specific power are very susceptible to losses. The literature indicates that the major losses associated with an external heat engine are: [5]

Dead Volume:

The unavoidable dead volumes associated with heat exchangers and the piston clearance volumes in an engine reduce the amplitude of the pressure ratio and with it the specific output of the engine. It is therefore vitally important to keep this void volume to a minimum, but at the same time still ensuring adequate heat transfer area.

Thermal Losses:

Thermal conduction along cylinder walls and shuttle heat transfer, due to the temperature gradient of the reciprocating displacer, reduce the temperature difference between the hot and cold end. Convection and radiation also serve to cool the hot end of the engine, further decreasing the temperature ratio. As the power and efficiency of an external heat engine are directly related to the temperature ratio, it is important to limit these heat losses with effective choice of displacer properties and suitable construction materials. Walker states that the shuttle heat transfer can be estimated from [5]:

$$Q_{sh} = 0.4L^2 \cdot k \cdot D \cdot \frac{(T_h - T_c)}{S \cdot Z}$$

Where L = Displacer stroke

k = Conductivity of gas

D = Displacer diameter

T_h = Heater temperature

T_c = Cooler temperature

S = Annular gap between displacer and cylinder

Z = Length of displacer

The design of the displacer should therefore be optimised to reduce this heat loss to a minimum.

Imperfect regeneration:

Due to heat losses, the exit temperatures from the regenerator will not be equal to the operating temperatures of the heater and cooler, resulting in a greater loading on these heat exchangers. It is therefore necessary to ensure that the heater and cooler are both capable of dealing with these extra requirements. The regenerator should also be designed for good heat transfer characteristics so as to keep this temperature difference to a minimum.

According to Rallis, for low regenerator effectiveness, the energy conversion efficiency for a Stirling cycle can be less than that for a near Stirling cycle that uses ploytropic processes. This is because a large fraction of the heat supplied during the isothermal expansion in a Stirling engine is rejected as the regenerator effectiveness decreases [11].

Mechanical friction:

Friction is a major problem with hot air engines, as the specific output is so low that it can easily be used entirely in overcoming friction. Rings, seals and bearings all serve to consume useful work from the engine. The number of seals in the engine should therefore be kept to an absolute minimum, and low friction seals should be used as far as possible. The mean rubbing velocity of sealing rings for a given size engine can be reduced by enlarging the bore, however that increases seal diameter and hence the friction and the leakage path length. A good compromise is therefore to make the stroke half the bore of the cylinder [5]. Senft also states that the buffer pressure, the shape of the thermodynamic cycle and the mechanism effectiveness also have a significant effect on the mechanical efficiency of an engine [12].

Fluid friction:

Pressure drops through heat exchangers and the work involved in compressing and displacing the fluid further reduces the engine output. Flow losses cause a decrease in the area of the expansion curve. The regenerator, heat exchangers and other ducting must therefore be designed so as not to obstruct the flow of air in the engine. In a cycle where compression and expansion are isothermal and there are no friction losses, the difference in the area of the expansion and compression space diagrams will be equal to the area of the P-V diagram for the total working space. In reality aerodynamic flow losses in the heat exchangers cause a difference in pressure of the working fluid in the compression and expansion spaces and thus a difference of about 40% can be expected.

In summary:

A Hot air engine is a cyclical heat engine that uses an external heat source, heat exchangers, pistons and a gaseous working fluid contained within the engine to convert heat to mechanical work by volumetric expansion. There are various cycles and design arrangements from which these engines can function, all of which are theoretically capable of high power output and efficiency, but in reality the performance can be considerably lower than expected.

2.2 Cycle analysis:

The main difficulty in analysing the cycle in an engine lies in the fact that the various working gas particles undergo widely different thermodynamic cycles. [5]

The analysis of engine cycles can be split into two categories:
Decoupled or second order analysis: This is where the working processes are considered independently, they are then summed at the end of the process and the loses subtracted to yield the predicted result.
Coupled or third order analysis: This is when all the thermodynamic and gas dynamic processes are combined to give a more realistic conceptual description of the working processes. [13]

The ideal cycle simulation contains many idealisations, and it is only suitable for the most eliminatory of design calculations [5]. It assumes that all the working fluid undergoes the same thermodynamic processes throughout the cycle and that the pistons move in some discontinuous fashion. It does however provide an opportunity for a comparison of different cycles.

A more realistic analysis was developed by Gustav Schimt in 1871, it soon became the classical analysis used. The Schimt analysis is still idealised however and the indicated performance can be expected to be no more than 60% of what is predicted using that analysis.[5]
The analysis assumes isothermal compression and expansion, when in the actual cycle the processes are nearer adiabatic, especially above 1000rpm. The analysis ignores all other losses and hence yields the carnot cycle efficiency, although it uses a continuous piston motion making the P-V diagram more realistic.

Finkelstein in 1960 developed a model that allowed the compression and expansion processes to be evaluated by means other than an isothermal calculation. The Schmit analysis then became a special case of this analysis, the Finkelstein adiabatic model being another. This is the preferred level for routine performance calculations [5]. This work culminated in the nodal analysis technique.

In the nodal analysis technique (the third order method), the machine is retained as a unified complex entity and subdivided into a set of one dimensional cells or nodes. Each cell is described by a set of differential equations of continuity, energy, momentum and state, the overall set of equations being solved numerically [14]. Within the nodal analysis technique, there are a myriad of different methods all differing slightly on the exact forms of the equations used and whether or not effects such as gas inertia are included.

However none have yet been validated against a fully documented publicly available industrial standard machine. [14]

In summary:
The analysis of hot air engine cycles is difficult because the various working gas particles undergo widely different thermodynamic cycles. A number of different techniques to carry out an analysis are available to the designer. Depending on the required accuracy or simplicity a highly theoretical analysis simplified by a number of assumptions, or a rigorous investigation taking into account all the major losses and imperfections in the real cycle can be used.

2.3 Furnaces

The majority of development in furnace design has been on large industrial types for large-scale power generation in boilers. These large furnaces generally burn high-grade coal and make use of one of two methods to fire the furnace. [15]

A chain grate or similar rotary type hearth: A long grate made up of interlinked chains runs the length of the furnace on which the charge burns. They use an auger type feeding mechanism to keep the grate stocked. Some smaller furnace designs also use an external pusher mechanism to move the charge through the furnace.

The other alternative is pulverised fuel firing: The coal is pulverised and then blown continuously into the furnace with a jet of air. This requires large amounts of processing and expensive equipment to handle it.

The small furnace types that are available are those used for domestic heating purposes and as such are elaborate designs and generally fed manually. Very little information is available on small multi-fuel furnaces.

Parker states that the selection of furnace designs and materials should be aimed at a minimal overall cost of construction, maintenance and fuel over a projected life [16]. Heat losses should be reduced by the use of effective insulation around the outside of the furnace. Furnaces vary widely in construction depending on the application of the heat released and whether it is direct or indirect heating of the medium.

The furnace design for a given performance involves the following processes [15]:

- Determination of the amount of liberated heat which must be utilised to meet requirements.
- Allocation of heat to be absorbed in the convection and radiant section of the furnace.
- Determination of the heat transfer rate and the surface area in the convection and radiant section.
- Determination of the type of fuel to be burned and the air fuel ratio. The theoretical flame temperature can then be determined from this using [16]:

$$\dot{Q}_f = \dot{m}Cp(T_f - T_0)$$

Where T_f is the theoretical flame temperature and T_0 is the inlet air temperature

In practice the actual flame temperature will be considerably lower than this value.

For highly pressurised hot air engines running at high speed, the limiting heat transfer is on the outside of the heater [5] thus a very good heat transfer is required. Radiation is the dominant heat transfer mechanism in most conventional furnaces as the combustion products are not dense or moving fast enough for adequate connective heat transfer [15]. However, with the use of fluidised beds, heat transfer coefficients as high as 600 W/m^{2}K are possible through conduction of the fluidised particles. [17]

The radiant heat transferred for a furnace can be estimated using [16]:

$$\dot{Q}_g = A_1\left(\frac{1}{\frac{1}{\varepsilon_1}+\frac{A_1}{A_t}\left(\frac{1}{\varepsilon_g}-1\right)}\right)\sigma\left(T_g^4 - T_1^4\right)$$

Where ε_1 and T_1 are the emissivity and temperature of the receiving surface

ε_g and T_g are the emissivity and temperature of the furnace air

A_1 is the area of the receiving surface and

A_t is the total surface area.

However, tube bank geometry and other factors should be taken into account when using this approach.

In Summary:

Little information is available on the design of small-scale furnaces, but the design procedure is much the same as it is for large industrial furnaces. The most effective heat transfer mechanism in most furnaces is radiation, therefore the design should be aimed at optimising this.

2.4 Heat Transfer

The difficulties in the design of heat exchangers for hot air engines arise from the lack of knowledge on the effect of the periodically reversing flow of the working fluid [18]. The flow direction can reverse two times a second or more, with the pressure, density and volume variations changing just as quickly. This is aggravated by the fact that no particle of fluid ever passes entirely from the compression space to the expansion space, or even right through the regenerator [5] see figure 2.4. The energy is passed by conduction from particle to particle instead. Thus it is often best to assume a reasonable value based on the average flow rate.

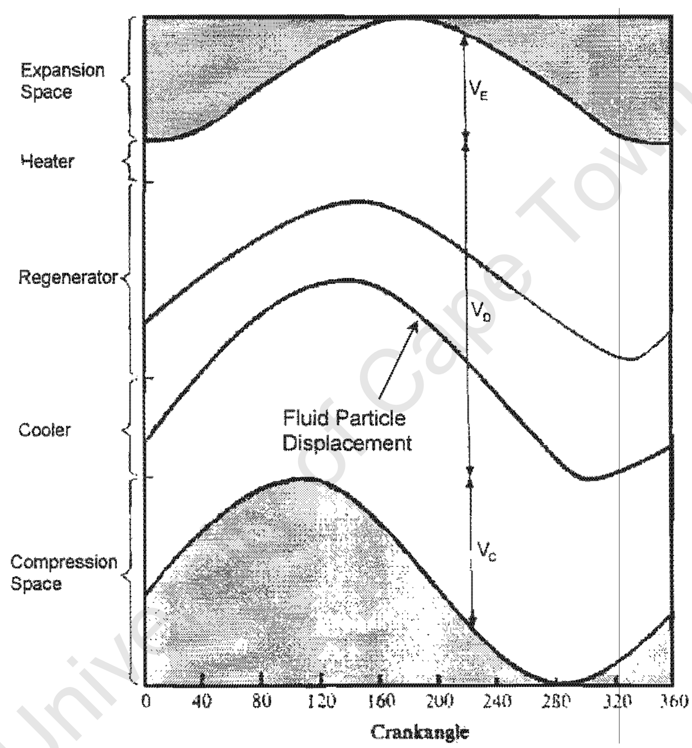

Figure 2. 4: Fluid Particle Displacement

(Source: "Stirling Engines", G. Walker, 1980)

Very little work is available on transient flow in hot air engines and the standard formula:

$$Nu = 0.036\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{\frac{1}{3}}\left(\frac{d}{L}\right)^{0.055} \qquad [19]$$

For calculating the heat transfer in a tube is inapplicable to this flow condition.

D. Donne et al suggested the use of the following formula to calculate the heat transfer in the heater tubes.

$$Nu = 0.021\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}\left(\frac{T_h}{T_w}\right)^{-0.5} \qquad [18]$$

Where T_w = Tube wall temperature

T_b = Mixed mean temperature of the fluid

Re = The Reynolds number for the flow regime

Pr = The Prandle number for air

Heat transfer in an engine cylinder is also a little understood topic, and as such, many of the formulas for calculating it give a very rough average value. R.S. Benson, suggested:

$$h = \frac{0.76 k_{air}\,\mathrm{Re}^{064}}{D} \qquad [20]$$

Where D is the cylinder bore

And the Reynolds number is obtained using the average piston sliding speed.

2.4.1 Regeneration

Most of the work done on regenerators is for air liquefaction or gas separation where the blow periods can be anywhere from ten minutes to several hours [5]. This work has been adapted to gas turbines, but is still inapplicable to hot air engines because the blow times in Stirling engines are very short, ten times less than the permissible minimum in gas turbines. This makes any assumptions about constant values questionable. Nodal analysis techniques can yield satisfying results by dividing the regenerator into many small cells then carrying out a rigorous thermodynamic analysis on each. The blow period is defined as the time taken for the total quantity of fluid to pass any point in the regenerator [5].

Ideal regeneration is the case when air entering or leaving the regenerator matrix does so at one of two constant temperatures [5]. This is only possible if the heat transfer or heat transfer area of the matrix is infinite. Alternatively the heat capacity of the fluid must be zero or the heat capacity of the matrix must be infinite. In reality the temperatures vary cyclically because the heat transfer in the regenerator is finite.

To simplify analysis, Nusselt presented 4 cases for regenerator operation [5]:

1) The thermal conductivity of the matrix is infinite and there is no temperature difference in the matrix. (This type would have poor performance).
2) The thermal conductivity of the matrix is infinite parallel to the direction of flow, and finite perpendicular to the direction of flow. (This condition would correspond to a short regenerator with thick walls)
3) The thermal conductivity of the matrix is zero parallel to the direction of flow, and infinite perpendicular to the direction of flow
4) The thermal conductivity of the matrix is zero parallel to the direction of flow, and finite perpendicular to the direction of flow

Saunders and Smoleniec made further assumptions for analysing regenerators:

- The specific heats of the working fluid and the matrix do not change with temperature.
- The fluids flow in opposite directions and have inlet temperatures constant over the flow direction and with time.
- The heat transfer coefficients and fluid velocities are constant with space and time.
- The rate of mass flow is constant during the blow period.

The most effective regenerators are based on experimental results and so the following guidelines are set out to aid design [5]. It is however impossible to satisfy them all.

- For maximum heat capacity – Large, solid matrix
- For minimum flow losses – Small, highly porous matrix
- For minimum dead space – Small, dense matrix
- For maximum heat transfer – Large, finely divided matrix
- For minimum contamination from oil etc – No obstruction
- The Ratio of heat capacity of the matrix to that of the gas should be a maximum

K. Mansoor and G. Rice found that a reduction in wire diameter for the same weight of matrix produced a more effective regenerator. Increasing the weight of the matrix for a given size of gauze also yielded better results, but at a diminishing rate. At lower speeds a matrix with a lower porosity was beneficial due to a gain in the heat transfer area. At higher speeds this was outweighed by increased fluid friction. [21]

Walker suggests that it is unnecessary to include a regenerator in small low speed engines (below 50mm bore, less than 5-6 mean pressure running at less than 1000rpm) [5]. He states that experience shows that the removal of a regenerator in such an engine nearly always improves its performance. This is because the gains due to the reduction in dead volume, and to a lesser extent the decrease in fluid friction effects, more than offset the loss of thermal capacity and the area for heat transfer in the regenerator matrix.

Walker further suggests the use of a displacer type regenerator with a very thin wall, and an air gap between it and the cylinder of between 0.4mm and 0.8mm. The limits of this annular ring type regenerator are not known, but it probably becomes less effective as the bore, pressure or engine speed is increased.

In Summary:
Little is known about the mechanisms of heat transfer in a hot air engine due to the periodically reversing flow conditions and fluid particle behavior. A number of formulas exist from which to obtain estimates of heat transfer coefficients, but the best results are obtained from actual experimentation

3. Initial Investigations

An experimental engine was devised at the beginning of the year, its purpose was to provide insight into the functioning of an unconventional external heat engine and to test the validity of theoretical predictions made with the thermodynamic model. Figure 3.1 shows this engine.

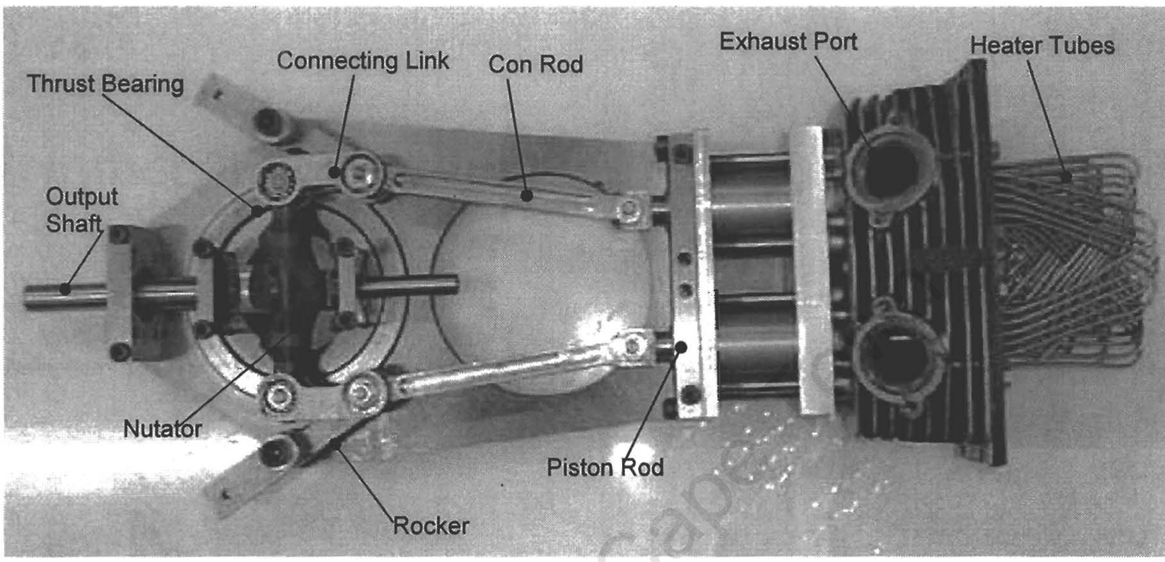

Figure 3. 1: Partially Assembled 3-Process Engine

This experimental engine was a scaled down version of the five kilowatt target engine, potentially producing at best only a few hundred watts. The engine was designed to function on a cycle devised by Dr A. Yates, consisting of a polytropic compression, a polytropic expansion and a constant volume process. Hereafter referred to as the 3-Process cycle.

Heat rejection was achieved by utilising a two-stroke principal to exchange the hot exhaust gas with cold recharge gas. This process would have less friction associated with it than the displacement of the gas used to achieve heat rejection in a Stirling type engine.

The engine consisted of two cylinders interconnected by a number of heater tubes, each piston alternated between compression and expansion and therefore needed an unconventional piston motion to attain the desired cycle. The motion allowed one piston to complete its compression and expansion stroke, while the other remained at top dead centre. Thus the engine would complete two full cycles per revolution of the driveshaft.

The theory was to maximise the indicated mean effective pressure, even at the expense of efficiency. This was due to the fact that the overall efficiency of an engine is given by:

$$\eta_o = \eta_m \cdot \eta_i$$

Where: η_m = Mechanical efficiency $= \dfrac{IMEP - FMEP}{IMEP}$

η_i = Thermodynamic efficiency

Thus a high IMEP will not only give a high engine output, but also improve the overall efficiency of the engine.

<u>Operation of engine</u> (See figures 3.2 and 3.3)

The piston moves up from bottom dead centre, closing the exhaust and transfer ports thereby compressing the air in the cylinder and forcing it into the heater tubes. Once at top dead centre, (2) the driving mechanism holds the piston stationary throughout the next 180 degrees rotation. On the underside of the pistons, the buffer zone is open to atmosphere, so a fresh charge is drawn in. Pressure in the heater tubes builds up due to the air heating and expanding. This expanding air causes the other piston to move down covering the inlet port and thus the air in the buffer zone is compressed.

Just before Bottom Dead Centre, the exhaust port is opened (3), which causes the hot air in the cylinder (still above atmospheric pressure) to exhaust (1). Shortly afterwards, while the exhaust port is still open, the piston uncovers the transfer port. This causes the compressed air in the buffer zone on the under side of the piston to flow into the cylinder. This helps to further purge the hot air and to fill the cylinder with a new charge of cold air. The piston moves upwards again to top dead centre and the cycle repeats itself using the other cylinder and piston.

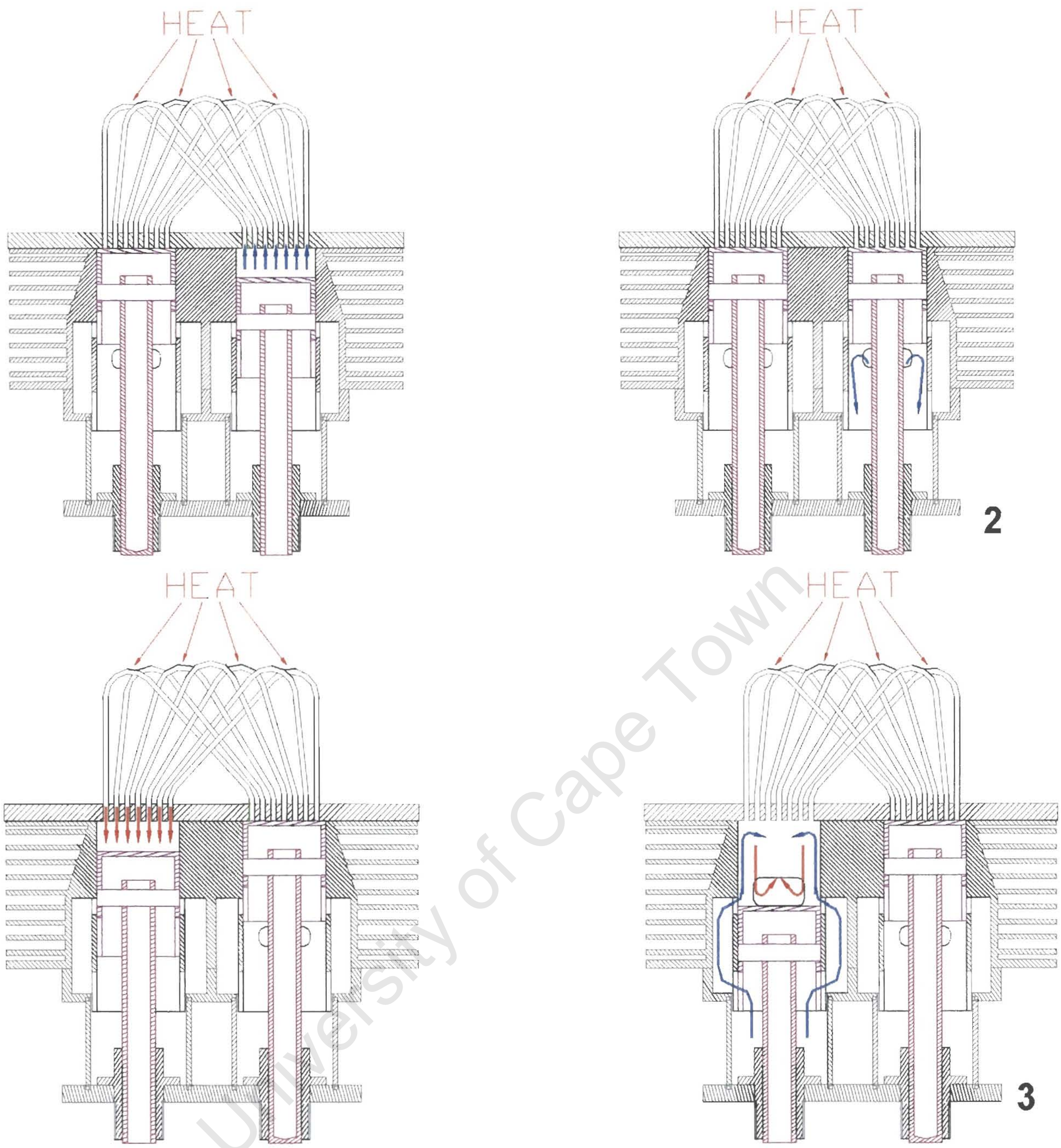

Figure 3. 1: Operation of the Cycle for the 3 Process Engine

This process produced the following Pressure – Volume Diagram:

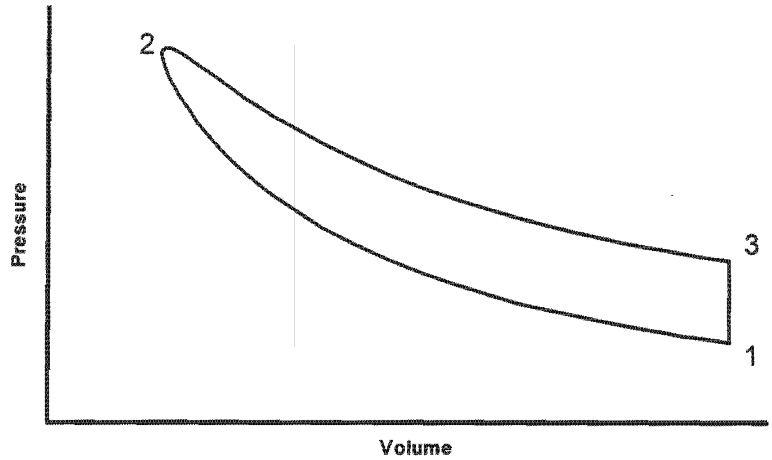

Figure 3. 3: Theoretical P-V Diagram for 3 Process Cycle

The engine was designed and constructed using standard two-stroke motorbike pistons and barrels. A nutrator being the key to the desired motion. Full details of the design are given in Appendix B.

Exhaustive tests were carried out to investigate the operating principals of this engine. Results were useful although the engine failed to run and the thermodynamic model did not reliably predict the pressure – volume diagram, see figure3.4. The experimentation, together with the results is presented in Appendix B.

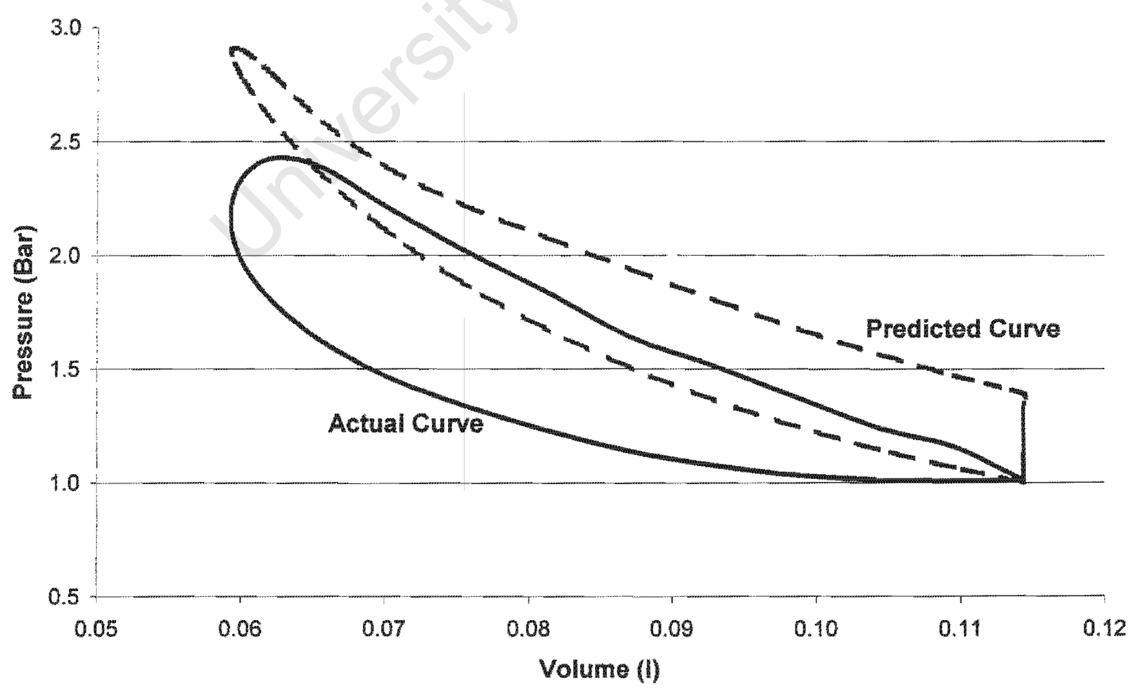

Figure 3. 4: Actual and Predicted P-V Curve

The experimentation with the 3-process engine revealed a number of weaknesses in its design and operating principals. These are summarised in table 3.1 together with recommendations for the next generation engine.

Certain key elements were valid however and should be retained:

- Optimisation of the mean effective pressure even at the expense of efficiency.
- The necessity to pressurise the engine.

Table 3. 1: Summary of Faults with 3-Process Engine

ITEM	COMMENTS	CONCLUSION
2-stroke scavenging	Very poor purging of air. Hot gas expanding from heater tubes into cylinder made situation even worse. (Initial temperature too high)	Forced intake / exhaust for open cycle, good cooling for closed cycle
Flow reversal	Average wall temperature bad for initial and final temperatures of cycle	Uniflow design (except in regenerator)
Friction and speed ratios	Sixty out of 180 degrees of crankshaft revolution for compression / expansion too low. High sliding speeds, so high friction	1:1 ratio as far as possible
Expansion ratio	Wasting useful pressure	Expand only whilst cylinder Pressure is greater than the FMEP
Furnace temperature	900 degrees Celsius is sensible, over 1000°C is too hot	Design for 800 degrees
Pressure and Friction	Only 1/3 crankshaft revolution used for compression and expansion.	Ensure buffer pressure balances system
Tubes and dead volume	$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$ Therefore dead volume is bad in the heater	No dead volume in heater if possible
Mechanism	Too complex, and excessive acceleration	Beta or gamma type engine preferable for high harmonics
Engine temperature	Rings too hot	Rings and valves in cool zone (regenerator type implicit)
Furnace	No need for double skin at this stage. Clinker removal essential. Difficult to dissemble.	No nuts or head bolts inside furnace

After careful consideration of these observations, it was decided that an entirely new engine concept and cycle would be more appropriate rather than attempting to correct the problems with the 3-Process engine.

4. Theoretical Development

Using the recommendations obtained from the first generation engine, a 'near Ericsson' cycle engine concept was conceived.

The design consisted of a compressor and an expander, having different bores but the same stroke, separated by a high pressure 'buffer' zone. The hot end of the expander was isolated with the use of an elongated displacer that also functioned as a regenerator. Thus the hot and cold zones of the engine were clearly separated. Flow between the compressor and expander was controlled by solenoid valves, with heat exchangers in the intermediate reservoirs to cool the air flowing between these two zones, see figure 4.1.

This design has a number of advantages over the 3-process engine and circumvents all the problems listed in table 3.1. It was further felt that the Ericsson cycle would be more appropriate than the Stirling cycle in meeting the requirement to function as a rural power source. This was for the following reasons:

- Being an open cycle, cooling for the Ericsson engine could take place outside the engine with no size or time limitations. The cooling required for a Stirling engine could only be carried out effectively and quickly by water cooling, a feature that was considered undesirable for rural Africa.
- With an open cycle, the cold inlet air is as cold as possible, thus increasing the temperature ratio and hence the efficiency and power output.
- The dead space for an open cycle is less than for a closed cycle, because the cooler is external to the engine.
- The drive linkage is much simpler, as the compressor and expander can operate in phase. This is because flow between the two is controlled by valves, unlike the Stirling engine, where a complicated linkage is required to ensure the piston and displacer function with a phase difference.
- The use of valves increases flexibility in flow control, timing and presents the possibility of higher pressure ratios than can be obtained using a Stirling type engine.

These advantages were considered to outweigh the requirement for inlet and exhaust control and a separate compressor necessary for the Ericsson engine.

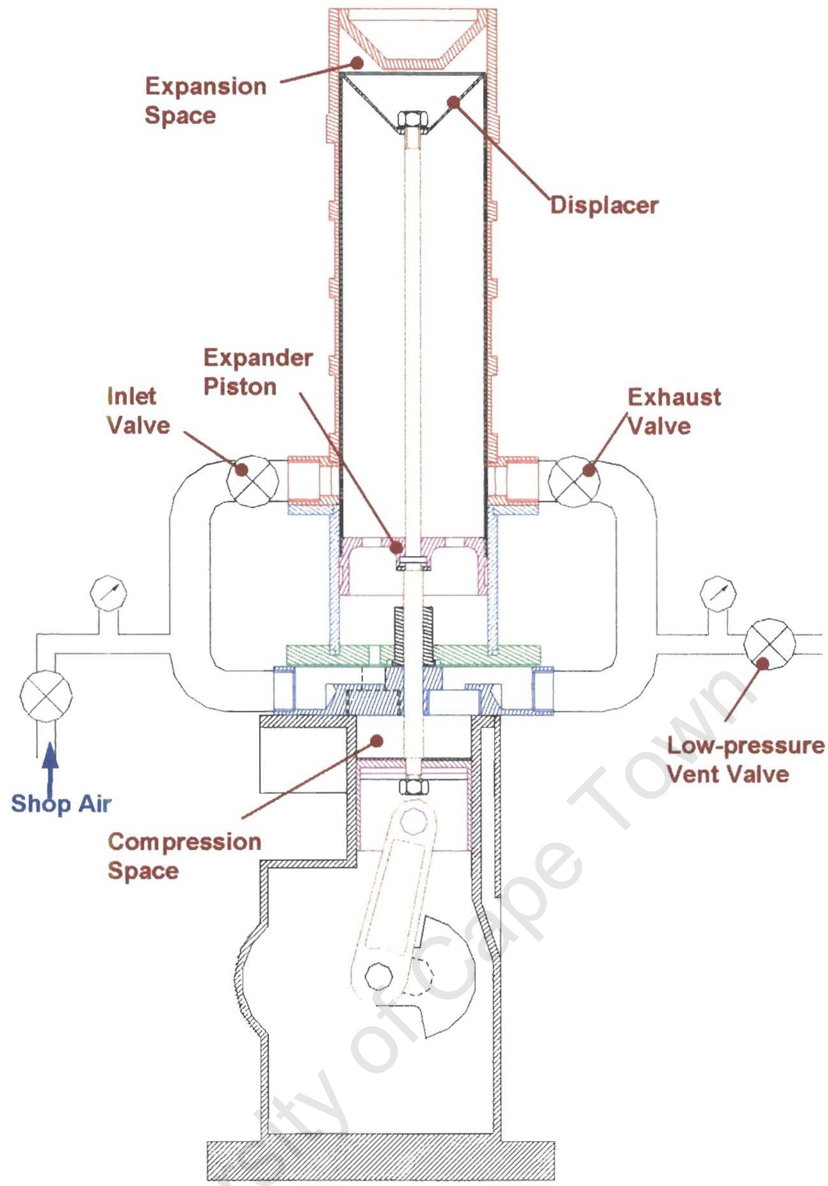

Figure 4. 1: Sketch of Assembled Truncated Ericsson Cycle Engine

4.1 Cycle Description:

The actual cyclic processes and piston motions, shown if figure 4.2, corresponding to the points in figure 4.3 are as follows:

At Bottom Dead Centre **(1)** the exhaust valve is open and purges hot air from the cylinder as the piston moves upward. Some time before Top Dead Centre, the exhaust valve closes **(2)** and thus the remaining air in the cylinder is compressed close to the delivery pressure. At TDC **(3)** the inlet valve opens, abruptly raising the pressure to that of the delivery pressure in the reservoir **(4)**. While the inlet valve is still open, the piston moves downwards, maintaining the cylinder at the delivery pressure. At point **(5)** the inlet valve closes, and the air expands polytropically to near the exhaust pressure. At BDC **(6)** the exhaust valve opens, dropping the pressure to the level of the low-pressure reservoir again **(1)**.

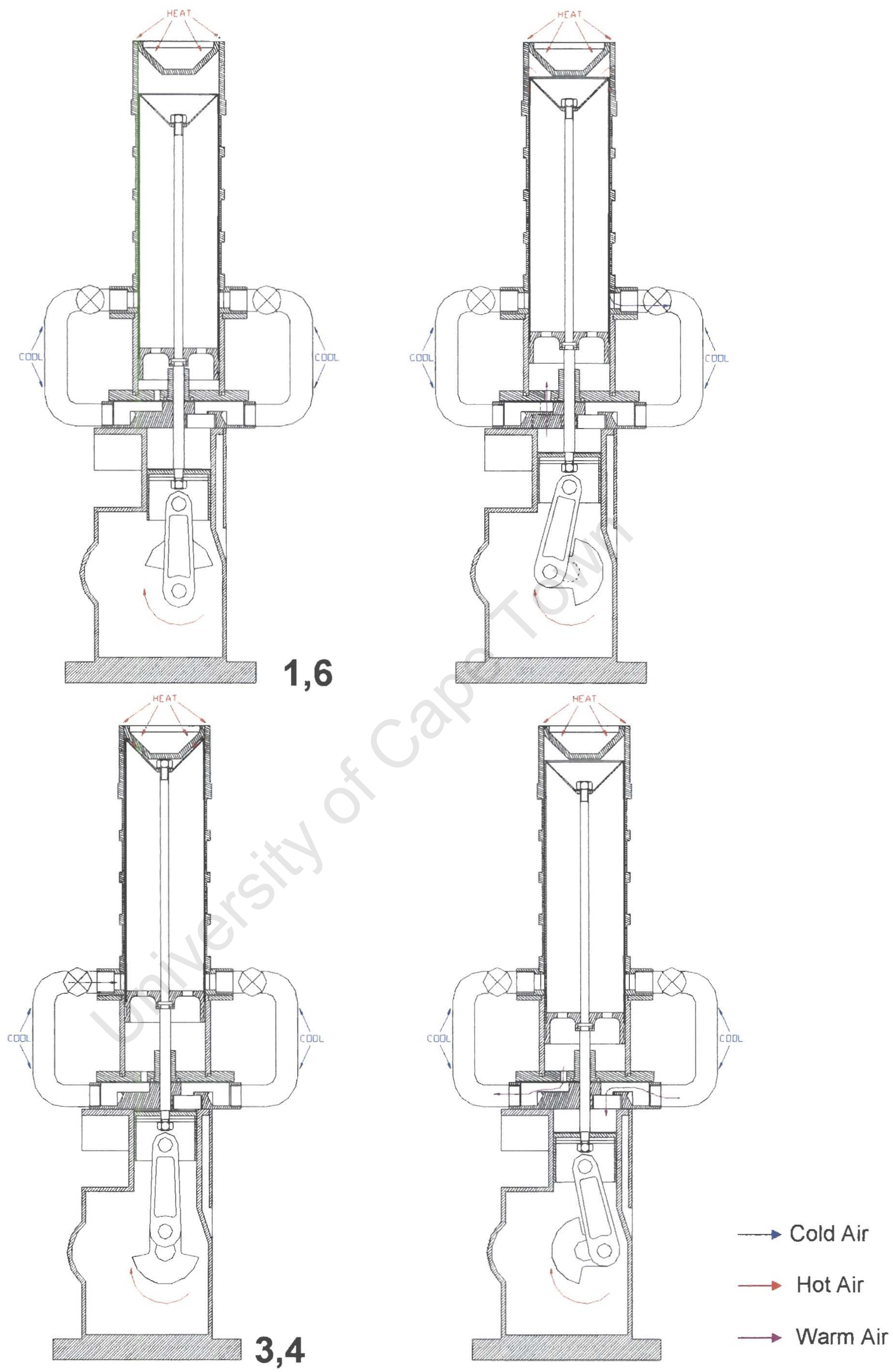

Figure 4. 2: Schematic of Cycle Operation

4.2 The Ideal Truncated 'Ericsson Cycle'

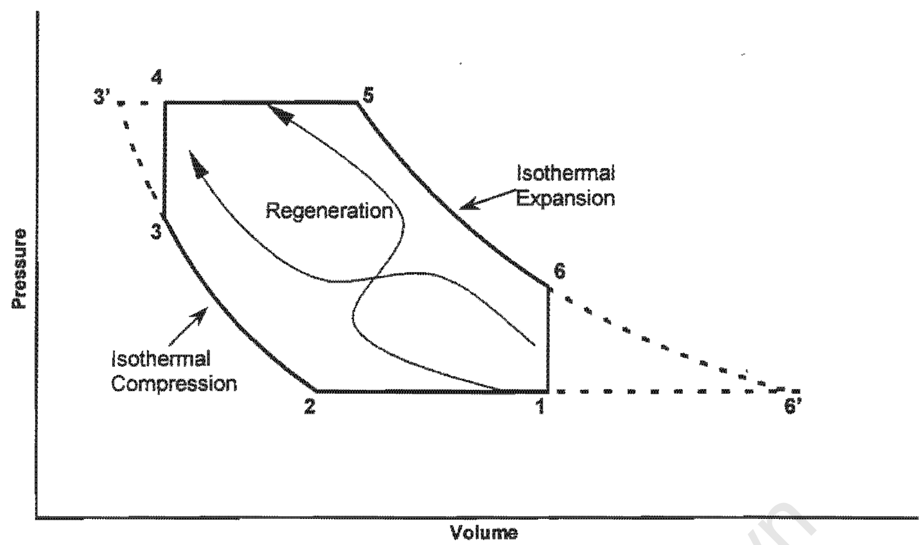

Figure 4. 3: The Ideal Truncated 'Ericsson Cycle'

For practical purposes, the expansion and compression processes were truncated by modifying the valve control. This significantly improves the utilisation of the working volume within the Ericsson engine with very little detriment to the thermodynamic efficiency. The expansion would be cut off at the frictional mean effective pressure of the engine because expanding to a pressure below that value does no useful work so

The idealised cycle is composed of six distinct thermodynamic processes. In reality, the processes are not completely isolated, but are simply identifiable parts of the one cyclic process.

1-2. Constant Pressure Heat Rejection (Regeneration)

Heat is removed from the working gas and stored in the regenerator.

2-3. Isothermal Compression

Heat produced by the compression is simultaneously removed at the cold-sink temperature.

3-4. Constant Volume Heating

The working gas is heated, as the volume is constant the increase in temperature causes the pressure to rise.

4-5. Constant Pressure Heating (Regeneration)

Heat stored in the regenerator is given up to the working gas.

5-6. Isothermal Expansion

The gas is heated at the hot-sink temperature and expands at a constant temperature.

6-1. Constant Volume Cooling

Heat is given up to the cold-sink, thereby reducing the pressure of the gas, as its volume is kept constant.

4.3 Computer Simulation:

The engine was modelled with the use of an Excel spreadsheet employing an approach similar to the standard adiabatic analysis technique used for Stirling engines. Appendix C contains the full details of the computer model.

For the analysis, the engine was assumed to made up of three distinct zones; a compressor zone, an expander zone and a regenerator zone. An analysis of the working gas in each of these zones was then performed for every degree of crankshaft revolution. See flow diagram in figure 4.3.

The following calculations were made for the expansion zone; heat transfer from the external environment, the work done, the change in internal energy and the enthalpy transferred to or from the neighbouring zone. The outlet temperature and the mass of the air were computed for the regenerator, and the flow through the valves and their response time were also modelled. The pressure, assumed uniform throughout the expander and regenerator was then adjusted to make the mass in the system constant.

The compressor was modelled separately using standard polytropic processes and the piston and crank linkage geometry from the standard engine base used.

The fundamental design variables and the principal results of the model are listed in tables 4.1 and 4.2

Table 4. 1: Design Variables

Parameter	Value	Unit
Delivery (High) Pressure	12	Bar
Exhaust (Low) Pressure	5	Bar
Initial Temperature	20	°C
Engine Speed	200	rpm
Heater Wall Temperature	700	°C
Diameter Displacer Piston	75.5	mm
Length Displacer Piston	250	mm
Diameter Expansion Cylinder	77	mm
Piston Clearance	3mm	mm
Regenerator inlet Temperature	30	°C
Inlet Valve Opening Position	0	Degrees from TDC
Inlet Valve Opening Position	80	Degrees from TDC
Inlet Valve Opening Position	170	Degrees from TDC
Inlet Valve Opening Position	280	Degrees from TDC
Diameter Valve Throat	12.7	mm
Valve Response Time	21	msec

Table 4. 2: Model Output Parameters

Parameter	Value	Unit
Compression Work	38	J
Expansion Work	67	J
Total Work Done	29	J
IMEP	1.7	Bar
Thermodynamic Efficiency	42	%
Power	144	W

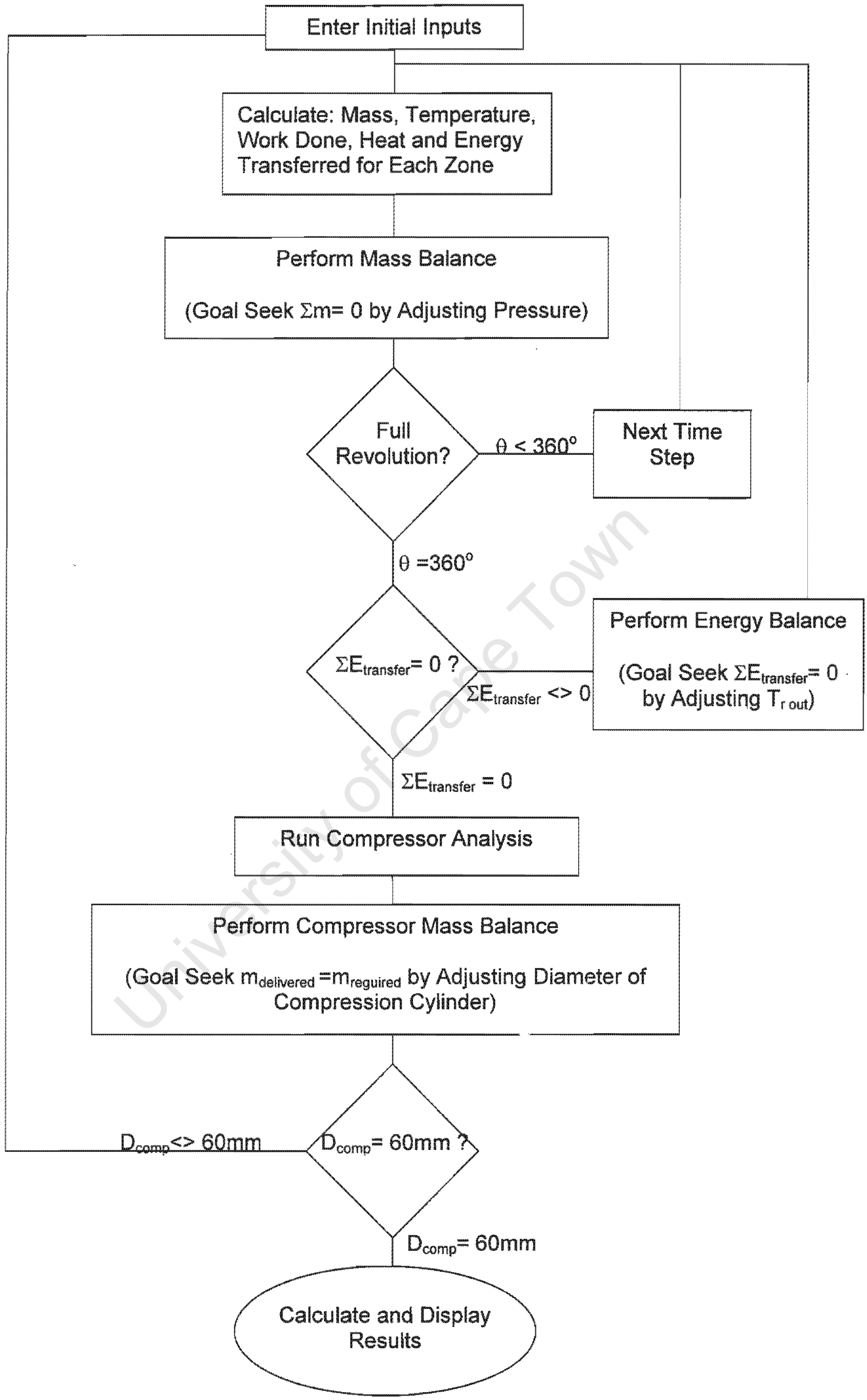

Figure 4. 4: Flow diagram of Computer Model Operation

4.4 Actual Cycle

The predicted Pressure-Volume diagram for the expansion space and compression space could be obtained from the model. These two were then combined by adding the two volumes corresponding to the same pressure. This produced a P-V diagram for the combined cycle.

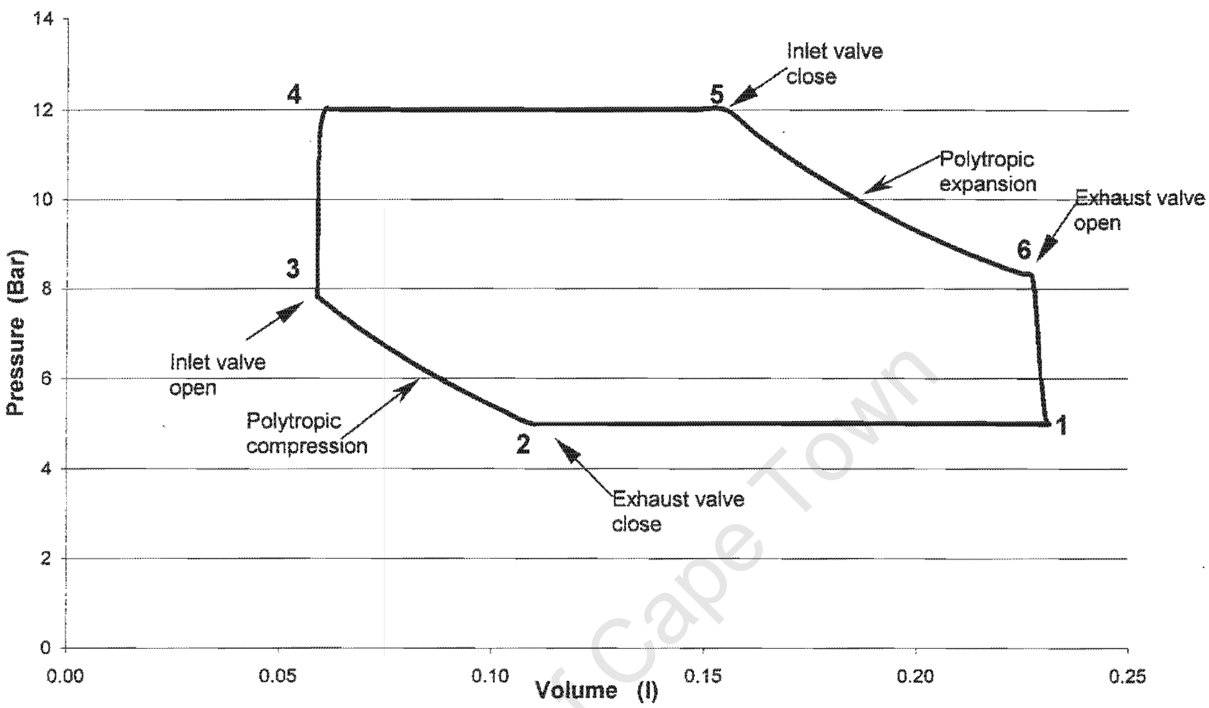

Figure 4. 5: Predicted P-V Diagram for Expansion Zone

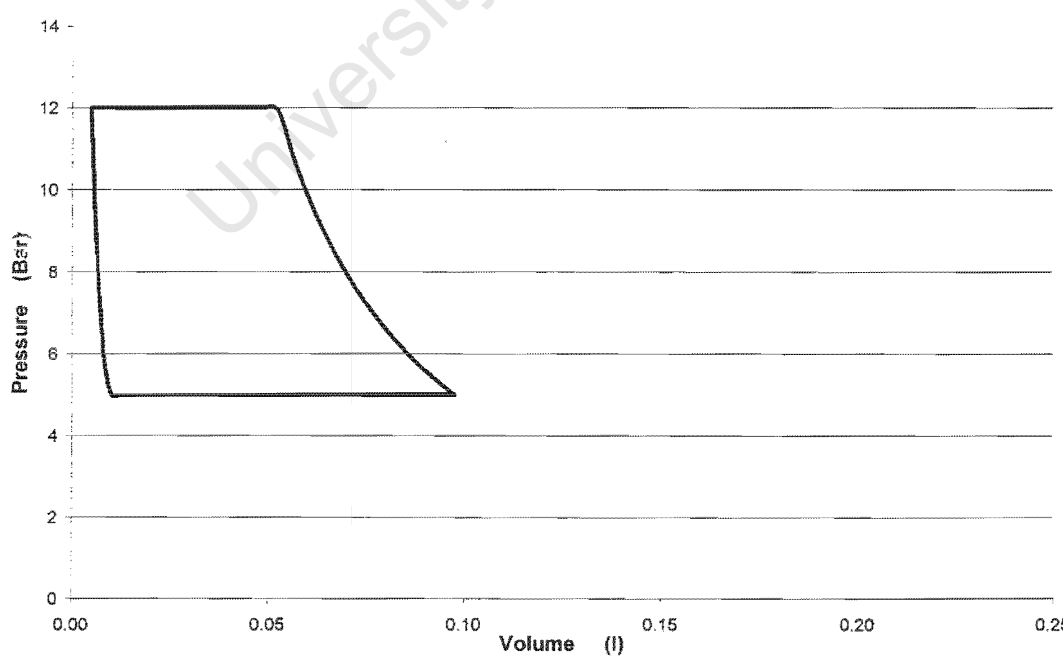

Figure 4. 6: Predicted P-V Diagram for Compression Zone

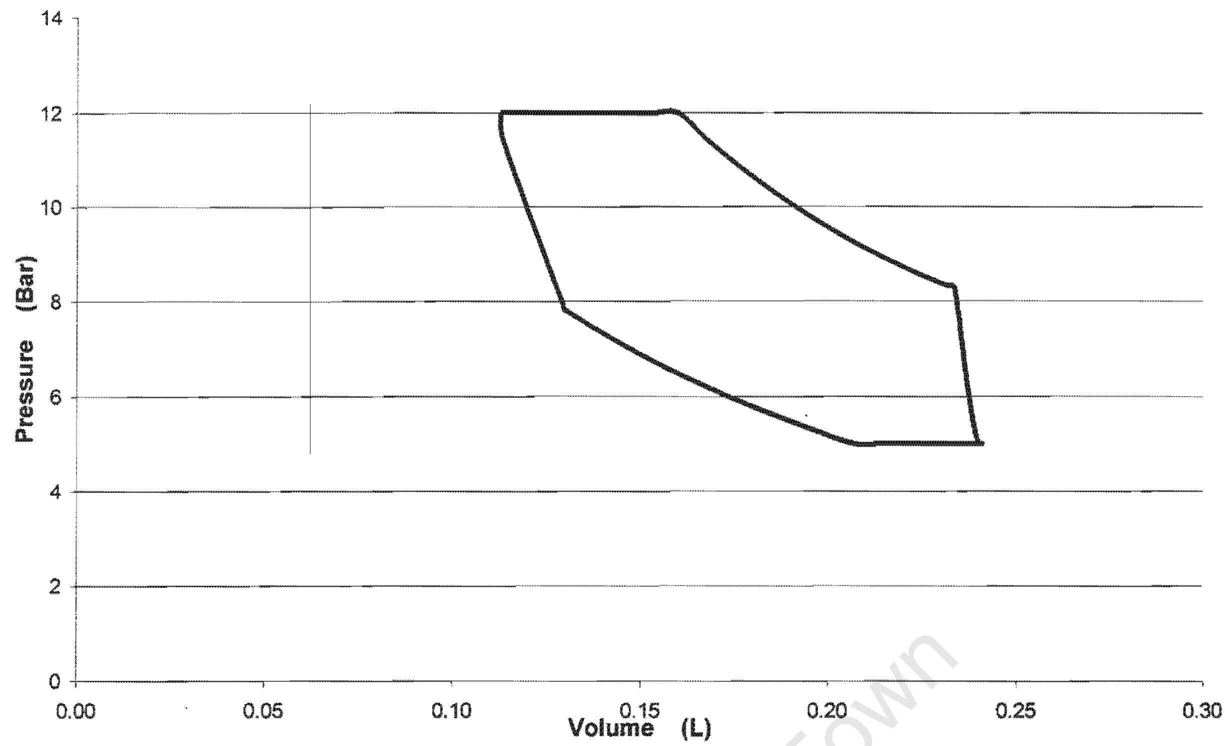

Figure 4. 7: Predicted P-V Diagram for Entire Cycle

Using the model, a number of different engine parameters could be investigated and their effect on engine performance examined.

Peak Cycle Pressure

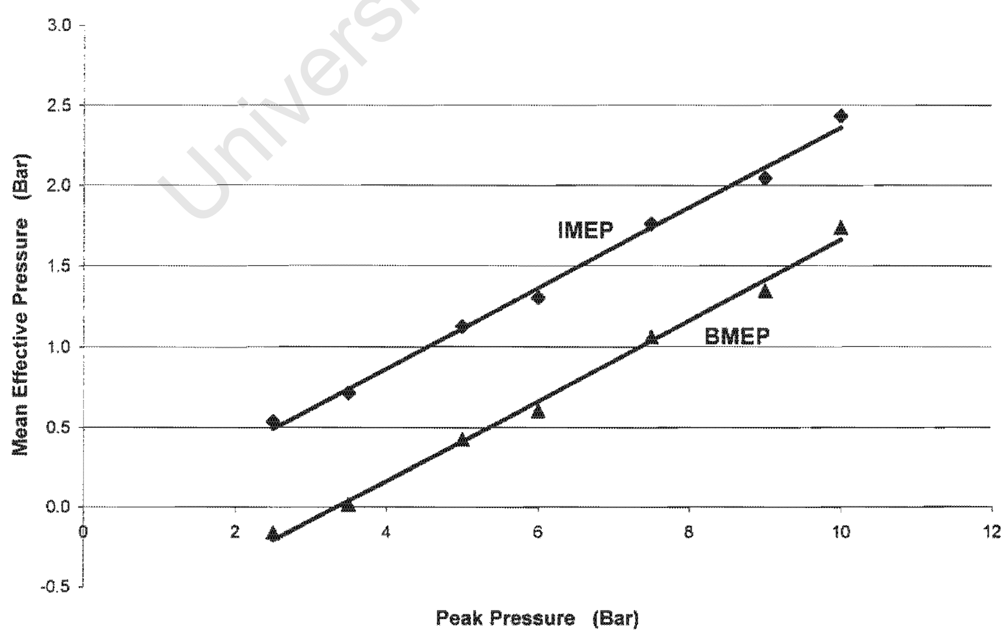

Figure 4. 8: MEP against Peak Cycle Pressure for Engine

From this linear relationship the Beale number for the engine could be calculated:

$$\frac{P}{p \cdot f \cdot V_0}$$

This produced a Beale number of 0.024, which is slightly higher than that predicted for a Stirling engine, about 0.015.

The BMEP line on this graph is not a true indication however, as the model assumed the FMEP to be a constant, but in reality this value would rise with the peak pressure value, reducing the slope of the line.

Heater Temperature

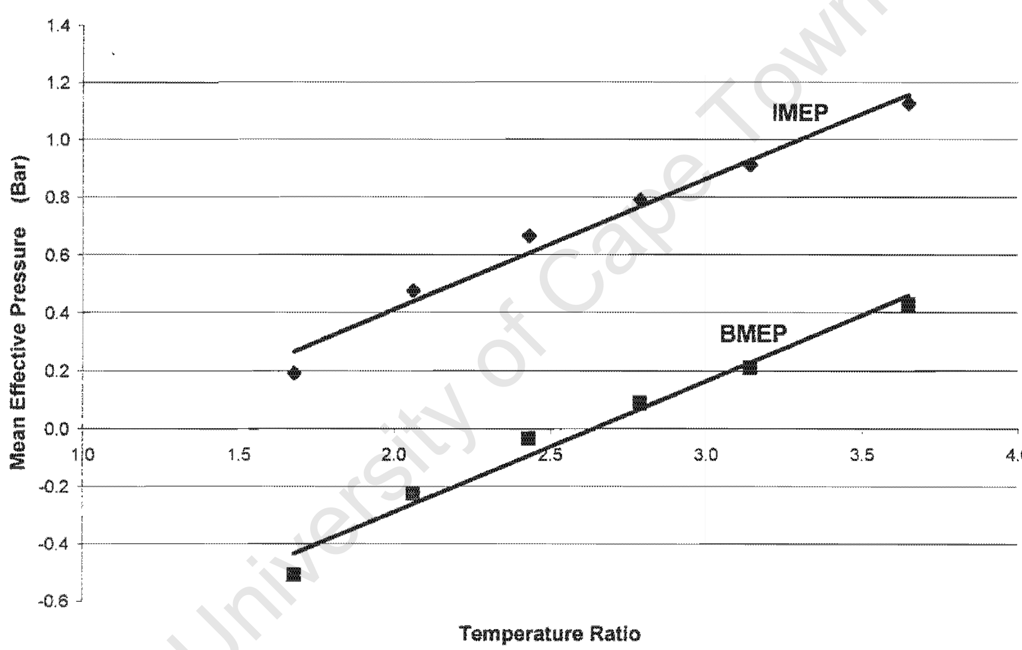

Figure 4. 9: MEP against Temperature Ratio for Engine

This is the most significant factor, as the gradient of the curve is slightly more than of the peak pressure line, indicating the importance of good heat transfer at the hot end and cooling of the inlet air.

Dead Volume

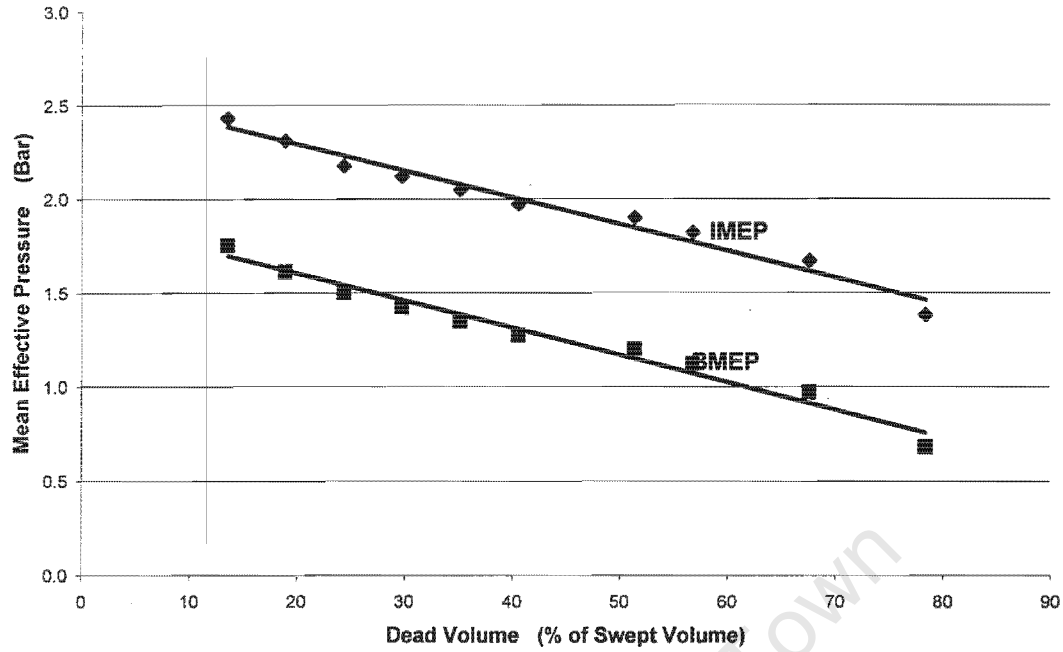

Figure 4. 10: MEP against Dead Volume for Engine

The engine must have the smallest possible dead volume for any piston size. Although not shown on this graph, dead volume in the heater should be particularly avoided (see table 3.1)

5. Design

5.1 Engine Design

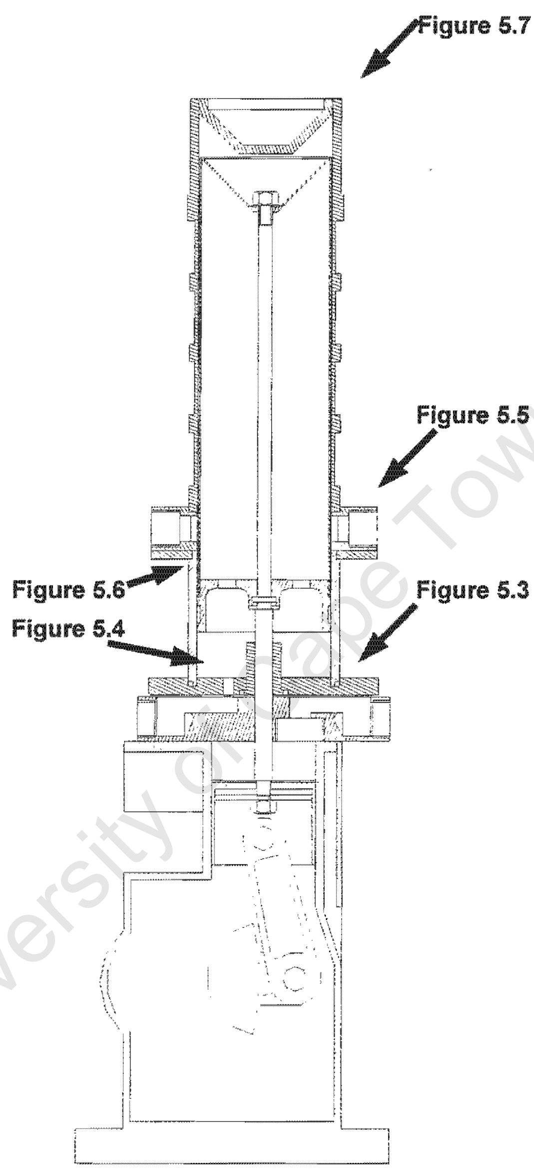

Figure 5. 1: Engine Showing Various Views

Figure 5.1 is a schematic of the engine showing the viewpoints of pictures in this chapter.

To save time and cost in the manufacturing stage of the second generation engine, it was decided to adapt the crankcase and compression cylinder of an existing engine. The use of air compressors, refrigeration compressors and lawn mower engines were all considered as a suitable starting point to build the engine.

A small Briggs and Stratton 2 horse power 4-stroke engine (figure 5.2) was decided on eventually as even though it ruled out the possibility of pressurising the crank case, the bore (60mm) and stroke (37mm) were in the region of the size required for the planned engine.

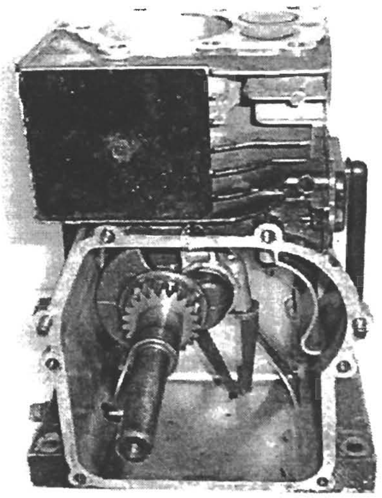

Figure 5. 2: Lawn Mower Engine to be used For Compressor

5.1.1 Compressor

The crankcase, crankshaft, connecting rod and cylinder were all left as standard. The valve gear was removed and a new cylinder head constructed.

This cylinder head had one way valves incorporated into it so that the lawn mower engine would now function as a compressor. It was also located in such a way as to seal off the existing inlet and exhaust valves on the engine block. See figure 5.3.

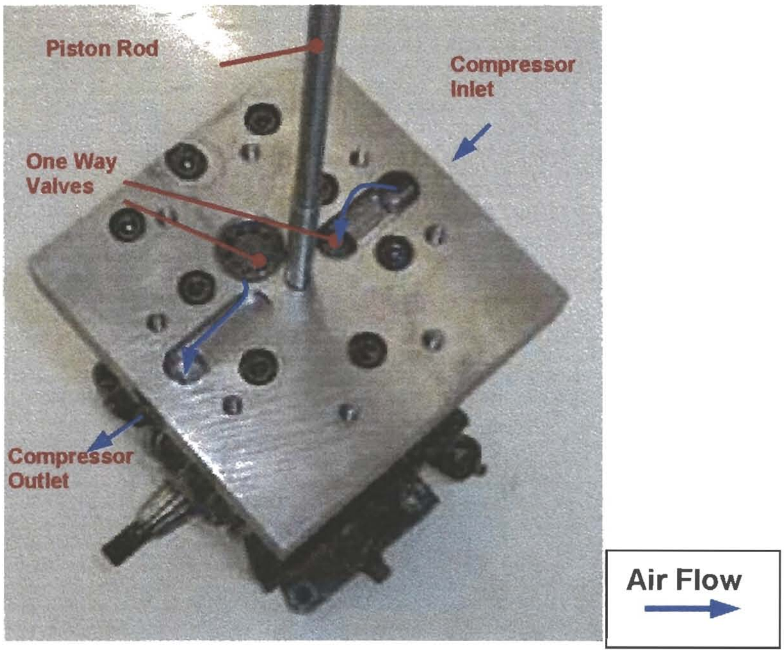

Figure 5. 1: Compressor Head Mounted on Engine

The piston was modified to allow the mounting of a rod on which to attach the expander piston. Another plate was then mounted on top of the cylinder head. The barrel for the expander piston was recessed slightly in this plate and sealed with an O-ring. The Piston Rod also ran through this plate, sealed by a Vesconite guide.

The compressor discharged into the region on the underside of this piston.

5.1.2 Expander

The expander piston was mounted on the rod attached to the compressor piston, so they move in phase with each other.

A lip seal fabricated of Rulon, a compacted PTFE material, separated the high-pressure buffer zone from the expansion space. The fact that the cylinder wall at that position would be hot and the complications involved in lubrication, meant that the seal would have to run unlubricated. See figure 5.4.

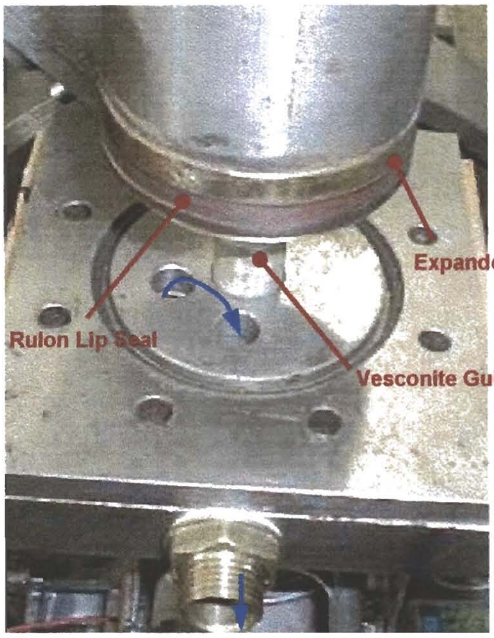

Figure 5. 2: Expander Piston and Lip Seal

The inlet and outlet of the expansion zone flowed through the same gallery. The air entering or leaving the region was forced past a 0.6mm gap between the cylinder wall and the displacer, as recommended by G. Walker [5]. This annular gap acted as a regenerator with a temperature gradient set up along its length. The cold air picked up heat as it moved from the bottom to the top of the cylinder where the heater was situated. Likewise, hot air leaving the cylinder was also forced through the same gap, thus cooling it.

The displacer was fabricated from a length of pipe with a wall thickness of 1mm, (a thin walled pipe was decided on in order to reduce inertia forces). The top was dished to provide stiffness and also to keep the end from bursting due to the high pressure in the buffer zone on the inside of the displacer, see figure 5.5. The displacer was mounted onto the expansion piston and sealed, it was tightened down with a torque of 25Nm. This torque produced a compression force equivalent to the pressure force that would try to separate the displacer from the expansion piston when pressurised.

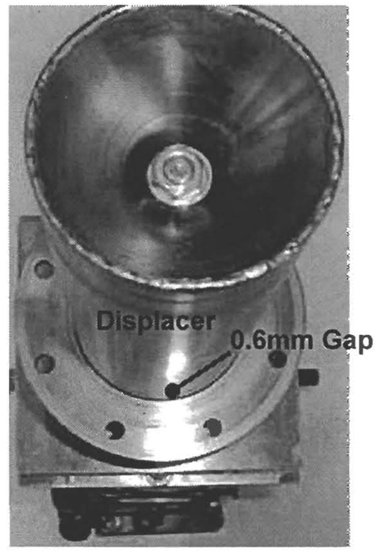

Figure 5. 5: Displacer and Barrel

Tests were carried out at relatively low speeds and thus it was unnecessary to balance the extra reciprocating mass added to the engine.

The expansion cylinder was made three times as long as its diameter. This was done for two reasons, firstly so there would be a significant temperature gradient setup along its length in order for it to function adequately as a regenerator; secondly in an attempt to isolate the rings, seals and valves from the hot end.

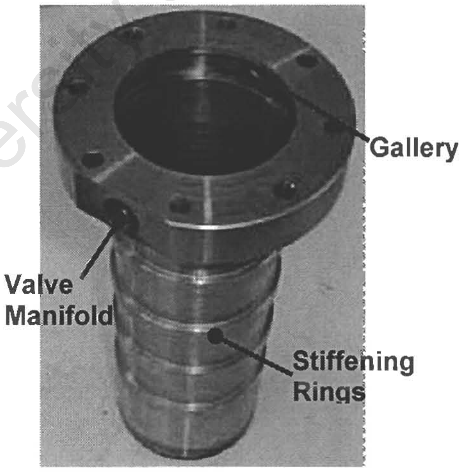

Figure 5. 6: Expansion Cylinder

The cylinder was made with a wall thickness of 2mm, with several stiffening rings along its length, see figure 5.6. The top of the cylinder was dished to match that of the displacer in order to minimise the clearance volume.

5.1.3 Valves

The valve timing for the engine posed a problem, as the computer model indicated that the valve timing changed considerably depending on the maximum pressure. It was therefore decided to use solenoid valves for the experimental version of the engine as this allowed flexibility to change the timing for different pressures. It also provided the ability to manually change the valve timing whilst the engine was running and thus investigate the effect of different timing.

A computer was used to control the valves via a PC30 card and the necessary circuitry.

A trigger pulse generated by the points on the existing engine was used to initiate timing on the program. The control software was written with Qbasic, it made use of the time interval between successive trigger pulses to determine the engine speed. This was then used to send a control signal to switch the valves. This program is explained in more detail in Appendix D.

5.1.4 Heater

The second generation engine, at least initially, was to be run using a gas heater. This was due to the fact that the engine's lubrication system did not allow for easy mounting into the existing furnace. The system would have to be mounted upside down, or on its side, in order for oil in the crankcase to flood the compression cylinder or leak from the engine. It was also felt that the theory of the coal furnace had been adequately proved in tests with the 3-process engine, and a gas burner would simplify temperature control and the speed of operation.

A simple gas burner was constructed, it consisted of copper tubing bent into a circle with several 1mm holes drilled into its inside face to act as gas jets. This heater was mounted on the top of the expansion cylinder. Figure 5.7 shows the heater mounted on the expansion cylinder.

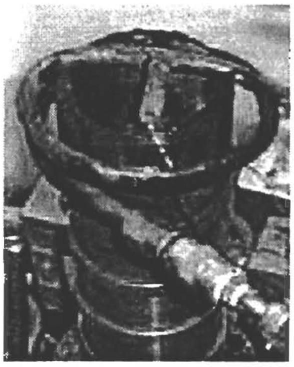

Figure 5. 7: Gas Heater Mounted on Expansion Cylinder

5.2 Furnace Design

Figure 5. 8: Furnace with Gravity Coal Feed

The fact that different materials require different fuel air ratios, grates, and feeding mechanisms precluded the possibility of making a general-purpose furnace that would allow for the engine to use heat generated from burning any combustible material. For example, wood needs a large amount of secondary air blown in above the fire to burn efficiently, while coal needs the majority of air to enter under the grid [16]. Thus, for simplicity, the furnace was designed to burn coal exclusively.

One aspect of the design that was considered important was the need for the furnace to be able to operate unattended for long periods of time. This meant that the furnace required some form of automatic fuel feed. Most large conventional furnaces make use of a chain grate system, or pulverized fuel injection and thus use a blower [22]. These systems were considered too complex and costly for this type of engine. Therefore the idea of a gravity feeder was explored.

Experimentation was carried out using what is commonly referred to as 'pea sized coal' and a grate constructed from a number of steel strips spaced evenly apart. It was found that an angle of 30 degrees for both the feeder and the grate worked well and provided a constant supply of coal for the fire for periods of over an hour.

When the engine was pressurised, a large heat transfer area would be necessary in order to heat the high-pressure air adequately. Due to this fact, the use of a fluidised bed was investigated as the relevant literature indicated figures in the order of $600W/m^2K$ could be expected with a fluidised bed. [17]

Investigations into the requirements of such a system revealed that the pressure drop necessary to fluidise the size bed required was too high to be obtained by natural convection alone. It therefore demanded the use of a high-speed rotary fan or a positive displacement pump, either system would consume a considerable proportion of the small engine output. The idea of a fluidised bed was thus rejected.

Due to the high temperature required to obtain reasonable thermodynamic efficiency, effective waste heat recovery would be desirable. This led to the concept of a double walled furnace design as this not only preheated the feed air, but also cooled the inner walls of the furnace as the air passed down. Thus the need for a refractory lining was avoided.

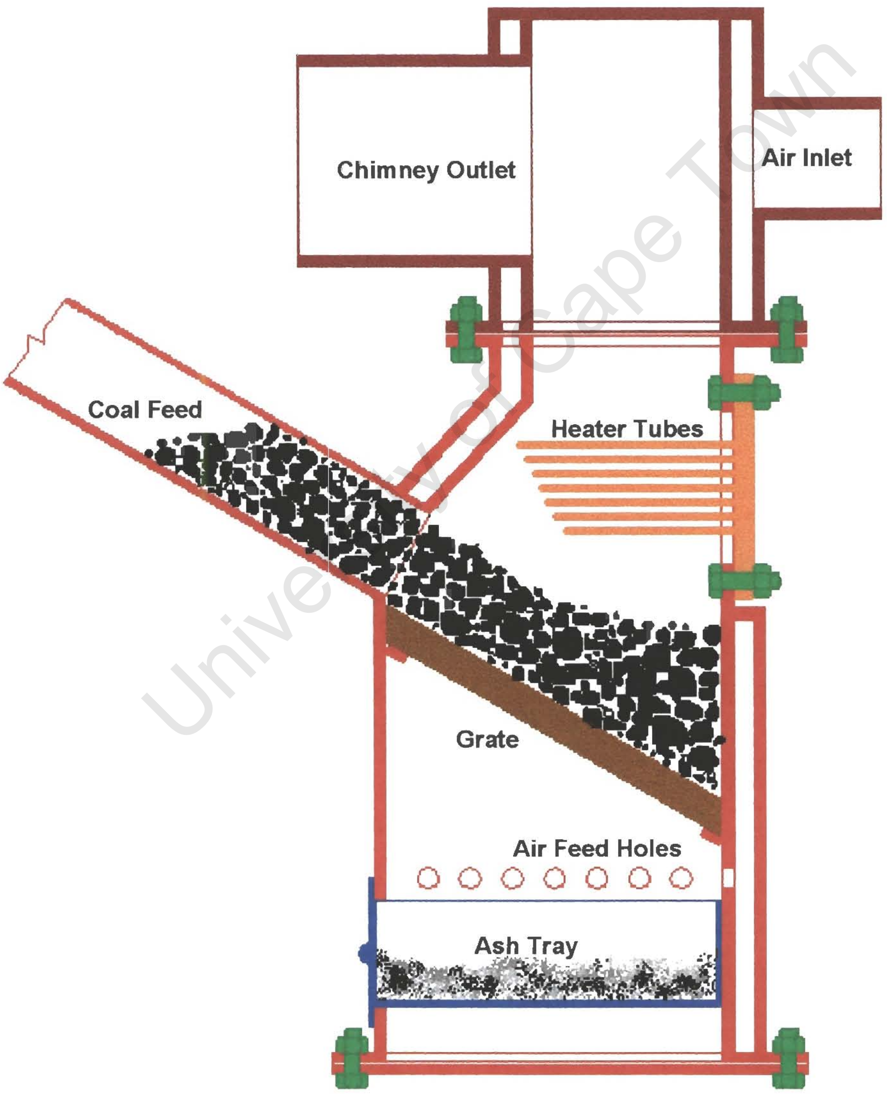

Figure 5. 9: Sketch showing fully Assembled Furnace

To simplify manufacture and reduce cost, construction was to be entirely of mild steel. This was felt adequate because of the comparatively short life requirement of the experimental furnace. Research revealed that it would be unnecessary to make use of expansion joints in a small furnace of this size [15].

The bolt-on chimney manifold was also designed to have a double skin as the inlet for the feed air was incorporated into it. See figure 5.9. The chimney was plumbed into an existing 5m high chimney.

Examination of small domestic coal heaters, revealed that for the design heat requirement of about 4kW, a grate area of about $0.03m^2$ was required. Thus the grate and furnace were sized accordingly. Figure 5.10 shows this grate and the double wall of the furnace.

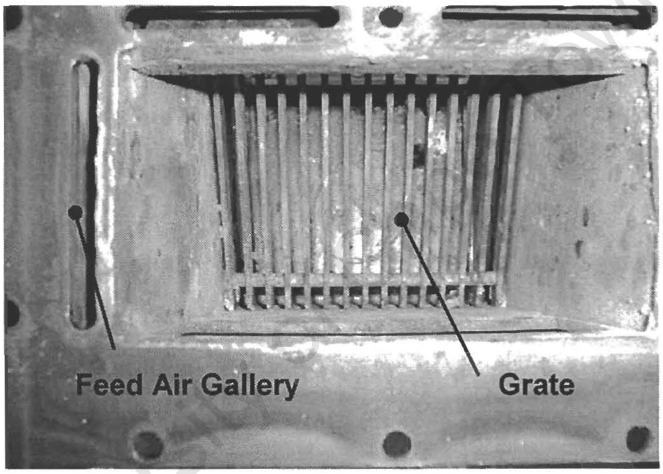

Figure 5. 10: Furnace showing Double Wall and Grate

5.3 Assessment and Commissioning

Few problems were encountered in the design or construction phase of the project. However, a number of major problems arose during testing.

The major problems discovered together with their necessary modifications are presented in table 5.1, Full details are given in Appendix E.

Throughout testing and experimentation, a supply of compressed air was continuously connected to the high-pressure reservoir to make up for the large number of leaks. These leaks occurred mainly around the joints and couplings of the copper piping that made up the reservoirs, where the nylon washers used were insufficient to seal the high pressure air.

The high temperatures that the expansion cylinder and displacer were subjected to combined with a high moisture content in the compressed air supply, caused serious corrosion on these mild steel components, requiring periodic cleaning.

Table 5. 1: Problems and Modifications Performed during Engine Testing

Problem	Cause	Solution / Modification
Valves not shutting properly	High-pressure air didn't allow valve to shut.	Valve piloted with internal pressure.
Engine develops a knock	Splash lubrication didn't operate at low engine speeds.	Cranks shaft and crank case replaced, forced lubrication system used.
Inadequate valve closure	Valves couldn't exhaust, vacuum formed under seat.	2mm hole drilled to allow trapped air to escape.
Air leak past Teflon seal	Teflon ring worn down.	Rulon lip seal and brass piston fabricated.
Air leak past lip seal	Piston off centre in bore.	Lip seal inner diameter machined oversize to enable it to self-centre.
Air leak past lip seal	Lip seal deformed plastically under load.	Rubber O-ring inserted under lip to force lip to seal against bore.
Engine seizure	Piston rod binding with compressor head.	Hole machined oversize.
Air leak between compressor inlet and outlet	Vesconite rod seal proud of its mounting plate	Seal machined flush and new gasket cut.

6. Experimental Investigation

The aim of the experimental investigation was to examine how closely the actual cycle resembled that predicted by the theoretical model. The effectiveness of the thermodynamic model could then be determined. This was done by recording actual pressure traces and comparing then with those predicted by the theoretical model.

The Mean Effective Pressure of the engine was also determined and compared with the frictional data. This was to ascertain engine performance and thus determine the feasibility of using this particular engine as a small stationary power source.

6.1 Experimental Apparatus:

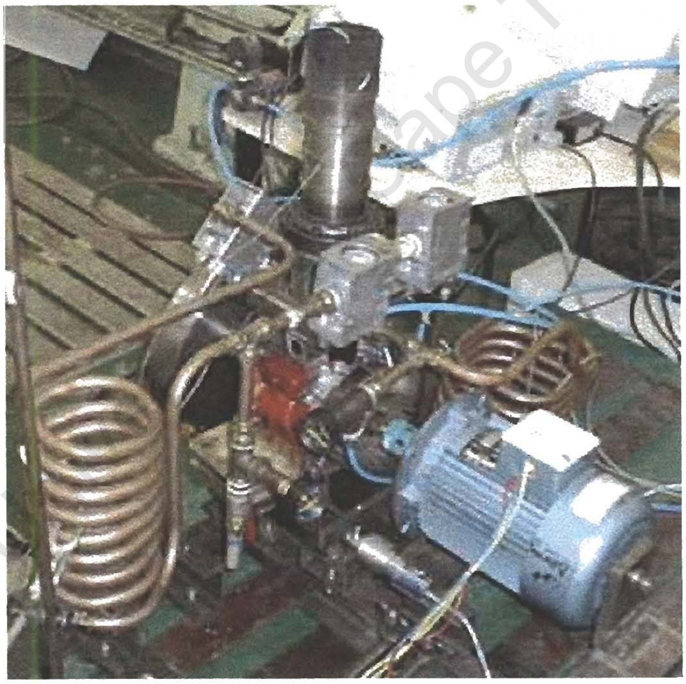

Figure 6. 1: Experimental Rig

a) **Test Engine**

   The engine to be tested was based on the truncated Ericsson cycle and built for this thesis and shown in figure 6.1.

b) Variable Speed Motor / Dynamometer

A motor mounted on gimbles was connected directly to the crankshaft of the engine by means of a flexible coupling. This allowed measurement of the torque the motor was supplying to turn over the engine. This was done by means of a lever arm attached to the motor that impinged directly onto a load cell, see figure 6.2. This was connected to a strain gauge amplifier that was calibrated to give a force reading in Newtons. The AC motor was driven by a frequency inverter that allowed the speed to be varied constantly.

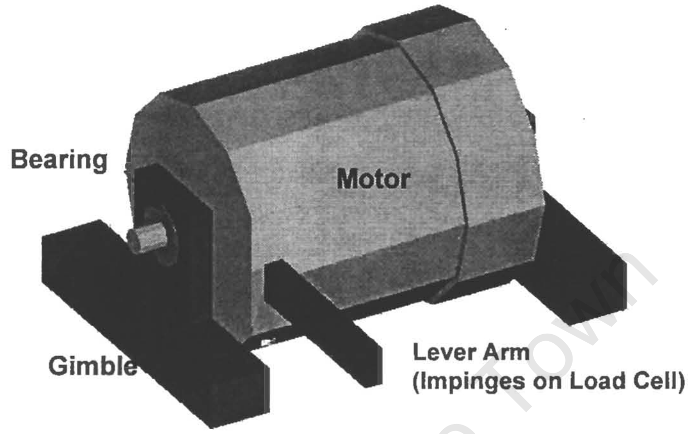

Figure 6. 2:Gimble Mounted Motor

c) Instrumentation

Instrumentation consisted of a water-cooled pressure transducer connected to a 2-channel Oscilloscope. The other channel of the oscilloscope was connected to an inductive sensor that indicated the Top Dead Centre (TDC) position and thus allowed a Pressure-Volume Diagram to be generated from the data. Another 2-channel oscilloscope was used to view the valve firing pulses in relation to the TDC indicator.

Thermocouples were connected to the expansion cylinder head and to various points along the copper tubing, before and after the valves.

d) Valve control system

This was achieved by a PC30 card installed into a desktop PC running the Qbasic control program (see Appendix D). A circuit board was used to feed a trigger (achieved through the existing points of the engine) into the program and switch the valves with the use of transistors. The valves were powered by a 12 volt power supply.

6.2 Test Parameters

The following parameters, summarised in table 6.1 were measured.

a) Engine Speed:

The engine speeds used for testing ranged from 100 to 300 rpm, these were governed by the rate at which the shop air could effectively recharge the high pressure region when the engine was running, whilst exhausting to atmosphere.

b) Expansion Cylinder Temperature

Initially tests were carried out with no heat addition, thus the expansion process caused a cooling of the cylinder wall. Temperatures as low as $-15^{o}C$ were reached. The heating was done using Liquid Petroleum (L.P.) gas and a circular burner. Thus the maximum wall temperature was governed by the design capability of this setup.

c) Valve Gallery Temperature

This was a dependent variable governed by the effectiveness of the engine's regenerator.

d) Delivery (High) Pressure

The high-pressure reservoir was maintained at a constant pressure by a pressure regulator connected to the workshop supply. Most tests were carried out between 2 and 3 bar, as this was a pressure at which the motor could easily turn the engine over without laboring.

e) Exhaust (Low) Pressure

This variable was set by the amount of expansion occurring in the expansion cylinder, this was determined in the initial design phase by the size of the expansion cylinder, and the possible heat input. The engine was designed to function with a pressure ratio of 2.5.

f) Instantaneous Expansion Cylinder Pressure

Dependent variable recorded by a pressure transducer connected to an oscilloscope.

g) **Torque Input**

Dependent variable measured with the aid of a load cell connected to a strain gauge amplifier.

Table 6. 1: Test Parameters

Test Parameter	Range
Engine Speed	100 – 300 rpm (steps of 25rpm)
Expansion Cylinder Temperature	-15 – 700 °C
Valve Gallery Temperature	20 – 60 °C
Delivery (High) Pressure	0.5 – 6 Bar Gauge Pressure
Exhaust (Low) Pressure	Dependant Variable
Instantaneous Expansion Cylinder Pressure	Dependant Variable
Torque Input	Dependant Variable

6.3 Experimental Procedure:

The power to the valves was turned on and the valve control software started. The lubricating valve was fired manually several times to lubricate the system and the burner at the top of the expansion cylinder was lit. The motor was started with the low-pressure vent valve closed.

The pressure of the high-pressure side was then slowly increased until it reached the desired operating range. Once at the desired operating temperature, the valve timing was adjusted until the resultant pressure trace was as desired and indicated that the engine was producing entirely positive work. This is implied when the majority of the area under the pressure trace is on the right hand side of the T.D.C marker.

The motor was run at various speeds (100 to 300 rpm) and for each speed, the corresponding torque was read off from the load cell and the pressure trace recorded for conversion to a Pressure–Volume diagram.

6.3.1 Friction tests

The above procedure was carried out with and without addition of heat from the burner, and the engine pressurised to a number of different operating values. Tests were also run with the engine unpressurised to gauge the friction in the system.

Next the engine was progressively stripped down, with torque and speed measurements taken at the different stages of disassembly. Firstly without the expansion cylinder and valve assembly. Then without the high-pressure Rulon seal, and finally without the compression cylinder head.

The effectiveness of the regenerator was also ascertained, by running the engine at constant speed, whilst heating up. The Expansion Cylinder temperature as well as the valve gallery temperature were recorded. The valve gallery temperature was a combination of the inlet and outlet air temperatures to the expansion zone.

7. Experimental Results

Figure 7.1 shows a typical pressure trace obtained whilst running the motor at 175 rpm, pressurised to 2.5 bar and 480 degrees Celsius heater wall temperature.

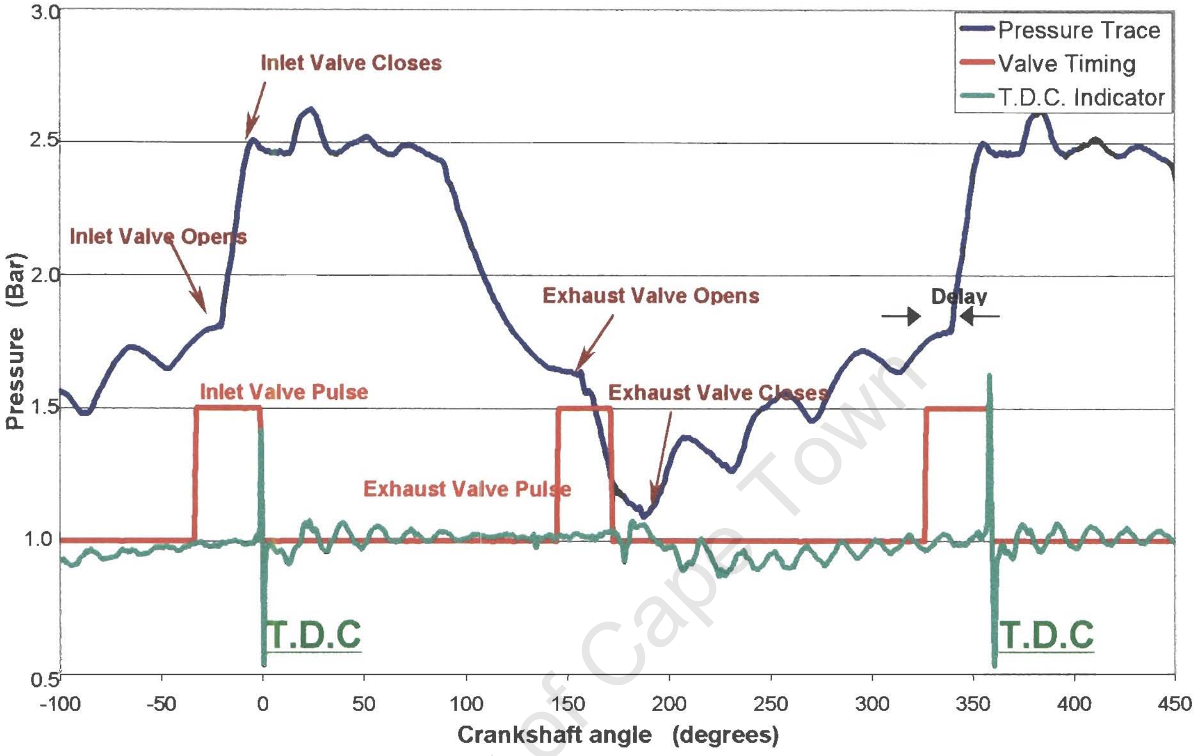

Figure 7. 1: Typical Pressure Trace obtained from Oscilloscope Data

Superimposed onto this are the traces obtained by the inductive transducer, indicating the Top Dead Centre position and the 12 volt pulses sent out to switch the valves.

As can be seen, there was a delay between the pulse and the valve actually opening, especially at high pressure when the valve struggled to close against the high back pressure and pressure waves setup. The pressure waves are evident by the amount of fluctuation in the trace, even rising above the maximum pressure in some cases.

With the aid of the Top Dead Centre indicator the corresponding crankshaft angles for each pressure reading could be obtained. These pressure traces could then be converted to a pressure-volume diagram by comparison of these crankshaft angles with the volume variations computed by the spreadsheet model.

The resultant pressure – volume diagram for the trace is illustrated below.

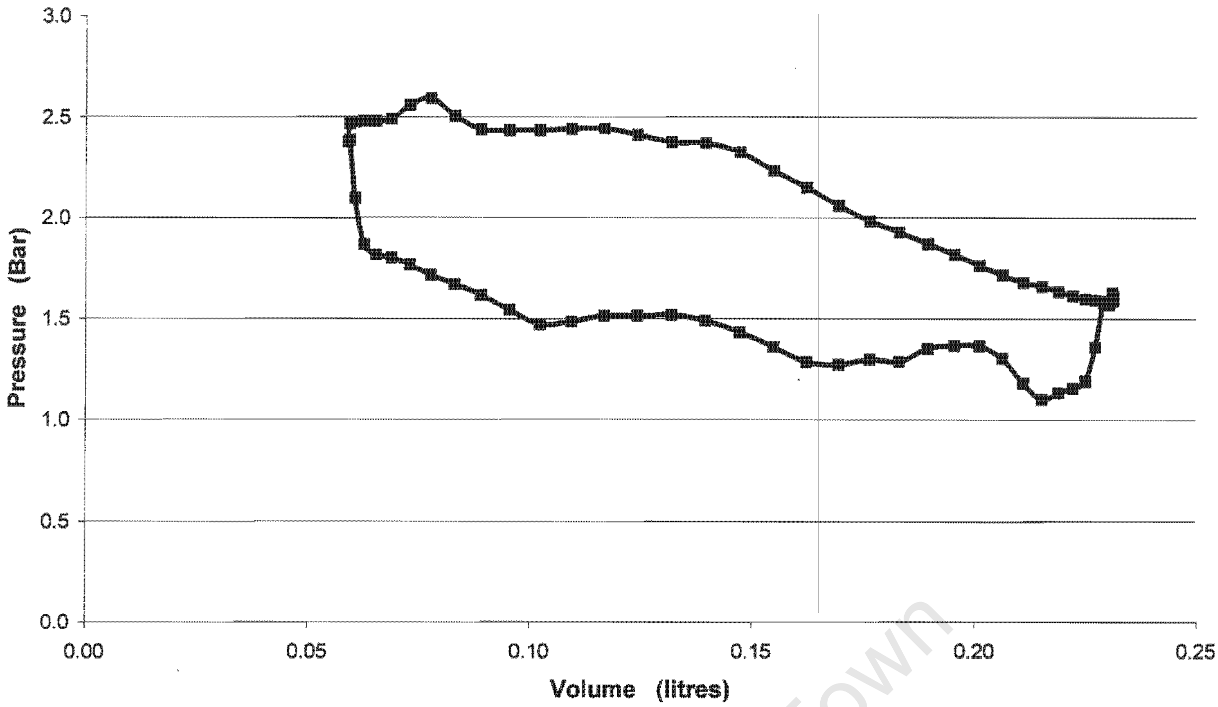

Figure 7. 2: Pressure – Volume Diagram for the Expansion Space of the Engine 2.5 bar Pressure

Many such traces were obtained throughout the testing period. These were used not only to indicate how much positive work the engine was producing, but also to determine the effect different parameters such as peak pressure and temperature had on the engine. This data could then be compared with the results obtained from the theoretical model. An overview of data from some of these tests is presented in table 7.3.

It was felt unnecessary to place a pressure transducer in the compression zone, as this was functioning as a simple compressor and therefore little would be gained by obtaining data from it.

Analysis of the torque data obtained from the motored engine revealed that the compression was apparently adiabatic, as expected. This was done by indirectly obtaining the indicated power of the engine. The method involved subtracting the power required to turn the engine over while pressurised with heat addition, from the power required when the engine was not heated.

The indicated power of the compressor was then deduced as being the difference between the overall indicated power and power calculated from the P-V diagram generated for the expansion cylinder. This was then compared with the value predicted using the standard adiabatic compression model used in the spreadsheet to predict performance.

Table 7. 1: Results of P-V Diagram in figure 7.2

	Expansion Space	Adiabatic Compression	Combined Engine
Work Done (J)	12.5	-8.0	4.5
Mean Effective Pressure (Bar)	0.72	-0.46	0.26
Power Produced (W)	38	-24	14

Below is the pressure –volume diagram obtained by pressurising the engine to 3.5 bar and heating it to 750 °C.

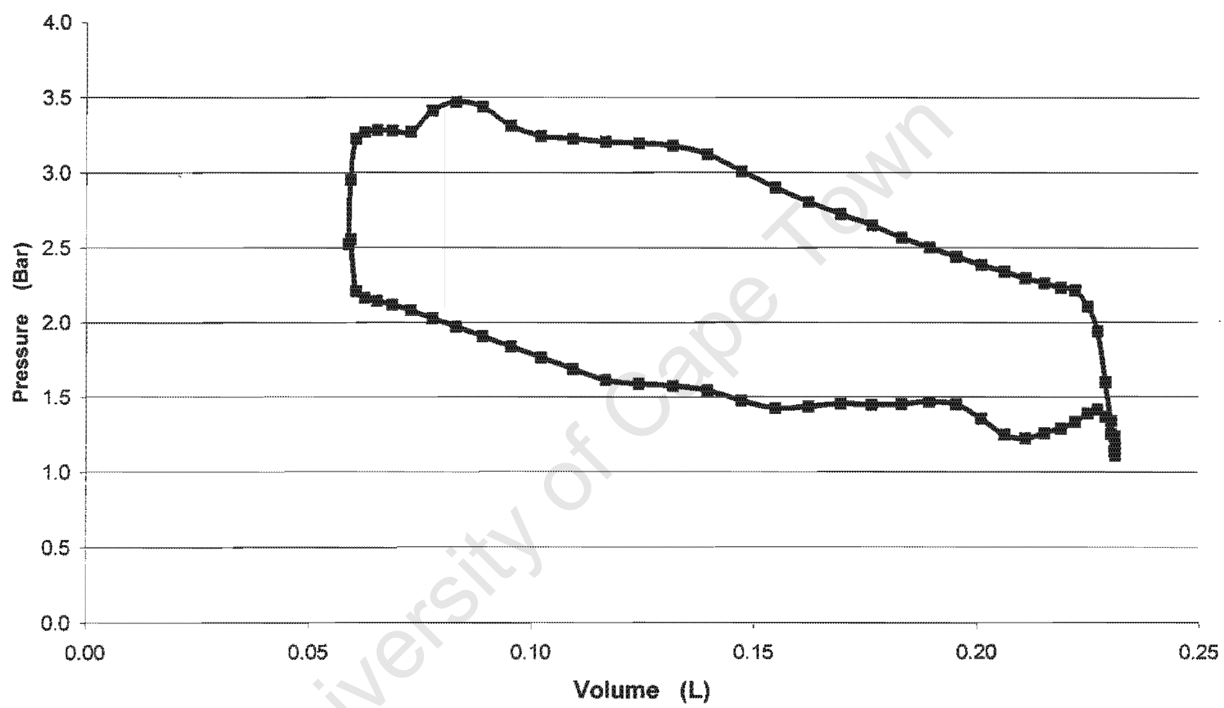

Figure 7. 3: Pressure – Volume Diagram for the Expansion Space of the Engine 3.5 bar Pressure

Table 7. 2: Results P-V Diagram in figure 7.3

	Expansion Space	Compression Space	Combined Engine
Work Done (J)	22.2	-14.0	8.2
Mean Effective Pressure (Bar)	1.3	-0.8	0.5
Power Produced (W)	72	-44	26

Table 7. 3: Overview of Tests

Engine Speed (rpm)	Expansion Work (J)	Expansion MEP (Bar)	Expansion Power (W)	Peak Pressure (Bar)	Heater Temperature (°C)
184	12.5	0.73	38	2.5	500
174	6.1	0.35	18	2.5	260
183	11.7	0.68	36	2.5	480
184	15.5	0.90	47.5	3.0	590
187	16.3	0.95	51	3.5	650
188	17.2	1.00	54	3.5	680
192	22.2	1.29	71	3.5	700
175	12.6	0.73	37	5.0	490

7.1 Friction Tests

The friction tests carried out for the various components of the engine, produced the following data, presented in the form of a graph.

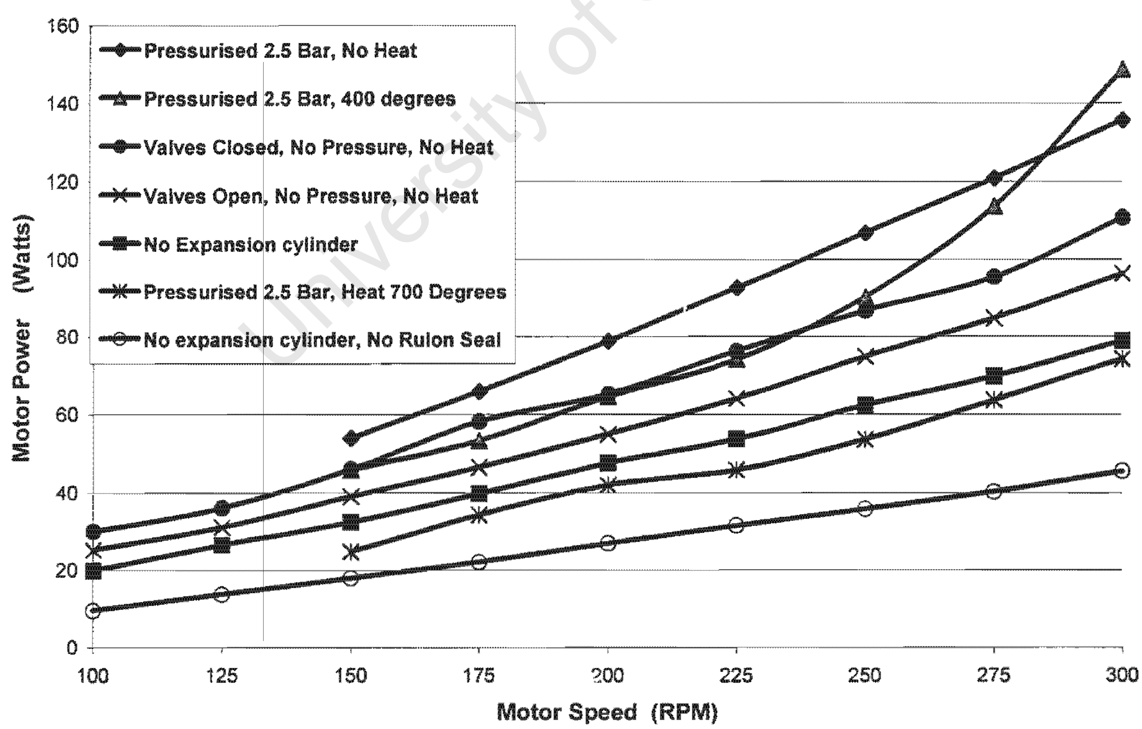

Figure 7. 4: Graph of Power against Motor Speed for Engine in Various States of Assembly

The main basic losses expressed as a percentage of the total loss are summarised in table 7.4.

Table 7. 4: Component Losses of Engine

Component Loss	Percentage of Total Loss
Pumping and Fluid Friction Loss at Atmospheric Pressure	17%
Friction Associated with Rulon seal	37%
Friction Associated with Compressor Piston Rings, Bearings etc	46%

As can be seen, the friction due to the Rulon Seal comprises a large percentage of the total loss.

7.2 Regenerator Effectiveness

The heater wall temperature and the valve gallery temperature were recorded throughout the running of certain tests. This gave some indication of how well the regenerator was working.

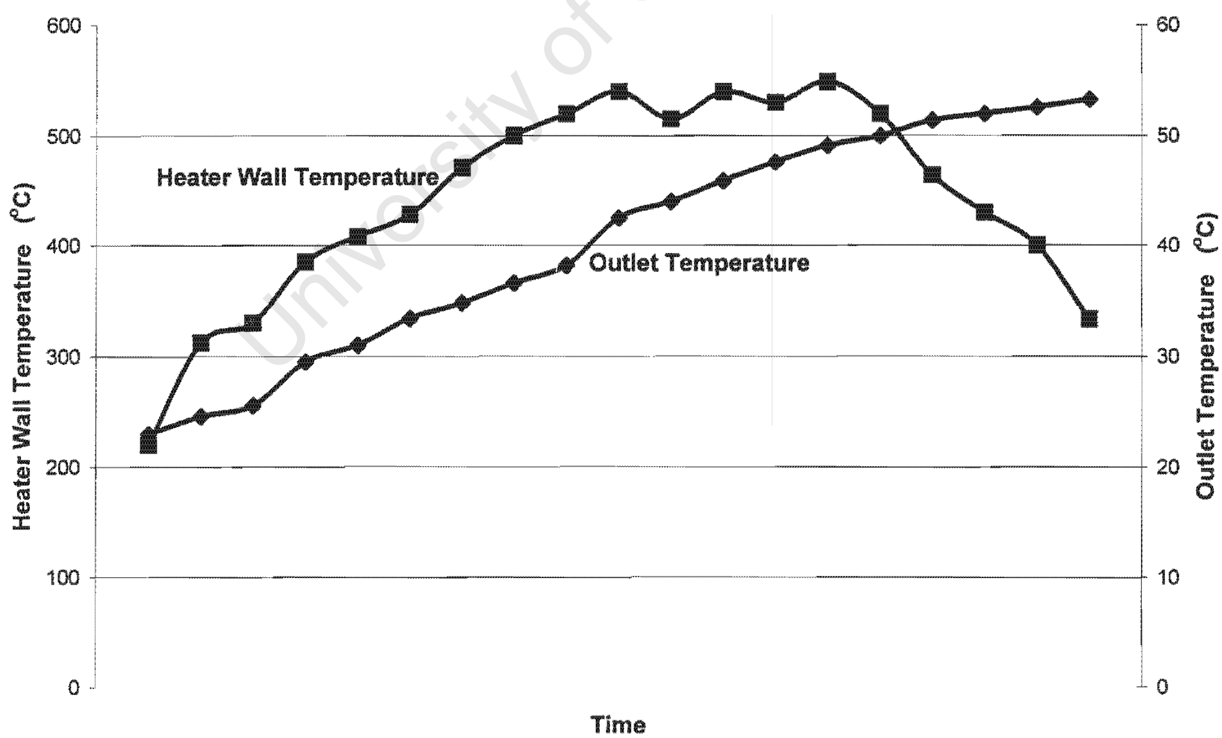

Figure 7. 5: Graph of Heater and Outlet Temperature against Time

8. Discussion of Results

In terms of the overall objectives for this thesis, the most obvious point is that the engine was unable to run unaided. This was disappointing, however, due to the large number of measurements taken and experimental data collected, it was possible to pinpoint exactly where the problem areas of the engine lay. Furthermore, these results allowed the evaluation of the engine design in terms of its ability to function as a small-scale power source for rural Africa, thus still fulfilling the overall aim of the project.

For the analysis of these results the measured Pressure–Volume diagram for the cycle was first compared to the theoretically predicted diagram. The performance of the engine in terms of power and friction was computed and compared with predicted values and results quoted in literature pertaining to similar sized engines. Reasons for discrepancies or insufficient performance were then examined. The engine's heat transfer and regenerator effectiveness were also considered in relation to predicted performance. Finally, methods of how the engine could be made to meet the requirements of the project were examined.

The truncated Ericsson cycle performed as expected, and closely matched the cycle produced by the original engine built by Ericsson in 1860 [1]. The thermodynamic processes involved matched closely those stated in the relevant literature [5], [1], and the actual pressure traces measured, closely conformed to the theoretical expectations for the cycle. Figure 8.1 shows the measured and theoretically predicted pressure-volume diagrams for the engine.

The pressure fluctuations evident in the measured Pressure-Volume diagram were thought to be due to pressure waves set up across the throat of the valve shortly after opening.

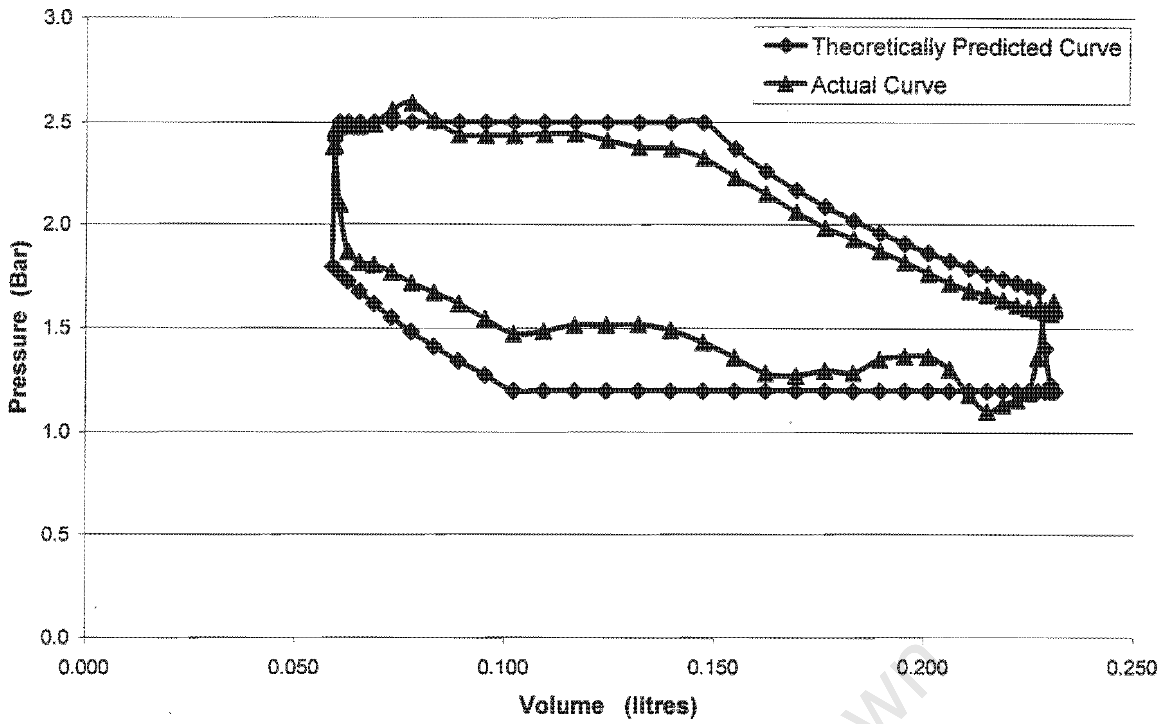

Figure 8. 1: Actual and Predicted Pressure-Volume Diagrams for 2.5 Bar Peak Pressure

From the friction data the actual power produced by the engine could be determined by subtracting the motor power of the test result concerned from the power for the same pressure and engine speed, but without heat addition.

This could then be converted to the mean effective pressure for the engine using:

$$MEP = \frac{Power \cdot 60}{SweptVolume \cdot rpm}$$

The IMEP for the expander could be calculated from the work done in that zone, obtained from the pressure–volume diagram. Subtracting these two gave the mean effective pressure for the compressor, which compared very favorably with that predicted by the thermodynamic model. This negated the need to investigate the compression part of the cycle further as it appeared to be performing as expected. The result also revealed that the compression work for the truncated Ericsson engine was large. The value was over 60% of the expansion work.

However, this value is only an estimate, and the error associated with this calculation could be great. A large experimental error was involved in the measurement of the power supplied by the motor due to fluctuations in the torque readings and other such factors.

The power calculated from the generated P-V diagram was also an estimate, and as such had errors associated with it. The combination of these two would produce and even larger error in the result.

Although the general trend of the two curves was similar, the difference between the actual mean effective pressure and the predicted value for most tests was large. This could partly be explained by the pressure waves that could not be predicted by the thermodynamic model. The higher pressure recorded during the compression stroke and the resultant decrease in the area of the P-V diagram for the cycle could also be explained by the continuous leakage problem experienced from the high-pressure buffer zone to the expansion space around the Rulon seal.

Table 8. 1: Actual and Predicted Data for Expansion Work at 2.5 Bar Peak Pressure

		Actual Data	Predicted Data	Deviation (%)
M.E.P Expansion	(Bar)	0.72	0.93	24
M.E.P Combined	(Bar)	0.26	0.54	52

8.1 Friction

The fact that the engine did not run despite the cycle operating as expected, leads to the conclusion that the friction in the system was considerably greater than first anticipated. Results from the 3-Process engine indicated that the frictional mean effective pressure was in the region of 0.7 bar, thus a value of about 0.7 bar was assumed for the truncated Ericsson cycle engine. Analysis of the frictional data obtained from this engine however revealed that the frictional mean effective pressure of the engine was 0.9 bar at atmospheric pressure, rising to 1.6 bar when pressurised to 3.5 bar.

The mechanical efficiency for the engine was obtained from the power output and the motoring power required for a similar pressure condition.

$$\eta_m = \frac{P_{out}}{P_{out} + P_{motor}}$$

This produced a figure of 33%, which is considerably lower than typical values in the region of 60% quoted for Stirling engines in the relevant literature [9]. However most of the Stirling cycle engines would have one less seal than the Ericsson engine, thus explaining the large difference.

The original Ericsson engine of 1860 had a mechanical efficiency of 50% [1], though this was a large engine with a piston diameter in excess of 250mm, so the friction would be a smaller percentage of the total engine output. This is because the ring friction of a reciprocating engine is proportional to the area of the piston, whereas the power is proportional to its volume. Therefore, a large engine will have a greater potential for a high mechanical efficiency.

According to Taylor piston ring friction accounts for 75% of the mechanical friction associated with the average multi-cylinder engine [23]. This finding applies equally to external heat engines, worsening if dry seals are used, thus an engine design with few seals would be extremely advantageous.

8.2 Pressurisation

The engine had initially been designed to run from a high pressure of 12 bar to a low pressure of 5 bar. This was to be achieved by pressurising the low-pressure side with shop air, then turning the engine over manually until the high-pressure side was pressurised to 12 bar by the engine's compressor. At that time the difficulty in sealing the high-pressure air was not fully appreciated and thus the leakage from the high-pressure reservoir did not allow for this.

Pressurising the engine to a greater degree would significantly raise the Mean Effective Pressure. Even though the fluid friction and pumping losses associated with a higher pressure would also increase, the effect is not as marked, therefore it is likely that the engine would have run when highly pressurise. The necessity to continuously supply air to the high-pressure reservoir meant that it was not possible to test at more than a peak pressure of 6 bar, thus this theory could not be explored.

Figure 8.2 shows the measured and predicted pressure-volume diagrams when the engine was pressurised to 3.5 bar.

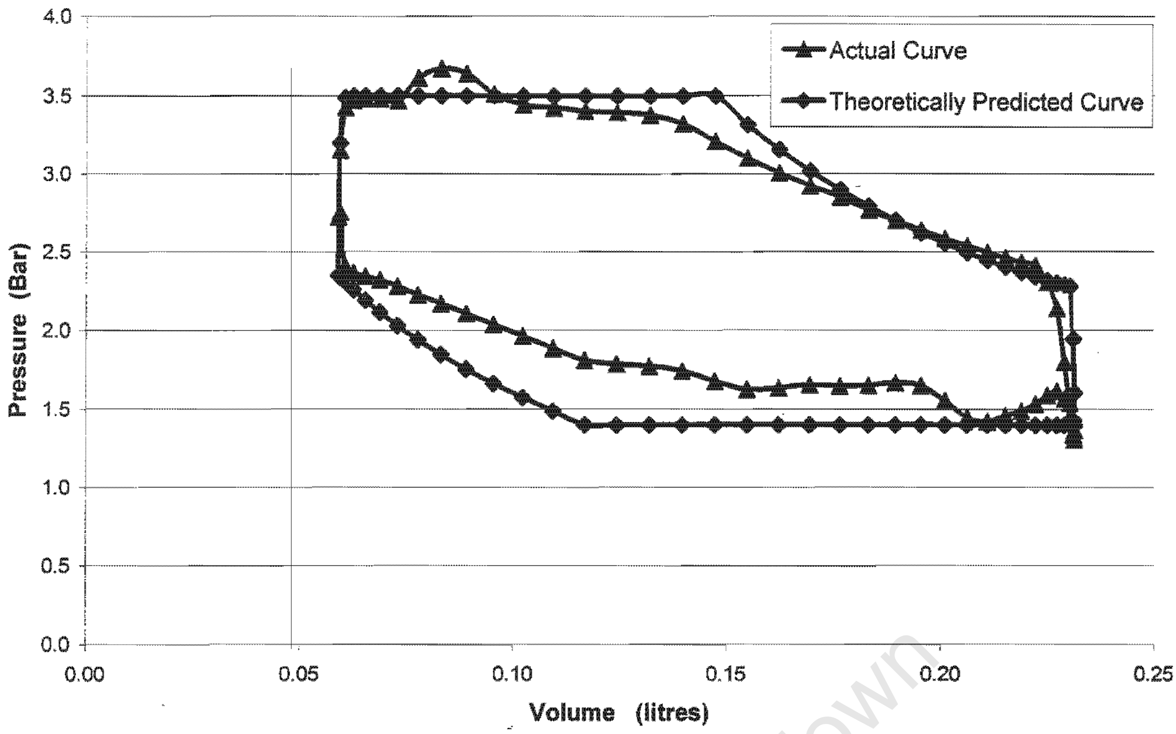

Figure 8. 2: Actual and Predicted Pressure-Volume Diagrams for 3.5 Bar Peak Pressure

Table 8. 2: Actual and Predicted Data for 3.5 Bar Peak Pressure

		Actual Data	Predicted Data	Deviation (%)
M.E.P Expansion	(Bar)	1.3	1.7	24
M.E.P Combined	(Bar)	0.5	0.9	44

Due to the correlation between actual and predicted data it is unlikely that other factors such as the heat transfer in the expander or regenerator were the cause of the poor performance experienced.

8.3 Heat Transfer

The heat transfer in the expansion cylinder, although not specifically measured because of the difficulty in inserting thermocouples into this zone, was assumed to be sufficient as after running at a high temperature, the top of the displacer showed heavy discoloration. This discoloration could only have been produced by the surrounding air being elevated to a high temperature, indicating that the heat transfer was good. See figure 8.3.

Before Testing After Testing

Figure 8. 1: Discoloration of Displacer due to the high Air Temperature

The regenerator appeared to be functioning well and its effectiveness, although not specifically measured was satisfactory and in general accordance with the performance predicted by walker [5]. As can be seen in the graph of figure 7.5, the temperature of the valve outlet gallery remained considerably cooler than the heater. A drop in the rate of rise of the outlet temperature on the graph can be seen to occur when the heater temperature starts to fall.

8.4 Findings

The thermodynamic model accurately predicted the performance of the engine and provided useful insight into the effect of many factors, in particular the peak pressure and dead space of the engine. The heat transfer predictions made using the model were very basic and could be improved with a more detailed analysis utilising more cells than were originally set. However, the accuracy with which the actual cycle was predicted using a basic model questions the need for such a complex analysis for the purposes of engine design.

With the use of low friction seals particularly for the expansion cylinder, the engine could produce useful work. If scaled up, and with further pressurisation, the engine could potentially meet the 5kw target. However, low friction seals such as the roll sock are extremely intricate and costly, and higher pressurisation would lead to major sealing difficulties. Although it would be of great academic interest to implement these ideas and produce a running engine, these efforts would compound to make the engine significantly larger and more complex, making it unsuitable to function as a rural power source for developing countries. Therefore these propositions were not explored.

The lack of success achieved in running the engine even in the face of its predicted performance questions the optimistic literature available on hot air engines. Theoretical efficiencies and power outputs are quoted without sufficient test data to prove these findings, this leads to unrealistic expectations on the performance of such an engine.

9. Conclusions and Recommendations

Based on the experimental observations and results the following conclusions may be drawn:

The thermodynamic model used accurately predicted the true cycle, therefore it could be concluded that such a basic model is adequate for the purposes of designing such an engine. This questions the need for highly detailed and complex differential equations in such an analysis.

The truncated Ericsson cycle functioned sufficiently well and has merit, however, the compression work was too large a portion of the work output in light of the small temperature ratio which this cycle was forced to operate between when used in an external combustion engine.

The necessity to reduce friction to an absolute minimum was not at first appreciated. The friction and losses associated with the engine were too large for the feeble output of the engine. The engine design had too many rings and seals.

The decision to construct the engine almost entirely out of mild steal was appropriate for experimental purposes. However the corrosion that appeared after a comparatively short period of testing indicates that the life of the engine would be unacceptably short.

The instrumentation was adequate for the requirements, however if the engine had proved successful a more detailed analysis would be necessary, requiring investigation of the compression part of the cycle and the heat transfer amongst other factors.

Although for test purposes, a supply of compressed air could be connected to make up for leaks, this would not have been acceptable for a prototype. There were too many joints and couplings that allowed leakage of the pressurised air.

If the engine was scaled up, low friction seals used and the system highly pressurised, the engine could potentially meet the 5kW design target. However, these ideas were not investigated because the aim of this thesis was to produce a simple, small-scale prime mover for use in rural Africa.

Although the hot air engine has the potential of to provide cheap power efficiently, in practice these engines need to be highly pressurised and run at temperatures close to the material limit in order to obtain useful work from them. This requirement and the need to reduce friction to a minimum means these engines need to be highly complex, this prevents them from effectively functioning as a power source for rural Africa.

The literature available on the subject of hot air engine design was overly optimistic about the potential of these engines. Few examples of the proposed engines could be found, and many others did not achieve their predicted results. Thus creating a false expectancy for such engines.

Engine Specific Recommendations:

The following recommendations are made for the purposes of improving this engine design.

- In light of the low specific output of an external heat engine, the number of seals used in an engine must be a minimum, as in an alpha type configuration. Alternatively low friction seals such as roll-sock seals should be used.

- Due to the need to optimise the engines upper temperature limit in order to improve performance, construction of the hot end should be out of stainless steal or another heat resistant material.

- Due to the problems of sealing high-pressure air, the number of joints and couplings in the air reservoirs and ducting in the system should be reduced to a minimum.

General Recommendations:

The following recommendations are made for rural power generation.

- The Stirling cycle should be examined as a potential prime mover for developing countries, as the losses with such a system could potentially be lower.

- Explore other alternatives to the hot air engine for a rural power source such as steam.

10. References

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Appendix A: Ideal Cycle Investigation

A number of ideal cycles were initially analysed in order to gain an appreciation for which cycles would be most applicable to a hot air engine.

The Otto, Diesel, Brayton, Stirling, Ericsson 1853, Ericsson 1860, and a cycle resembling half a Carnot cycle were all analysed using highly idealised equations.

The Brake Mean Effective Pressure and overall efficiency was obtained for each cycle operating between the same temperature differentials of 300K to 900K. The Frictional Mean Effective Pressure for all the engines was assumed constant at 0.25 Bar and the mechanical efficiency was calculated from:

$$\eta_m = 1 - \frac{FMEP}{IMEP}$$

Otto Cycle

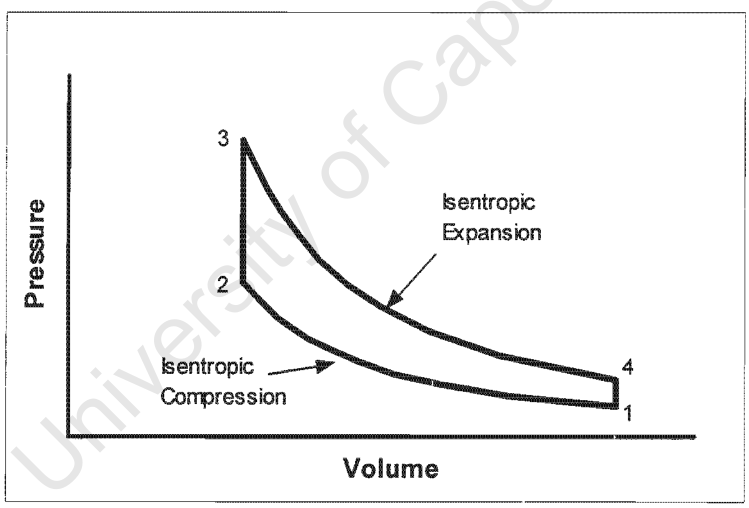


Figure 4. 1: P.V. Diagram of the Ideal Otto Cycle

This is the cycle approximated by the modern day petrol engine. It consists of two isentropic and two constant volume processes.

Diesel Cycle

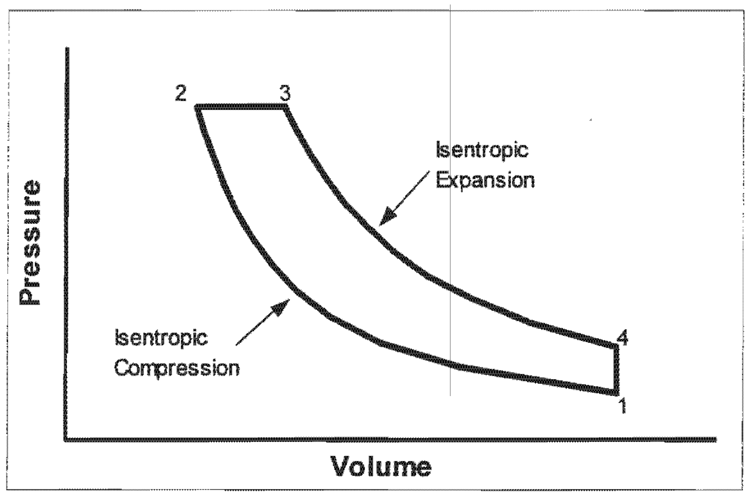


Figure 4. 2: P.V. Diagram of the Ideal Diesel Cycle

The Diesel engine operates off this cycle. It consists of an isentropic compression and expansion, a constant pressure heat addition and a constant volume heat rejection process.

Brayton Cycle

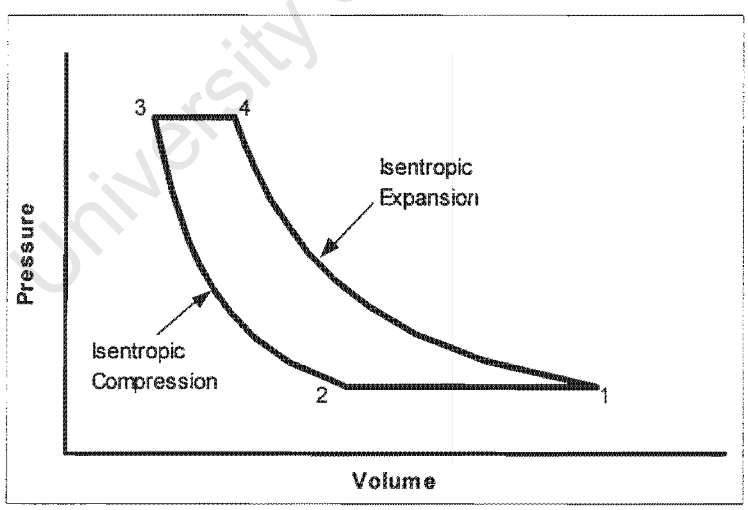


Figure 4. 3: P.V. Diagram of the Ideal Brayton Cycle

The cycle employed by the jet engine. It comprises an isentropic compression, isentropic expansion and two constant pressure heat exchange processes.

Stirling Cycle

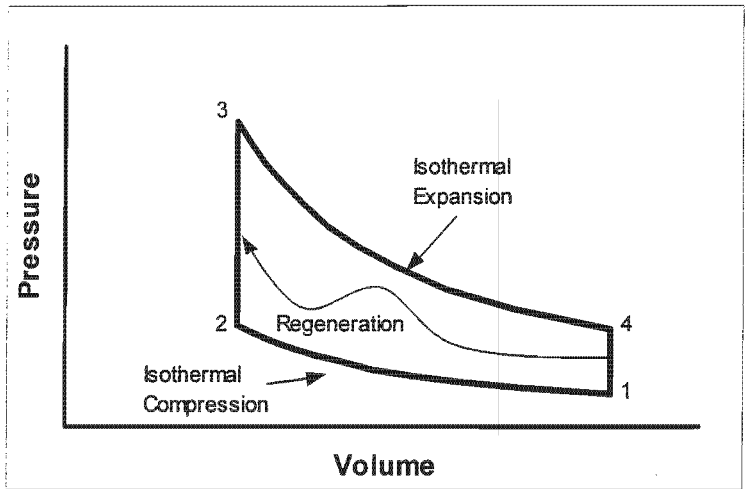


Figure 4. 4: P.V. Diagram of the Ideal Stirling Cycle

This is the most commonly used hot air engine cycle. The compression and expansion processes are isothermal instead of isentropic. It also contains two constant volume heat exchange processes with regeneration.

Ericsson Cycles

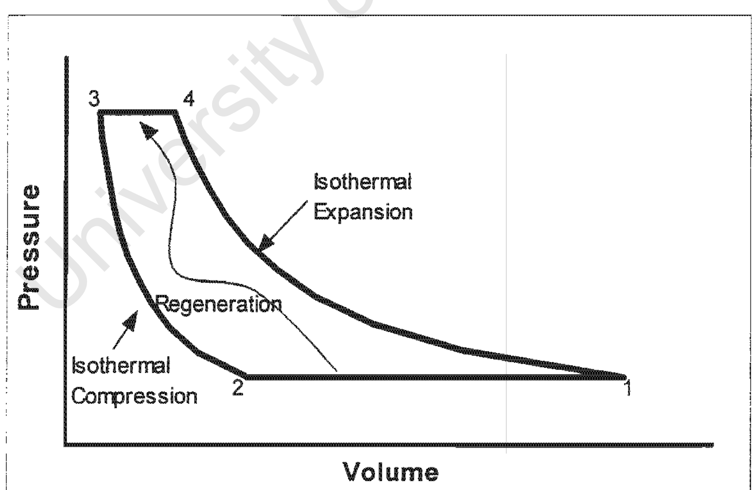


Figure 4. 5: P.V. Diagram of the Ideal Ericsson Cycle of 1853

This cycle common known as the Ericsson cycle is much the same as the Stirling cycle, but makes use of constant pressure instead of constant volume heat exchange processes.

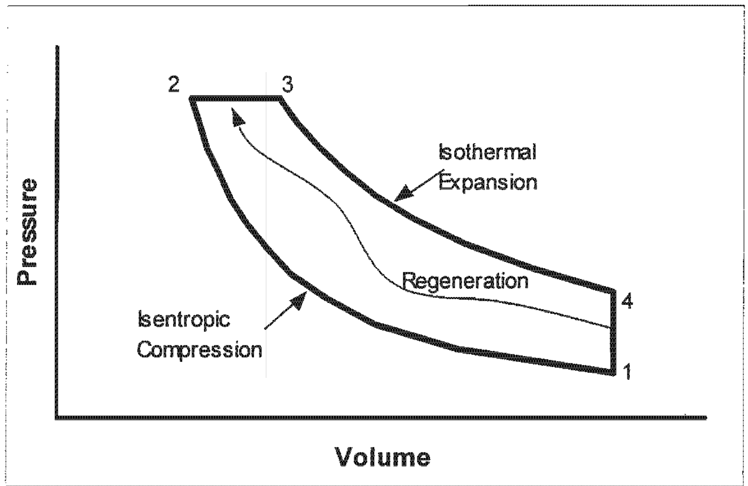


Figure 4. 6: P-V Diagram of the Ideal Ericsson Diesel Cycle of 1860

This cycle consists of an isentropic compression and an isothermal expansion, with a constant pressure heat addition and a constant volume heat rejection process.

Three Process Cycle

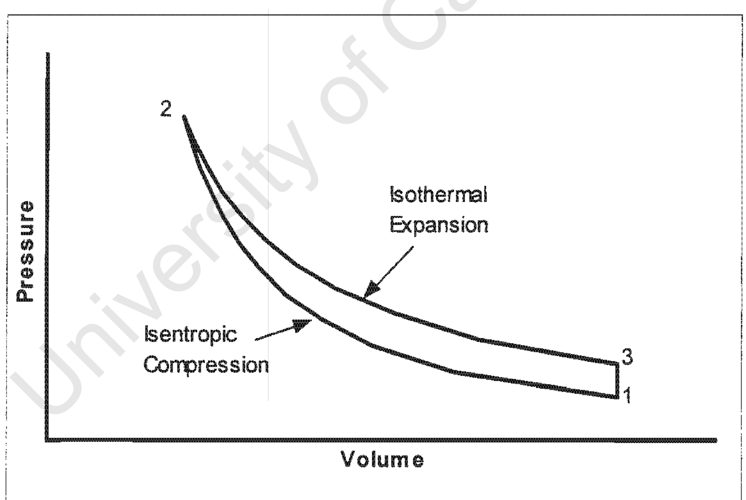


Figure 4. 7: P-V Diagram of Ideal 3 Process Cycle

This cycle conceived by Dr A. Yates features and Isentropic compression and an isothermal expansion, but only one constant volume process, heat rejection takes place during this process. Heat addition takes place during the expansion process.

Comparison of cycles

Mean Effective Pressure, was considered the most important factor when choosing the cycles. As can be seen from the graph below, the most favorable cycles are the Stirling, Ericsson 1860 and the 3-process cycle. The Otto and Diesel cycles needing a much higher temperature differential than would be feasible with external combustion.

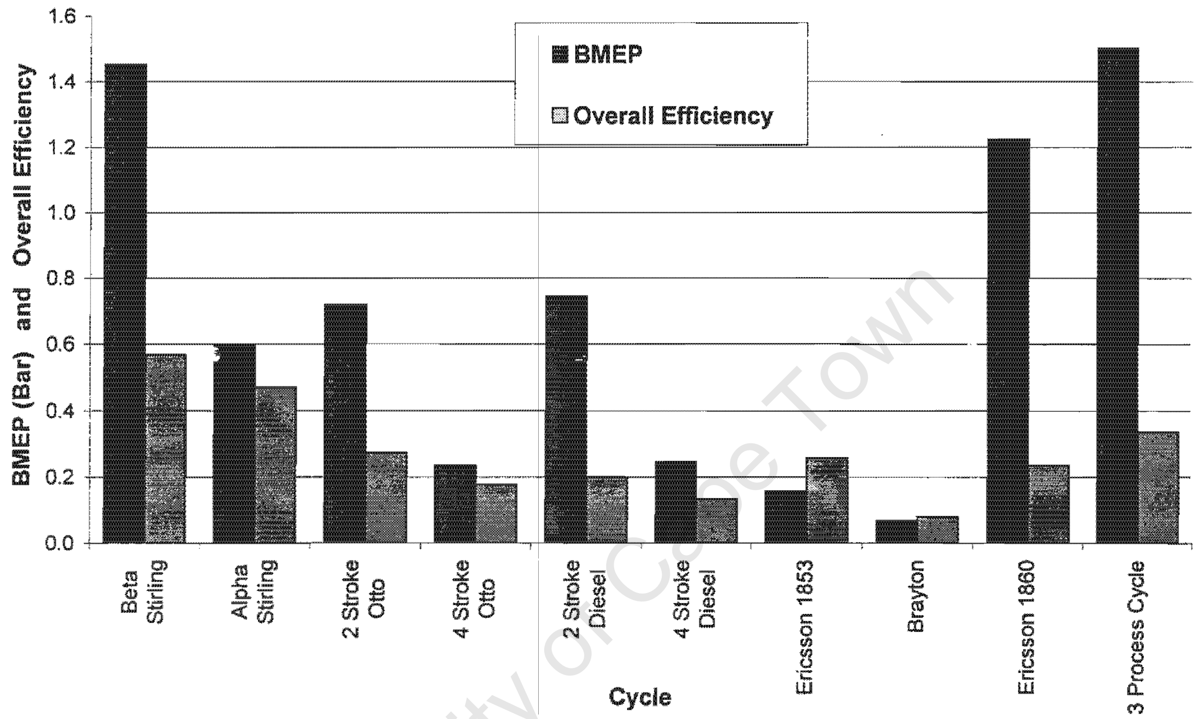


Figure A 1: BMEP and Efficiency for Idealised Cycles

From the above information and the perceived simplicity with which the cycle could be executed, the 3-process cycle was chosen as the basis for the initial engine.

Appendix B: Preliminary Engine Investigations

The 3-Process engine was conceived by Dr A Yates, it's cycle compromised a polytropic compression, followed by a polytropic expansion with heat addition, then a constant volume heat rejection process.

It was originally designed to be an open cycle, making use of the two-stroke principle for it's exhaust and inlet processes, but the possibility existed to pressurise the engine at a later stage and hence increase its output.

The purpose of this experimental engine was to provide insight into the functioning of an unconventional external combustion engine and to test the validity of theoretical predictions made using a thermodynamic model of the cycle. It was also constructed to give an indication of the requirements for such an engine to run effectively and help decide the criteria for the next generation engine.

Thermodynamic model

The engine was modelled with an Excel spreadsheet using an approach very similar to the technique described in Appendix C.

The cycle was analysed by considering the engine to consist of three zones, a compression zone, an expansion zone and a heater zone. Each zone was considered to be at a distinct uniform temperature throughout.

An analysis was then performed for every degree of crankshaft rotation.

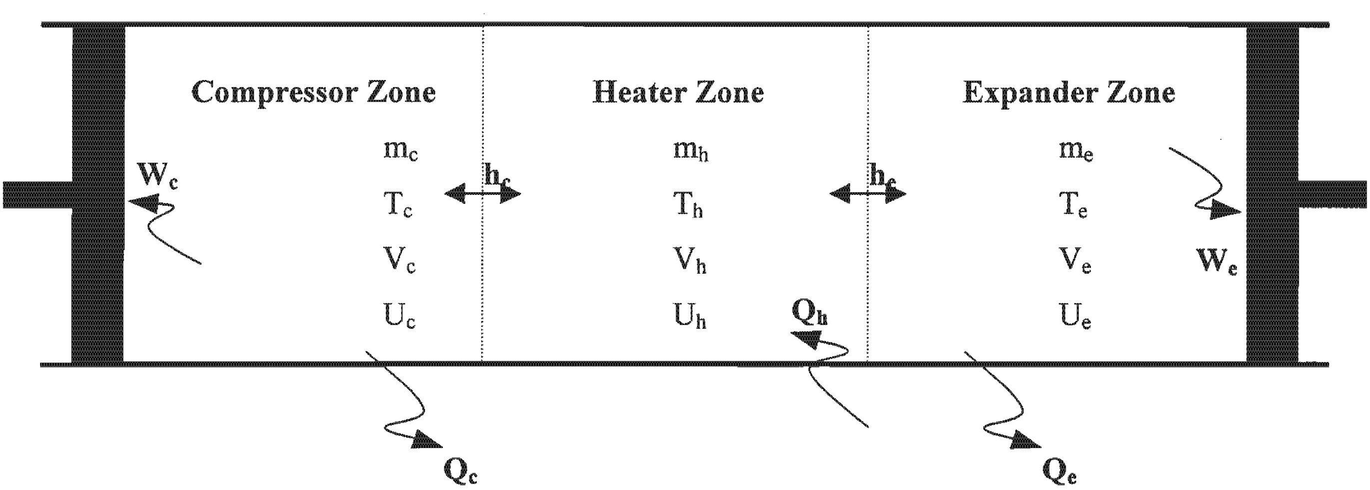


Figure B 1: Schematic of Thermodynamic Model for 3 Process Cycle

The following calculations were made for the compression and expansion zones: The heat transfer to the external environment, the work done by the gas, the change in internal energy of the gas and the enthalpy transferred to or from the neighbouring zones.

The heat transfer to the gas in the heater was calculated and used in conjunction with the enthalpy transfers from the above zones to obtain each successive temperature in the heater zone.

The pressure, which was assumed uniform throughout the system, was then adjusted at each time step to ensure conservation of mass. This calculation was done with the aid of a visual basic macro.

The heat transfer coefficient in the heater tubes for the next step was then calculated
The two stroke scavenging process was taken into account by assuming perfect mixing in the cylinder when the exhaust and transfer ports were open and applying the formula. [24]

$$m_1 CpT_1 + m_2 CpT_2 = (m_1 + m_2) CpT_{final}$$

to obtain the initial temperature for the cycle.

An example of the input parameters for the engine model for a particular configuration are presented in table B1.

Table B 1: Example of Parameters for Engine Model

Parameter	Value	Units
Engine Speed	350	rpm
Ambient Air Temperature	60	°C
Cylinder Wall Temperature	150	°C
Heater Wall Temperature	700	°C
Volume of Cylinder when Transfer Port Uncovered	0.059	l
Compression ratio	2.7:1	
Frictional Mean Effective Pressure	0.5	Bar

The frictional mean effective pressure for the engine was an estimation obtained from “Stirling Engines” By G. Walker for similar sized engines [5]

Using the above Inputs yielded the following outputs when the engine was operating from atmospheric pressure:

Table B 2: Output Parameters for Above Configuration

Parameter	Value	Units
Peak Pressure	3.3	Bar
Compression Work	-12	J
Expansion Work	16	J
Work Done	3.7	J
Indicated Mean Effective Pressure	0.66	Bar
Brake Mean Effective Pressure	0.16	Bar
Thermodynamic Efficiency	24	%
Mechanical Efficiency	13	%
Overall Efficiency	3.2	%
Power Output	13	Watts

The pressure – volume diagram of figure B2 was predicted for the above configuration

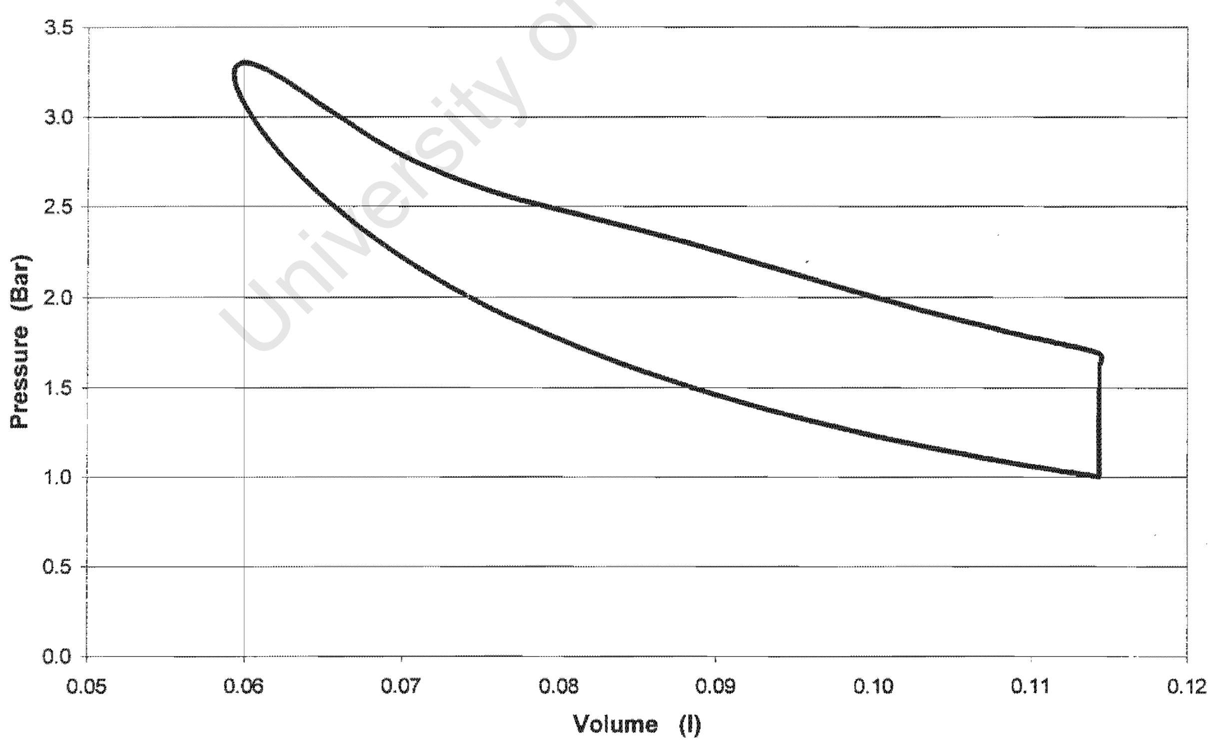


Figure B 2: Theoretically Predicted P-V Diagram

When pressurised to a base pressure of 7 Bar (the maximum limit of the shop air supply) the analysis indicated the following parameters.

Table B 3: Output Parameters when Pressurised to 7 Bar

Parameter	Value	Unit
Peak Pressure	22.8	Bar
Compression Work	-79	J
Expansion Work	101	J
Work Done	22	J
Frictional Mean Effective Pressure	0.7	Bar
Indicated Mean Effective Pressure	2.0	Bar
Brake Mean Effective Pressure	1.5	Bar
Thermodynamic Efficiency	14	%
Mechanical Efficiency	75	%
Overall Efficiency	10	%
Power Output	400	W
Average Torque	20	Nm

The FMEP was assumed to increase due to the additional friction and pumping losses associated with higher base pressure.

Design of 3-Process Engine

To save time and cost in the manufacturing stage and because of the difficulties in machining the ports, the decision was made to use standard two-stroke motor bike pistons and barrels with the two stroke porting already built in.

The basic design parameters, summarised in table B4, were then estimated based on what could feasibly be achieved and on which parameters produced the best overall results. A detailed design was then carried out to produce the 3-process cycle engine.

Cylinders and Pistons

Due to the fact that a two stroke engine normally pressurises the crankcase with the inlet mixture, a 'buffer' cavity had to be created below the pistons to accomplish this. This required the cylinders to be mounted on an aluminium plate with the necessary ducting to convey air to the cavities under the pistons. Two tubes were therefore mounted between this aluminium plate and a base plate, then sealed with rubber O-rings. The pistons were mounted on two highly polished silver steel rods that ran through brass guides to seal the bottom end of the cavity. See Figure B3.

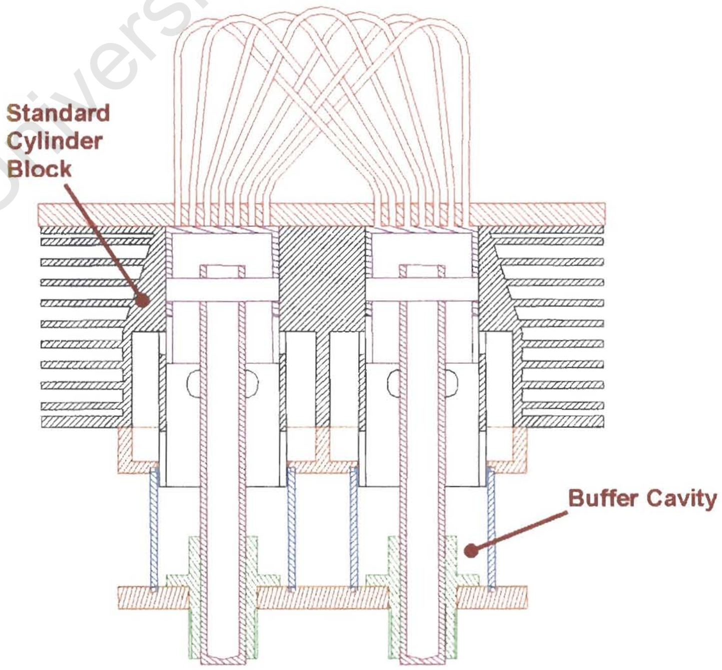


Figure B 3: Cylinder Head and Block

A cylinder head was fabricated, incorporating 40 stainless steel tubes connecting the two cylinders, see figure B4. These tubes were mounted directly in the furnace. The tubes were individually bent to ensure that each was the same length and that as far as possible, the tubes would be exposed to the radiant heat from the fire.

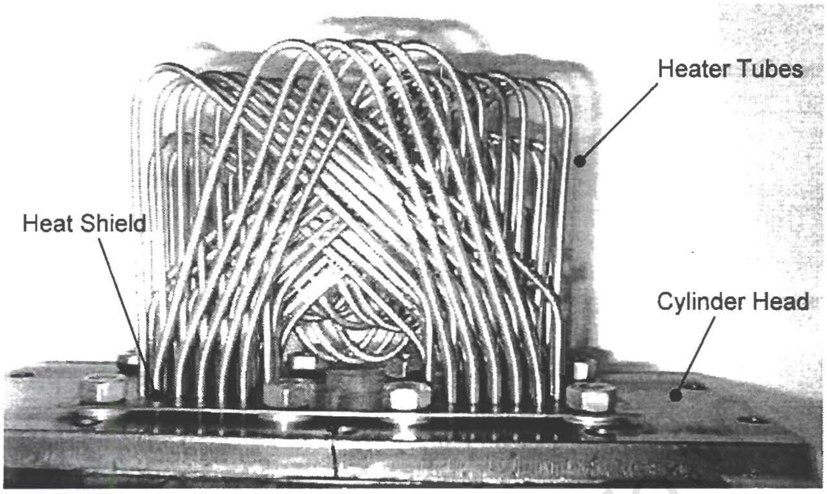


Figure B 4: Cylinder Head with Heater Tubes

Table B 4: Cylinder Block Parameters

Parameter	Value	Units
Bore	50	mm
Stroke	34	mm
Dead Volume	0.0019	l
Outer Diameter Heater Tubes	3.1	mm
Inner Diameter Heater Tubes	1.75	mm
Number of Heater Tubes	40	-
Length Heater Tubes	200	mm

Mechanism

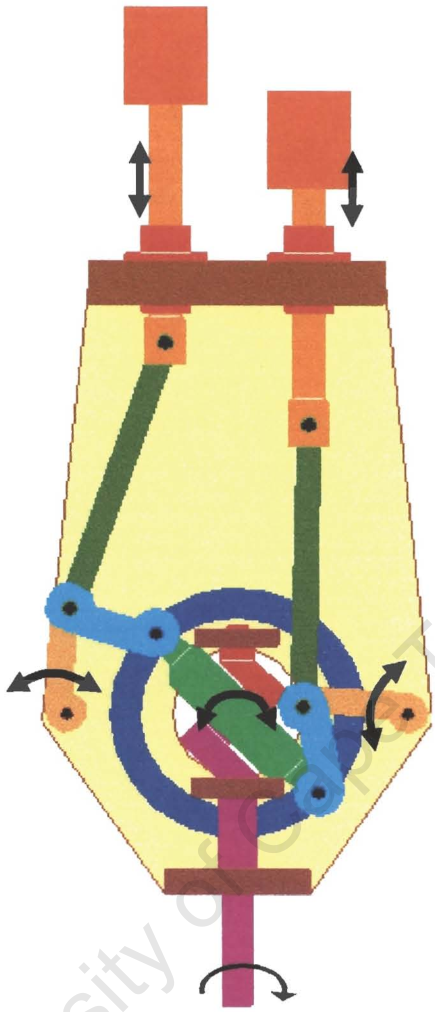


Figure B 5: Motion of Links in Engine Mechanism

The mechanism itself was conceived by Dr A. Yates. It made use a nutator, which revolved in one plane with simple harmonic motion, whilst the driveshaft rotated. This reciprocating movement was used to drive a rocker mechanism by means of a universal swivel linkage. The back and forth motion of the rocker arms translated to the piston rods by means of a connecting link, and by suitable sizing of the elements, produced the desired motion at the pistons, shown in figure B6.

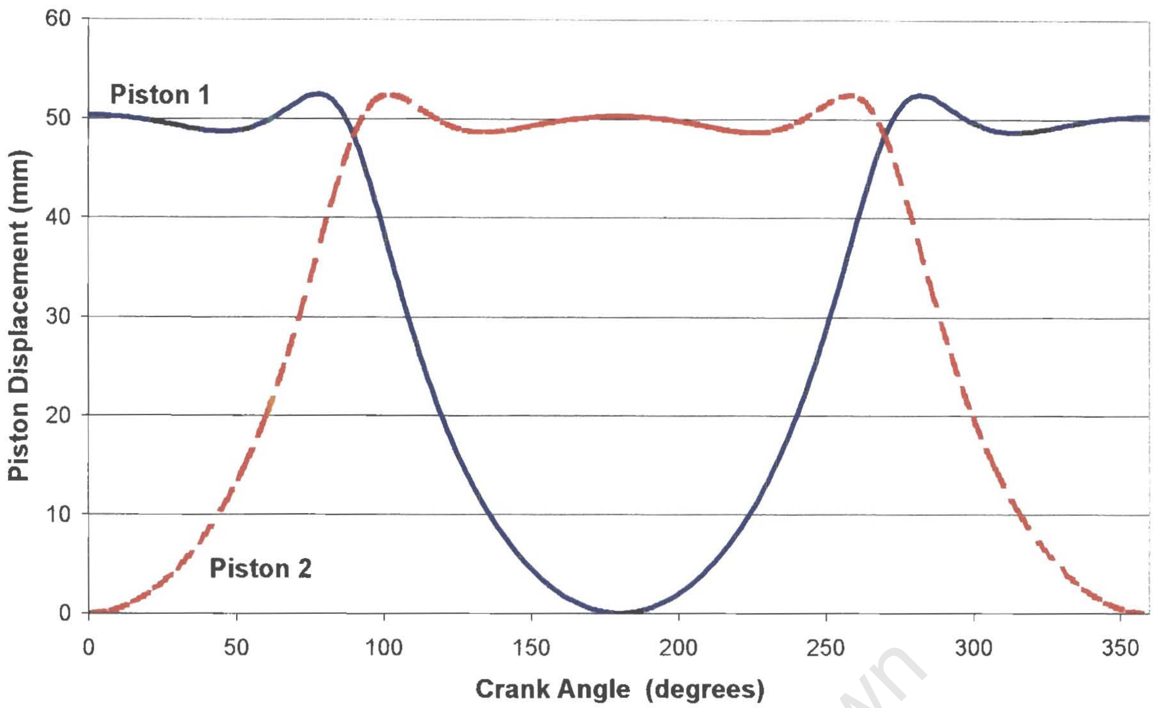


Figure B 6: Piston Displacement Vs Crank Angle

The nutator was mounted onto the driveshaft at 45degrees to the direction of rotation via two taper roller bearings. These were necessary to support the large forces exerted on the nutator. The driveshaft was split to allow easy assembly of the system. The bearings ran on shaft 1 (the output shaft) which was located onto shaft 2 by means of a key. See figure B7.

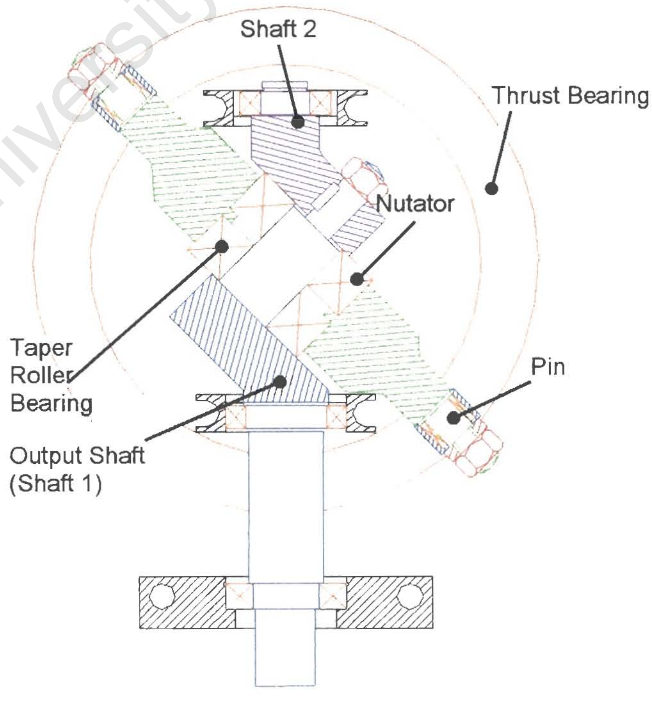


Figure B 7: Sketch Showing Sectional View of Nutator

At either end of the nutator there were two pins that were mounted with needle roller bearings running directly on the hardened shaft of the nutator. These pins located the universal connecting links with the aid of ball bearings. These pins also butted against a large thrust bearing which ensured that the nutator was retained in the desired plane of operation. At the other end of each connecting link was a compound joint where the rocker and conrod were attached with needle roller and deep groove ball bearings respectively, see figure B10. The rockers were secured with plain bearings to the housing. Each rocker extended beyond the pivot point and was fitted with counter balance weights. See figure B8.

The conrods, which were attached to the piston rods were constrained linearly with brass guides; the guides also served as a seal for the buffer cavity created by the underside of the piston.

The output shaft was secured to the housing with three deep groove ball bearings mounted in bearing blocks. See figure B9.

Lubrication was of the splash type, with the output shaft providing the splash as it rotated. Grooves milled into the top bracing plate of the housing allowed oil to run and drip onto the piston rods to lubricate them in their brass guides.

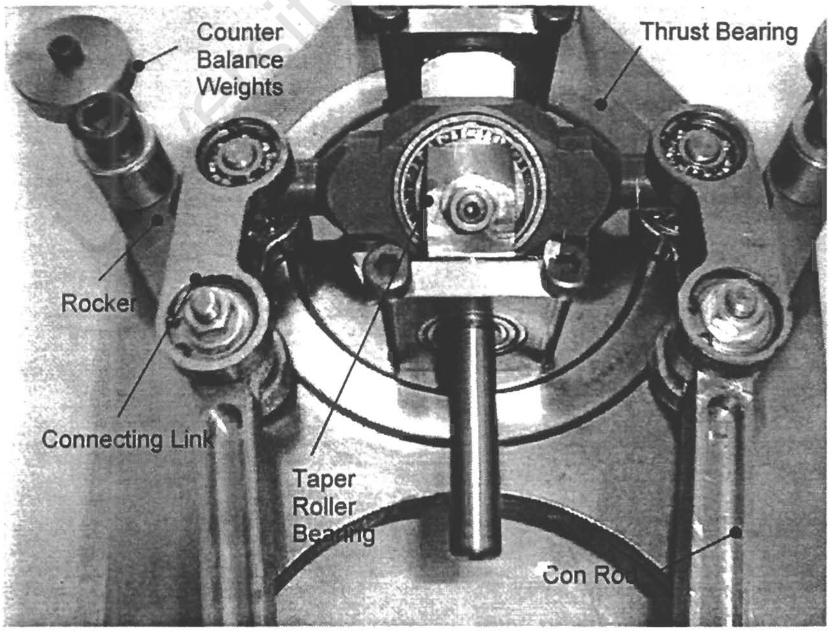


Figure B 8: Nutator and Associated Components

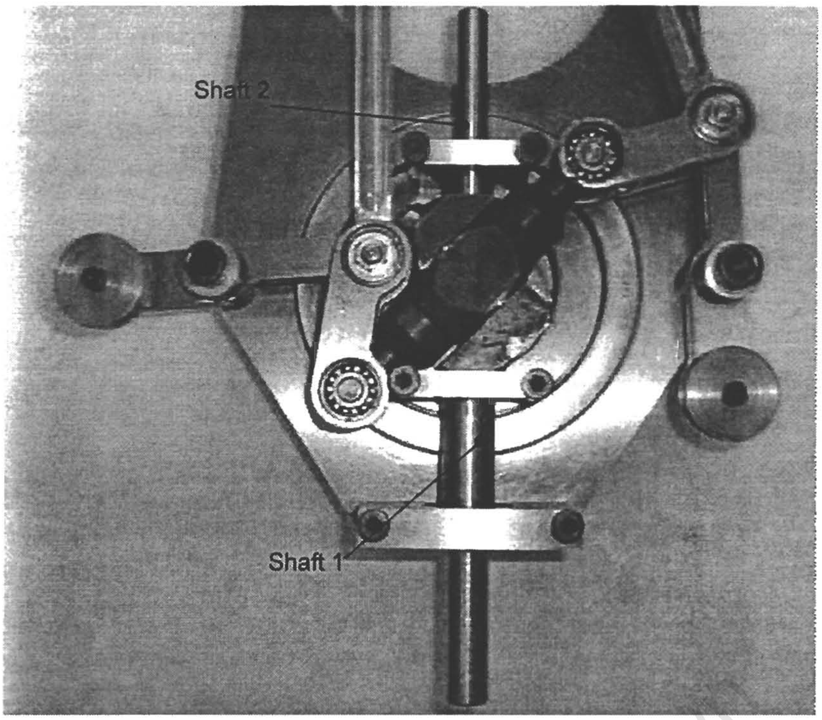


Figure B 9: Nutator (Revolved Through 90 Degrees)

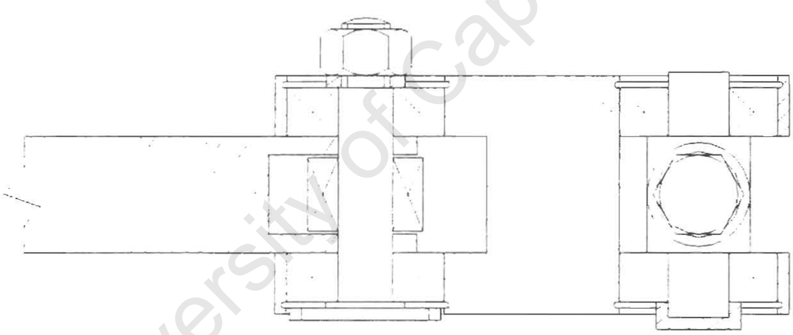


Figure B 10: Sectioned Sketch of Connecting Link

Heat Exchanger:

When pressurised, the exhaust air from the engine needed to be cooled before being fed back into the inlet port. An air to air heat exchanger was thus commissioned from Cape Heat Exchange.

It was planned to feed the air, which had already been used to cool the exhaust air into the furnace inlet. It would then be further heated as it passed through the double skin of the furnace, this would allow complete waste heat recovery. The heat exchanger was sized using the standard extruded section sizes available from Cape Heat Exchange. The external surfaces were shrouded and baffles put in to direct the flow of air through the heat exchanger and into the furnace. See figure B11.

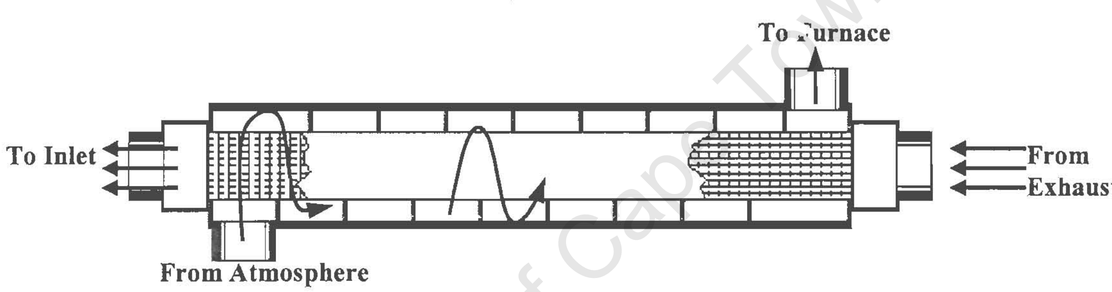


Figure B 11: Heat Exchanger

Experimental Investigation with Initial Engine

The thermodynamic model indicated that the engine should have sufficient power to turn itself over when operating from atmospheric pressure. Hence initial tests were carried out without pressurising the engine. Instrumentation consisted of a pressure transducer connected to a small hole drilled into the top of the cylinder head. An inductive transducer was used to signal the top dead centre position from a notch made in the flywheel. These transducers were coupled to an oscilloscope and a computer based data acquisition system so that pressure traces could be recorded and transformed into a Pressure - Volume diagram. Thermocouples were attached to selected heater tubes and onto the cylinder wall in order to monitor the temperatures.

First Phase Testing

The initial trials were not successful as the peak pressure was lower than expected once the engine was running at temperature (only 1.5Bar above atmosphere). This was only slightly more than when running cold, indicating that the gas temperature at the start of compression was higher than expected. Investigations with the computer model confirmed this hypothesis. It was concluded that the engine was unable to fully purge all the hot air. Heater tube wall temperature was about 500 ^{0}C.

To increase the operating temperature a blower was connected to the furnace. However, shortly afterwards, the engine suddenly lost compression when the temperature rose to over 1000 degrees. Upon disassembly, it was found that a large number of the heater tubes had melted due to the excessive temperature. See figure B12.

Scorched oil on the top of both pistons indicated that the air leaving the tubes was extremely hot. This was the case with all tubes, not just the ones that had melted through.

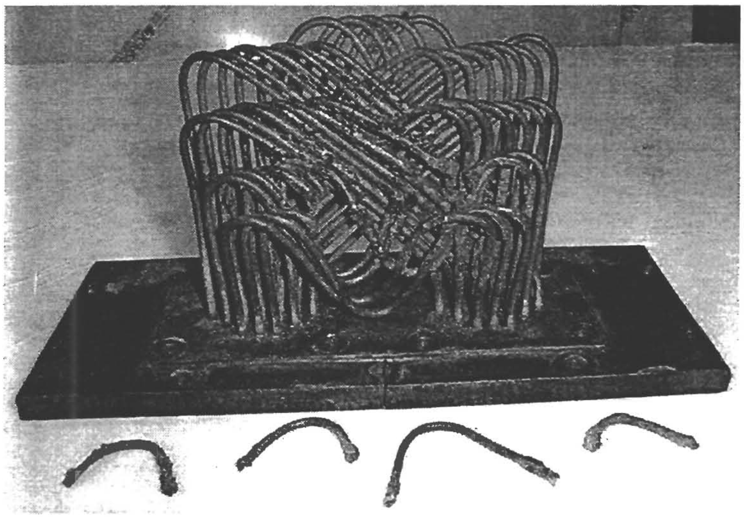


Figure B 12: Melted Heater Tubes

Second Phase Testing

A redesigned heater head was manufactured, which raised the tubes above the fire some 50mm higher than the earlier model in order to prevent them from getting too hot. See figure B13.

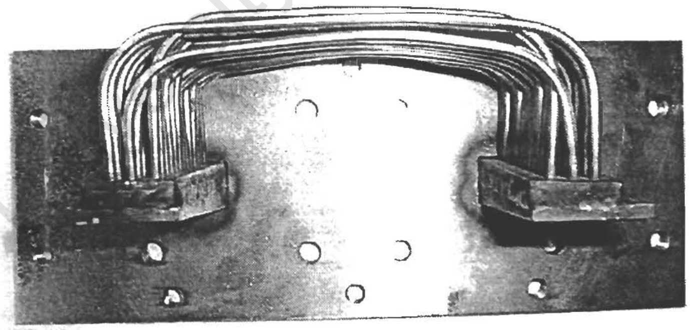


Figure B 13: Redesigned Heater Head

The engine was again tested, and although a slightly higher peak pressure was reached (1.8 Bar), the engine still showed no signs of producing positive work. At 300rpm the pressure drop across the heater tubes was measured to be about 0.2Bar.

Investigations into the friction of the engine were carried out. A piece of rope was wound around the driving pulley, which was attached to the flywheel and pulled using a spring balance to assess the mean rope force.

Table B 5: Friction Data for Engine

Configuration	Force (N)	Torque (Nm)
Without Barrels	6	0.57
With Barrels:	12.5	1.18
With cylinder head and barrels:	20N	1.9

Further tests were carried out in an attempt to improve the scavenging process. Initially an extraction fan was connected to the exhaust ports to help purge the hot air. Compressed air was then blown into the exhausts at an upward angle in an attempt to force cold air to the top of the cylinder and thus expel the hot air in that region. This showed a slight but not significant improvement.

The possibility of injecting small quantities of water into the inlet was considered. Initial investigations using adaptations in the form of an increased temperature in the computer model showed that, provided sufficient heat transfer could be attained, the performance would greatly increase.

Tests were conducted by spraying a fine jet of water into the transfer port through a hypodermic needle mounted on the side. This produced a slight improvement, but not of the magnitude that was expected. This concept was abandoned because the heat transfer was not considered to be high enough for flash boiling of the injected water. On some occasions, the amount of water building up in the heater tubes and cylinder caused additional problems in the form of hydraulic lock.

Investigations with soapy water sprayed around the joints between the heater tubes and cylinder head showed at least 2 leaks when the engine was pressurised. Investigations and modeling a leak with the spreadsheet model showed that a 0.2mm hole could have a significantly negative effect on the performance of the engine.

Third Phase Testing

A third heater head was constructed. This time, for ease of swaging and welding and due to the large pressure drop associated with the small diameter tubes, a decision was made to use 3/16 inch tubes instead of the previous 1/8 inch.

In order to assist the scavenging process, it was decided to obtain another pair of pistons, leaving the top domed. Previously the pistons had been machined flat in order to reduce the clearance volume. In order to ensure the clearance volume was still a minimum, a corresponding dished hollow had to be machined into the cylinder head. It was thought that the domed pistons would help scavenging by deflecting the incoming air towards the top of the cylinder and thus expel the hot air in that region.

An electric motor was mounted to the frame in order to turn the engine over at a constant speed. The new cylinder head was mounted and pressure tested. No leaks were found around the welds, but the engine still lost pressure. It was determined that the leakage was from around the piston rings.

The engine was run in for about 2 hours. During operation a severe knock developed that seemed to emanate from the pins attached to the ends of the nutator. This knock was due to excessive end float on the system.

The furnace was fired and pressure traces were recorded whilst running the engine with the motor. On this occasion, most diagrams indicated that the engine was producing positive work, however it still showed no sign of being able to run without assistance.

The actual pressure – volume diagram from the engine differed significantly from that which had been theoretically predicted

This notable difference was thought to be due to hot air expanding out of the heater tubes during, and for some time after the scavenging process. Among other problems this would cause the starting temperature of the compression part of the cycle to be higher than predicted. In addition the thermodynamic model did not adequately take into account the time taken for the gas to heat up as it passed through the heater tubes.

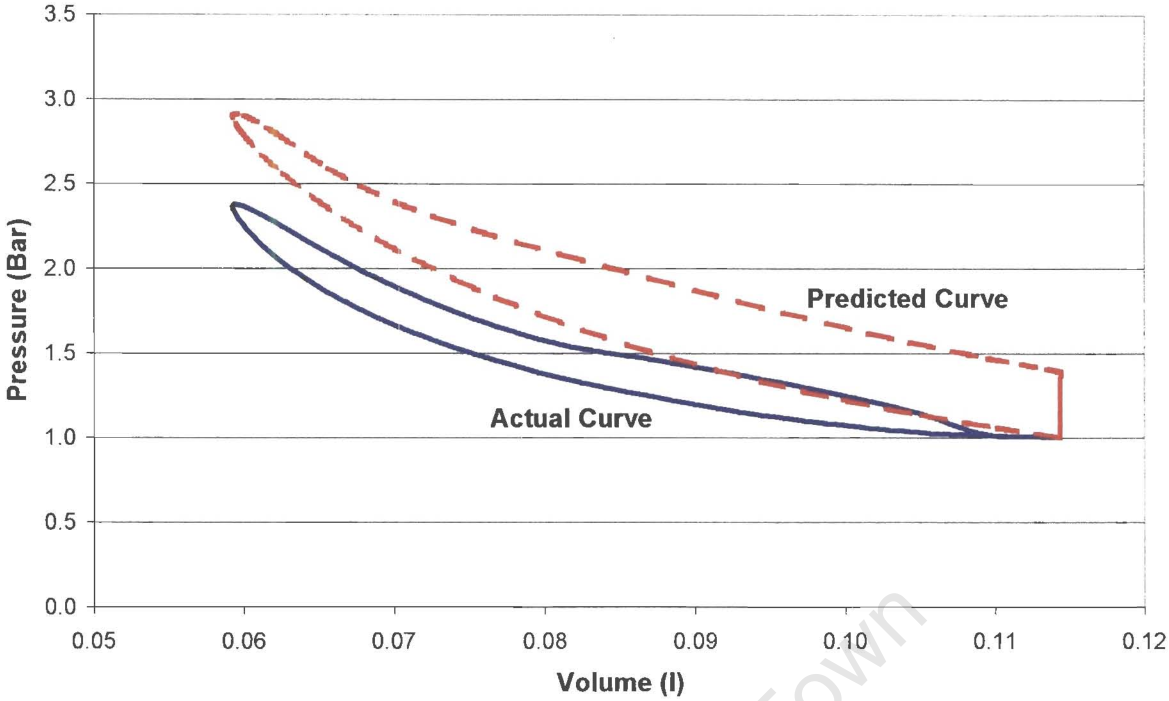


Figure B 14: Actual and Predicted P-V Diagram

Fourth Phase Testing

During the next phase the engine was pressurised. The heat exchanger was plumbed into the system, connecting the exhaust to the inlet manifold. A hose directly from the high-pressure shop air was connected to make up for leaks in the system, and control the pressure. The experimental setup is shown in figure B15.

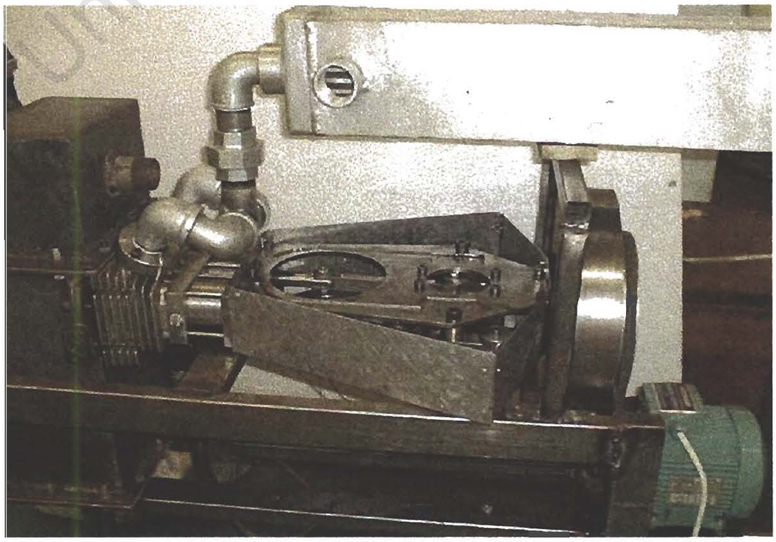


Figure B 15: Experimental Rig

The measured curves differed considerably from those predicted.

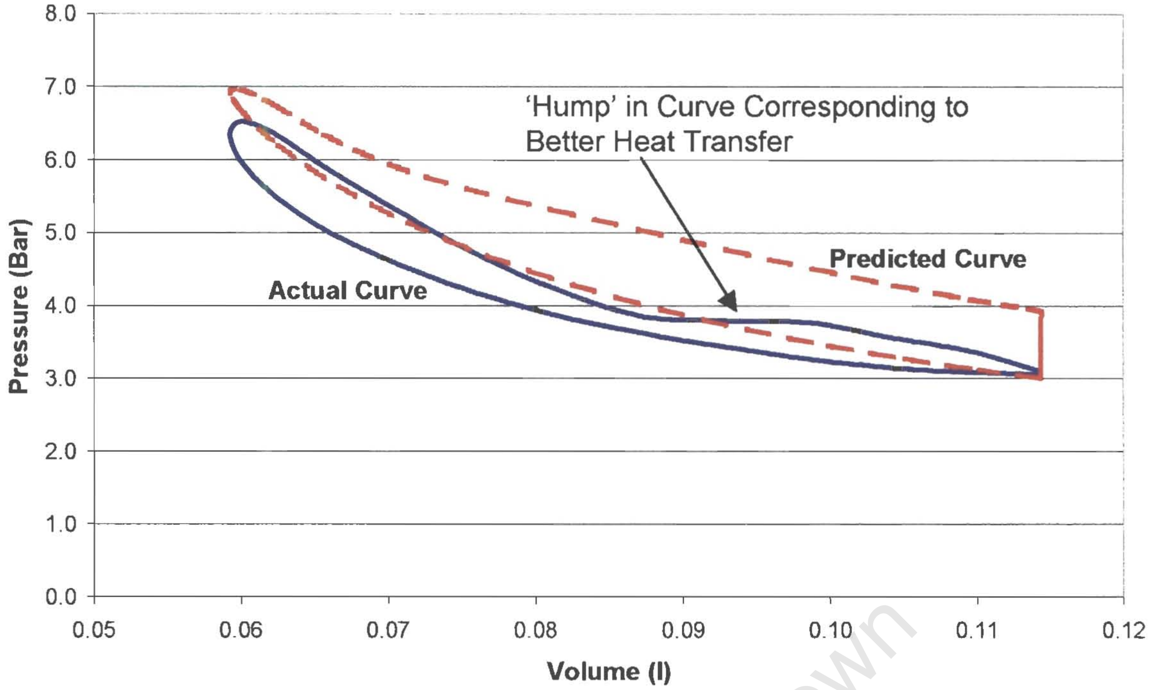


Figure B 16: Pressurised P-V Diagram

Nearly all the curves obtained had a 'hump' in the middle of the expansion stroke, (figure B16) which could correspond to an increase in heat transfer at that point. This would seem to indicate that the heat transfer up until that point was too low and then began to improve.

The Mean Effective Pressure did not rise significantly, but did reach 0.8 Bar for one set of data. The friction in the system would also tend to increase with pressure, thus still not allowing the engine to run.

Heat Transfer Investigations

Investigations were carried out to determine the heat transfer inside the heater tubes. This consisted of mounting the cylinder head onto the furnace with a modified plate bolted to it so as to allow air to be blown through the heater tubes. Air was blown through at different velocities measured with a rotameter. Velocities were chosen which would be close to those experienced while the engine was running under normal conditions. The inlet temperature, outlet temperature and the pressure difference measured with a U tube manometer were recorded for each flow rate.

Flexible rubber hoses were used to connect the manometer and the inlet air supply. This limited the temperature at which the tests could be run. Nevertheless comparisons could still be made by setting the temperatures in the spreadsheet model similar to those tested.

The heat transfer coefficient was then calculated from:

$$hA(T_{wall} - T_{mean}) = \dot{m}\,Cp(T_{out} - T_{in})$$

Table B 6: Results of Heat Transfer Investigation

Flow Rate	T_{wall} (Average)	T_{in}	T_{out}	T_{mean}	Pressure Difference	Mass Flow	velocity	h_{Actual}	$H_{Predicted}$
ℓ/s	°C	°C	°C	°C	bar	g/s	m/s	W/m^2°C	W/m^2°C
1.22	134	19	124	71	0.0034	1.23	13.43	87	149
1.22	190	19	134	76	0.0034	1.21	13.43	62	127
1.50	107	19	101	60	0.0043	1.56	16.52	114	189
1.50	214	19	143	81	0.0048	1.47	16.52	67	143
1.67	191	19	139	79	0.0054	1.64	18.37	74	163
1.22	146	19	122	70	0.0031	1.23	13.43	71	143
1.22	105	19	102	60	0.0030	1.27	13.43	100	162
1.67	136	19	118	68	0.0050	1.69	18.37	105	188
1.67	120	19	112	65	0.0048	1.71	18.37	123	199
1.67	126	19	114	66	0.0048	1.70	18.37	115	194
1.67	123	19	114	66	0.0049	1.70	18.37	120	196

The values obtained were about 45% lower than those predicted with the model. This was deemed acceptable as the turbulence generated by the piston motion would cause the Reynolds number and hence the heat transfer coefficient to be higher when the engine was running. Figure B17 shows the normalised heat transfer coefficients for a number of the tests run.

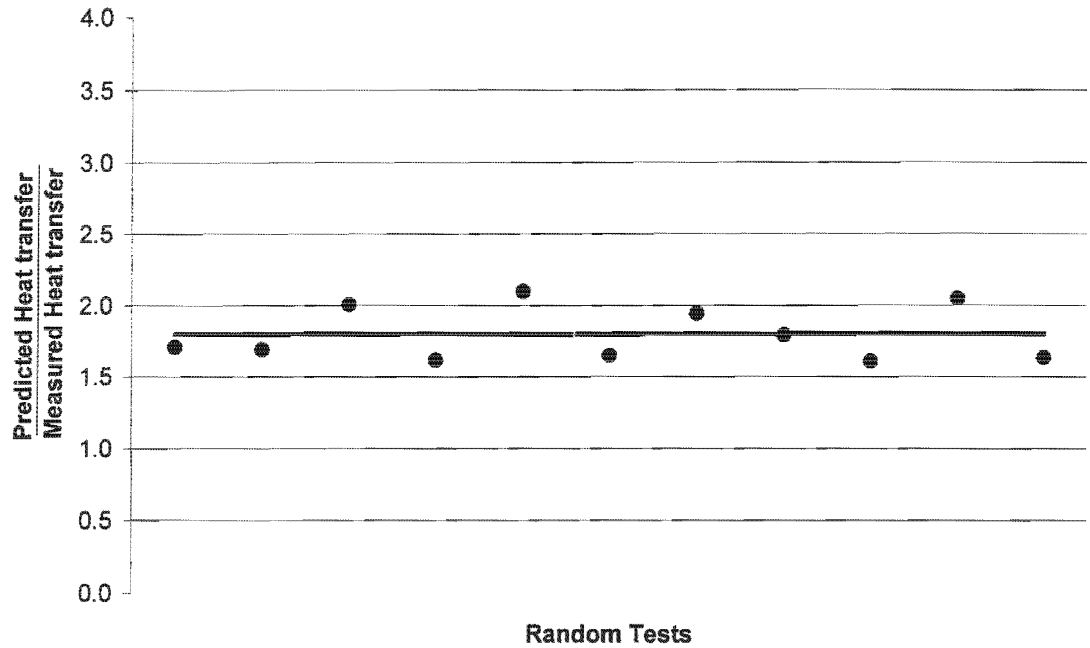


Figure B 17: Normalised Heat Transfer Coefficients for a Number of Different Test Setups

Discussion of Results

The experimentation with the 3-process engine revealed a number of weaknesses with its design and operating principles, which are examined below in more detail below. However, certain key elements were proved valid and should be retained.

The idea of optimising the mean effective pressure, even at the expense of efficiency, proved worthwhile in the light of the low specific performance of a hot air engine.

The necessity to pressurise the engine also showed itself to be an intrinsic requirement of this type of engine, if any reasonable power is to be produced.

2-stroke scavenging

The 2-Stroke principle in an internal combustion engine is imperfect at best, and was found to be inapplicable to the design of a hot air engine. The lack of turbulence, (which is created by the combustion process in an internal combustion engine) means that the cylinder is unable to purge the hot air effectively. To make this situation worse, hot air in the heater tubes did not flow through as quickly as was envisioned. Instead, the air seeped slowly out of the heater tubes throughout most of the piston's scavenging process and for some of the compression stroke. This meant that the temperature of the air in the cylinder at the beginning of the compression stage was excessively high. This was thought to be the reason that the peak pressure was never as high as had been predicted.

It was therefore concluded that a proper intake and exhaust process would be essential in a hot air engine that incorporates valves.

Flow reversal

The concept of reversing the flow direction in the engine, each cylinder alternating between expansion and compression, involved certain serious disadvantages. The most severe penalty related to the condition where the cylinder wall temperature was unacceptable both in terms of the compression, when it would be too hot and the expansion, when it would be too cold.

A Uni-flow design would circumvent this drawback, comprising a hot or expansion zone and a separate cold or compression zone.
Obviously this would not apply to a regenerator if one was to be incorporated, since the principle relies on heat energy recovery by virtue of the flow reversal.

Friction and speed ratios

For the 180 degrees of driveshaft revolution that made up a complete cycle, only one third was used for the compression and expansion strokes. This was due to the fact that a large proportion of the pistons stroke occurred whilst the transfer or exhaust port was open.
The fact that there were two complete cycles per revolution meant that the instantaneous piston sliding speeds and accelerations were high, thus producing excessive friction and mechanical losses.

A one to one ratio for crankshaft revolution and piston stroke should be used as far as possible.

Expansion ratio

The engine expanded the high-pressure air down to atmospheric pressure. This was inefficient because expansion below the frictional mean effective pressure of the engine does no useful work. The elevated pressure would be better put to use purging the hot air from the cylinder when the exhaust valve was uncovered.

For an Ericsson type engine, the exhaust valve should be opened when the cylinder pressure equals the FMEP. In a Stirling type engine, expansion should be terminated at that point.

Pressure and Friction

The buffer pressure of the engine was at the minimum cycle pressure, (diagram A) meaning that during the compression and expansion stroke the pressure differential across the piston was large. This had implications with sealing, as the high-pressure difference caused greater leakage past the piston rings.

There was also a higher piston ring friction due to the differential, and greater loading on the conrod from pressure forces causing increased losses in the drive mechanism.

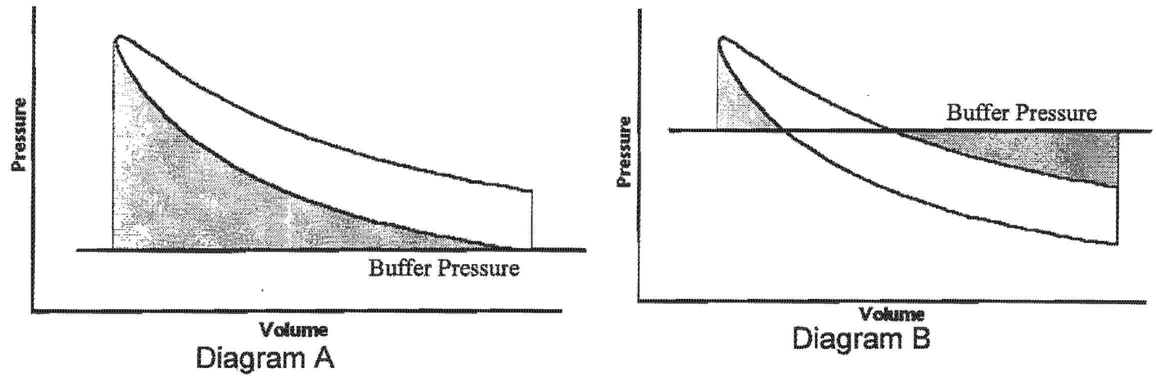


Figure B 18: Effect of Buffer Pressure on Work Done

Therefore, if an engine is buffered at the mean cycle pressure, (diagram B) the pressure differential and hence the friction will be at a minimum for the entire cycle. There will also be the added advantage of decreased leakage from the working space [12].

Tubes and dead volume

Heater tubes are beneficial in that they greatly increase heat transfer, but they also have a drawback in the dead volume associated with them. This increase in dead volume results in a decrease in the pressure ratio with a resultant drop in engine performance. Dead volume in the heater zone is particularly limiting due to the following:

The basic equation governing the temperature change in a polytropic process is:

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

Therefore when the pressure is increased, the temperature of the already hot dead space gas is raised even further by the heat of compression. This would cause the temperature to rise above that of the heater tubes, and thus lose heat! This is not the case with a regenerator as the temperature gradient ensures heat transfer in a manner that is useful to the engine performance.

There should be minimal dead volume in the heater of any future designed engine. The concept of heater tubes should be carefully considered, due to their propensity to fail under conditions of light engine load where overheating of the tubes can easily occur.

Mechanism

The mechanism itself, although adequately fulfilling the requirements in terms of the desired motion, was too complex. The manufacturing and packaging of the system was extremely difficult and involved some compromises in the factor of safety that applied to certain elements.

The resultant motions of the various components also served to produce excessive acceleration.

A beta or gamma type engine would be preferable especially if the motion of the piston involves higher order harmonics.

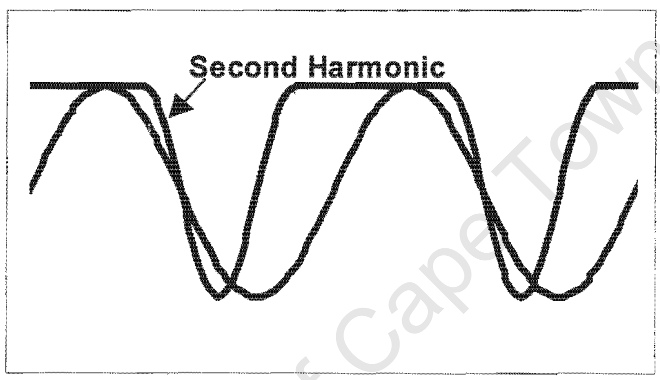


Figure B 19: Piston Displacement Showing Higher Harmonic

It would be preferable to use a conventional crankshaft for the piston mechanism.

Engine temperature

The reversing flow direction and the fact that the cylinder barrels were bolted directly onto the furnace all served to raise the cylinder wall temperature too high. This not only caused the temperature of the working air to be too high, but also raised the temperature of the rings to an unacceptable level, on occasion causing them to swell and stick in their grooves, thus not allowing them to seal properly.

All rings and valves should be kept in the cold part of the engine. A regenerator would enable this type of design.

Furnace

The furnace on the whole worked well, especially the gravity assisted coal feed.

It was possible for the furnace temperatures to become excessively high, causing material failures in the heater tubes. The fire had to be carefully controlled on a continuous basis to ensure temperatures did not rise above 1000 degrees Celsius. The excessive pressure drop through the double skin did not allow it to function as desired, thus raising questions on this added complexity. The furnace design would have been significantly simpler using insulation to recover waste heat instead of a double skin.

The mounting bolts for the engine, which protruded into the furnace seized when hot, causing problems when disassembling the system.
After about half an hour of operation clinker build up on the grate became a problem, starving the fire and eventually after extended periods, extinguishing it.

Future furnaces should be designed with some form of clinker ejection system, and all nuts and bolts should be kept external to the furnace.

Appendix C: Thermodynamic Engine Model

For the analysis, the engine was considered to consist of an expansion zone, a regenerator zone and a compression zone which was modelled independently. The compressor and expander were assumed to work between a high-pressure and low-pressure reservoir. Flow between these reservoirs and the expansion zone was controlled in the model by a variable diameter valve. See figure C1.

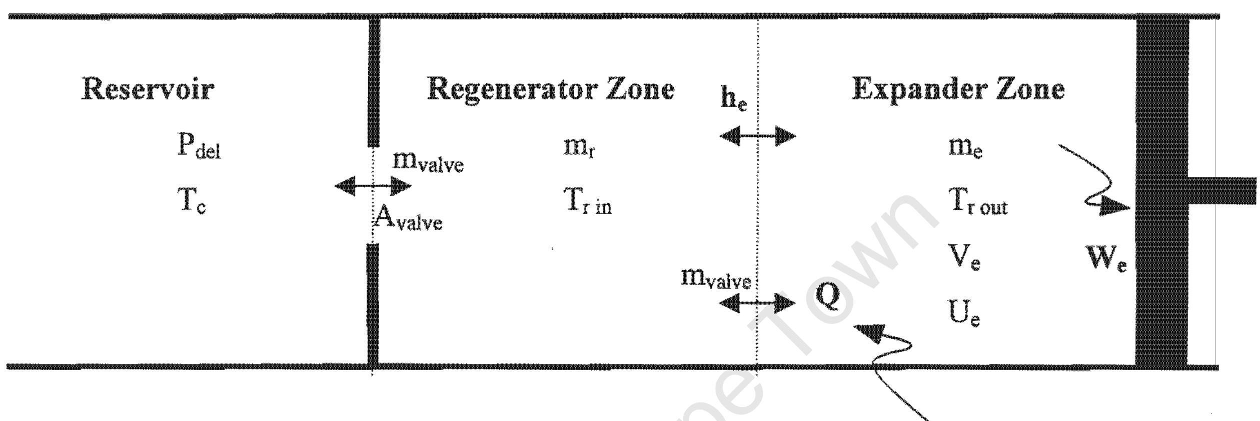


Figure C 1: Schematic of Model Operation

Expansion and Heater Zone

A separate spreadsheet was set up to model the mechanism of the engine.
From this simple crank and slider simulation, it was possible to calculate the volume in the expander (V_e) at any given time.

The work done by the gas was then calculated using:

$$W = \frac{P_1 + P_2}{2}(V_2 - V_1)$$

The heat transfer to the zone from the external environment was then found using

$$Q = A_{se} h_e \Delta t (T_2 - T_1)$$

The heat transfer coefficient was estimated with the following empirical formula:

$$h_e = \frac{0.76 * k_{air} \operatorname{Re}^{0.64}}{D} \qquad [20]$$

Where:

$$Re = \frac{\rho_{air} v_{ave} D}{\mu_{air}} \text{ (The Reynolds Number)}$$

$$v_{ave} = \frac{Stroke}{\Delta t_{cycle}} \text{ (Average Piston Speed)}$$

$$\Delta t_{cycle} = \frac{\theta_{end} - \theta_{begin}}{360} \frac{60}{rpm} \qquad \text{(Time for each cycle)}$$

The internal energy of the zone was then found:

$$U = \frac{P_2 V_2 - P_1 V_1}{\gamma - 1}$$

Rearranging the basic energy equation:

$$Q - W + \Delta m C p T^* = \Delta U$$

to obtain:

$$\Delta m = \frac{(\Delta U - Q + W)}{Cp \cdot T^*}$$

The change of mass in the expander could be determined depending on the direction of mass flow.

If (U-Q+W)<0 — Δm is positive (air is leaving the zone) therefore $T^* = T_{ic}$ (temperature of expander zone)

If (U-Q+W)>0 — Δm is negative (air is entering the zone) therefore $T^* = T_{ir}$(temperature of the regenerator zone)

This was then added to the previous mass found to give the mass of air in the expander for that step.

The initial mass being:

$$m_1 = \frac{P_{e,1} V_{e,1}}{R T_{e,1}}$$

The initial temperature of the zone being an input made equal to the final temperature by the calculation macro.

The energy transfer could then also be calculated: $\Delta m_e C p T^*$

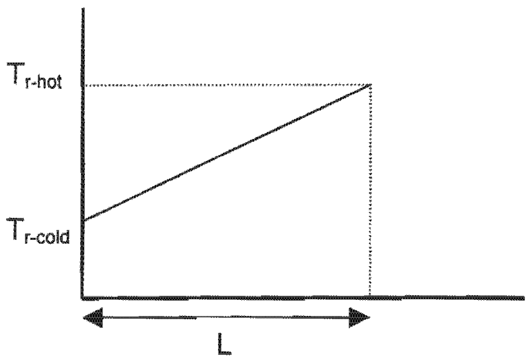
The temperature of the air was computed using the ideal gas equation.

$$T_e = \frac{P_e V_e}{m_e R}$$

Regenerator Zone

The volume of the regenerator zone was constant and thus the work done was zero.

The regenerator was assumed to have a linear temperature gradient.



$$T = T_{r-cold} + \left(\frac{T_{r-hot} - T_{r-cold}}{L} \right) x$$

The mass in the zone was computed using:

$$m_r = \frac{PV}{R(T_{r-hot} - T_{r-cold})} LN \left(\frac{T_{r-hot}}{T_{r-cold}} \right)$$

Obtained from the integration of the ideal gas equation.

$$m_r = \int_0^L \frac{P \cdot A \cdot dx}{R \cdot T}$$

The change in mass of air for the regenerator could thus be found

$$\Delta m_r = m_{r2} - m_{r1}$$

The outlet temperature of air from the regenerator was assumed to be constant. This was felt to be accurate because the regenerator would have a fairly large thermal inertia and thus its overall temperature gradient would not change significantly during the intake or exhaust process. The outlet temperature was determined using the conservation of energy principal. The energy transfer in the regenerator would be constant, so the sum of the individual energy transfers in the expander zone for each step must be equal to zero. A goal seek was thus used to make the sum equal to zero by adjusting T_{r-hot}.

Valves

The velocity of air flowing through the valves was calculated using:

$$|v| = \sqrt{\frac{2|\Delta P|}{\rho_{air} K}}$$

Where

K = The entry loss coefficient for the valve, assumed to be 0.5 [25]

The pressure differential was also found

$$\Delta P = P_i - P_{external}$$

Where $P_{external}$ = Same as high-pressure reservoir, if inlet valve is open.
Same as low-pressure reservoir in exhaust valve is open.
P_i if both valves are closed.

From the velocity, the mass flow rate could de calculated

$$\dot{m} = |v| A_{valve} \rho \qquad \text{if } P_i < P_{external}$$

$$\dot{m} = -|v| A_{valve} \rho \qquad \text{if } P_i > P_{external}$$

and hence the change in mass:

$$\Delta m = \dot{m}\,\Delta t$$

$\Delta m_{delivery} = \Delta m$ if $\Delta m > 0$

$\Delta m_{delivery} = 0$ if $\Delta m < 0$

The density of the air flowing through the valve had to be determined.

$$\rho_{air} = \frac{P}{RT_{regen}}$$

Where

T_{regen} = Regenerator temperature, assumed to be constant and the average of $T_{r\text{-}hot}$ and $T_{r\text{-}cold}$

For the speeds the engine was to run at, the time to open or close the valves would be influential, and hence had to be modelled, this was done by working out the area of the valves at each time step.

$$\delta A = \frac{A\Delta t}{\Delta t_{valve}}$$ Positive if θ is after the valve opening pulse

Negative if θ is after the valve closing pulse

Where

Δt_{valve} = Valve opening time (Supplied by manufacturers)

The valve area could then be obtained from the previous area, and the change in area.

$A_{valve,i} = \delta A + A_{valve,i-1}$ if $A_{test} < A$

$= A$ if $A_{test} > A$

Where A is the maximum area of the valve

The mass balance for the system then becomes $\Delta m_e + \Delta m_r - \Delta m_{valve}$

The pressure, assumed constant throughout the system was adjusted to make the mass balance for each time step equal to zero. This was done using a goal seek command in a macro that ran through each time interval.

Compressor

The volume variations for the compressor were obtained using the same mechanism analysis spreadsheet that was used for the expander.

The spreadsheet was designed to model different compression processes. A polytropic coefficient was be chosen for the compression process and a pressure calculated for each volume, using:

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^{\gamma}$$

Adiabatic compression was assumed for this engine due to the high speeds it would run at.

The upper and lower bound on the compression curve were set to a delivery pressure P_{max} and an inlet pressure P_{inlet}.

The work at each step could then be found

$$\Delta W = \left(\frac{P_1 + P_2}{2}\right)(V_2 - V_1)$$

The compression temperature was also calculated using,

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

This allowed the mass in the compressor to be found using the ideal gas equation.

$$m = \frac{PV}{RT}$$

Another goal seek macro was then used to make the mass delivered by the compressor equal to the mass required by the expander. This was achieved by adjusting the diameter of the compressor cylinder.

The mass delivered by the compressor was found by subtracting the maximum mass from the minimum mass in the compressor. The mass required by the expander was found by summing the masses at each time step in the regenerator zone.

The diameter of the expander then had to be adjusted to get the diameter of the compression piston equal to 60mm as this was the size of the existing cylinder from the lawn mower engine chosen.

Appendix D: Valve Control

The switching of the valves was accomplished using a Qbasic program to control a PC30 card. Outputs from this card were connected to transistors used to switch the valves.

A 5 volt output from the PC30 card was connected across the points of the engine. This was used to trigger a timer in the program to calculate the engine speed.

A valve-timing array was generated for a number of different speeds in order to ensure the valve timing changed proportionately with the engine speed. A variable called 'valve' was also setup, corresponding to the state of each valve.
1 = Inlet Valve Open
2 = Inlet Valve Close
3 = Exhaust Valve Open
4 = Exhaust Valve Close

The valve timing array consisted of a time for each valve state shown above, for a number of different speeds. For each run of the loop, a counter was compared to the value in the array for that valve state at the calculated speed. If the count exceeded the value in the array, then a variable 'valve' was increased. This allowed the valve timing to be changed while the engine was running by increasing or decreasing the value in the array corresponding to the desired valve state.
The output was set to 1 to open the inlet valve, 2 to open the exhaust valve or 0 to close both valves. These numbers corresponded to a voltage from one of two ports on the PC30 card. The input port of the card was then read and the counter incremented.

For each revolution the system was armed when the points were open. As soon as the points closed again, the value for the speed was calculated, the variable 'valve' set to 1 and the system disarmed in preparation for the points to open and arm the system again. See figureD2.

The output signal from the PC30 card consisted of a 5 volt pulse that was used to switch a transistor. Two identical circuits were used for the inlet and exhaust valves, each was connected to a digital output port on the PC30 card and a 12volt power supply to shift the solenoids.

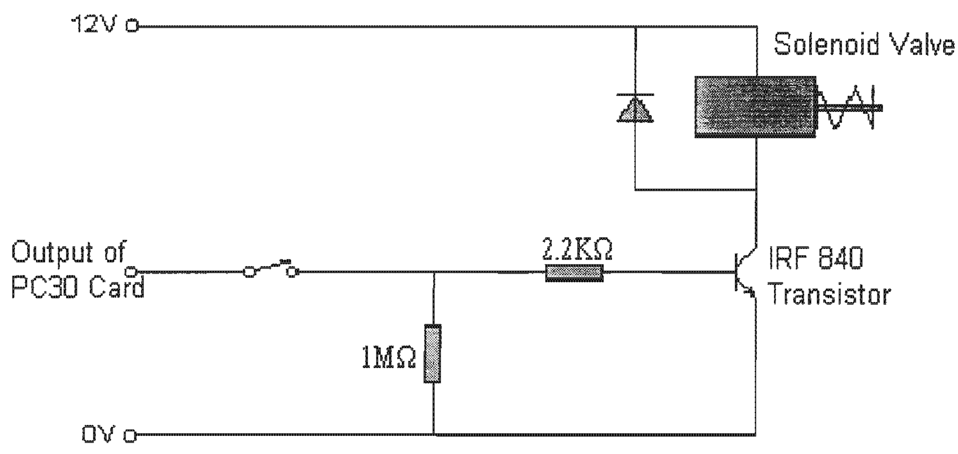


Figure D 1: Valve Control Circuit

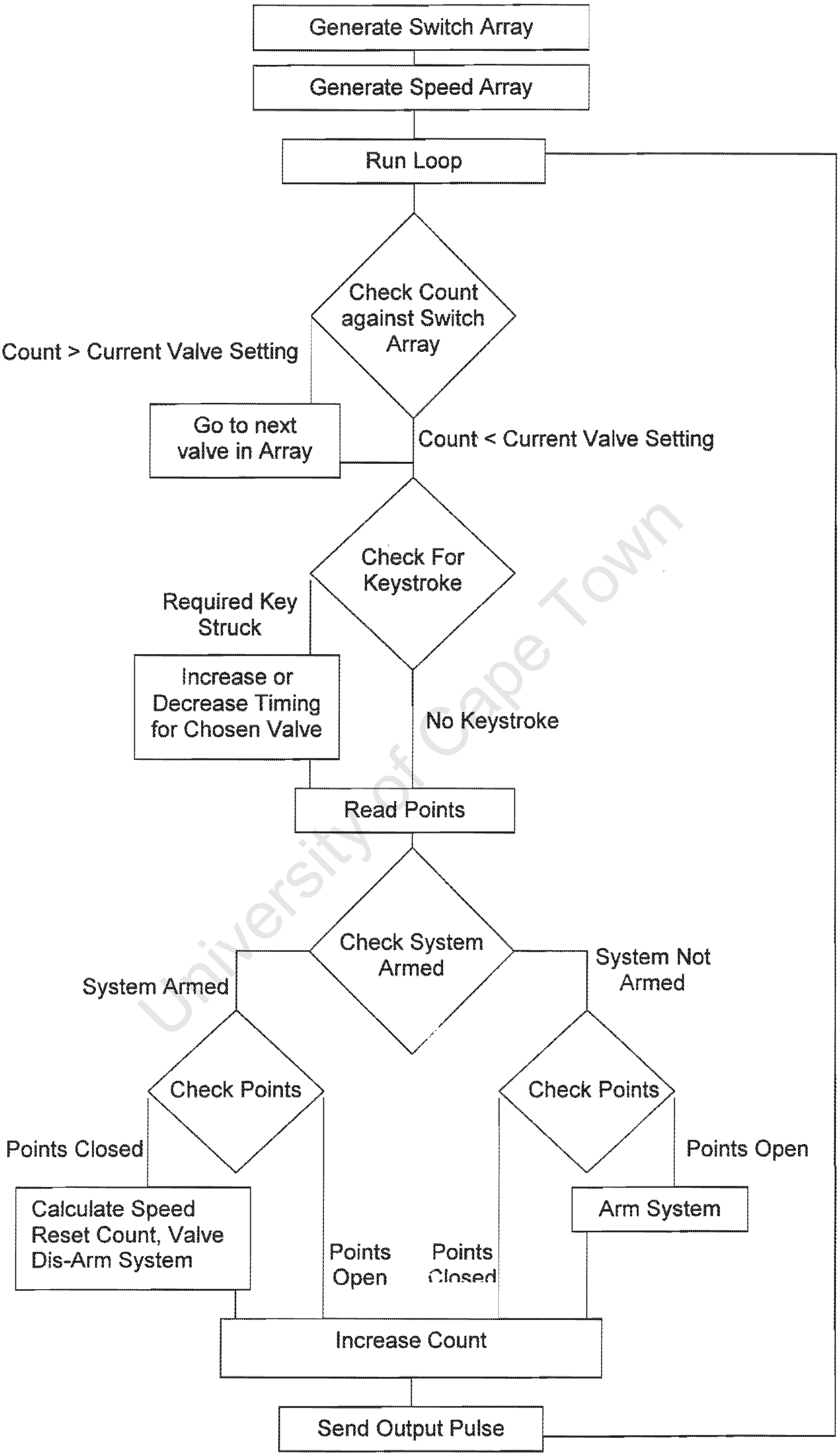


Figure D2: Flow Diagram for Valve Control Program

Appendix E: Initial Tests

For the initial tests and for the running-in stage, the engine was operated using compressed air. The high-pressure region was pressurised using shop air controlled by a pressure regulator, while the low-pressure reservoir was vented to atmosphere.

First Phase Testing

During the initial trials the engine ran well, although the pressure traces obtained were not as predicted. The valves did not appear to be shutting properly. This was initially thought to be due to the high (unregulated) pilot air, preventing the valves from exhausting properly. The valves were therefore modified to utilise the internal air pressure for piloting purposes. The resultant pressure trace seemed slightly better but there was still evidence of a long delay in valve opening. As such, the valve timing had to be advanced considerably.

Running in this manner with no heater, the air cooled considerably during the expansion stroke and the engine thus functioned as a refrigerator. Temperatures well below zero were reached (-15°C) and ice crystals could be seen forming on the top of the expansion chamber, Thus indicated that the thermodynamic principles behind the design were sound.

With continued testing, the engine began to develop a knock, which became more severe as time progressed. Upon disassembly, it was evident that the splash lubrication system had not been functioning properly at the low engine speeds (about 250rpm maximum). The crankshaft had worn severely on its plain bearing as had the connecting rod, and there was a large amount of play in each. The Teflon piston ring had also worn allowing the expansion piston to come into contact with the cylinder wall. As a result of this unlubricated binding, deep scouring on both components had taken place.

A new crankshaft and crankcase had to be sourced.

Second Phase Testing

The new engine was assembled, this time making use of a forced lubrication system. It consisted of a solenoid valve that used the same timing and circuitry as the exhaust valve. This valve charged a small reservoir with high pressure shop air, which was then discharged from a tube below the level of oil in the crankcase, causing a 'spurt' of oil to lubricate the piston and small end bearing. See figure E1.

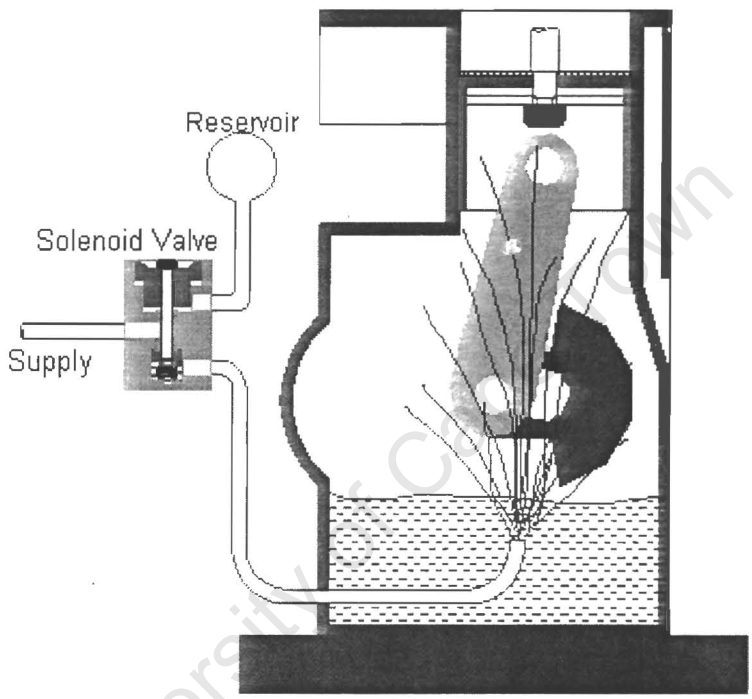


Figure E 1: Forced Lubrication System

The engine was once again run on compressed air, but still failed to give a satisfactory P-V diagram. The pressure traces indicated that the engine was not attaining an adequate compression or expansion after the valves shut. This evidence again pointed to inadequate valve closure.

The engine was allowed to heat up whilst running on compressed air, the low-pressure vent valve on the exhaust reservoir was then slowly closed to pressurise the system so that the compressor would be taking air from the low pressure reservoir. The shop air was left connected to the high-pressure region to make up for any leaks.

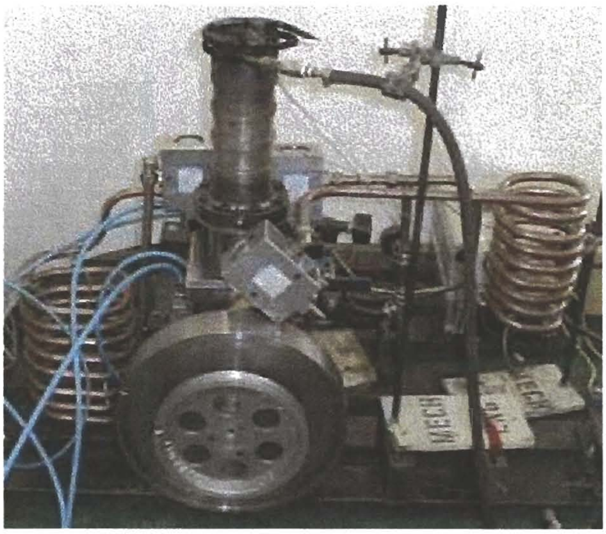


Figure E 1: Engine as Tested

The engine continually stalled just before the valve could be fully closed.
Mr Kerry Motherwell from Festo (the valve suppliers) was called in to advise on the valve closure problem. He established that the valves were failing to close properly because they were unable to exhaust adequately. This was due to the fact that the exhaust port was plugged to prevent pressure loss in the system when the valve switched. Hence the top seal could not lift off its seat due to a vacuum formed in the region directly underneath it as air could not enter this section.

A modification (figure E3) was added in the form of a small 2mm hole drilled in the valve to allow the trapped air to escape. The pilot air was then reconnected externally to the high-pressure shop air.

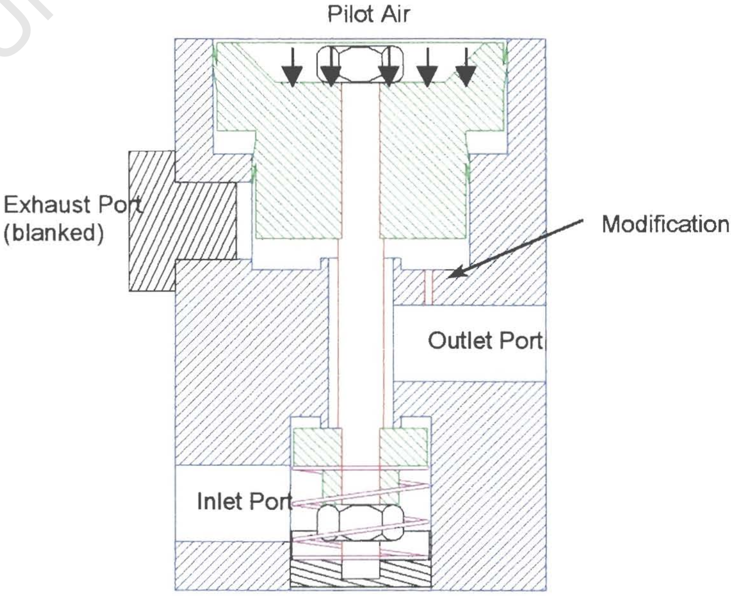


Figure E 2: Valve Modification

The engine was tested again, but still no satisfactory pressure trace was achieved. This time however, a better expansion was obtained, down to a lower pressure that had been previously unobtainable. The valve on the low-pressure reservoir was closed and the engine cranked over by hand. A number of P-V diagrams were obtained in this manner, but all were unsatisfactory and all had negative work associated with them.

It was deduced that the poor engine performance was due to air leaking past the Teflon seal from the high-pressure buffer zone to the expansion region. The pressure trace confirmed this, as there was a slight rise in pressure at the end of the expansion.

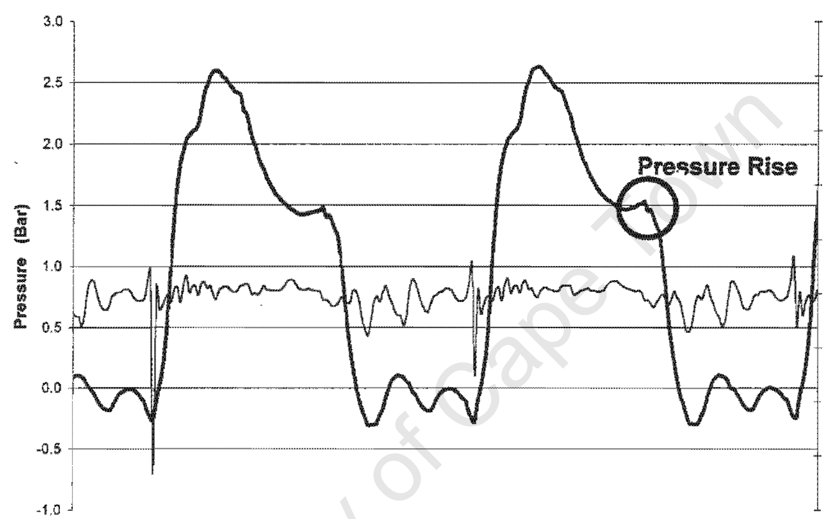


Figure E 4: Pressure Trace Highlighting Leak

This diagnosis was tested by pressurising the engine with both the exhaust valve and the low-pressure vent valve open. The low-pressure vent valve was then shut and the pressure rise in the reservoir timed (this pressure rise corresponded to the leak). It was found that the low-pressure reservoir pressurised in about 5 seconds when shop air at 3 bar was used. This was a considerably greater airflow than the compressor throughput at normal operating speeds.

Upon disassembly it was discovered that the Teflon seal had worn considerably and its diameter was now less than the diameter of the aluminium piston. As a result, the unlubricated sliding fit between the aluminium and steel had caused gouging on the bore.

A lip seal made of Rulon, as recommended by Walker was commissioned.

Rulon is a compacted PTFE material, which has a coefficient of friction similar to that of Teflon, but its rate of wear is several orders of magnitude better. The seal was manufactured by Wright Seal and Plastics of Cape Town.

The piston had to be modified to allow the seal to be fitted. As an added precaution, it was decided to fabricate the piston from brass to prevent binding in future operation.

Third Phase Testing

The engine was assembled with the new seal and piston, but the leak problem persisted.
It was found that the piston was sitting slightly off-centre in its bore, and thus air could leak past it on one side. The leak was tested, by tightening the barrel onto the cylinder head without the expansion cylinder, then pressurising the engine. The air leaking past the Rulon seal could be felt escaping on one side of the piston only.
The inner diameter of the Rulon seal was thus machined slightly oversize to allow it to move and find its own centre in the cylinder bore.
Despite this, there was still a leak identified. The Rulon tended to deform plastically under load, so the leak was believed to be caused by the lip deforming and not springing back to seal against the bore. A rubber O-ring was thus inserted under the lip to force it to expand against the barrel. The engine was then left pressurised overnight to try and encourage further expansion. This process significantly reduced the leak.

A variable speed motor was connected to the engine via a belt drive to allow the engine to be rotated while pressurised with the low-pressure vent valve closed. The engine was run several times with and without heat. It was also run with the low-pressure vent valve ranging from fully open, to fully closed, to provide different amounts of back-pressure on the system. Pressure traces were obtained for each trial.

The pressure traces still indicated an air leak. The expansion cylinder was removed again and the engine pressurised. No air was found to be leaking past the Rulon seal, instead investigations revealed that air was now leaking between the inlet and outlet ports of the compressor head. This was found to be due to the Vesconite rod seal sitting proud of its mounting plate. The seal was thus machined flush with its plate, and a new gasket cut. The engine was then reassembled and tested for leaks, this time the leak was found to be infinitesimal.

The engine was reassembled and readied for friction tests. See chapter 7.

Appendix F: Furnace Tests

Before the 3-Process engine was connected to the furnace, tests were carried out on the furnace to gauge its performance and maximum possible temperatures.

Thermocouples were attached to the blanking plate where the engine would normally mount, to measure air temperature about 40mm above the bed. The furnace ashtray and coal feed were sealed with rope seal, the coal feed stocked, and the furnace lit with the aid of a gas torch.

The furnace failed to draw through the double skin properly. However, with the ashtray open so the fire could draw air through that opening the furnace temperature rose to over 1000 ^{0}C. The furnace was unable to sustain this temperature because excessive ash and clinker build up on the grate began to extinguish the fire after about half an hour of operation.

The furnace was dismantled and the grate removed. The air feed holes were drilled bigger and the spacing between steel strips on the grate doubled.

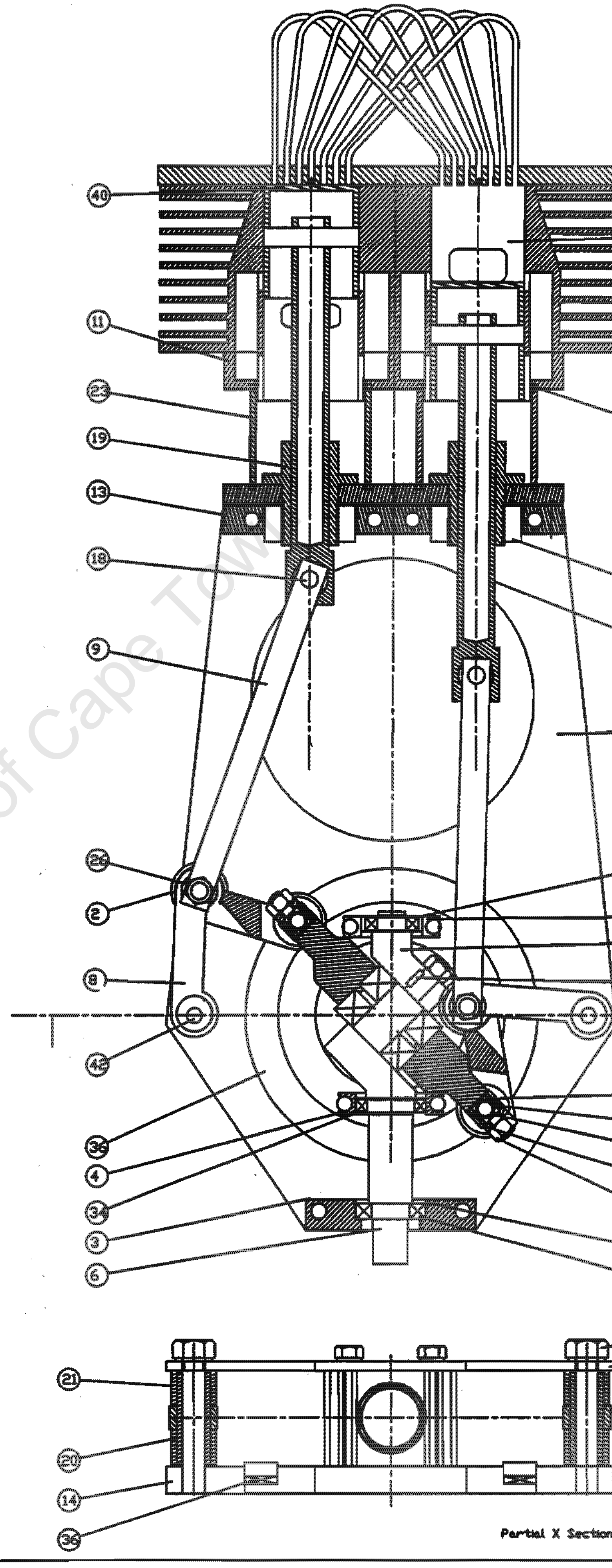
The furnace was fired up again, but still no significant improvement was noticed. The temperature was maintained at 800 ^{0}C drawing through double skin, but the fire was starved of air. A blower was then attached to furnace air inlet and air was blown through at about 6m/s. This produced a temperature in excess of 1200 ^{0}C. Clinker buildup was still a problem as the grate was soon clogged which choked the fire.

Due to the size coal used, the grate spacing could not be increased anymore without the unburned pieces of coal falling through.

'Riddling' the grate with a rod temporarily alleviated the problem. This was sufficient for testing purposes and thus the decision was reached to leave the design unchanged due to the complex nature of a clinker ejection mechanism.

Appendix G: Design Drawings

List of Drawings



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42
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21
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14
36
Partial X Section

PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS
45	M12 Bolt	M.S.	1	
44	4mm Dowel	M.S.	1	
43	Lock Nut	M.S.	2	KMT6
42	Oring	M.S.	1	ø2mm
41	Cylinder	1	2	
40	Piston	–	2	
39	Circlip	M.S.	1	ø26
38	Ball Bearing	–	4	629
37	Needle Bearing	–	4	NKI19/12
36	Needle Thrust Bearing	–	1	AXK120155
35	Ball Bearing	–	1	61905
34	Ball Bearing	–	1	61805
33	Ball Bearing	–	4	6000
32	Needle Bearing	–	2	HK1010
31	Ball Bearing	–	1	61804
30	Ball Bearing	–	1	61903
29	Ball Bearing	–	1	16002
[illegible]	Circlip	M.S.	1	ø15
27	Rocker Support Bottom	M.S.	2	
26	M8 Nut	M.S.	8	
25	Key	M.S.	1	3 x 3 x 10
24	Washer	Bronze	2	
23	Cylinder Base	M.S.	2	
22	Swash Pin End Cap	M.S.	2	
21	Rocker Support Top	M.S.	2	
20	Rocker Bush	Bronze	2	
19	Piston Rod Guide	Brass	2	
18	Conrod Pin	EN8	2	
17	ConLink Pin	EN8	2	
16	Swash Pin	EN8	2	
15	Top Engine Support	M.S.	1	
14	Bottom Engine Support	M.S.	1	
13	Cylinder Support Plate	M.S.	1	
12	Cylinder Backing Plate 2	M.S.	1	
11	Cylinder Backing PLate 1	Al	1	
10	Piston Rod	M.S.	2	
9	Connecting Rod	M.S.	2	
8	Rocker	M.S.	2	
7	Swash Support Shaft	M.S.	1	
6	Output Shaft	M.S.	1	
5	Bearing Mount 3	M.S.	1	
4	Bearing Mount 2	M.S.	1	
3	Bearing Mount 1	M.S.	1	
2	Con Link	M.S.	2	
1	Swash Plate	EN8	1	

UNIVERSITY OF CAPE TOWN
DEPARTMENT OF MECHANICAL ENGINEERING

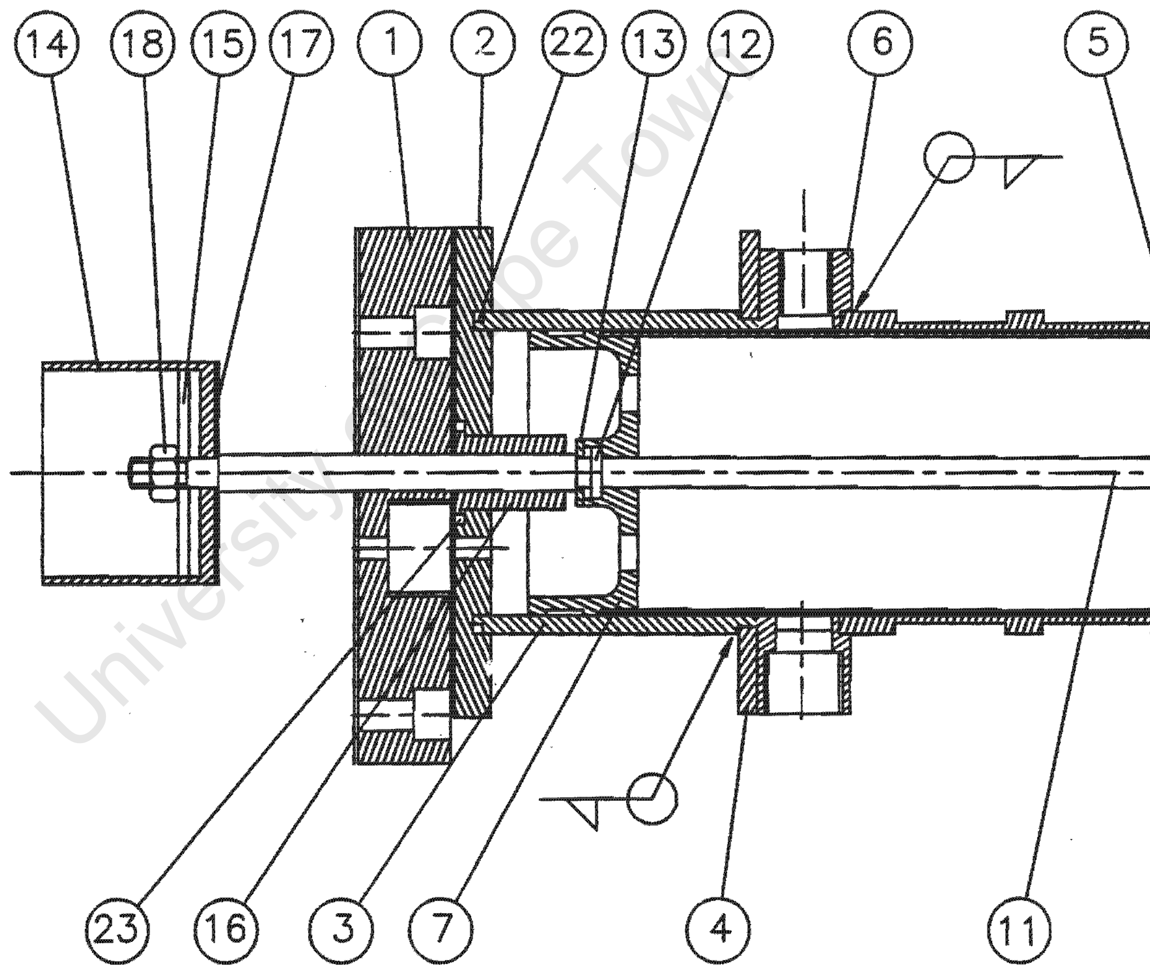
TITLE
Engine Assembly

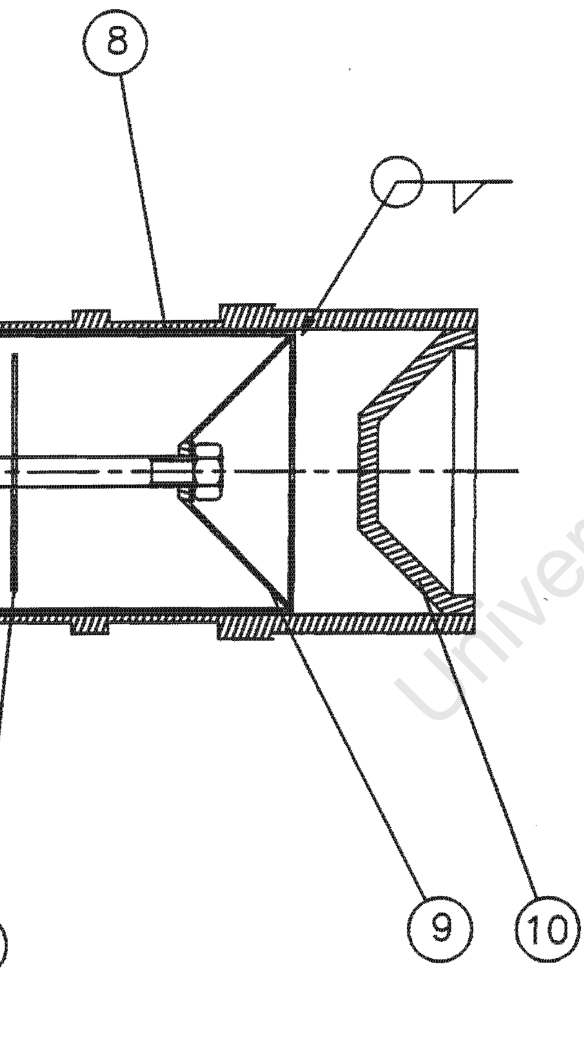
DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED 0.1

SCALE	DATE	SHEET	OF
1 : 3	30/03/00	1	17

DRAWN BY: J.HUSSEY

DRAWING NUMBER: 01-MECH-001-00


14
18
15
17
1
2
22
13
12
6
5
23
16
3
7
4
11



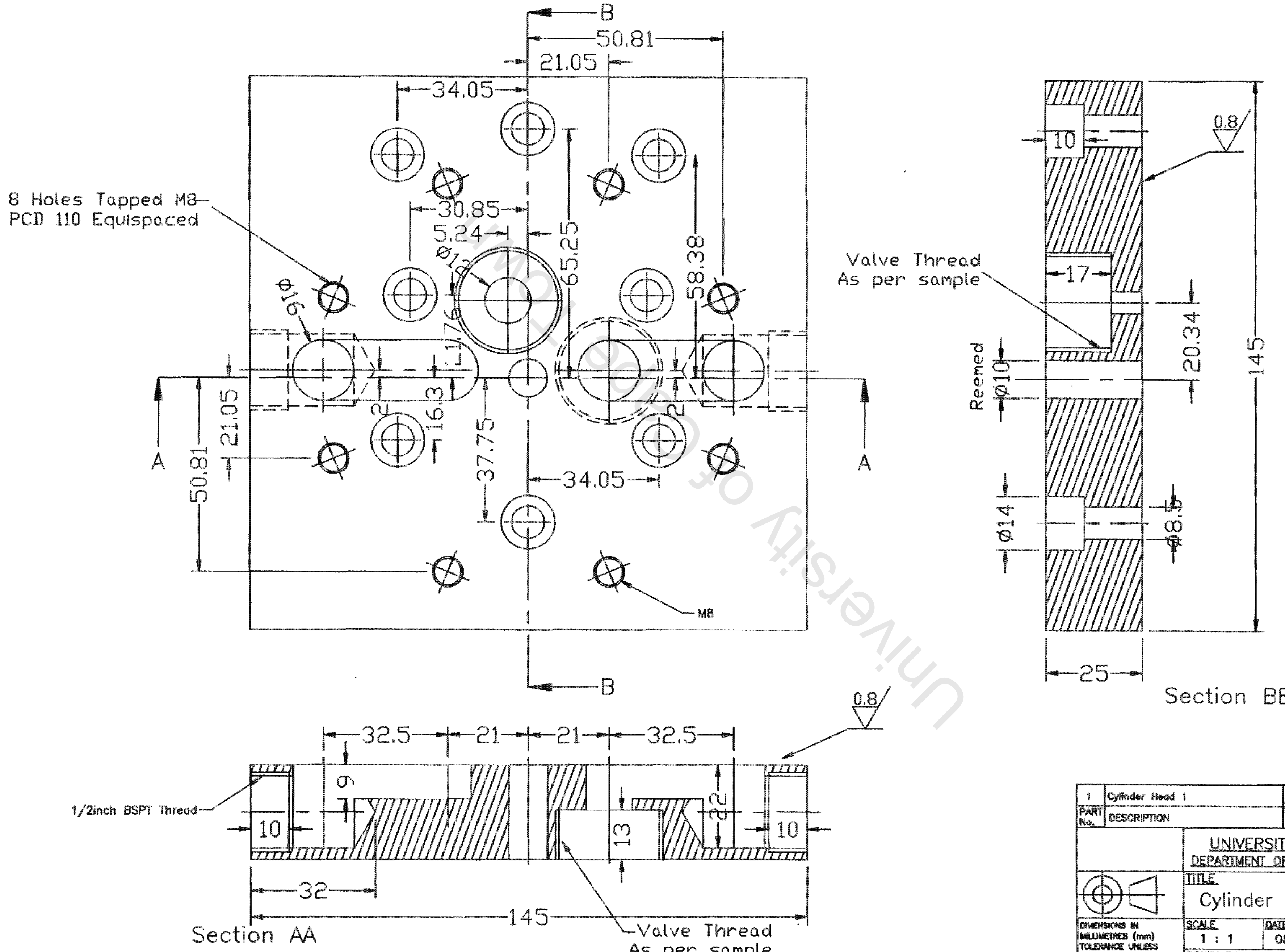
PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS
24	Lip Seal	Rulon	1	
23	3mm Oring	–	1	ø20
22	3mm Oring	–	1	ø83
19	Heat Shield	Al	1	
18	M8 Nut	M.S.	2	
17	Washer	M.S.	1	
16	Bush	Vesconite	1	
15	Rod Support	M.S.	1	
14	Compressor Piston	–	1	
13	Washer	M.S.	1	
12	Pin	–	1	ø3
11	Piston Rod	S.S	1	
10	Cylinder 2 Top	M.S.	1	
9	Regenerator Top	M.S.	1	
8	Regenerator	M.S.	1	
7	Piston	Al	1	
6	Cylinder 2 Flange	M.S.	1	
5	Cylinder 2	M.S.	1	
4	Cylinder 1 Flange	M.S.	1	
3	Cylinder 1	M.S.	1	
2	Cylinder Head 2	M.S.	1	
1	Cylinder Head 1	M.S.	1	

UNIVERSITY OF CAPE TOWN
DEPARTMENT OF MECHANICAL ENGINEERING

TITLE
Partial Engine Assembly

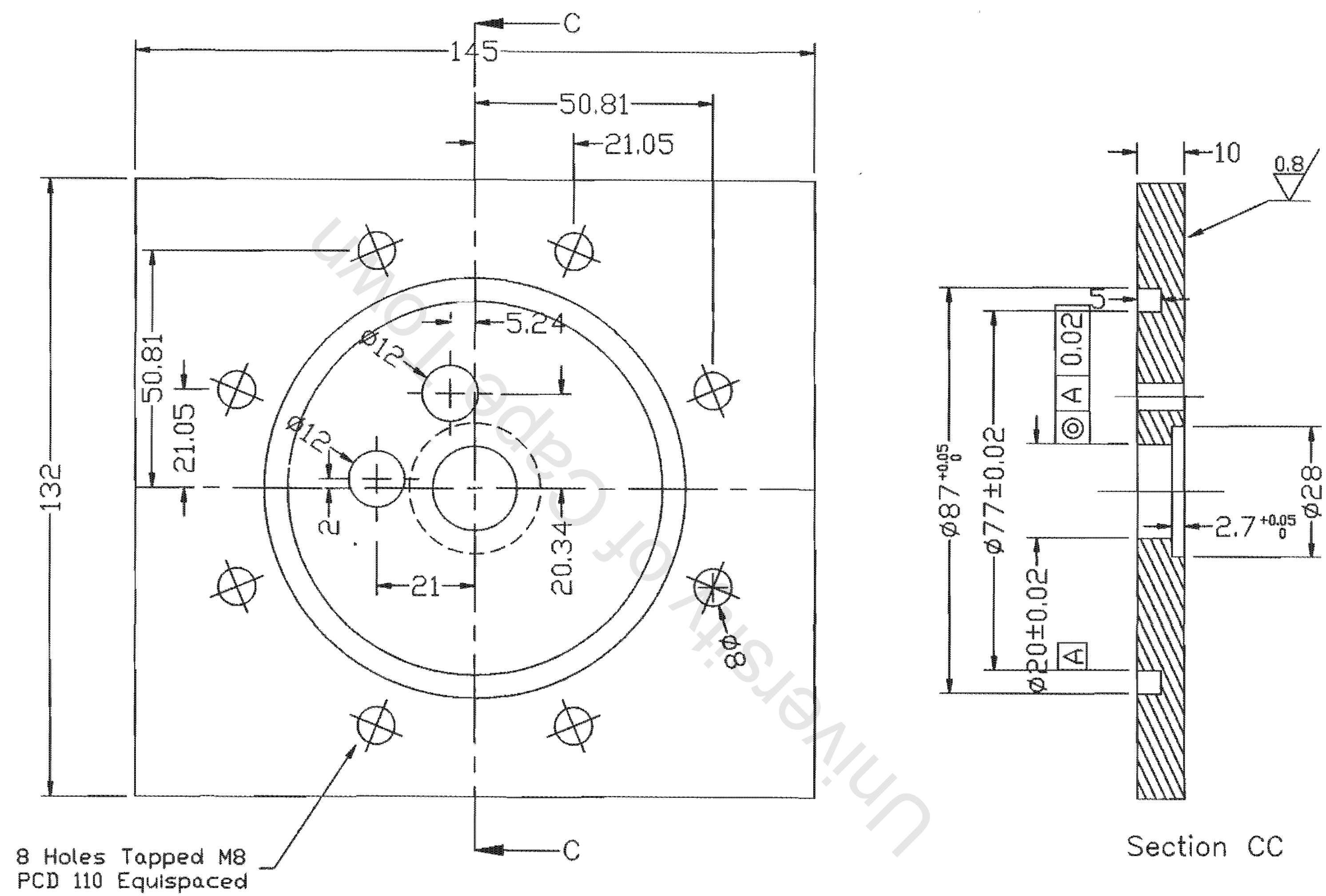
DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.1

SCALE	DATE	SHEET OF
1 : 2	05/07/00	1 12
DRAWN BY		DRAWING NUMBER
J.HUSSEY		01-MECH-001-00



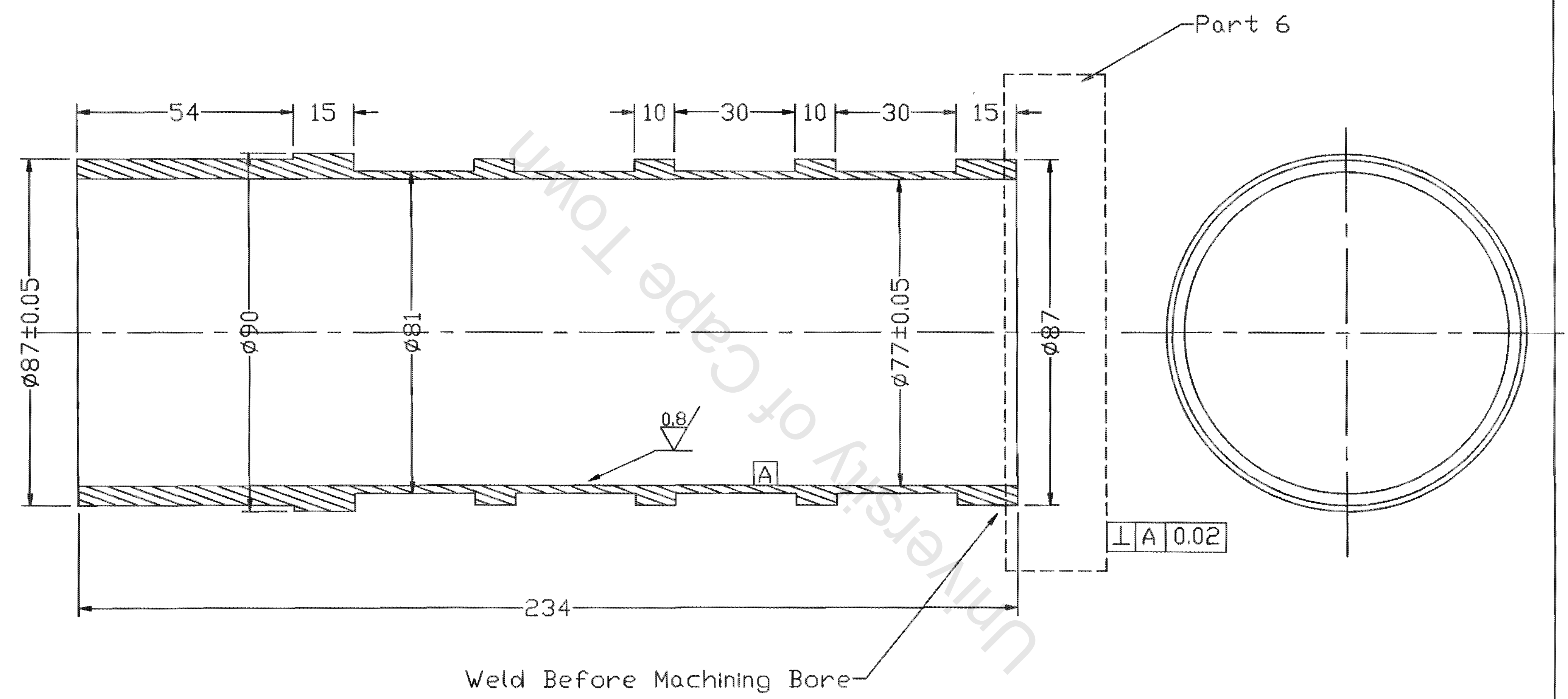
1	Cylinder Head 1	M.S.	1	
PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS

	UNIVERSITY OF CAPE TOWN DEPARTMENT OF MECHANICAL ENGINEERING		
	TITLE Cylinder Head 1		
DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.08	SCALE 1 : 1	DATE 05/07/00	SHEET 2 OF 12
	DRAWN BY J.HUSSEY		DRAWING NUMBER 01-Mech-002-00



2	Cylinder Head 2	M.S.	1	
PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS

	UNIVERSITY OF CAPE TOWN DEPARTMENT OF MECHANICAL ENGINEERING		
	TITLE Cylinder Head 2		
DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.08	SCALE 1 : 1	DATE 05/07/00	SHEET 3 OF 12
	DRAWN BY J.HUSSEY		DRAWING NUMBER 01-MECH-003-00

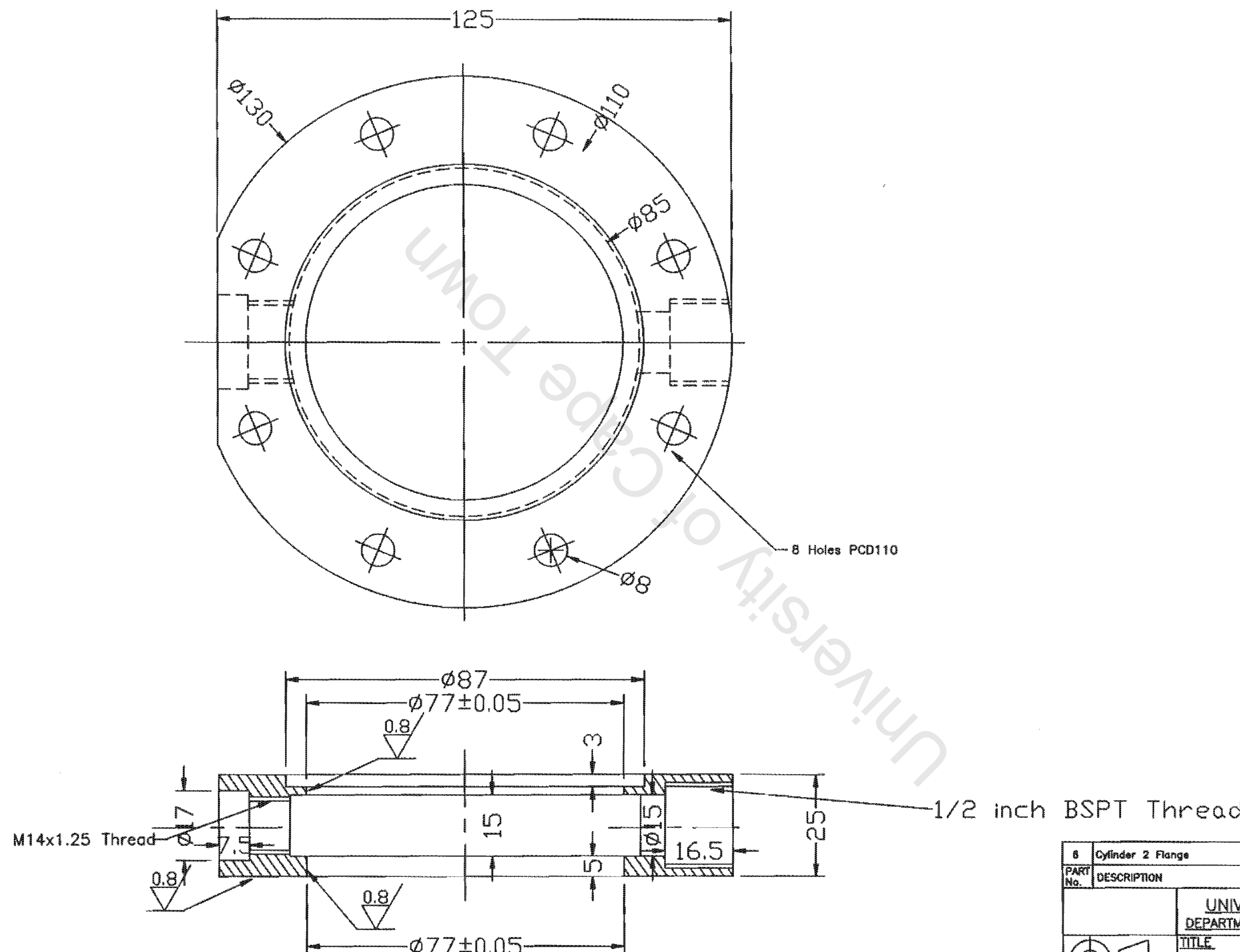


5	Cylinder 2	M.S.	1	
PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS

UNIVERSITY OF CAPE TOWN
DEPARTMENT OF MECHANICAL ENGINEERING

TITLE
Cylinder 2

DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.08	SCALE 1 : 1	DATE 05/07/00	SHEET 4 OF 12
	DRAWN BY J.HUSSEY		DRAWING NUMBER 01-MECH-004-00

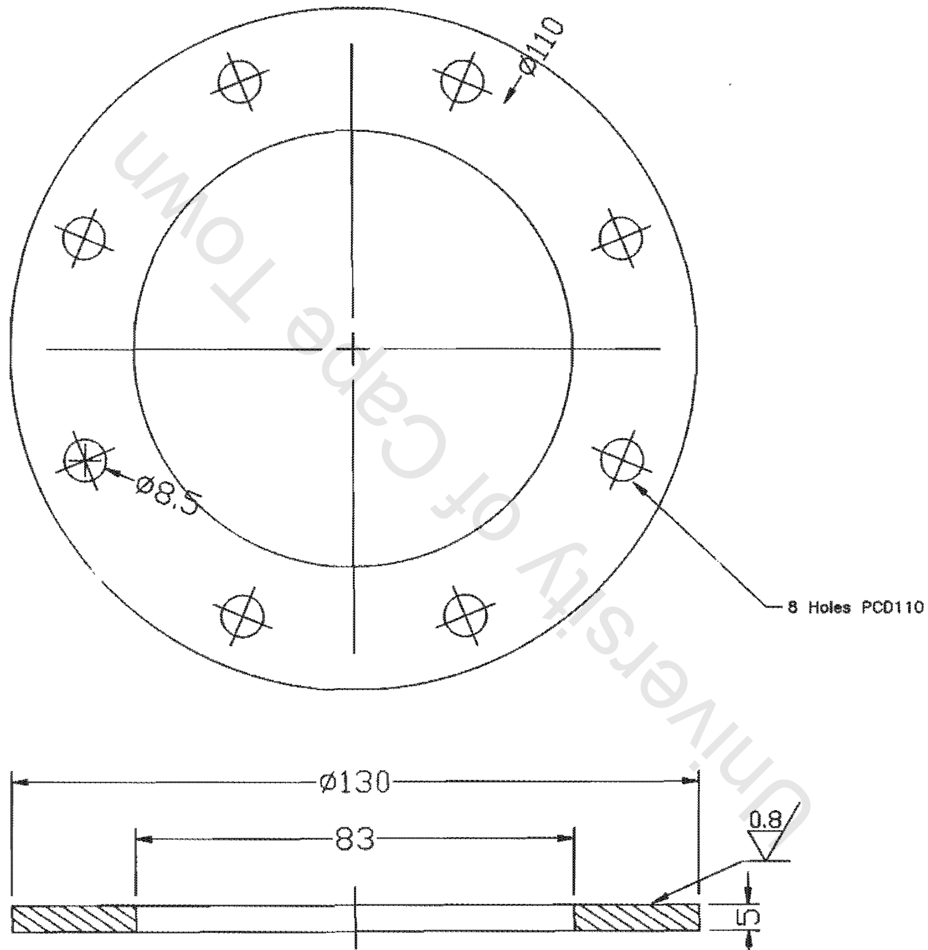


6	Cylinder 2 Flange	M.S.	1	
PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS

UNIVERSITY OF CAPE TOWN
DEPARTMENT OF MECHANICAL ENGINEERING

TITLE
Cylinder 2 Flange

DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.08	SCALE 1 : 1	DATE 05/07/00	SHEET 5 OF 12
	DRAWN BY J.HUSSEY		DRAWING NUMBER 01-MECH-005-00

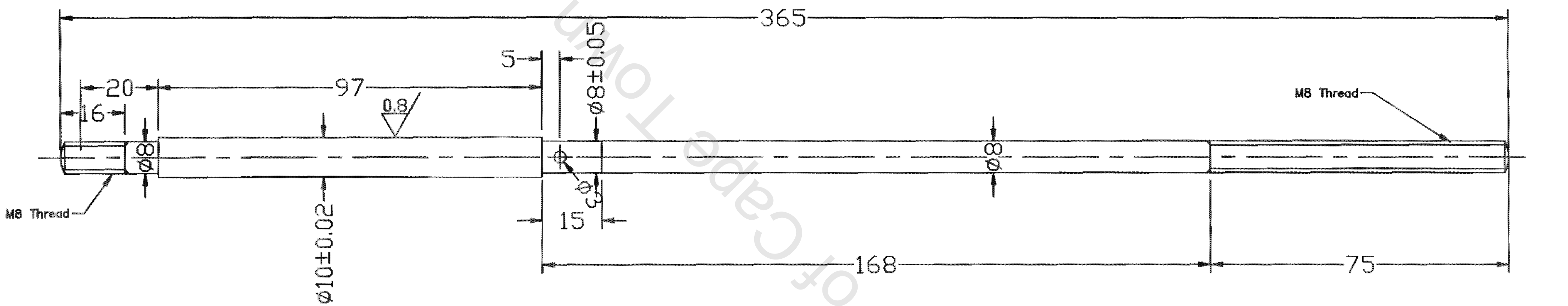


4	Cylinder 1 Flange	M.S.	1	
PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS

UNIVERSITY OF CAPE TOWN
DEPARTMENT OF MECHANICAL ENGINEERING

TITLE
Cylinder 1 Flange

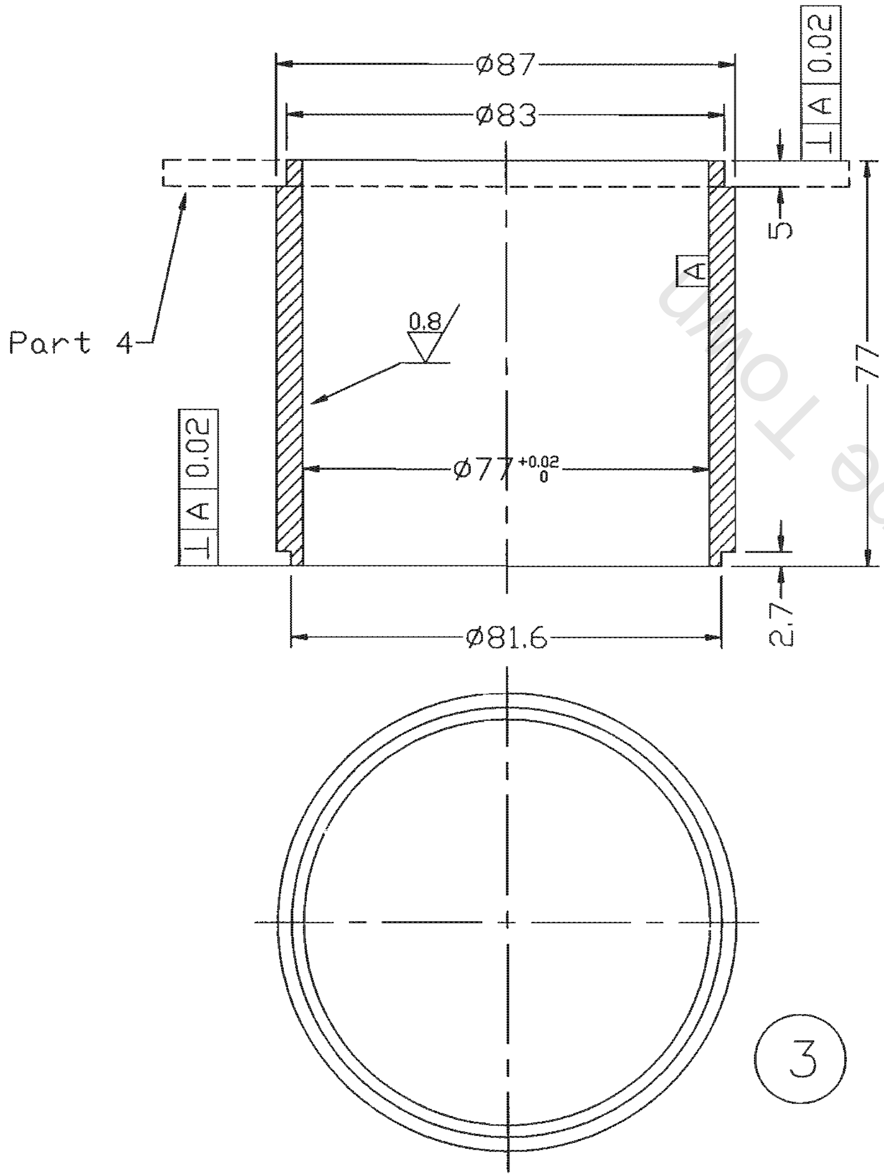
DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.08	SCALE 1 : 1	DATE 05/07/00	SHEET 6 OF 12
	DRAWN BY J.HUSSEY		DRAWING NUMBER 01-MECH-006-(.0

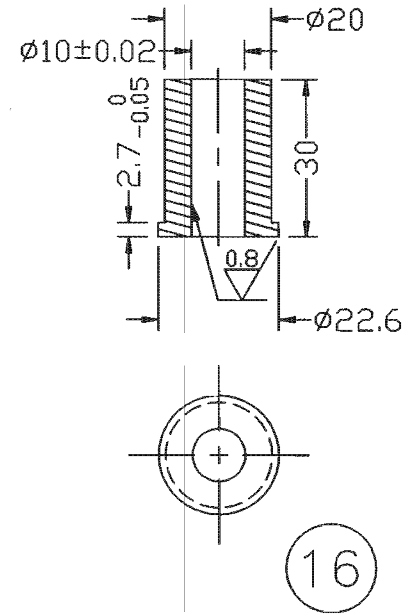


11	Piston Rod	Silver Steel	1	
PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS

UNIVERSITY OF CAPE TOWN
DEPARTMENT OF MECHANICAL ENGINEERING

	TITLE Piston Rod		
DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.08	SCALE 1 : 1	DATE 05/07/00	SHEET 7 OF 12
	DRAWN BY J.HUSSEY		DRAWING NUMBER 01-MECH-007-00

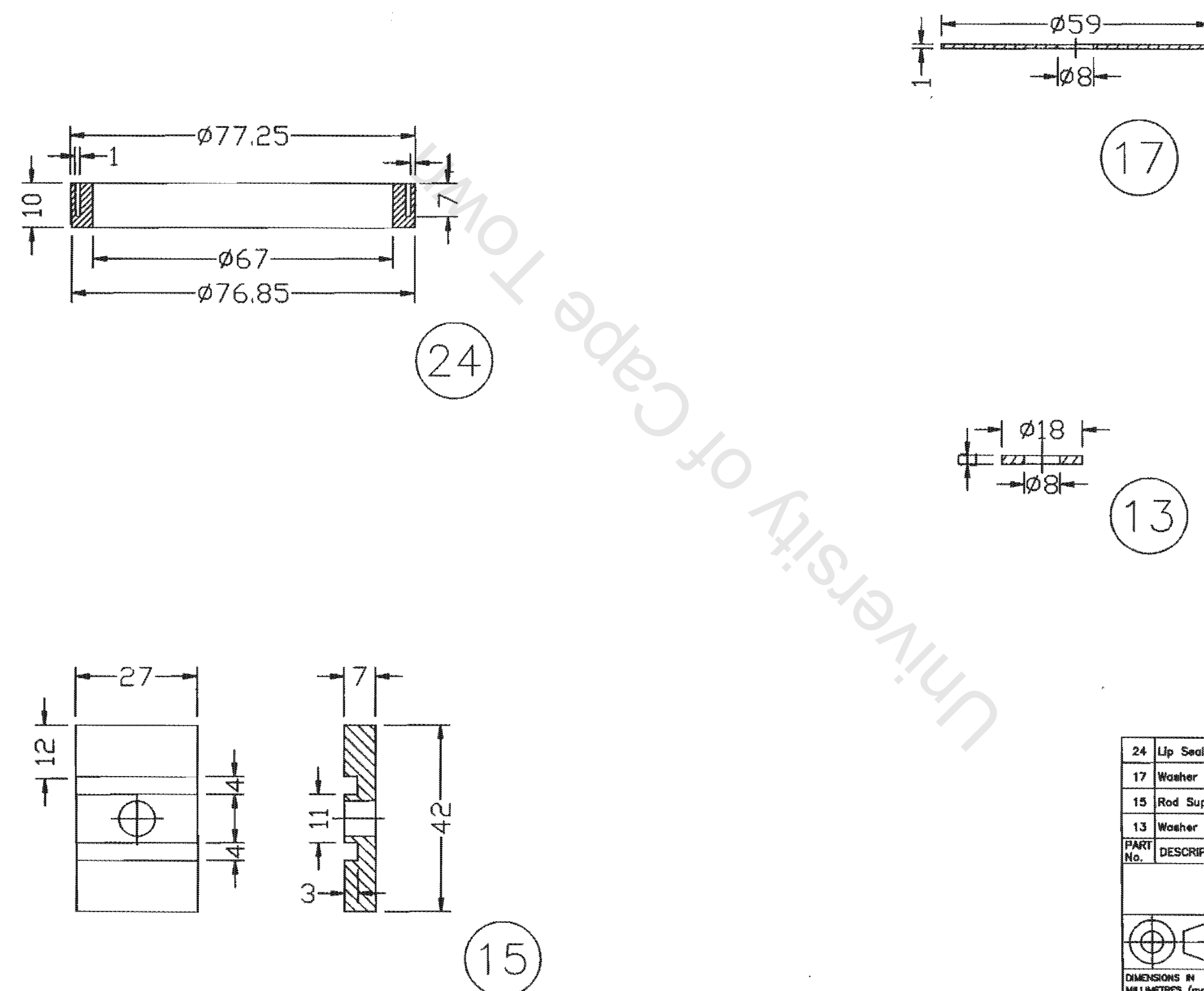




Note: This part to be welded to part 4
Before finishing machining

PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS
16	Bush	Vesconite	1	
3	Cylinder 1	M.S.	1	

UNIVERSITY OF CAPE TOWN DEPARTMENT OF MECHANICAL ENGINEERING			
TITLE Cylinder 1 and Bush			
DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.08	SCALE 1 : 1	DATE 05/07/00	SHEET 8 OF 12
	DRAWN BY J.HUSSEY		DRAWING NUMBER 01-MECH-008-00



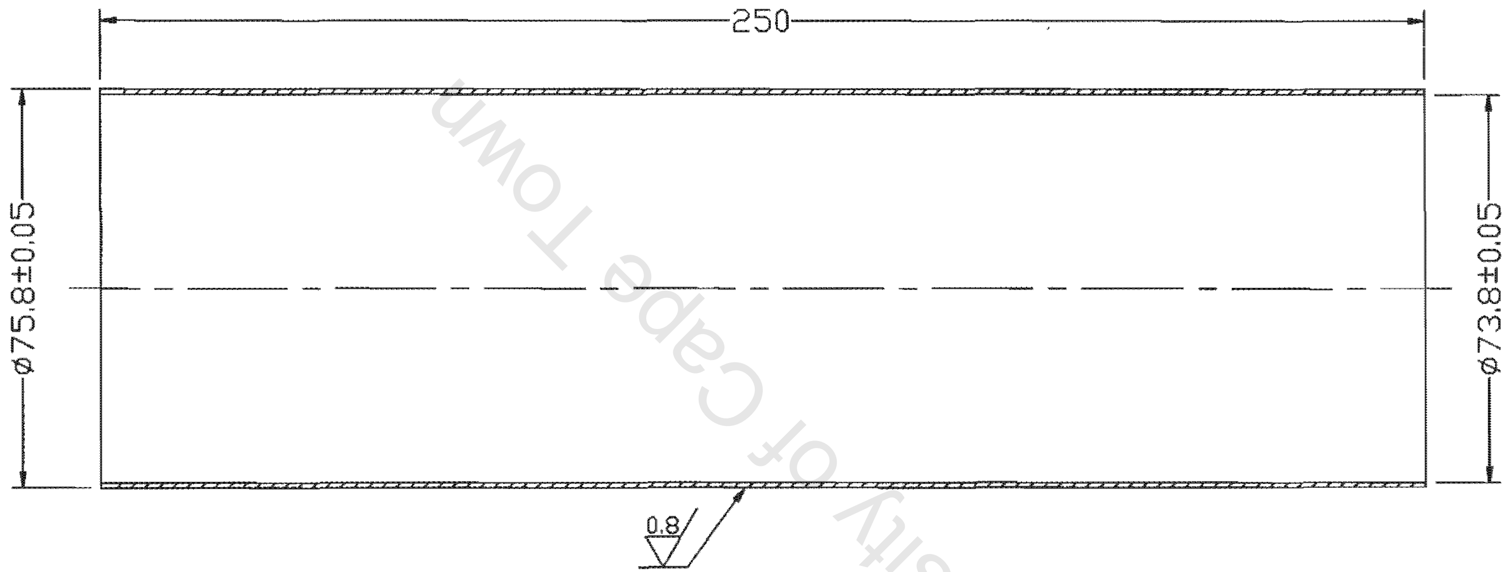
PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS
24	Lip Seal	Rulon	1	
17	Washer	M.S.	1	
15	Rod Support	M.S.	1	
13	Washer	M.S.	1	

UNIVERSITY OF CAPE TOWN
DEPARTMENT OF MECHANICAL ENGINEERING

TITLE
Seals and Washers

DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.08

SCALE	DATE	SHEET	
1 : 1	05/07/00	9 OF 12	
DRAWN BY	J.HUSSEY	DRAWING NUMBER	01-MECH-009-00



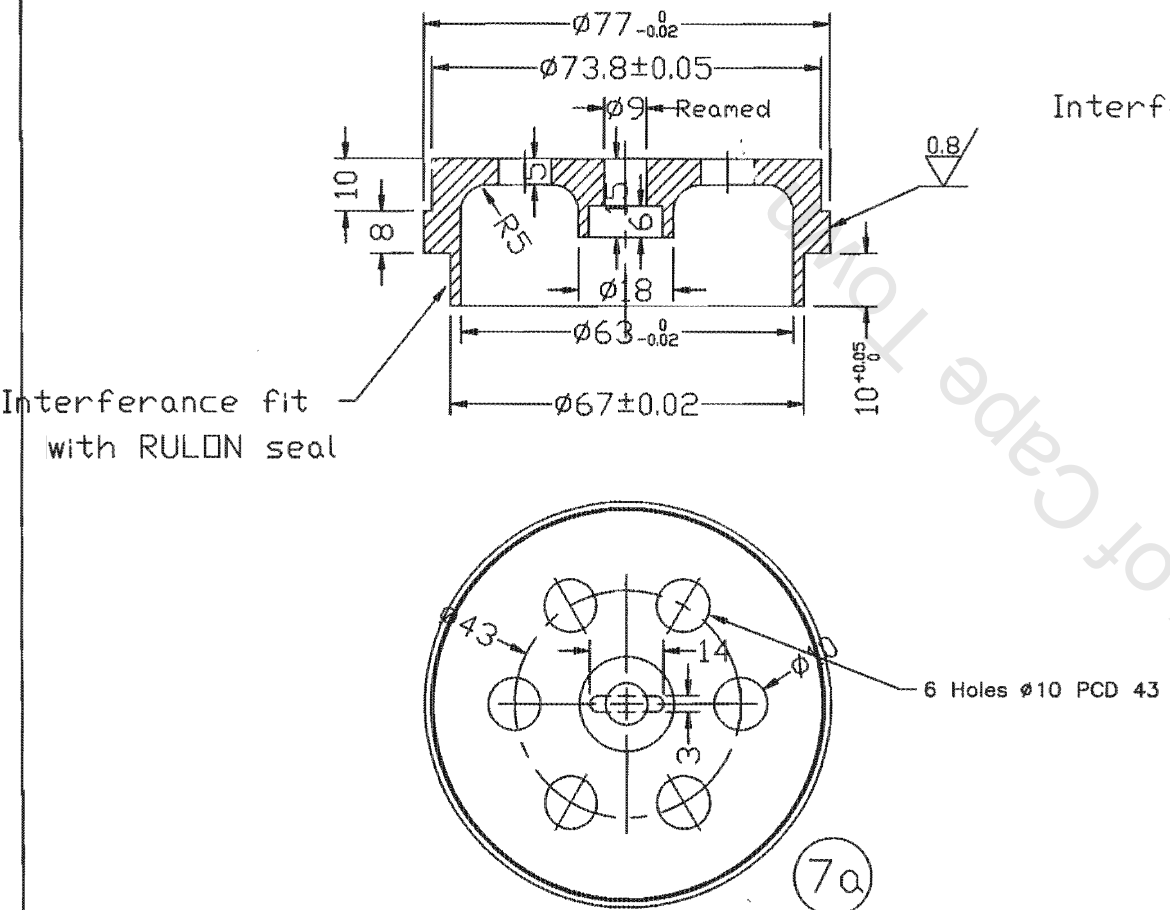
8	Regenerator	M.S.	1	
PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS

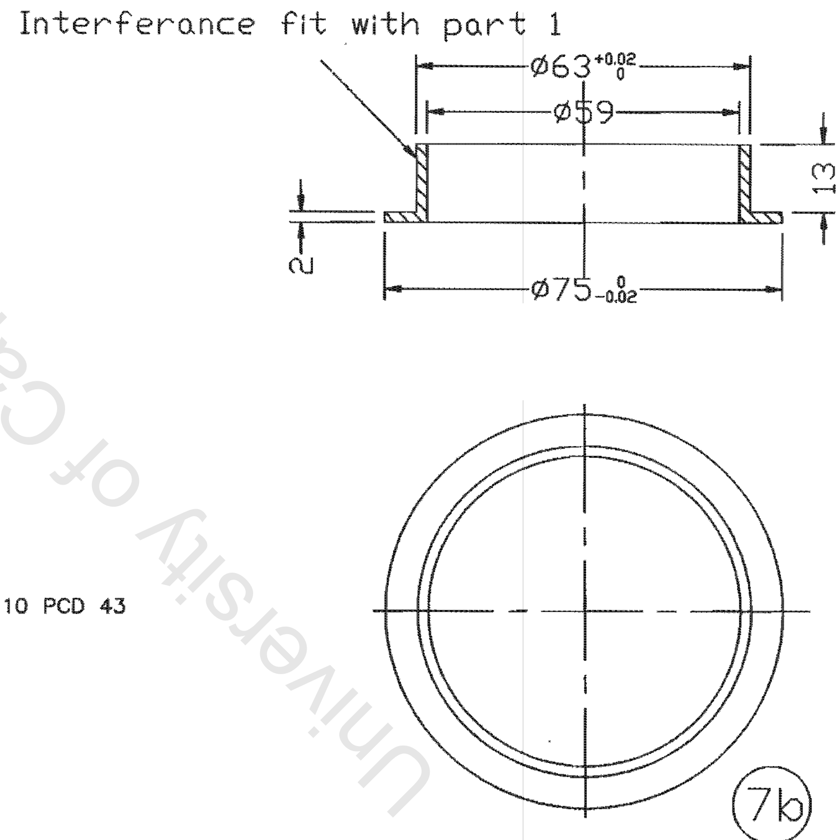
UNIVERSITY OF CAPE TOWN
DEPARTMENT OF MECHANICAL ENGINEERING

TITLE
Regenerator

DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.08

SCALE 1 : 1	DATE 05/07/00	SHEET 10 OF 12	
DRAWN BY J.HUSSEY		DRAWING NUMBER 01-MECH-010-.0	



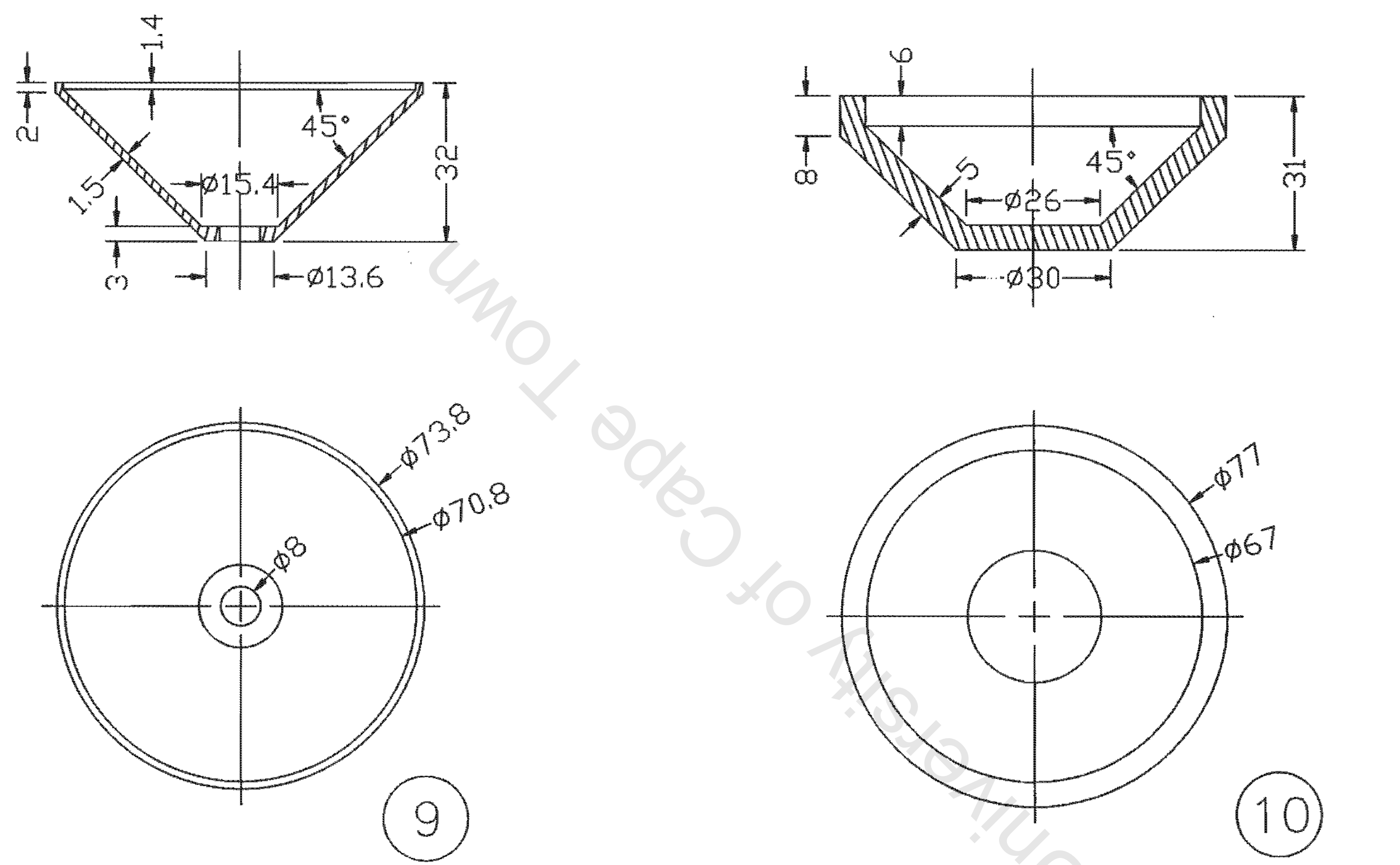


PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS
7b	Piston Bottom	Brass	1	
7a	Piston Top	Brass	1	

UNIVERSITY OF CAPE TOWN
DEPARTMENT OF MECHANICAL ENGINEERING

TITLE
Piston

DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.08	SCALE 1 : 1	DATE 04/09/00	SHEET 11 OF 12
	DRAWN BY J.HUSSEY		DRAWING NUMBER 01-MECH-011-00



PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS
10	Cylinder 2 Top	M.S.	1	
9	Regenerator Top	M.S.	1	

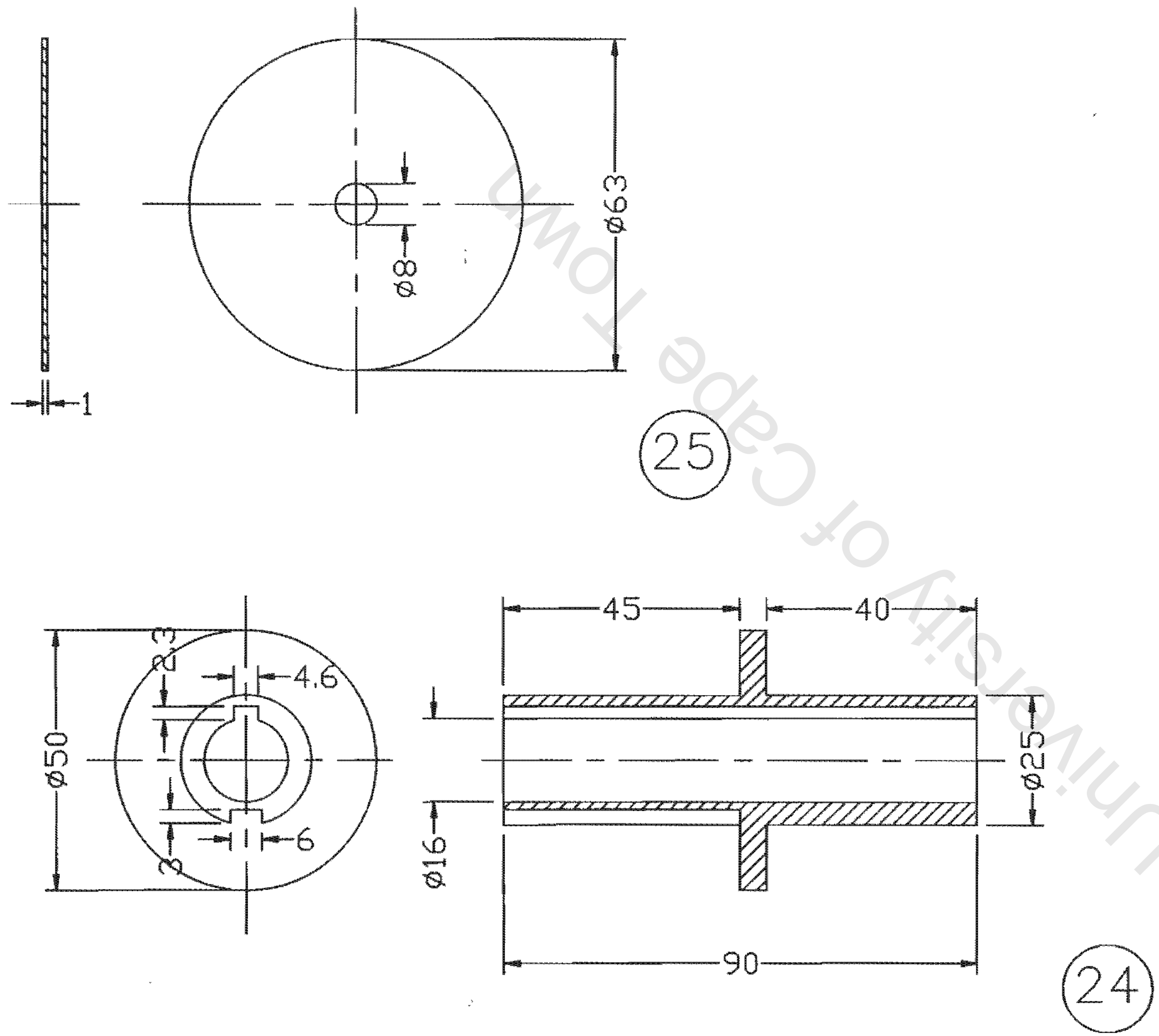
UNIVERSITY OF CAPE TOWN
DEPARTMENT OF MECHANICAL ENGINEERING

TITLE
Regen and Cyl 2 Tops

DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.08

SCALE 1 : 1 | DATE 05/07/00 | SHEET 12 OF 12

DRAWN BY J.HUSSEY | DRAWING NUMBER 01-MECH-012-00



PART No.	DESCRIPTION	MATERIAL	No. OFF	REMARKS
25	Heat Shield	M.S.	2	
24	Flywheel adaptor	M.S.	1	

UNIVERSITY OF CAPE TOWN
DEPARTMENT OF MECHANICAL ENGINEERING

TITLE
Adaptors

DIMENSIONS IN MILLIMETRES (mm) TOLERANCE UNLESS OTHERWISE STATED +-0.08

SCALE	DATE	SHEET OF
1 : 1	12/07/00	13 12

DRAWN BY: J.HUSSEY

DRAWING NUMBER: 01-MECH-013-00