CHAPTER 1

INTRODUCTION

1.1 Background

In man’s quest to improve his quality of life he has developed many technologies that use his knowledge of the universe and it’s workings to his advantage. Among the most important is the discovery of the heat engine, a machine that converts heat into useful work.

Over the course of human history, many different kinds of heat engines have been invented that operate on many different concepts. A. Slaby classified them into three broad categories; internal combustion open cycle, external combustion open cycle and external combustion closed cycle engines[2]. The first category includes regular automotive engines and turbine engines. The second category was popular about 100 years ago but is now obsolete. The focus of this project is the third category, a closed cycle external combustion engine specifically a Stirling cycle engine.

A Stirling engine is a kind of heat engine invented by Rev Robert Stirling in 1816 to show the potential use of another of his inventions, the regenerator[2]. Due to the lack of previous patents, it can be assumed that Robert Stirling was the inventor of the closed cycle air engine[3]. Soon after the approval of his patent a revised model was installed in a nearby quarry where it ran for two years[2]. After a few years of development Robert Stirling and his brother James Stirling developed an engine with 45 brake horsepower and an efficiency of 18%[3].

After a period of interest in the early 19th century progress on the Stirling engine diminished and slipped into a period of obscurity until development was picked up by the Phillips Company in the 1930s. Originally, the Stirling engine was intended to power small electrical generators for radios in remote areas. After a systematic comparison of many prime movers it was decided that the air engine offered the best prospects[3].

1.2 Problem Statement

Over the last few years there has been growing interest in renewable energy. The Stirling engine is very attractive due to its ability to run on any heat source and low maintenance. Low grade low temperature energy is widely available but rarely utilized. Is it possible to run an Alpha type Stirling engine on a low temperature difference?

1.3 Objectives

The objective of this study is to determine the feasibility of running an Alpha type Stirling engine on a low temperature differential (~80^{o}C).

1.4 Scope of Studies

The project is split into three main components:

- Design of an Alpha type Stirling engine
- Development of the designed engine
- Testing of the engine

CHAPTER 2

LITERATURE REVIEW

The Stirling engine predictably operates on the Stirling cycle. This cycle is characterized by isothermal heat addition and extraction and constant volume regeneration of a gas[1]. Figure 1 below shows the T-s and P-v diagram for the Stirling cycle.

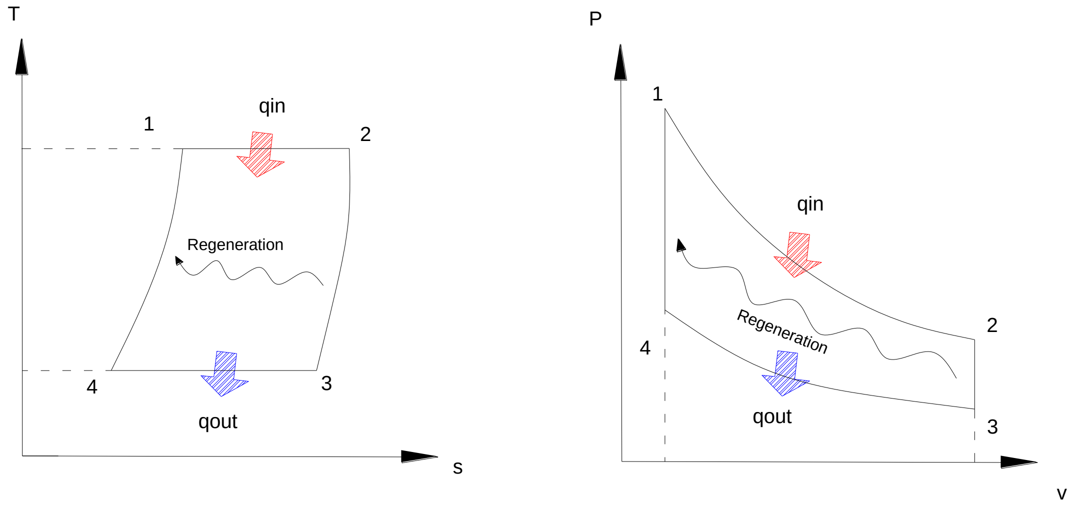

Fig 2.1: T-s and P-v diagram of a Stirling cycle

The Stirling cycle is made of four completely reversible process:

1-2 constant temperature heat addition

2-3 constant volume heat rejection (regeneration)

3-4 constant temperature heat rejection

4-1 constant volume heat addition (regeneration)

The theoretical efficiency of a Stirling engine is equal to the Carnot efficiency of a heat engine running on the same thermal difference. It must be noted that the practical efficiency of a Stirling engine will be significantly lower than the theoretical efficiency due to losses and the fact that the textbook cycle down not give a good representation of what actually happens inside an actual engine[9].

The basic Stirling engine can be categorized into three configurations, alpha, beta and gamma. The configurations are not designated by their order of historical appearance. The alpha type has two sealed pistons both of which have an effect on the volume and pressure in the engine. The beta type has one sealed piston and one displacer piston in the same cylinder. Only the sealed piston has an effect on the pressure and volume in the engine and the displacer piston is only to shuttle the working fluid to and from the hot and cold areas of the cylinder. The gamma type is similar to the beta type but the sealed piston and the displacer piston is not in the same cylinder[2]. A schematic diagram of the types of engines are show below.

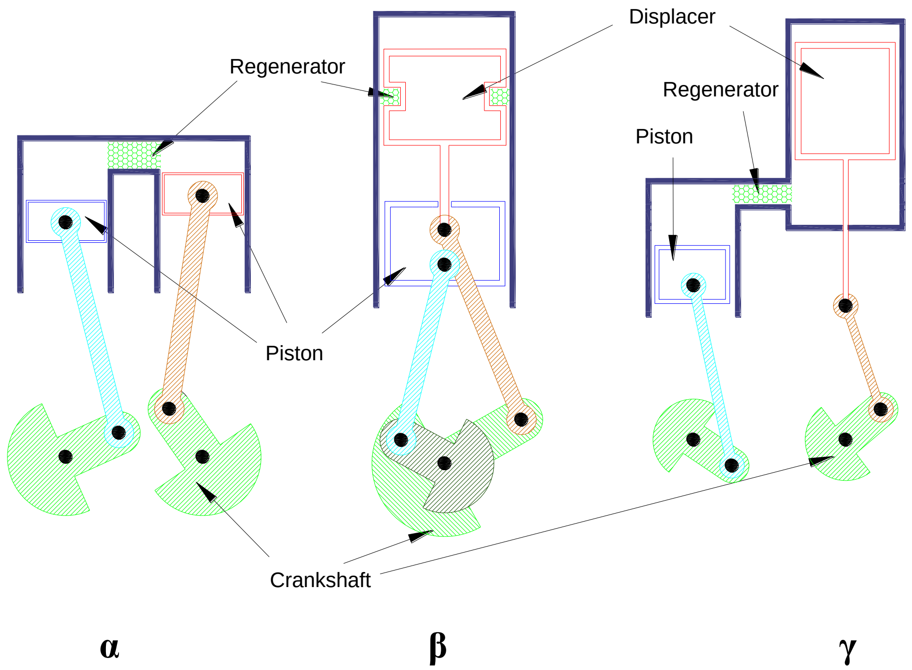

Fig 2.2: Types of Stirling engines

An alpha type engine was selected due to simpler linkage and possibility of using the same size of pistons and cylinders.

The power available from a heat engine is proportional to the mass of working fluid it can process per second[2]. For example a gas turbine engine with an inlet diameter of 2m processing air at a density of 1.2kg/m^3 and a velocity of 200m/s consumes

700kg/s. This explains the high specific power output of gas turbine engines and why reciprocating engine cannot compete. To overcome this in a Stirling engine the mean pressure of the working fluid in the engine can be raised to raise the fluid mass per working cycle. Doubling the working fluid pressure can potentially double the specific power but increasing the pressure poses new problems in sealing and heat transfer because the heat transfer area remains the same[2]. Power can also be increased by increasing the cycles per second or the rpm.

The Stirling engine is also known as a hot-air or hot-gas engine because it produces work through the alternate heating and cooling of a contained mass of gas. The first Stirling engines used pressurised air as the working fluid but later developers experimented with other gasses. One reason is safety, during one recorded incident at the Phillips laboratory an engine with pressurised air exploded when the lubricating oil in the presence of the high pressurized oxygen self ignited resulting in the death of one person[2]. After the accident all Phillips engines were charged with inert gasses such as nitrogen or helium. It has been proven that although lighter gasses like helium and hydrogen offer lower flow losses, with appropriate scaling to the engine the type of gas used does not matter[2].

One of the most important components of a Stirling engine is the regenerator. The regenerator was the reason of Rev. Robert Stirling's patent application in the first place[2]. The regenerator or economiser is a device that greatly improves on the efficiency of the Stirling engine. It can be thought as a sequential heat exchanger where instead of the hot and cold fluids running through separate channels, the hot and cold fluids alternately pass through the same thermal mass. A 1% increase in regenerator efficiency will result in a 3% increase in engine efficiency[2]. A stylized representation of regenerator operation is shown below.

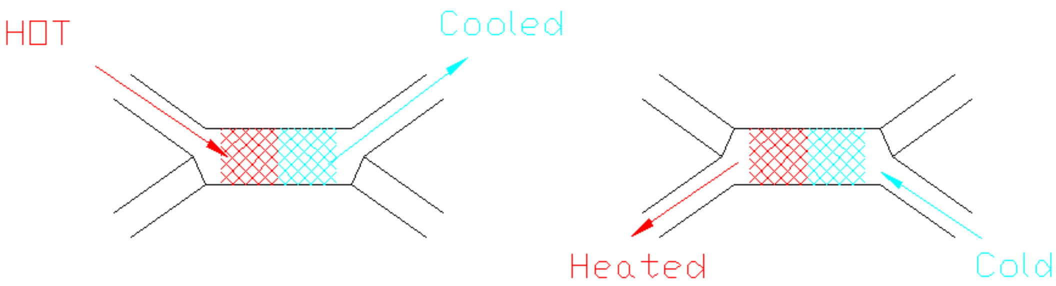

Fig 2.3: Stylized representation of regenerator operation

It is interesting to note that the idea of using the sun to power a Stirling engine has been around since at least 1872 when John Ericsson added a parabolic mirror to the hot end of the cylinder[2]. In recent years this concept has been perfected to the point of practical and economic viability with the two large solar thermal power plants in the United States supplying 1.7GW of power produced by 70,000 solar thermal dish Stirling engine concentrators[8]. Research has shown that a Stirling engine can be made to run from heat generated by a flat plate solar collector that is integrated into the engine and supplying a temperature difference of 80^0C[3].

Stirling engines do not need very high temperature differentials to run, engines that run on temperature differentials as low as 0.5^0C have been developed[2].

To the authors knowledge there has been no research in low temperature alpha Stirling engines.

CHAPTER 3

METHODOLOGY

The scope of this project will be spread out over four main areas research, design & analysis, development and testing of the Stirling engine.

3.1 Research Methodology

The process of researching this project first begins with conducting a cursory search on the internet for relevant webpage's to get a better understanding of the Stirling process. Next is the reviewing related titles that are available in the university library. From this a deeper understanding of the Stirling engine and its history is gained. Journals, books and articles referenced in the books are reviewed if possible for any other useful information

3.2 Project Activities

Once the bulk of the literature has been reviewed the next step of the project can begin, the design of the engine. The appropriate engine dimensions are selected and a thermodynamic analysis is run on the design to check feasibility. In parallel potential sources of the components are sourced.

The finalization of the design signals the start of the next phase, fabrication. This begins with the sourcing of various materials that can be used in the engine design. Some parts might need to be manufactured in house due to non-availability of components.

Testing of the finished engine will be the final stage of this project. Items to be tested include if the engine can run at the desired temperature differential and running speed.

The work process flow is shown graphically below.

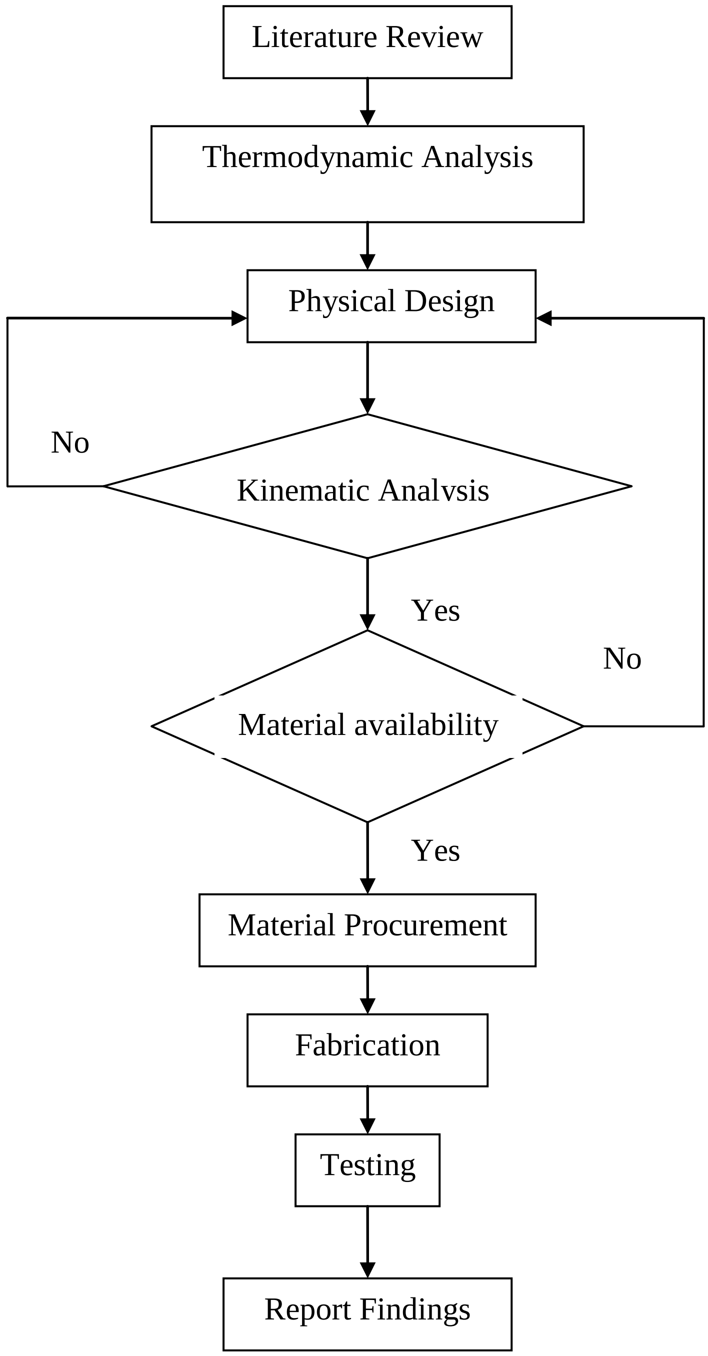

Fig 3.1: Flow diagram of activities

3.3 Tools

Among the tools that might be needed are:

- Autodesk AutoCAD
- Microsoft Excel
- Fabrication Equipment

CHAPTER 4

RESULTS & DISCUSSION

4.1 Design

The design goal of the engine is to produce approximately 10 Watts of power at 60rpm while running on an input temperature of 380K and an output temperature of 300K. This equates to a work produced per cycle of 10 Joules.

4.1.1 Thermodynamic Analysis

4.1.1.1 Finkelstein-Schmidt

This is an analytical solution based on the Schmidt equation and derived by Finkelstein for a two cylinder alpha Stirling engine[4]. The assumptions used in this solution are:

- Sinusoidal motion of parts
- Known and constant gas temperatures on all parts of the engine
- No gas leakage
- Working fluid obeys perfect gas law
- At each instant in the cycle the gas pressure is the same throughout the working gas

The work per cycle in given by

$$W1 = \frac{(2\pi)(K)(1-AU)(\sin(AL))(M)(R)(TC)}{(AU+K+(2)(S))^2\sqrt{1-(DL)^2}\,(1+\sqrt{1-(DL)^2})} \qquad \text{(Eq 4.1)}$$

Where:

W1	=	work per cycle, J
VT	=	VL + VK + (1 + K)VL
VL	=	swept volume in the expansion space, cm^3
VK	=	swept volume in the compression space cm^3

K = swept volume ratio = VK/VL

AU = TC/TH

TC = compression space gas temperature, K

TH = expansion space gas temperature, K

$$DL = \frac{((\mathrm{AU})^2 + 2(\mathrm{AU})(\mathrm{K})\cos(\mathrm{AL}) + \mathrm{K}^2)^{1/2}}{AU + K + 2S}$$

TR = dead space gas temperature = (TC + TH)/2

AL = angle by which volume variation is the expansion space lead those in compression space, degrees

$S = \dfrac{2(RV)(AU)}{AU + 1}$ = arithmetic average temperature for the regenerator

RV = VD/VL, dead volume ratio

VD = HD + CD +RD = total dead volume, cm^3

The design engine has the following values:

TH	=	380K
TC	=	300K
Bore	=	6.0cm
Stroke	=	5.5cm
AL	=	90^0
Charge pressure	=	100kPa

With the design values used as shown in the previous segment, this equation gives a value of 9.71 J per cycle.

The spreadsheet used for solving the equation is shown in appendix A.

4.1.1.2 Numerical Analysis

With known values of an engine such as bore, stroke and charge pressure it is possible to plot the relationship between the pressure within the engine and the volume resulting in a P-V diagram. The P-V diagram can be graphically integrated to find the work generated per cycle[4].

The assumptions used in this calculation are:

- Constant instantaneous pressure throughout the engine
- The gas behaves as an ideal gas
- No gas leakage
- Known and constant gas temperature in all parts of the engine

The volumes and the pressure on the engine are given by the following equations:

Hot volume

$$H(N) = \frac{VL}{2}[1 - \sin(F)] + HD \qquad \text{(Eq 4.2)}$$

Cold volume

$$C(N) = \frac{VK}{2}[1 - \sin(F - AL)] + CD \qquad \text{(Eq 4.3)}$$

Total volume

$$V(N) = H(N) + C(N) + RD \qquad \text{(Eq 4.4)}$$

Engine pressure

$$P(N) = \frac{(M)(R)}{\frac{H(N)}{TH} + \frac{C(N)}{TC} + \frac{RD}{TR}} \qquad \text{(Eq 4.5)}$$

Figure 4.1 and table 4.1 below shows the nomenclature of the engine.

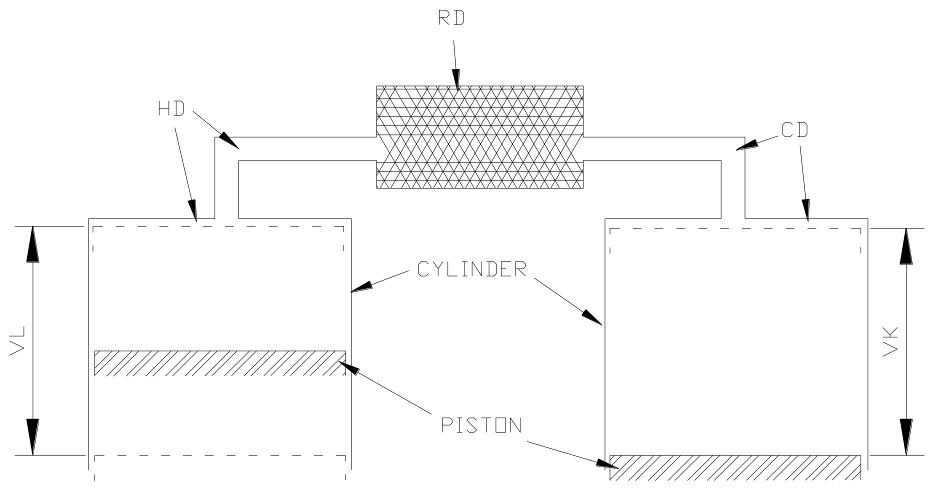

Fig 4.1: Nomenclature for Dual-Piston Engine

Table 4.1: Dual Piston Engine Nomenclature

Symbol	Definition	Units
HD	Hot dead volume	cm^3
RD	Regenerator dead volume	cm^3
CD	Cold dead volume	cm^3
VL	Hot piston live volume	cm^3
VK	Cold piston live volume	cm^3
TH	Effective hot gas temperature	K
TC	Effective cold gas temperature	K
TR	Effective regenerator gas temperature	K
M	Engine gas inventory	g mol
R	Gas constant	J/gmolK
P(N)	Common gas pressure	MPa
F	Crank angles	Degrees
ND	Crank angle increment	Degrees
AL	Phase angle	Degrees

The design engine has the following values:

TH = 380K

TC	=	300K
Bore	=	5.5cm
Stroke	=	6.5cm
AL	=	90^0
Charge pressure	=	100kPa

Fig 4.2 and 4.3 below show the relationship between pressure, volume and crank angle.

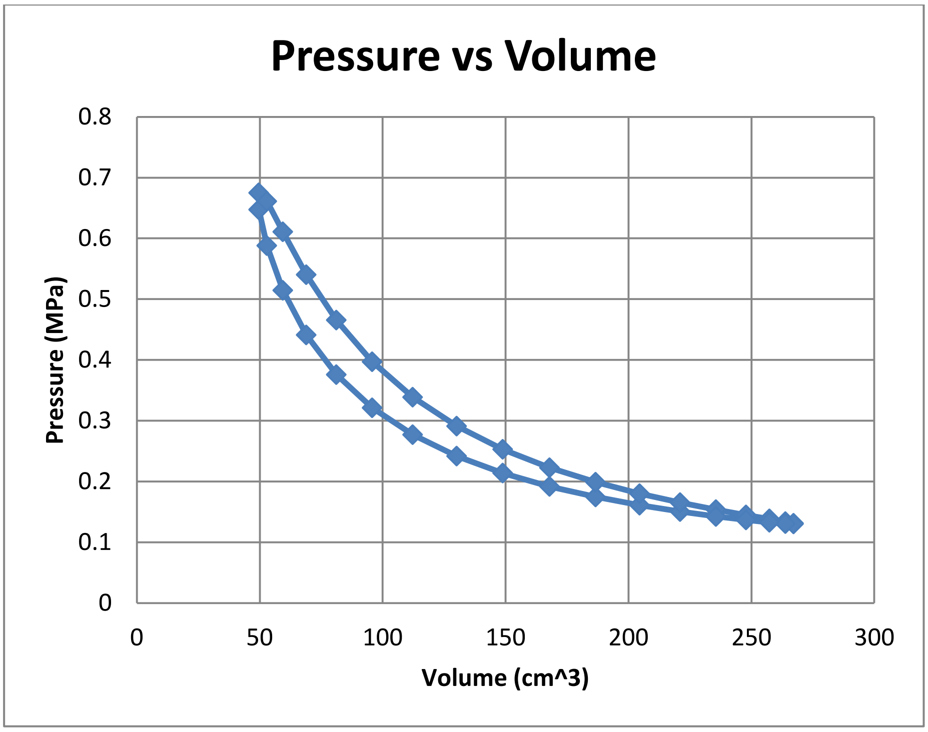

Fig 4.2: P-V Diagram

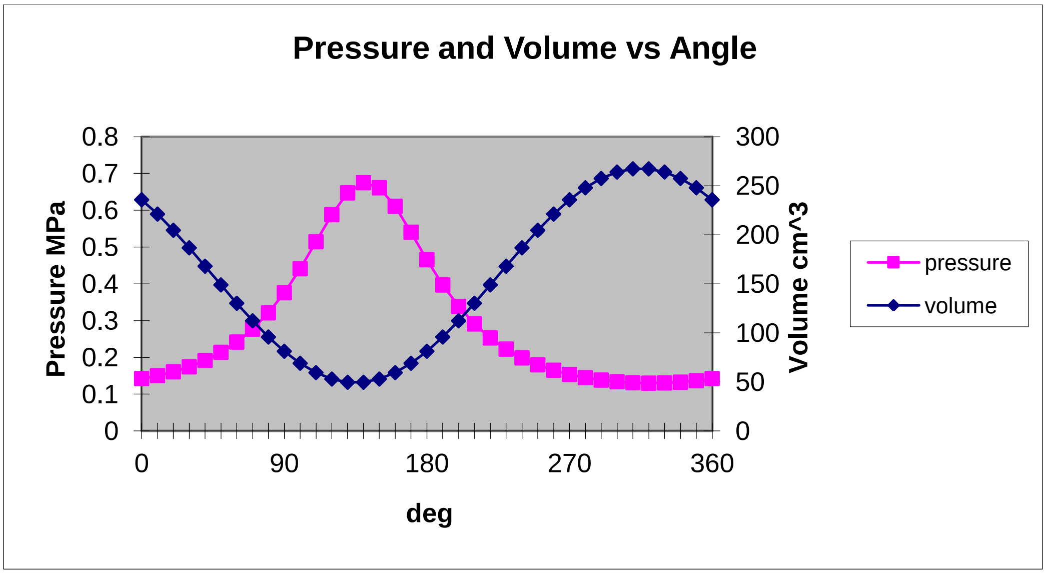

Fig 4.3: Pressure and volume vs angle

The spreadsheet used for the numerical analysis is shown in appendix B.

4.1.2 Kinematic Analysis

The kinematic analysis of the engine is very important to understand the forces acting on the engine components. The forces experienced by the parts are dependent on the acceleration achieved. This is given by Newton's law of motion;

$$F = ma$$

Where F is the force applied, m is the mass and *a* is the acceleration.

The kinematics of a piston connected to a crankshaft are given by the equations;

$$displacement, x = rcosA + \sqrt{l^2 + r^2 sin^2 A} \qquad \text{(Eq 4.6)}$$

$$velocity, v = r\omega \sin(\omega t) + \frac{r^2 \omega \sin(2\omega t)}{2l} \qquad \text{(Eq 4.7)}$$

$$acceleration, a = r\omega^2 \cos(\omega t) + \frac{r^2 \omega^2 \cos(2\omega t)}{l} \qquad \text{(Eq 4.8)}$$

Where

x is the displacement of the piston from crankshaft centre in meters

v is the velocity of the piston in m/s

a is the acceleration of the piston in m/s^2

r is the crank radius in meters

A is the angle of the crank from top dead centre

l is the length of the connecting rod in meters

ω is the rotational speed in rad/s

t is the time in seconds

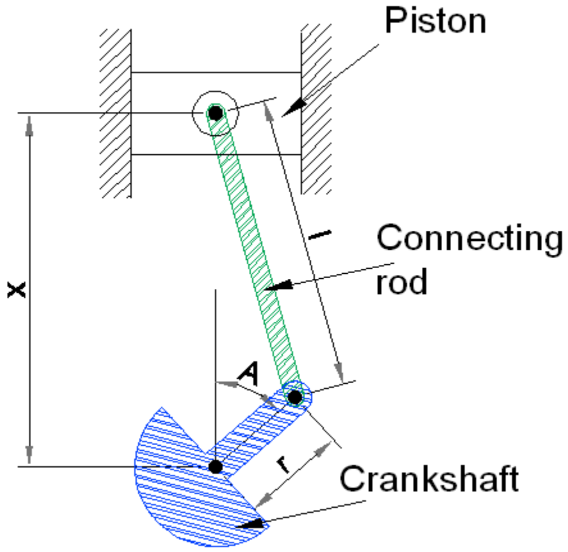

Fig 4.4: Diagram showing naming convention of piston equation

Under constant angular velocity of the crankshaft, the piston is the component that experiences the greatest acceleration. The length of the connecting rod determines the magnitude of the acceleration. A longer connecting rod results in a smoother acceleration curve. Below is shown the acceleration curve of a piston at 60rpm with connecting rod lengths of 15cm, 10cm and 7cm. A connecting rod of 10cm was chosen as a balance between smooth acceleration and low eccentric mass.

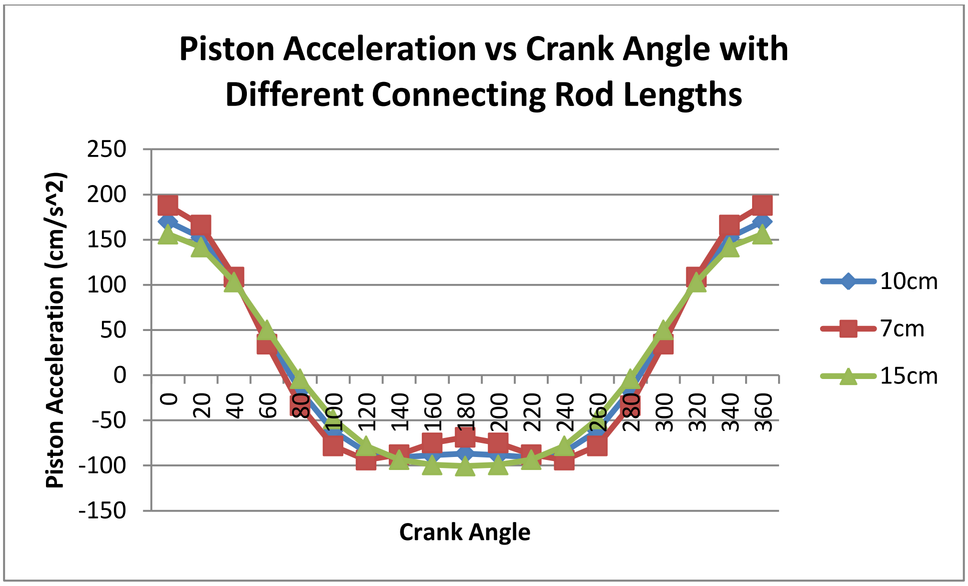

Fig 4.5: Acceleration of piston with different connecting rod lengths

The spreadsheet used for the kinematic analysis is shown in appendix C.

4.1.3 Physical Design

The basic design of the engine has been determined based on the dimensions from the numerical and Finkelstein-Schmidt equations. The basic design went through several revisions before the current design was finalized and any material was fabricated.

4.1.3.1 Purchased Components

It was decided early in the project that as much of the components should be sourced off-the-shelf as possible to reduce the fabrication time. The main component that was bought was the piston-cylinder assembly. The piston-cylinder assembly was sourced from a high pressure foot pump rated to 700kPa. The pump used is shown below.

Fig 4.6: Foot pump sourced for piston-cylinder assembly

The dimensions of the engine had to be changed because the cylinder and piston from the pump has a smaller bore than in the original design. The initial bore and stroke of the engine was 6.0cm and 5.5cm respectively but the bore and stroke of the pump is 5.5cm and 7.0cm. To compensate for the smaller bore the stroke was lengthened to 6.5cm. This gives a theoretical energy per cycle of 9.7J using the Finkelstein-Schmidt equation.

4.1.3.2 CAD Model

A CAD model was created detailing all relevant information of the dimensions of the engine to minimize the fabrication work needed. Because all aspects of the design are inputed into the model, rework of fabricated parts is kept to a minimum because all parts fit together correctly. A rendering of the designs are shown below.

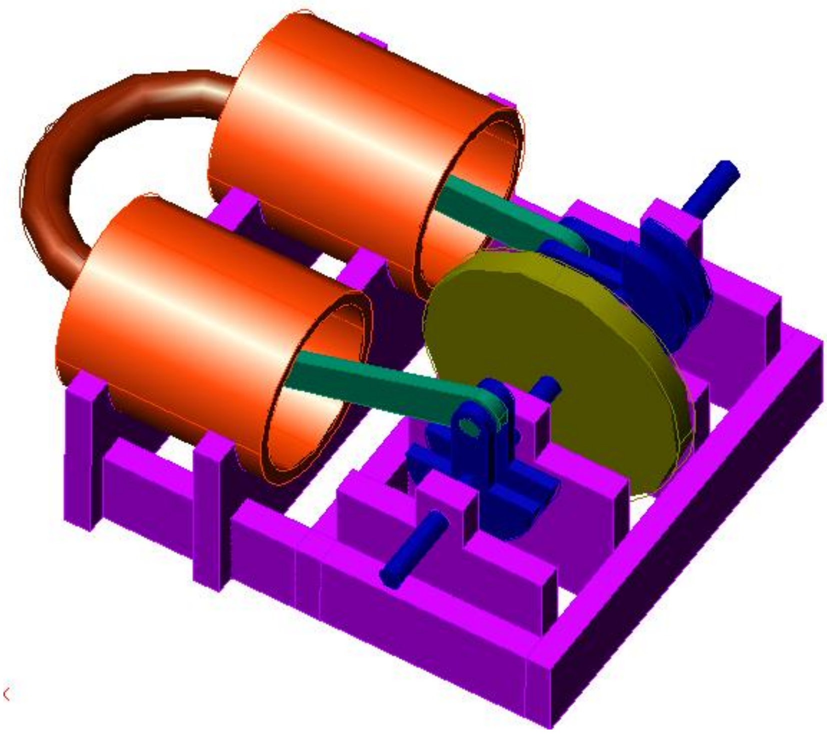

Fig 4.7: Initial design

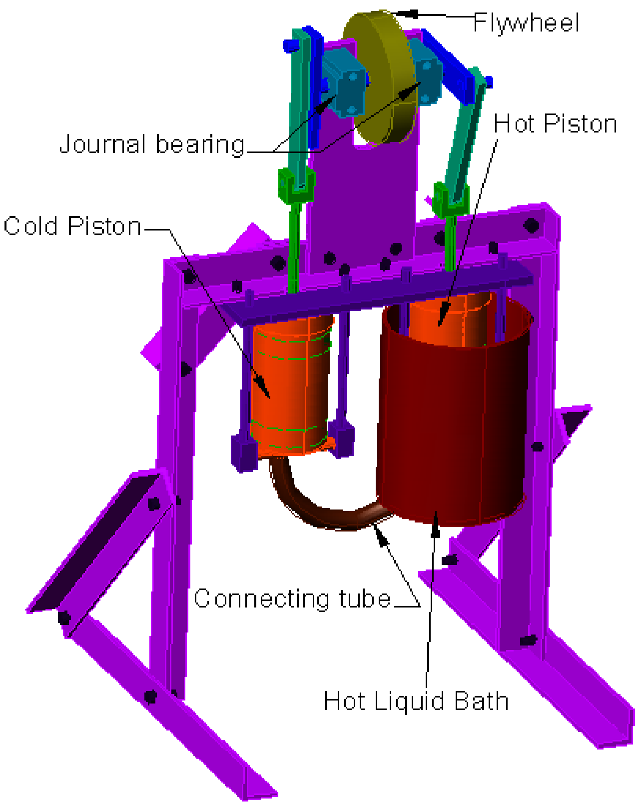

Fig 4.8: Final design

A vertical orientation was chosen to allow the hot cylinder to be immersed in a liquid bath. The use of a liquid bath to heat the cylinder will allow for the accurate control of the heat input temperature.

Supporting the cylinders provided a unique challenge due to the need to allow fluid flow around the cylinder and the very thin cylinder walls. Clamping the cylinder to hold it is unacceptable because clamping with any reasonable force would distort the cylinder walls and cause the piston to seize. It was decided to use the pre-existing supports that were used to anchor the cylinder when it was used as an air pump.

The main components are:

2 x Cylinders (orange)
2 x Pistons (light green)
1 x Crankshaft (blue)
2 x Connecting rods (dark green)
1 x Connecting tube (brown)
1 x Flywheel (bronze)
1 x Support frame (purple)
2 x Journal bearings (blue-green)
1 x Hot liquid bath (maroon)

A full set of engineering drawings is attached in Appendix D.

4.2 Fabrication

Most of the components of the engine are hand machined using conventional lathes and milling machines. Where possible aluminium was used due to its high machinability and low weight. Mild steel was used for high stress rotating components including the crankshaft and connecting pins. All rotating surfaces were polished to a mirror finish using 1200 grit sandpaper. All fabrication material was sourced from UTP labs. Fig 4.9 to 4.11 below shows the various stages of fabrication.

Fig 4.9: Assembled crankshaft

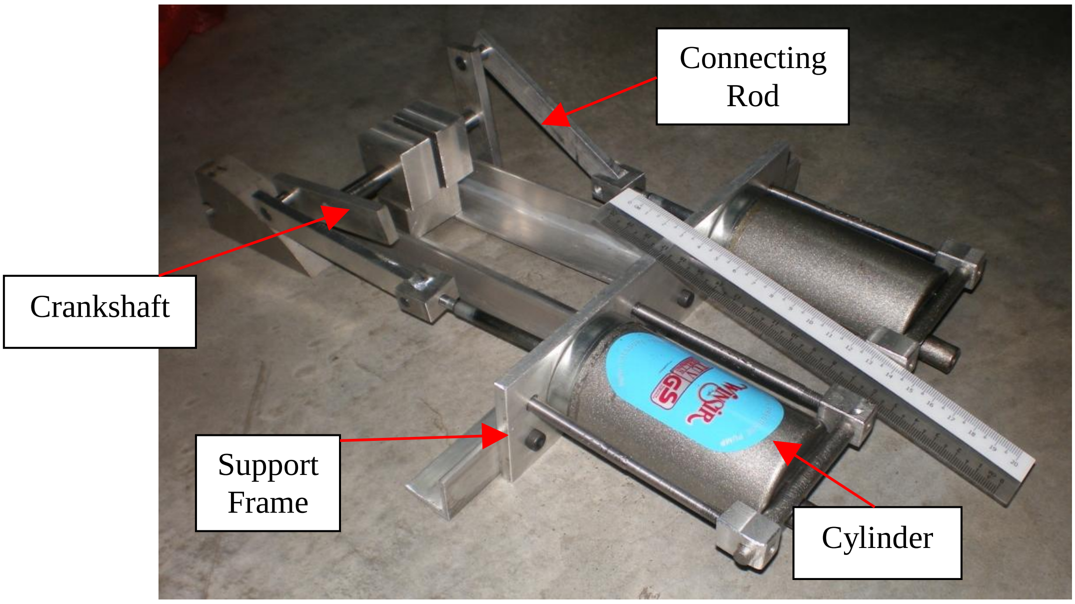

Fig 4.10: Mock-up showing fabricated components and approximate positions

Fig 4.11: Fully assembled engine

The piston had to be modified because it was designed with a moving piston ring that enables air to flow into the cylinder on the out-stroke while compressing air on the in-stroke. In a Stirling engine the working fluid in the cylinders are not ejected or sourced from the atmosphere but a re contained in the cylinders. Fig 4.12 shows the moving piston ring and how it works.

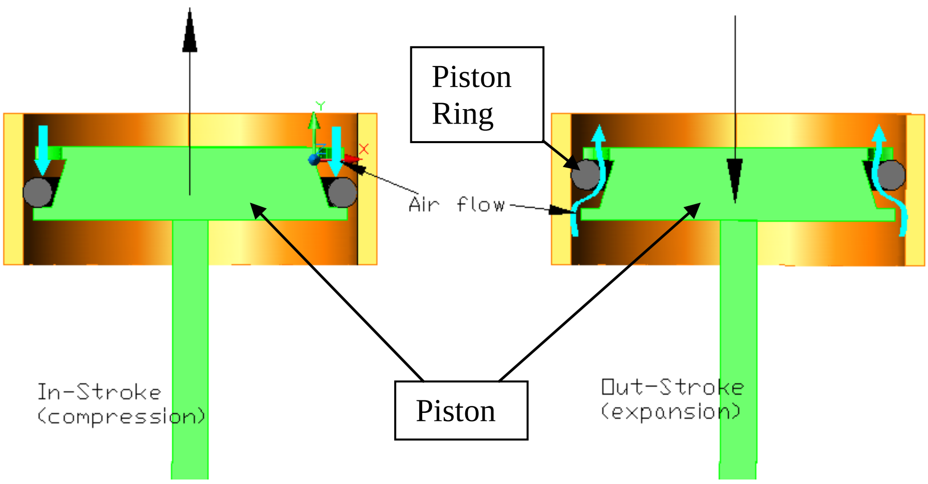

Fig 4.12: Operation of moving piston ring in foot pump

To overcome the problem of the moving piston ring, a spacer disc was machined from Perspex to hold the piston ring close to the bottom of the piston lip at all times. Fig 4.13 shows the schematic effect of the piston ring and fig 4.14 shows the completed and installed spacer ring.

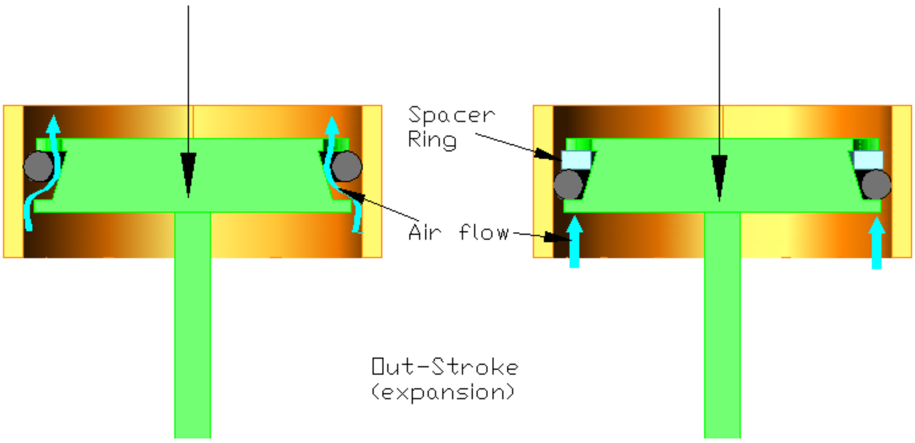

Fig 4.13: Piston outstroke with and without spacer ring

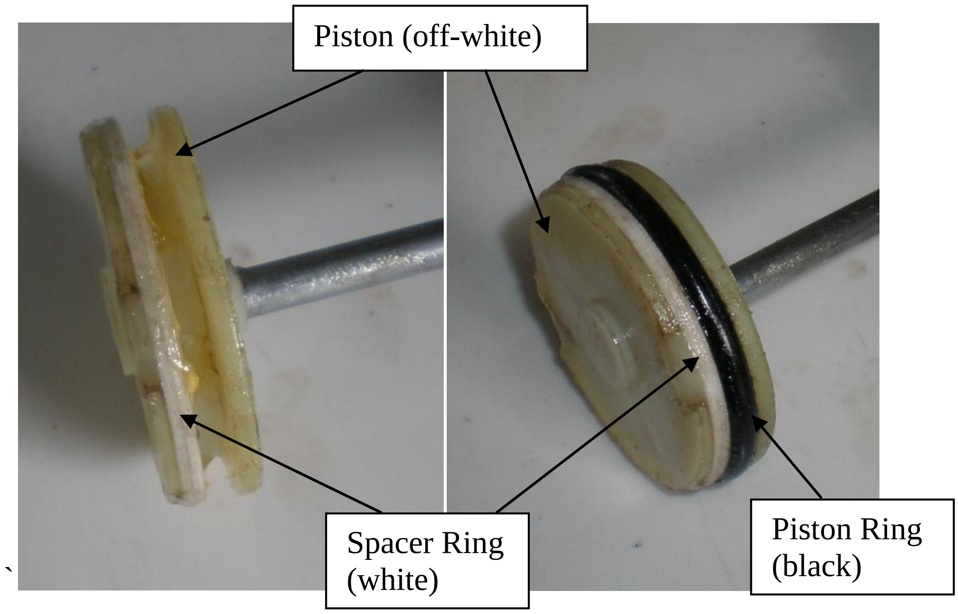

Fig 4.14: Completed and istalled spacer ring

4.3 Testing

The testing of the engine was originally planed to proceed in 3 stages. The first stage was to set the engine at design parameters of 380K input temperature and 300K output temperature and see if the engine could run.

If the engine runs, the second stage is to determine the operating characteristics of the engine, runaway speed, and speed and power output under different loads. The runaway speed is determined by marking a spot on the flywheel and recording a video of the engine under runaway speed. By viewing the video frame by frame the revolution of the engine can be determined. A diagram of the equipment setup used to measure torque is shown below.

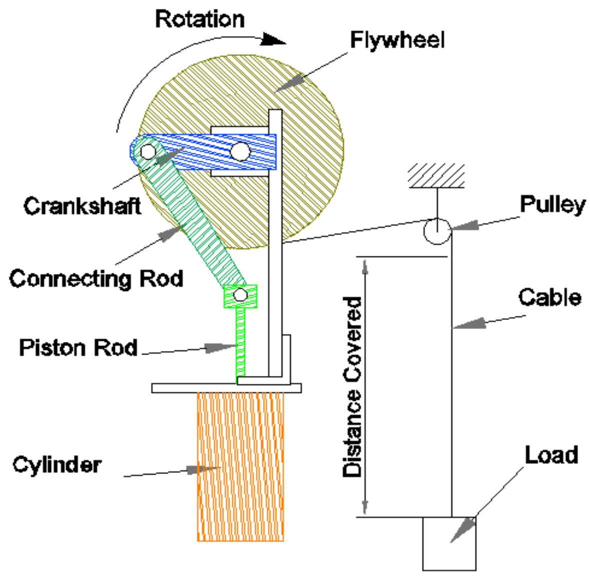

Fig 4.15: Diagram of torque measuring setup

Torque is given by the equation;

$$\tau = r \text{ x } F \quad \text{(Eq 4.9)}$$

Where:

τ is torque in N.m

r is the crank length in meters

F is the weight of the load in Newton

Power output is given by the equation;

$$P = \frac{W}{t} \qquad \text{(Eq 4.10)}$$

$$P = \frac{mgh}{t} \qquad \text{(Eq 4.11)}$$

Where:

P is the power output in Watt

m is the mass of the load in kg

g is the acceleration of gravity ($9.81 m/s^2$)

h is the distance covered by the load in meters

t is the time taken to cover distance *h* in seconds

The third stage of testing is to see if the engine can run under different operating conditions mostly higher or lower input temperatures and how it affects the engine characteristics.

The engine did not pass the first stage of testing due to poor sealing between the piston and cylinder. This prevented adequate pressure build-up in the cylinders to move the pistons.

4.4 Discussion

The main cause of the failure of the engine to run is because of the piston-cylinder assembly. Changing the design of the piston-assembly should solve the problem and achieve the objective of this project.

The use of a piston-cylinder assembly from a foot pump had both good and bad points. The benefit of buying the piston-cylinder assembly is that it reduces the fabrication time needed on the project. The attributes that made the piston-cylinder assembly are:

- Thin cylinder walls – enable very good conduction of heat to and from the working fluid
- Rated pass the maximum pressure generated by the engine – safety aspect
- Approximately the right size – design is 6.0cm bore and 5.5cm stroke, bought piston-cylinder assembly is 5.5 cm bore and 7.0cm stroke
- Availability

The downsides of using a bought piston-cylinder assembly are:

- Because a smaller bore is used, less force is applied on the crankshaft (F=PA)
- Seal around the piston is poor allowing air to escape from the cylinder and causing insufficient pressure to be built in the cylinder
- Low quality pistons are not perfectly perpendicular with the cylinder walls

Possible modifications to the piston to make the engine run are:

- Use a bigger diameter piston and cylinder to increase the force acting on the crankshaft
- Source or fabricate pistons with a closer fit with the cylinders (possibly high pressure pneumatic cylinders)
- If pistons are custom fabricated, dry piston seals are advised. Sealing from oil-free compressor technology using carbon or graphite based materials are commercially available and have superior performance to PTFE-based materials[2].

Excessive side loading of the pistons due to connecting linkage might also be a cause of sealing problems of the cylinder. An alternative linkage might be used to remedy this situation. Careful consideration must be made when considering more complex linkages because increasing complexity means increasing friction and energy loss. A Ross-yoke linkage is shown below which produces very little side thrust on the cylinder walls.

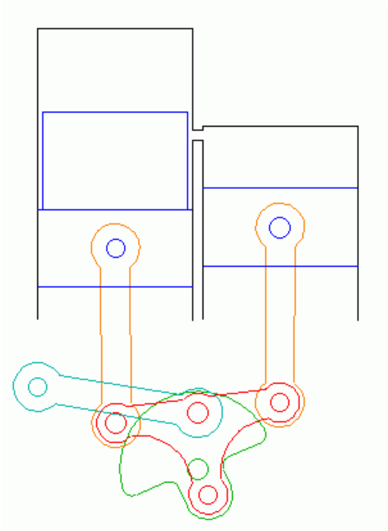

Fig 4.16: A Ross-yoke linkage used in a gamma Stirling engine[10]

CHAPTER 5

CONCLUSION

Stirling cycle engines are one of the many heat engines that can be used to harness useful power. Mere decades after its inception, the sun was used to power Stirling engines. Of the three types of Stirling engines the type alpha was chosen due to its simplicity. Low temperature differential Stirling engines have been built and can be powered by flat plate solar collectors. The design criterion was to produce about 10 Watts at 60 rpm or 10 Joules per cycle with an input temperature of 380K and an output temperature of 300K. Due to the size limitation of the purchased piston-cylinder assembly sourced from a high pressure foot pump the piston size was changed from 6.0cm bore and 5.5cm stroke to 5.5cm bore and 6.0cm stroke. This would produce 9.7J work output per cycle to give an output of 9.7W at 60 rpm with an input temperature of 380K and an output temperature of 300K. The fabricated engine failed to run due to poor sealing between the pistons and cylinders. It is recommended is to replace low quality pistons used with pistons and cylinders from high pressure pneumatic actuators. This will ensure extremely good sealing between the cylinder and piston. Redesign of crank linkage to reduce side thrust can also reduce gas leakage. The design offers several interesting properties including simplicity, low operating temperature and silent running if the problems are solved.

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