Design and Development of a Liquid Piston Stirling Engine

E90 Senior Design Project Report
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BY: FRANK KYEI-MANU
ALOYSIUS OBODOAKO

Advisor: Professor Carr Everbach

Abstract

An alpha-type Liquid Piston Stirling engine with a maximum power output of 23W and an efficiency of 3.4% was designed, built and instrumented for demonstration purposes. Based on in-depth research and design optimization, engine parameters were determined and a particular design was drafted. All the engine parts were manufactured in the Engineering Department's machine shop. The design was modified following initial testing and modified to improve performance. During the final two weeks of the project different sensor types were added to enable real-time data collection and processing. Furthermore, sufficient data was taken to characterize the fluidyne system. There is still a lot of work to be done in order to realize our initial project goals and it is our hope that the engine will be improved upon extensively in the coming years.

Acknowledgements

We owe Professor Carr Everbach a great deal of gratitude for his expertise, support and advice during the entire process but more especially during the initial stages of the project in getting us onto the right track. Secondly, we would like to thank Mr. Grant Smith 'Smitty' for his help in machining several system components, in ordering several system parts and directing us when it came to using the machines in the departmental shop. Professor Fred Orthlieb was also very invaluable in moving the project forward and his help in machining some of the system pieces when Smitty was away is very much appreciated. To our E14 students, Paul Agyiri '07 and Lauren '07, we say thank you for testing the regenerative materials in order suggest what the best material is for the purposes of our project. Finally to our peers, engineering faculty members and friends who kept pushing us on, we say thank you.

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1. INTRODUCTION

1.1 Background Information

There is an ongoing campaign for the need for alternative energy sources to meet the demands of today's world. The abundance of solar energy especially in sub-Saharan African is a resource that cannot be overlooked. This ever-present energy source is however underutilized despite the many uses to which it can be put. It is with this in mind that we intend to address one of the pressing needs in developing countries.

Residents in developing countries often cannot count on the availability of clean drinking water due to the pollution of surface water sources such as rivers and lakes. (see fig 1.1 overleaf). Thousands of deaths occur every year from water-borne diseases alone. In countries with plentiful sunlight, heat energy powered by a constant supply of solar energy could be used to pump well water. In addition, the water that is pumped could be boiled by the same focused sunlight, thereby providing a continuous source of clean water.

The purpose of this project is to design and implement a liquid piston Stirling engine that outputs enough power to pump water from a depth of at least 7 feet. We also intend to include a parabolic collecting mirror that will focus the sun's energy to heat the system. The system we plan to implement will use fluidyne technology, which is currently underappreciated.

1.2 Project Objectives & Goals

The primary objectives of the project are:

- To build a liquid piston Stirling engine with a power output of at least 5W, capable of pumping water to a height of at least 7 feet.
- To boil the water that has been pumped using focused sunlight.
- To choose a suitable design that incorporates mechanical simplicity with sustainability within the limitations of a third-world society.
- To raise awareness about fluidyne technology as an alternative, low cost energy source.

Fig 1.1 below shows the crude methods most rural folks in Africa obtain their water and attempt to purify them. As is evident for one or more of these images, water supply is generally unhygienic and the means of accessing water is not as efficient as it could be.

Slow flowing pond as source of water

Village water source. Girl with bucket in a ditch, fetching her water.

Woman pouring water into pot to be purified/sterilized by sunlight

Women standing next to a well, typically 20-30 feet in depth

Fig 1.1: Pictures of rural water supply and storage

1.3 Why a Liquid Piston Stirling Engine

A very important objective of this project, as was mentioned in the goals section in this report, is to design and develop a system that can easily be constructed given the limitations of a developing society. With this in mind, there is the need to choose a design that incorporates constructional simplicity; a fluidyne system provides this. It can be constructed using relatively simple and inexpensive materials. In our case, PVC tubing, which are primarily cheap and also come in different standard sizes, can sufficiently accommodate the needs of a Fluidyne System. A liquid piston Stirling engine can therefore be built without the need for sophisticated machining which is definitely a plus.

A second major advantage of liquid piston Stirling engines is that they are silent during operation. Compared to mechanical-piston Stirling engines as well as other pumps, fluidynes are extremely silent during operation which is an added benefit. One does not have to concern themselves with losses as a result of moving parts (mechanical pistons). In fact the only predominant losses that lower the efficiencies considerably of fluidyne systems are viscous losses. The oscillating liquid must be viscous enough to be able to sustain oscillations got a long period of time. The engines' efficiency ranges from 3-6%. Despite the low efficiency, the constant supply of solar energy all year round will be enough to power the engine to serve the needs of villages in a typical rural setting.

1.4 Liquid Piston Engines: A Historical Overview

Internal-combustion liquid pistons have been built and sold since the early 1900's. The first of these, popularly referred to as the Humphrey pump, is more-or-less an internal combustion engine, using either a two- or four-stroke cycle in which the conventional solid pistons are replaced with a liquid column. Inlet and outlet valves for the water, as well as a high-grade fuel to provide the energy to power the engine - usually petroleum – are used.

The major advantage of liquid pistons is clearly depicted by the aforementioned engine: its simplicity. Liquid pistons do not require accurately dimensioned cylinders and they permit great flexibility in mechanical design with relatively simple construction. For most fluidyne systems, the engine and the pump can be made as a single system. The use of a liquid-output piston avoids the need for a sliding mechanical seal, which has been a continuing problem for crankshaft Stirling engines.

Most of the early interest has centered around the use of Fluidyne machines to pump water. Like any other Stirling engine, the liquid piston engine can also be operated as a refrigerator or a heat pump and several people have proposed exploiting this.

1.5 Basic Operation of the general Stirling Engine

The basic principle of the Stirling engine is a simple one: it relies only on the fact that when a gas is heated, it tends to expand or, if confined, to a rise in pressure. There are currently three configurations of the Stirling engines – alpha, beta and gamma – available in the market. Our choice will depend on the power output we expect as well as on efficiency.

Stirling engines work by the repeated heating and cooling of a sealed amount of working gas which in our case will be air. The gas follows the behaviour described by the gas laws which describe how a gas' pressure, temperature and volume are related. When the gas is heated, because it is in a sealed chamber, the pressure rises and this then acts on the power piston to produce a power stroke. When the gas is cooled, the pressure drops and this means that less work needs to be done by the piston to recompress the gas on the return stroke, giving a net gain in power available on the shaft. The working gas flows cyclically between the hot and cold heat exchangers.

The Stirling cycle is an idealized thermodynamic cycle that involves two isothermal and two isochoric (constant volume) processes. The ideal Stirling cycle is shown below, and proceeds in the same direction - from 1 → 2 → 3 → 4 → 1.

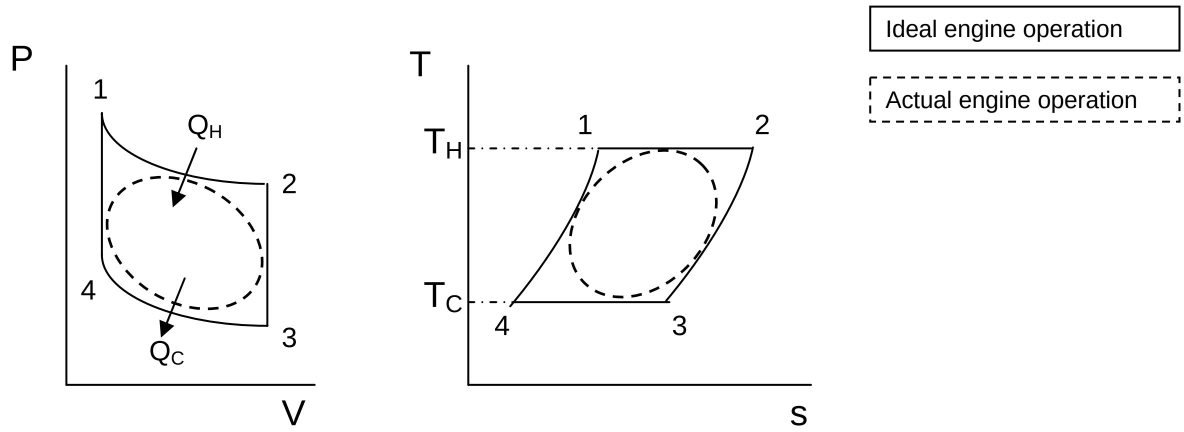


Figure 1.4.1: *Stirling cycle Pressure-Volume and Temperature-Entropy diagrams*

1 – 2: *Isothermal Expansion*

The working gas (air in our case) expands as heat Q_H is transferred to the expansion space of the engine. The gas expands and does work (usually work is done on a power piston), causing the engine volume to increase and the pressure to decrease. Assuming isothermal conditions ($T=T_C$), the heat transferred to the working gas is exactly $Q_H=W_E$, where W_H is the work done on the power piston.

2 – 3: *Isochoric Displacement (Cooling)*

The working gas is moved through the regenerator at the maximum engine volume. Heat is transferred from the working gas to the regenerator, causing the pressure, temperature and entropy of the gas to decrease.

3 – 4: *Isothermal Compression*

The cooled working gas is compressed (usually by a power piston) in the compression space, and heat Q_C is sunk to the cold reservoir at constant temperature T_C. Consequently, the engine volume decreases, while the engine pressure increases. Assuming isothermal conditions ($T=T_C$), the heat sunk to the surroundings is exactly $Q_C=W_C$, where W_C is the work done by the power piston on the working gas.

4 – 1: Isochoric Displacement (Heating)

The working gas is moved through the regenerator at the minimum engine volume. Heat is transferred from the regenerator to the working gas, causing the pressure, temperature and entropy of the gas to increase.

Types of Stirling Engines

The Stirling cycle can be implemented in practice through various types of engines. A breakdown of the key parts of a typical engine follows. The key parts of the engine are the pistons, connection rods, the crankshaft assembly, the heat source and heat sink, and the regenerator. All engines contain at least one power piston, which must have very low friction and near-perfect sealing for satisfactory engine operation. The engine output per cycle is the net work that the power piston does on the working gas over a cycle (i.e. one rotation of the engine shaft). The linear motion of the power piston is transformed into rotational motion by a drive mechanism. Usually the drive mechanism consists of connection rods. However, other configurations such as Ross yokes are possible[1].

The three basic types of Stirling engines are alpha, beta and gamma. Figure 3.1 contains basic schematics of each type. Alpha engines have two power pistons, while beta and gamma engines have a power piston and a displacer. Beta engines have a power piston with a coaxial displacer; while gamma engines consist of a power piston and displacer in separate cylinders. The three basic types are well described in the literature [1,4], and we will outline only their main characteristics here.

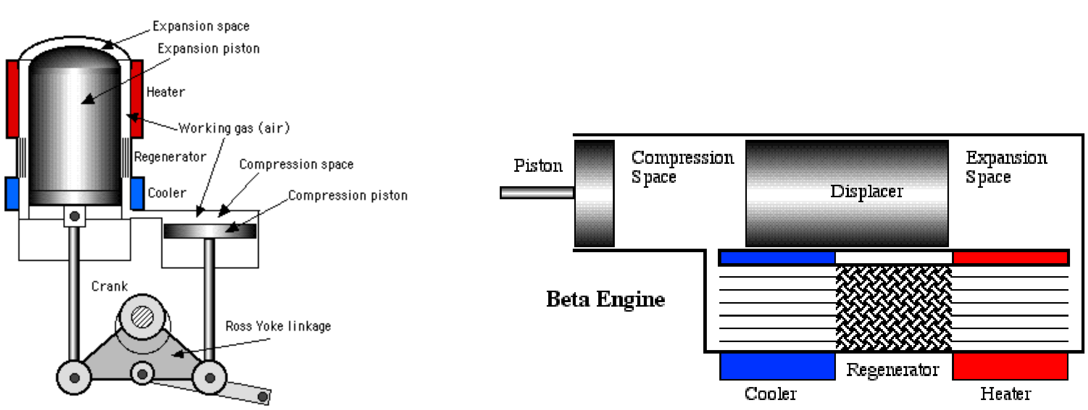
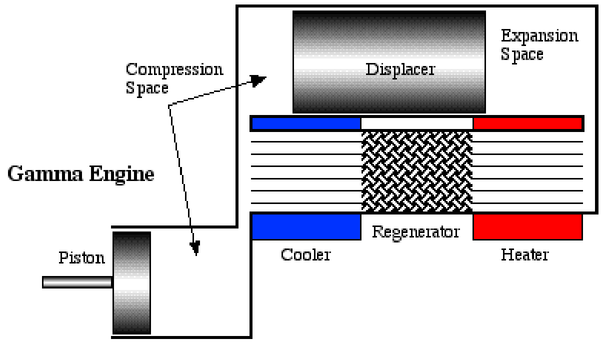


Figure 3.1.a) A typical alpha-type Stirling engine. b) A typical beta-type Stirling engine.

[1] The Ross yoke drive mechanism is discussed at http://www.ent.ohiou.edu/~urieli/stirling/engines/engines.html



source: http://www.ent.ohiou.edu/~urieli/stirling/engines/engines.html

Figure 3.1. c) A typical gamma-type engine.

Each of the three engine types has certain advantages and disadvantages. The primary disadvantage of alpha type is a requirement for perfect sealing for two pistons. Also, the configuration of the two pistons at an angle to each other can be cumbersome for construction, especially for demonstration engines.

Beta engines offer potential significant advantages, especially in terms of efficiency and size, but are technically complex. The requirement of a coaxial displacer and power piston makes for difficult machining of the engine. This requirement also increases friction in the engine because the displacer necessarily slides in and out of the power piston. Moreover, the mechanical drive mechanism used to convert translation of the pistons into rotation of the output shaft is a difficult design problem to say the least. Finally, while a beta demonstration engine would be interesting, the coaxial piston motion and regenerator position make it difficult to see Stirling cycle principles at work.

Gamma engines are best equipped for educational purposes. With two parallel cylinders attached to a common crankshaft (see Figure 3.2 below), it is relatively easy to discern the four stages of the Stirling cycle, and to study the functions of the displacer, power piston and regenerator in each thermodynamic process.

1.6 Basic Operation of the Liquid Piston Fluidyne Engine

The liquid piston Stirling operates quite differently from the generic Stirling engine described above. The most obvious is the fact that the mechanical pistons are replaced by water. Therefore, as the hot side is heated, the increased air pressure raises water on the cold side and lowers water on the hot side. The left-hand U tube which has one end heated and the other end cold functions as a displacer; and the right-hand tube, which has one end open to the atmosphere, works as the output, or power, piston; this configuration is generally known to be the gamma configuration.

When the water in the displacer is set oscillating- by manually rocking the system to jumpstart-from one limb of the U tube into the other limb and back, it is obvious that at one point in the cycle, top dead air in the cold end will correspond to bottom dead air in the hot end; this situation is illustrated in the left hand part of fig 1.5.1 overleaf, in which most of the air trapped above the water in the displacer is in the hot left-hand limb. Most of the air is therefore hot, so its pressure will rise, which tends to force the tube to move from right to left as the arrow indicates in the figure below;

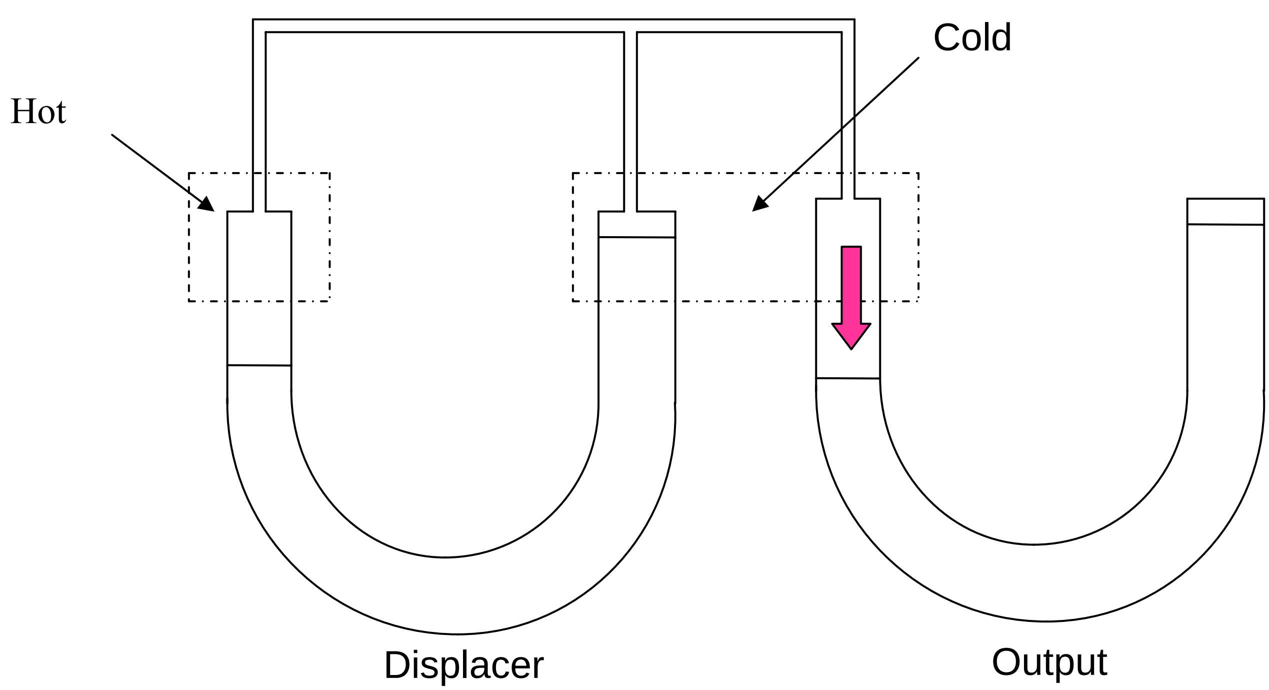
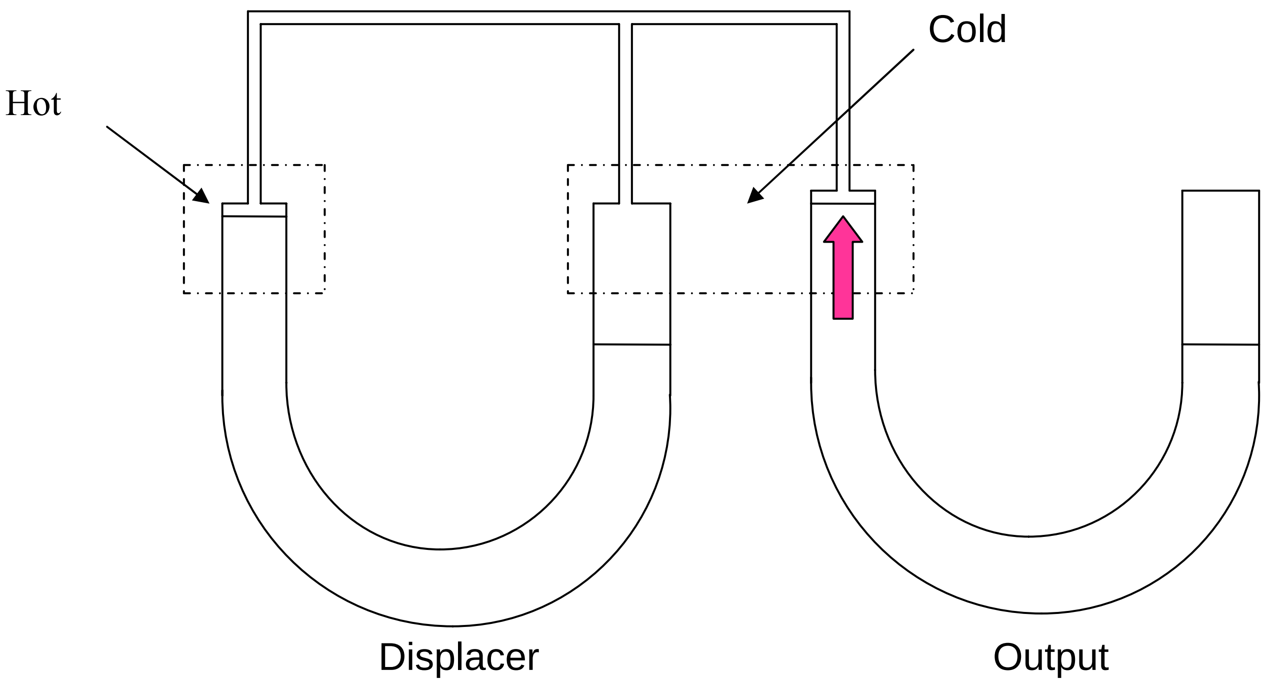


Fig 1.5.1: Basic schematic of a fluidyne

Half a period later, the displacer water will have swung back into the other limb, so that the cold surface is at the bottom dead center, which is the situation at the right hand side of fig 1.5.1. Most of the air is therefore in the cold side of the machine, so its pressure will fall, pulling the water in the output column back from left to right.



The most important factor in Stirling engine design is the efficiency losses due to non-idealities. Due to imperfections, the efficiencies of Stirling engines are significantly lower than the ideal ones. The non-idealities include adiabatic losses and heat losses from mechanical components.

1.6 Effects of Evaporation and Mean Pressure

An implication of evaporation is to increase the heat input required since latent heat must be supplied, which also leads to an increase in the power output. However, the water system may be limited to efficiency of 1 percent or less or less at these high temperatures. It has been observed that evaporation has a marked effect at high temperatures in Fluidyne systems leading to lower efficiencies. By suppressing evaporation, at least 10 times greater efficiency can be achieved. To deal with this problem, we intend to increase the mean pressure of the working fluid to increase the pump's pumping capability and its efficiency and minimize the evaporation by designing a float which will rest on top of the liquid column in the hot side of the engine.

1.7 Applications

The engine output power has been extracted and used for pumping, including irrigation pumping. Three simple ways to use the Fluidyne output to pump water have been identified.

The first is known as series coupling which simply requires a T piece placed at the end of the output column and two non-return valves. On the inward stroke of the output liquid when the gas pressure inside the engine is low, liquid is drawn in through the lower non-return valve. On the outward stroke, liquid is forced out through the upper valve. However, the presence of the T piece (pumping line) in series with the output column of the non-return valves may upset the tuning of the system because with the non-linearities associated with the valves, the behavior of the output tube as a resonant, oscillating system may be seriously affected.

The drawbacks that have just been highlighted could be remedied by placing the pumping system in parallel with the output column. The volume of liquid moving in the output tube can be greater than the volume passing through the pump, so that there need not be a close correlation between the pumped volume and the engine stroke which in the series configuration, has to be the case. Secondly, nonlinearities in the flow through the pump will have relatively small effects on the oscillation of the larger amount of liquid in the output line. In this case, the output line does not do any work directly, except to overcome its own losses, but merely oscillates at a frequency tuned to that of the displacer, thus giving rise to relatively large pressure variations within the engine for pumping purposes. For this reason, the "output tube" is usually referred to as the "tuning line" or the "tuning column" highlighting the fact that its main function is to have a large, resonant oscillation and not to provide a direct output mechanism.

1.8 Report Organization.

The report has been broken down into several sections. This first part has been a general introduction to the broader topic of Stirling engines and has narrowed our scope focus to Fluidynes. The theory section (section 2) explains the theory underlying our design as well as the model equations and concepts that are pertinent to our system .The next section (section 4) discusses the design process and the considerations that were important in arriving at our system's parameters. Section 5 discusses the construction and assembly of the different parts of the engine after which this paper presents and discusses our results, draws relevant conclusions and recommends ways to improve on the existing design. The appendices then follow.

2. THEORY

2.1 Working Fluid and pressure

To reach our target our target performance, there is the need to keep evaporation at a minimum and the working fluid must always be in the gaseous phase. To this end, and for constructional simplicity, air is usually used at a mean pressure equal to that of atmospheric pressure (0.1Mpa).

To suppress the losses that come with evaporation, a float, made from an insulating material is used. If this is done, the displacer liquid is largely isolated from the heater and the working gas. It should be strong enough to withstand the hot side temperatures.

2.2 Operating Temperatures

High powered, high-pressure Stirling engines typically operate with the heat exchanger at 700 to 800 º C, but more sophisticated materials are needed for this.

An ideal fluidyne would be able to run with a temperature difference of less that 1º C.[2] However, for real machines, there are flow and power losses which require that the hot end be kept as hot as possible, while the cold end be kept cold; that is, to maintain a steep temperature gradient.. With that said, a convenient upper temperature figure should be

[2] Elrod, 1974; Geisow, 1976.

between 120 - 300 º C for a machine which incorporates design simplicity like ours and uses relatively available and inexpensive materials for construction, insulation and jointing. The cold side temperature can be maintained at water's ground level temperature of about 10 º C.

2.3 Displacer Frequencies

To get the maximum amplitude of oscillation in the liquid columns, the flow losses should be low and the frequency of operation should be close to the natural, or resonant, frequency of the columns themselves.

Derivation of the operating frequency

To illustrate this, let us first consider the displacer alone: in this case, we can imagine both ends of the column to be open to the atmosphere. To start the oscillation going, we raise the level slightly in one arm of the tube- for example by blowing on the other end. If the liquid surface rises by a distance x on one end, it must fall by the same amount on the other. One end of the liquid column now has more weight of liquid than the other, by an amount $2\chi A_d\rho$ where: χ = the amount by which the liquid is displaced; A_d = the cross sectional area of the U-tube; ρ = density of the liquid. (see fig 2.3.1 below).

The pressure arising from this is $2\chi\rho g$, where g is the acceleration due to gravity and hence the resulting force is $2\chi A_d\rho$. The mass of the liquid column is $A_d\rho L_d$ and so the acceleration induced by this force (in a direction to reduce χ) is given by

$$A_d \rho L_D \ddot{\chi} = -2\chi A_d \rho g \quad (2.3.1).$$

$$\ddot{\chi} = \frac{-2gx}{L_D} \quad (2.3.2)$$

This is the equation for undamped simple harmonic motion and the natural frequency omega, ω, is given as:

$$\omega = \sqrt{\frac{2g}{L_D}} \text{ rad/s} \quad (2.3.3)$$

$$\text{or} \quad f = \frac{1}{2\pi}\sqrt{\frac{2g}{L_D}} \text{ Hz} \quad (2.3.4)$$

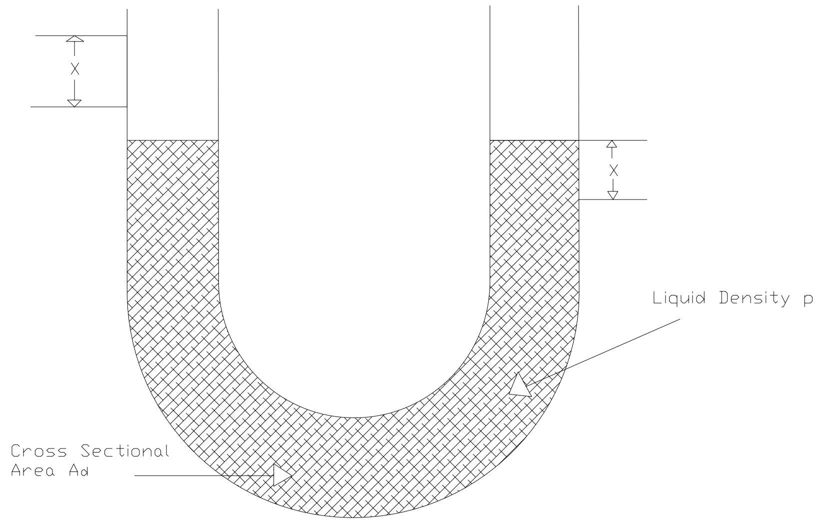


Fig.2.3.1 Simple Displacer U-Tube

A displacer length of less than 1 ft may be impractical for the however, one in the range of 1-2m, corresponds to frequencies of 0.5-0.7Hz, which are common in most practical engines.

2.4 Tuning of Liquid Columns

As in any oscillating system, the maximum amplitude of movement in the output column will be achieved if the frequency of the pressure variations, ie, the driving force upon it, is about equal to the natural or resonant frequency of the water oscillating in the output column. These pressure variations are due to the oscillations of the displacer water, so it follows that for maximum movement, the two natural frequencies have to be equal.

As long as the water in the displacer can be kept oscillating back and forth in its U-tube, the water in the output tube/tuning line will also move back and forth, taking the pressure that it needs from the changing air pressure in the machine.

<u>Derivation of tuning column length given displacer frequency</u>

The calculation of a natural frequency for the tuning column is slightly more difficult because the forces caused by compression, or expansion, of the gas above the liquid in the machine are not canceled out by acting equally on both ends of the liquid column as they are in the displacer.

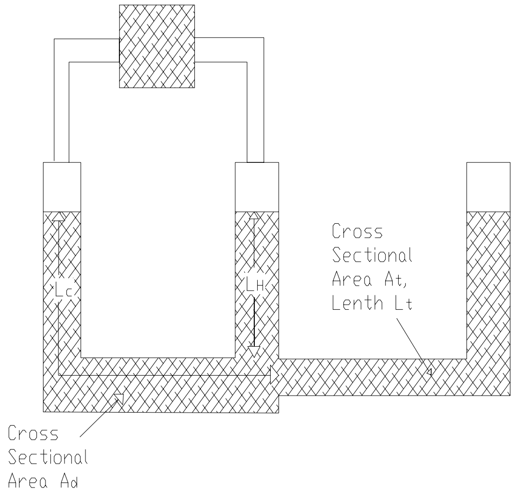


Fig 2.4.1: Tuning column configuration with merged cylinders

Fig 2.4.1 shows a representation of the tuning column that represents a merged cylinder machine. It has liquid length L_t and a cross-sectional area, A_t. One end is open to the atmosphere. The other end terminates in the displacer.

Displacing the water in the open end of the tuning column downward by an amount χ does two things. First, it raises the liquid at the other end by an amount χ. Secondly, it reduces the value of gas above the working space by an amount $A_t\,\chi$. Both effects give rise to a pressure difference across the tuning column tending to force it back toward equilibrium position.

If the gas is initially at a pressure P_m and at a volume where the tuning column is at a mid-stroke, V_m, the space above the liquid is isothermal as we would like it to be in an ideal Stirling engine, then the pressure will rise by an amount p, where:

$$P_mV_m = (P_m + p)(V_m - A_t\,\chi) \qquad (2.4.1)$$

according to the ideal gas law. Therefore,

$$V_mp = (P_m + p)A_t\,\chi \qquad (2.4.2)$$

Stirling engines usually have a relatively low compression ratio, so that p is generally fairly small compared to P_m. Consequently, an approximation for $p = A_t\,P_m\,\chi/V_m$.

For a merged cylinder configuration as has been shown in Fig. 2.4.1, the pressure difference between the liquid surface in the displacer and the open end of the tuning column is:

$$\triangle P = \frac{P_m A_t \chi}{V_m} + \rho g \chi + \frac{\rho g \chi A_t}{2A_d} \qquad (2.4.3)$$

Gas compression | Tuning liquid level lowered | Displacer liquid level raised

Most of the pressure difference will act across the tuning column, generating an angular velocity which is equal to

$$\omega = \sqrt{\left(\frac{A_t P_m}{V_m \rho L_t} + \frac{[1 + A_t / 2A_d]g}{L_t}\right)} rad/s \quad (2.4.4)$$

Or

$$f = 1/2\pi \sqrt{\left(\frac{A_t P_m}{V_m \rho L_t} + \frac{[1 + A_t / 2A_d]g}{L_t}\right)} Hz \quad (2.4.5)$$

The compressibility of a perfectly isothermal gas is equal to V_m/ P_m where V_m and P_m are, respectively, the initial volume and pressure. The compressibility of a perfectly adiabatic gas is $V_m/ \gamma P_m$, where γ is the specific ratio of the gas. For a mixed isothermal-adiabatic volume, we can simply calculate the overall compressibility of the working gas from the weighted average of the mean isothermal and adiabatic volumes, V_i and V_a.

$$\frac{\Delta P}{\Delta V} = \frac{-P_m}{V_i + {}^{V_a}\!/_{\gamma}} \qquad (2.4.6)$$

Substituting equation 2.4.6 into equation 2.4.5 gives an approximate formula for the natural frequency of the tuning line in a Fluidyne machine with mixed isothermal and adiabatic spaces.

$$f_t = (1/2\pi) \sqrt{\frac{1}{L_t} \left[\frac{\pi R_t^2 P_m}{\rho (V_i + {}^{V_a}\!/_{\gamma}} + g\left(1 + \frac{R_t^2}{2R_d^2}\right)\right]} \qquad (2.4.7)$$

Where R_d is the diameter of the displacer and R_t, the diameter of the tuning column. Equation 2.4.7 can be rearranged to give a simple relation between the tuning column length, L_t and its radius R_t for any given frequency of operation f.

$$L_t = \frac{\{\frac{\pi R_t^2 P_0}{\rho[V_i + V_a/\gamma]}\} + g[1 + \frac{R_t^2}{2R_d^2}]}{4\pi^2 f^2} \qquad (2.4.8)$$

2.5 Power Output

For a machine with separate displacer and tuning column cylinders, the power output, as approximated by Cook and Yarbor[3], is given by:

$$W_0 = P_m V_o f \pi \frac{V_e}{4V_m} \frac{T_e - T_c}{T_e + T_c} \sin\theta \qquad (2.4.9)$$

where V_o = volume swept out by the surface of the tuning column.

V_e = volume swept out by either surface of the displacer.

V_m = mid-stroke volume.

P_m = mean pressure.

T_e and T_c = temperatures of the hot and cold spaces respectively.

θ = phase angle between the displacer and tuning column

f = frequency of operation.

Ideally, the goal is to get the two movements fairly near to 90˚ out of phase, i.e θ = 90˚, in order to maximize the value of $\sin\theta$. Data collected on solid piston Stirling engines suggests that the actual net power (i.e after losses) is usually 0.3 to 0.4 times the output calculated ideally. The very limited experience available with well designed liquid piston Stirling engines is consistent with this. We therefore use the figure 0.3 as a suitable correction factor, recognizing that at lower temperatures or for small machines, where various losses become more important, the formula will tend to overestimate the power available. Therefore a more accurate equation for the power output is given as:

$$W_{net} \simeq 0.3 P_m V_o f \frac{\pi}{4} \frac{V_e}{V_m} \frac{T_e - T_c}{T_e + T_c} \qquad (2.4.10)$$

To realize an increased power output without the need for a proportionately high input power to overcome the latent heat of evaporation, an insulating float is necessary as was briefly discusses in section 1.6 of this report In addition the need for an extremely high input power should again be taken care since heat losses will be accounted for by using

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[3] By West, 1971.

thermal insulation around the heat-exchanger as well as other system parts that are good thermal conductors.

2.6 Losses

The liquid piston Fluidyne engine has no rotating or sliding solid parts, and therefore no mechanical friction. It, however, suffers from the viscous and other losses associated with flowing fluids, especially flowing liquids and air. In addition, just like in the more conventional Stirling engines, it suffers from the fact that the gas spaces are, in general, neither perfectly isothermal nor perfectly adiabatic.

2.6.1 Viscous Losses:

To determine viscous losses due to oscillating flow in the tubes, it is necessary to determine whether the tube is to be treated as narrow or wide. It is "wide" if the dimensionless quantity R * which is the radius parameter and can be expressed as,

$$R_t\sqrt{2\pi f\rho/\eta} \qquad (2.4.11)$$

is much greater than unity. f is the frequency of the oscillation, R_t, the diameter of the tube and ρ, η are the density and viscosity, respectively, of the fluid. Substituting the parameters for water at room temperature, and using a frequency of 0.52 Hz, which is the operating frequency of our fluidyne, we find that the radius parameter becomes

$$R_t\sqrt{2\pi * 0.52 * 10^3/0.001} \qquad (2.4.12)$$

With a design choice where the radius of the tuning column is 0.05m, the radius parameter, R * = 90.37 which implies that our cylinders can be treated as wide when we calculate viscous effects of water flow.

Closely related to viscous flow is the resistance coefficient which is defined as the pressure drop per unit length divide by the mean flow velocity. For nonturbulent oscillating flow in wide tubes, as we have determined above, the resistance coefficient is given as:

$$R = \sqrt{2\rho\omega\eta}/R_t \qquad (2.4.13)$$

It is obvious, therefore, that the pressure drop for a given flow rate increases only as the square root of the viscosity η.

2.6.2 Power Losses in Fluid flow:

The instantaneous rate of power loss in the flow is equal to the volume flow rate multiplied by the pressure drop. The volume flow rate is equal to the mean flow velocity multiplied by the cross-sectional area of the tube. The rate of power loss can easily be calculated from the resistance coefficient given that, by definition, the resistance coefficient is the pressure drop per unit length. Therefore:

$$\Delta P\dot{V} = \sqrt{2\rho\omega\eta}\,\frac{L_t \dot{V}^2}{\pi R_t^3} \qquad (2.4.14)$$

Equation 2.4.14 shows that the instantaneous value of the rate of power loss is proportional to the square of the volume flow rate.

In developing the afore-derived equations, the implicit assumption has been that the flow is not turbulent. At higher Reynold's number, the transition may take place from laminar to turbulent flow and there is therefore the need to determine a critical Reynolds number which we can express as:

$$R_e(critical) \simeq 375\left(\frac{R_t^2\omega\rho}{\eta}\right)^{2/3} \qquad (2.4.15)$$

2.6.3 Kinetic Flow Losses

In addition to viscous losses, there are minor pipe losses that occur when the fluid must change speed or direction, for example at a bend or a pipe exit. The pressure drop caused by a minor pipe loss can be expressed as a factor which depends on the type of obstruction multiplied by the velocity:

$$\Delta P = K\frac{1}{2}\rho\upsilon^2 \qquad (2.4.16)$$

Where K represents the minor pipe loss coefficients. Table 2.4.1 presents K values for different types of bends and exits:

Element	K
90˚ smooth bend	0.15-0.25
90˚ mitre bend	1.0
Sharp edge contraction	0.5
Sharp edge enlargement	0.2

Table 4.2.1: Minor pipe loss coefficients

The total pressure drop due to all the bends, constrictions, enlargements, etc, in the system will be obtained by summing the contributions from each of them individually. With this in mind, the minor pipe loss can be expressed as:

$$E_k \simeq 0.42\Sigma K \frac{\pi\rho f^3 V_o^3}{2R_t^4} \qquad (2.4.17)$$

The minor pipe losses increase with the cube of the frequency and the cube of the swept volume. Also, as the tube diameter is decreased, the kinetic losses increase more rapidly than do viscous losses.

2.6.4 <u>Heat Losses:</u>

A fluidyne operating at atmospheric pressure is a low power density machine and for a given cylinder size, the power output is much lower than for a high-pressure, high speed Stirling engine. Consequently, heat losses by conduction through the insulation and other engine components are significant and must be minimized if a high efficiency is to be attained.

The main heat components (hot cylinder, heat exchanger and regenerator) will be roughly, cylindrical in shape. This implies that our equations will be developed to apply to such cylindrical devices. The heat loss Q_i from a cylinder of diameter D_1 and length L surrounded by insulation of diameter D_2 is given by:

$$Q_i = \frac{2\pi k L \Delta T}{\ln\left(\frac{D_2}{D_1}\right)} \qquad (2.4.18);$$

where k is the thermal conductivity.

2.6.5 Shuttle Losses

This is the heat loss due to the motion of the displacer piston. In the stationary state, the temperature of the piston will be approximately equal to that of the adjacent cylinder wall. When the piston moves, each section of its surface moves to confront a new part of the cylinder wall, at a different temperature. Heat is transferred between the two adjacent surfaces at different temperatures. The formula for calculating the shuttle losses[4] is:

$$Q_s = \frac{\pi s^2 k \Delta T D}{8Lg} \qquad (2.4.19);$$

where s = stroke

k = thermal conductivity of the gas between the piston and cylinder

ΔT = temperature difference between the hot and cold end of piston

D = piston diameter (or cylinder insider diameter)

L = Length of piston

g = gap between piston and cylinder

It is evident that this equation is frequency-independent. The reason is that if the frequency is, for example, increased, the time available for heat transfer is during each motion of the piston is proportionally reduced. Therefore, the amount of heat transferred during each cycle is inversely proportional to the frequency. The total amount of heat transferred per unit time is therefore equal to the amount transferred per cycle multiplied by the number of cycles. One of these factors varies inversely with frequency, and the other, directly with frequency. The overall effect is therefore independent of the frequency.

2.6.6 Other Losses that could affect system functionality

The loss parameters that have been described above represent the more significant losses we expect to encounter with our design. However, there are a few other avenues for losses which we will neglect because we do not think they will significantly affect our systems performance. An example refers to pumping losses, which comes about as a result of the fact that with a solid float on the surface on the liquid surface in the hot side of the machine, the gap between the cylinder wall and the lower end of the liquid

[4] Equation is discussed in Martini's *Design Manual,* (1978)

meniscus is close while the gap is open at the top. As the pressure in the cylinder varies, gas flows into and out of this volume. Since the lower end of the gap is kept cold by the oscillating liquid column, extra heat must be added to this gas as it leaves the space. This loss is however pretty small in Fluidyne machines; we will therefore ignore it for the purposes of our design.

Another minor heat loss source comes from the fact that extra heat input may be needed because of the inefficiency of the regenerator. The regenerator, as it will be explained shortly, reheats the gas as it returns to the hot cylinder. Since there is a great possibility that it will not be perfectly effective, extra heat must be supplied from the heater which could lead to the system being less efficient.

Finally, the heat that is stored in the hot components of the engine must be supplied by the heat source which means that if it does not go into heating the air but rather the hot pieces of the device, then it represents a lost energy source which could make the system a less efficient.

2.7. The Regenerator and its Operation[5]

Although not absolutely necessary for engine operation, the regenerator is a key element of the Stirling engine, and it distinguishes the Stirling engine from other external combustion engines. The action of the regenerator can be described in simple terms as that of a 'thermal sponge' – it absorbs and releases heat during different stages of the cycle. The idea of regeneration is that if some of the heat added to an engine can be stored within the system between the heating and cooling stages, the efficiency can be significantly increased. Thus, if there is an element in the engine that absorbs heat during the cooling stage and releases it during the heating stage, the compression space will cool faster and less heat will need to be added to the hot space in order to keep the engine running.

There are different ways of making regenerators in practice, and most of them employ significant surface area to enhance the heat transfer rates and thus the amount of heat stored and released by the regenerator during a cycle while minimizing the

[5] Description of the regenerator adapted from external source. E90 Report by Milos Ilak'04 and Jesse Hartigan '04.

impedance to the flow of the working gas through the regenerator itself. This is usually achieved by using arrays of tubes, wire meshes, or by using porous materials with high heat capacity through which the flow of the working gas is forced. In practical engines, there is often a significant tradeoff between the gains in efficiency due to the regenerative action and the flow losses. The flow rate of the working gas can be reduced greatly, which causes losses to the power output. For small engines, these losses can be higher than the potential benefits of a regenerator. Also, the passages in the regenerator may add to the dead space in the engine, reducing the power output and efficiency further. However, even with no regenerator, some regenerative action will still be provided by different engine elements. In the case of an engine that contains a displacer for example, the thin annulus of air around the displacer can provide some regenerative action.

Figure 2.7.1 shows a sample regenerative displacer piston which would not cause an increase in the dead space or flow losses, since the displacer would have to exist in order to provide reciprocating motion to the power piston as described earlier in this section. The temperature gradient is considered negligible in the longitudinal direction, and high heat transfer rates to and from the regenerator are assumed in the radial direction. One way of achieving this highly anisotropic thermal conductivity is to make the regenerator matrix out of light plastic in which radial highly conductive wires are embedded.

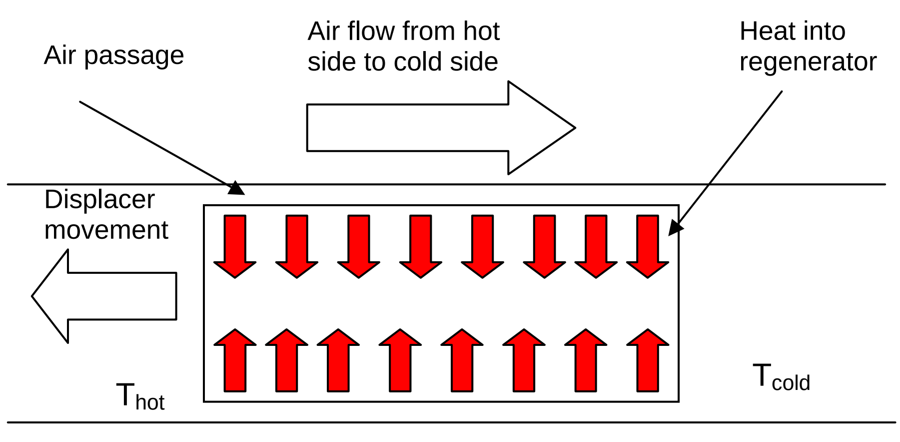


Figure 2.7.1. a) Heat transfer occurring from the hot air to the regenerator as the air moves towards the cold space.

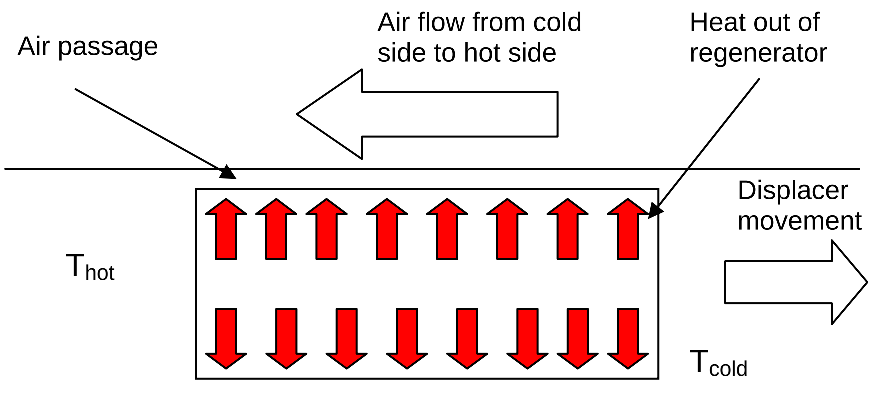


Figure 2.7.2. b) Heat transfer from the regenerator to the working gas as the cold air moves towards the hot space.

3. DESIGN PROCESS

In this section, we discuss the design process, the reasons for the design decisions made and the design parameters we chose and calculated, based on the equations derived under the theory section.

3.1 Our Design

One of the advantages of a liquid piston system is constructional simplicity. A liquid piston always fits its cylinder exactly; no piston rings are needed for sealing; no bearings are required to support oscillating components. Consequently, Fluidyne engines can be quickly and easily built with the simplest of tools and materials.

There are several practical examples made from different types of materials. Examples include the plastic engine, wooden engine and glass fluidyne pump among others, which implement the Stirling cycle. The design we chose is an alpha type-high temperature difference liquid Stirling engine with the following general features:

1) *Construction Simplicity:*

For a project of this scope, with severe time and budget constrains, it was crucial to pick a design that would be well constructed and tested on time. The major components of our design include the displacer, tuning column, heat exchanger and the regenerator, all of which can easily be constructed from relatively simple and inexpensive materials

2) *Machining Simplicity*

As stated above, liquid pistons generally incorporate constructional simplicity which implies that machining the various pieces in place is not necessarily a difficult task. Unlike with Stirling engine designs which utilize mechanical pistons, there is really no reason to achieve picture-perfect machining on any part of the system. In some mechanical pistons for instance, it is imperative that the displacer and the output piston be aligned perfectly so that no significant friction effects develop.

3) *Instrumentation Easiness*

Our design incorporates various testing devices that make it easy to collect relevant data on our system as well as to characterize it. The design incorporates various sensors as well as other analog measurement devices in order to realize the above mentioned objective.

4) *Ability to improve upon design*

A feature of a great design is one that allows future work and improvements to be performed on it. The materials and parts that are implemented in the system definitely lend themselves to future improvements and further work. It is our hope, actually, that future students will find it easy in developing the existing model; this factored immensely into our design choice.

Fig 3.1.1 below shows a crude 2-D AutoCAD model of our system. The section below discusses and obtains quantitative values for several system parameters.

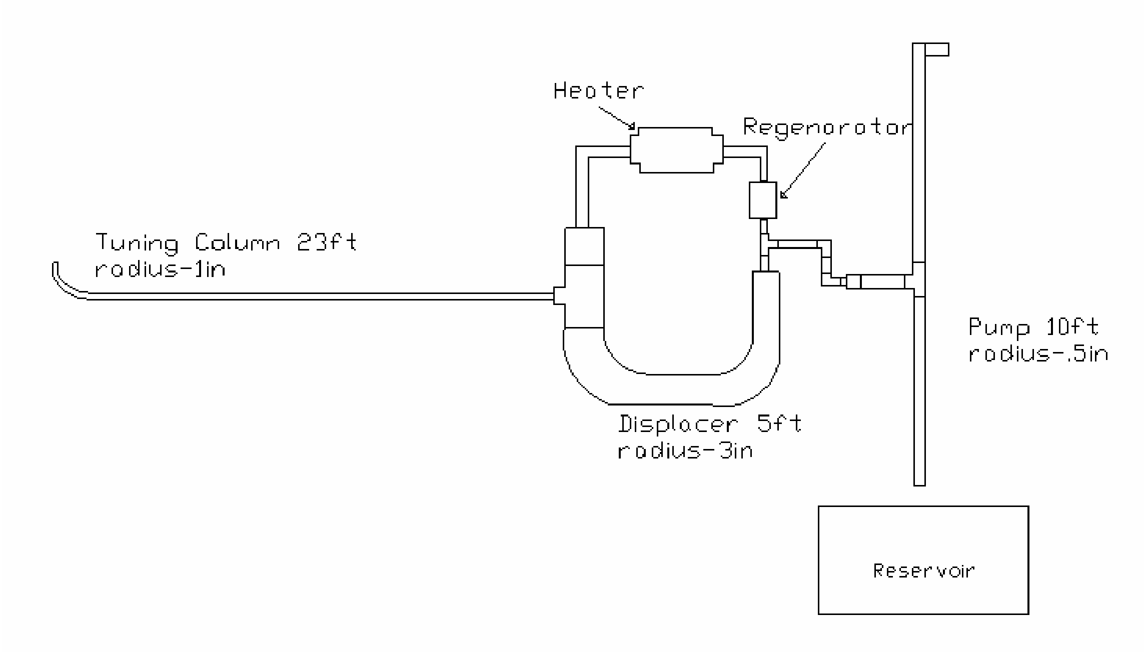


Fig 3.1.1: Design diagram of our fluidyne pump

3.1.1 Engine System Parameters

For conveniently sized tubes/design, the frequency of fluid oscillation varies from 0.5 to 0.7 Hz. Thus a reasonable approximation for our design is 0.5 Hz. With is in mind, we apply equation 2.3.4 to obtain the length of the displacer, L_D.

$$L_D = \frac{g}{2\pi^2 f^2} \qquad 3.1.1$$

L_D = 183 cm (6 ft) long and 15.2 cm (6 in) inner diameter. With this diameter, the liquid in the tube will theoretically oscillate at 0.5Hz.

Swept Volume

The next design parameter to determine is the engine-displacement, or swept , volume. The Beale number[6] B_n enables us to generate a fairly good estimate. It offers a simple relationship between mean cycle pressure, operating frequency, engine displacement and power output.

Power = Beale number * mean pressure * frequency * displacement

$$W = B_n P f V_o \qquad (3.1.2)$$

[6] Walker, 1979

For our system, we have a target performance of $0.5m^3/hour$. Therefore, with a pumping arm of length 10 feet (3.05 metres), the theoretical power output through this head for our system is given as:

$$0.5m^3/hour * 10^3 * 3.05m * 9.81/3600 = 4.16Watts \quad (3.1.3)$$

Fluidyne machines tend to be approximately 4-6% efficient. We therefore expect to provide an input power of approximately 70 watts.

The swept volume, from equation 3.1.2 is therefore expected to be:

$$\frac{4.16}{0.005 * 1 * 0.52} = 1131.37cm^3 \quad (3.1.4)$$

This is the total volume change during each cycle of the engine. However, in the case of an alpha configuration machine with the hot and cold pistons moving with approximately equal strokes and about 90° out of phase, the total volume change is $\sqrt{2}$ * (the swept volume of either piston). Therefore for our particular design, each piston should have a displacement of 1131.4 cm^3.The projected stroke, which is the amplitude of the oscillations can then be expressed as

$$\frac{s\pi 0.15^2}{4} = 1131.37 * 10^{-6} \ \ m^3 \quad (3.1.5)$$

This equation yields a stroke of 6.23cm.

Dead Volume

The optimum compression ratio in a Stirling engine is about 2:1. For an alpha configuration machine, if both pistons have a swept volume V and differ in phase by 90°, the total volume of the working gas in the cylinders, excluding clearance spaces is given by:

$$V(1 + \frac{1}{\sqrt{2}}\cos(\omega t - \frac{\pi}{4})) \quad (3.1.6)$$

If the unswept volume available for clearance space, heater and regenerator is V_D, then the compression ratio, i.e., the ratio of the maximum to the minimum is equal to 2 when V_D = 1.12V. The unswept/dead volume of our system is therefore calculated to be approximately 1267cm^3.

Other miscellaneous design parameters

At a frequency of 0.52 Hz, the number of strokes per hour is expected to be 0.52*3600 = 1877/hour. To be able to pump $0.5m^3$/hour requires 0.5/1877 = $266cm^3$/stroke.

The pumping system is gas-coupled to the Fluidyne by means of an air-filled pipe. The mean volume of air in this pipe must be at least $133cm^3$ because this is the anticipated volume swept in each column during half cycle. To minimize dead volume so as to maintain a relatively high compression ratio, the air-filled couple was designed to have a volume close enough to $133cm^3$. The mean volume must be at least this value, rising to $266cm^3$ and falling to zero during each stroke. This implies that we have a remaining dead volume of about $1134cm^3$ to allocate between the heat exchanger, regenerator as well as for clearances and connecting pipes.

3.2 Components to the System

The fluidyne system is composed of five main parts. These component parts include the heat exchanger, regenerator, tuning column, displacer column, and pump column. The respective components of the system function in such a way as to bring about pressure variations, thus allowing for the suction and pumping of water from some depth.

3.2.1 Heat Exchanger

The heat exchanger is the device in the system where the moving fluid, in our case air, is heated. The exchanger is the device where heat transfer occurs between the heat source, which is a coiled piece of nichrome wire and the air. Although the heat exchanger is the means by which the system's hot side is being heated, the moving fluid is constantly entering and exiting the exchanger every cycle at different temperatures.

For testing purposes a gas powered torch will be used as a means to bring the exchanger to an appropriate high temperature (350 – 450 F). The gas powered torch will be utilized because it allows the heat exchanger to reach the desired temperatures in a rather short period of time. Furthermore, it is a very simple apparatus to use for the purposes of heating the exchanger, thus triggering oscillation. To improve the heat transfer from the copper cylinder to the air inside, a 6 in long piece of thin copper was rolled into a coil and placed inside the exchanger. The outer surface of the coil was placed in such a manner at to be in contact with the inner surface of the cylindrical 8in copper tube. The thermal expansion of copper is $17*10^{-6}$/C.

Nichrome wire will be used as a heat source within the heat exchanger during the actual operation of the system. The gage of the wire being used is 29 AWG (0.0113 in dia.) and it will be to maintain the temperature inside the exchanger between 350 – 450 degrees Fahrenheit. Fluidyne machines tend to be 3-4% efficient so it was determined that 70 watts input power would be needed. To maintain the desired temperature, 100 – 150 watts will be the power input to the wire to achieve the temperatures desired within the exchanger.

The nichrome wire will be wrapped around a 1.5 in x 1.5 in x 6 in long piece of ceramic block, which will be centered inside of the copper heat exchanger. The ceramic will be utilized because it can withstand temperatures of up to 1400 F. In addition, the ceramic provides a way to situate the nichrome wire inside the heat exchanger without the wire touching the copper casing, thereby; shorting the wire and limiting the amount of heat that can be generated.

3.2.2 PVC

The tubing used in constructing the major part the system is PVC (Polyvinyl chloride) material. This material is a widely-used plastic that is commonly used for similar applications as that of this project. In addition, PVC is relatively easy to assemble and cheap. The melting point for PVC is 212 C and has a heat transfer coefficient of 0.16 W/m k. The three parts of the system that consist primarily of PVC are the displacer, tuning, and pump tubing.

3.2.3 Displacer

To achieve a flow rate of 0.5 m^3/h the displacer tubing was constructed to be. To move 0.5 m^3/h of water the number of strokes per hour for the hot and cold spaces was calculated to be 1877. The displacer has two chambers; one chamber is the hot space the other chamber is the cold space. The volumes of both chambers were derived to be 231 cm^3. The hot and cold chambers are adiabatic and isothermal volume spaces, respectively.

3.2.4 Pump column

The pumping column is 305 cm (10 ft) long and has a diameter 3.81 cm (1.5 in), thus the volume of the tubing is 133 cm^3 (8.11 in^3). The pump is an adiabatic volume space. The volume of water per stroke was determined to be 266 cm^3/stroke. The total volume change during each cycle of oscillation was derived to be 1131 cm^3. The volume of water drawn per stroke would be deposited in the air-filled couple. The air-filled couple connects the pump to the rest of the system. The volume of the air-filled couple must be at least half the volume of the max volume of water/stroke, 266 cm^3/stroke, but for the purposes of this project it was constructed to be 231 cm^3/stroke or 86% of 266 cm^3/stroke.

3.2.5 Regenerator

The liquid piston Stirling engine is a alpha configuration model; therefore, a regenerator component will be included in the system operation. The regenerator typically consists of a mass of wires and is located between the reservoirs (hot space and cold space). Two types of regenerative material will be tested and analyzed and the most efficient of the two materials will be in the final system operation. The materials being tested are small rocks and steel brillo pads. When the air is moving between the hot and cold sides its heat is transferred to and from the regenerator. The regenerator contributes to the efficiency of the Stirling cycle by storing and releasing the heat to and from the air. The regenerator has a volume of 1400 cm^3 (85 in^3) and will be situated above the cold space and to the right of the heat exchanger. The regenerator is an isothermal volume space.

3.2.6 Connections

The volume for the tubing used to connect the respective components and the clearances in the system was calculated to be 210 cm^3 (12.8 in^3). The total volume consisted of an adiabatic space of 109 cm^3 (6.62 in^3) and an isothermal space of 101 cm^3 (6.18 cm^3).

Table 1: Mean Volume of Adiabatic and Isothermal Spaces

Component Parts	Isothermal cm^3	Adiabatic cm^3
Hot Cylinder		266
Cold Cylinder	266	
Regenerator	1400	
Heater	925.8	
Pump		133
Clearance/Connections	101.3	108.5
Total	2693.1	501.5

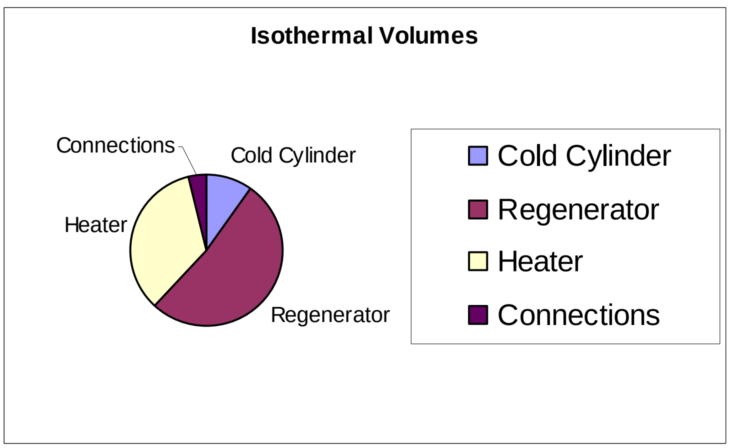


Fig 3.1.2: Isothermal Volumes

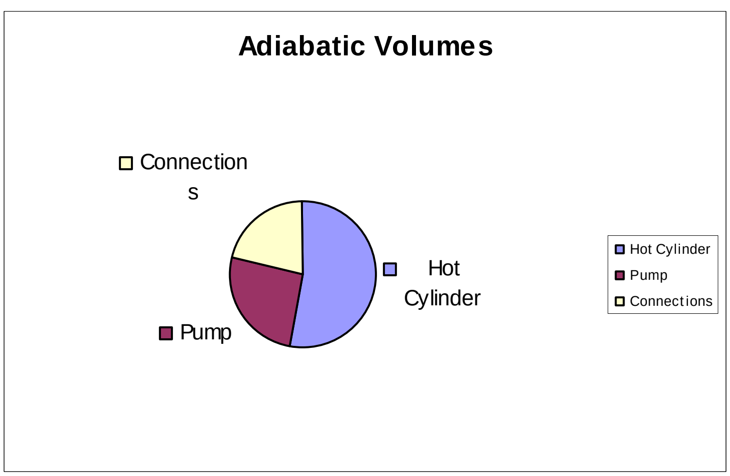


Fig 3.1.3: Adiabatic Volumes

3.2.7 Tuning/Output Column

The tuning line will reach it maximum amplitude of movement in output when the frequency of the pressure variations is about equal to the natural or resonant frequency of the water oscillating in the output column. To achieve the appropriate length of the tuning line the total isothermal and adiabatic volume spaces, table 1, were used in equation 2.4.8. The length of the tuning line was calculated to be 7.13 cm^3 (23.4 in^3).

3.2.8 Unistrut Structure

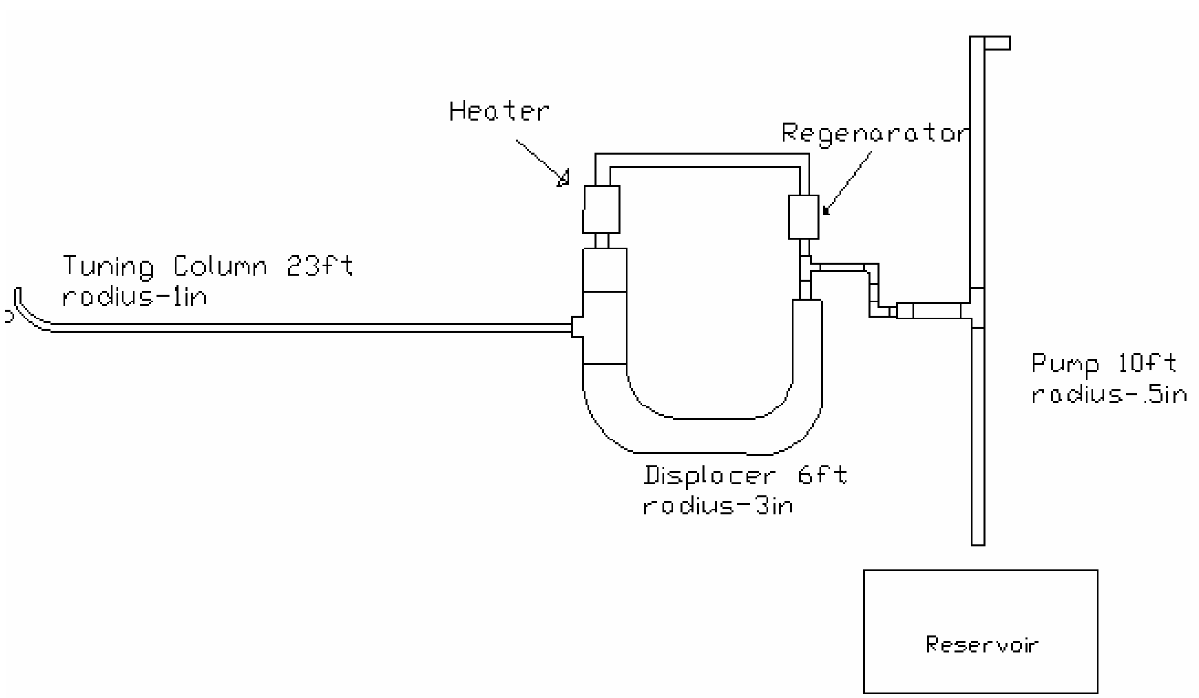
The fluidyne system was not designed to maintain its own weight, so a simple structure was created to provide the needed support. The support structure is made of unistrut metal framing material. The unistrut system is primarily used for mechanical support and is designed to greatly simplify and reduce the cost of structural framing. The metal support system has the advantage of being able to be adjusted and connected into numerous different support structures. In addition, there are many different types of mounting methods, accessories, and fittings that are used to construct the structure, without necessarily using the methods of welding and/or drilling.

3.2.9 Summary of Design & Parameter sizes

Parameter	**Value**
Working Fluid	Air
Material for displacer, tuning column, pump	Polyvinyl Chloride (PVC)
Support Frame	Unistrut
Insulating Float	Styrofoam
Oscillations Phase Angle	90°
Cold Cylinder Stroke	TBA
Cold Cylinder Diameter	6 inches (isothermalized)
Hot Cylinder Stroke	TBA
Hot cylinder diameter	6 inches (adiabatic)
Hot cylinder Float length	4 inches (above water level)

Float-cylinder gap	3.5mm
Tuning column diameter	50mm
Tuning column length (determined from equation 2.4.8)	23 feet
Pump-arm mean volume	133cm^3
Regenerator internal volume	1440 cm^3
Heater inner volume	1440 cm^3
Dead space Volume	3250 cm^3
Swept volume	7000 cm^3
Heater Temperature	Max. 400 °C
Cold Temperature	10 °C

The Final Design



4. CONSTRUCTION & ASSEMBLY

4.1 Introduction

This section will provide details on the design, fabrication and construction process as well as the decisions that we made to improve on the original design. A couple of pages are been dedicated to explaining exactly how we constructed the various engine parts because we believe there is the need to document this process should there be a desire, in the future, to improve on the existing design. Please refer to the appendix to see the various crude sketched, with dimensions, of the different engine parts.

The construction of the engine parts, after the design decisions had been made, took the most time primarily because one member of the group had no prior experience with the machine shop which meant that we needed to spend some time learning how to use the machines in the shop. We got under way around the second the second week of the semester and completed all the engine parts that constituted our initial design by mid April. We are currently in the process of testing several engine parameters to see if there are any design decisions we may have to alter. As an example, we have just decided to set up a better and more effective heat exchanger and have machined pieces that enable us to utilize a nichrome wire rather than our original idea which will be explained shortly. We have also incorporated a pressure transducer into the design to enable us get a sense of the pressure variations within the system. The circuitry for this device is however, yet to be completed. Additionally, thin wire thermocouples have been placed in the hot and cold spaces inside the system.

4.2 Machine Shop Work

The engine was constructed and the pieces assembled in the departmental machine shop in Papazian with the help of Grant Smith, the machinist .We ordered almost all of our polyvinyl chloride (PVC) pieces from outside. We also needed caps that would enable us make the transition from higher diameter pipes to the smaller diameter ones. We therefore ordered end caps which we machined to accommodate our desired PVC size. There were several other pieces we had to order since they were not readily available in the shop.

All the parts were fabricated using standard machinery – Bridgeport milling machines and lathes, band-saws, drill presses, a belt sander, etc. For the more sophisticated parts, we sought the assistance of Grant Smith and Professsor Orthlieb. However, for most of the engine components, we made them and gained useful insights and experience in the process.

The machinist, Mr. Grant Smith, ordered and or provided the materials that were necessary in advancing the project. The table below shows the ordered system components and the approximate cost.

Material/Part	**Cost/$**[7]
PVC Tubes (6”, 2”, 1”,¾”)	70.00
Transparent PVC	10.00
Six inch T	15.00
Carbide Silicate Board	15.00
High Temp. RTV	15.00
Copper Foil	20.00
Nichrome Wire	15.00
Phenolic	20.00
Plastic End Caps	10.00
Pressure Transducers	10.00
Thermocouples	15.00
TOTAL	**220.00**

Table 4.2.1: Engine Part costs

4.2.1 Displacer

The displacer, U-tube, which is made of 6 in ID PVC was cut to the desired lengths using the band saws, after which the ends were beburred with the belt sander. The length of six inch tubing on the hot side is 3 ft while that on the cold side of the displacer is 2ft. A 6 inch ‘T’ connects from one of the two six inch elbows at the lower ends of the displacer,

[7] The figures given here are all approximate.

which give the U-shape of the displacer, to the tuning line and the rest of the hot engine column

4.2.2 Six inch to three quarter inch end caps

To minimize the dead space and to maintain consistency with our design calculations, there was the need to transition from the six inch tubes to 3⁄4 inch tubes. To make this transition, we machined two circular discs and center drilled them so the 1 inch OD connecting tubes would fit in snugly. In addition, we had to ensure that the discs would fit into the 6 inch ID of the bigger pipe and so half the thickness of the circular discs was machined accordingly on the lathe. A third disc was machined in a similar fashioned with the only difference being that, since it transitioned from 6 inches to the 2 inch tuning column, the hole was made to accommodate the outside diameter of the pipe.

4.2.3 The tuning column

The length of the liquid column is about 23 ft. We therefore have a 21ft 2in PVC that runs across horizontally. Through a 90 degree elbow, a 2-3ft long clear PVC was attached vertically making it possible to view the oscillations. We also attached an outlet to the tuning column which makes it easy to drain out the liquid in the machine after it has been used.

4.2.4 The Regenerator

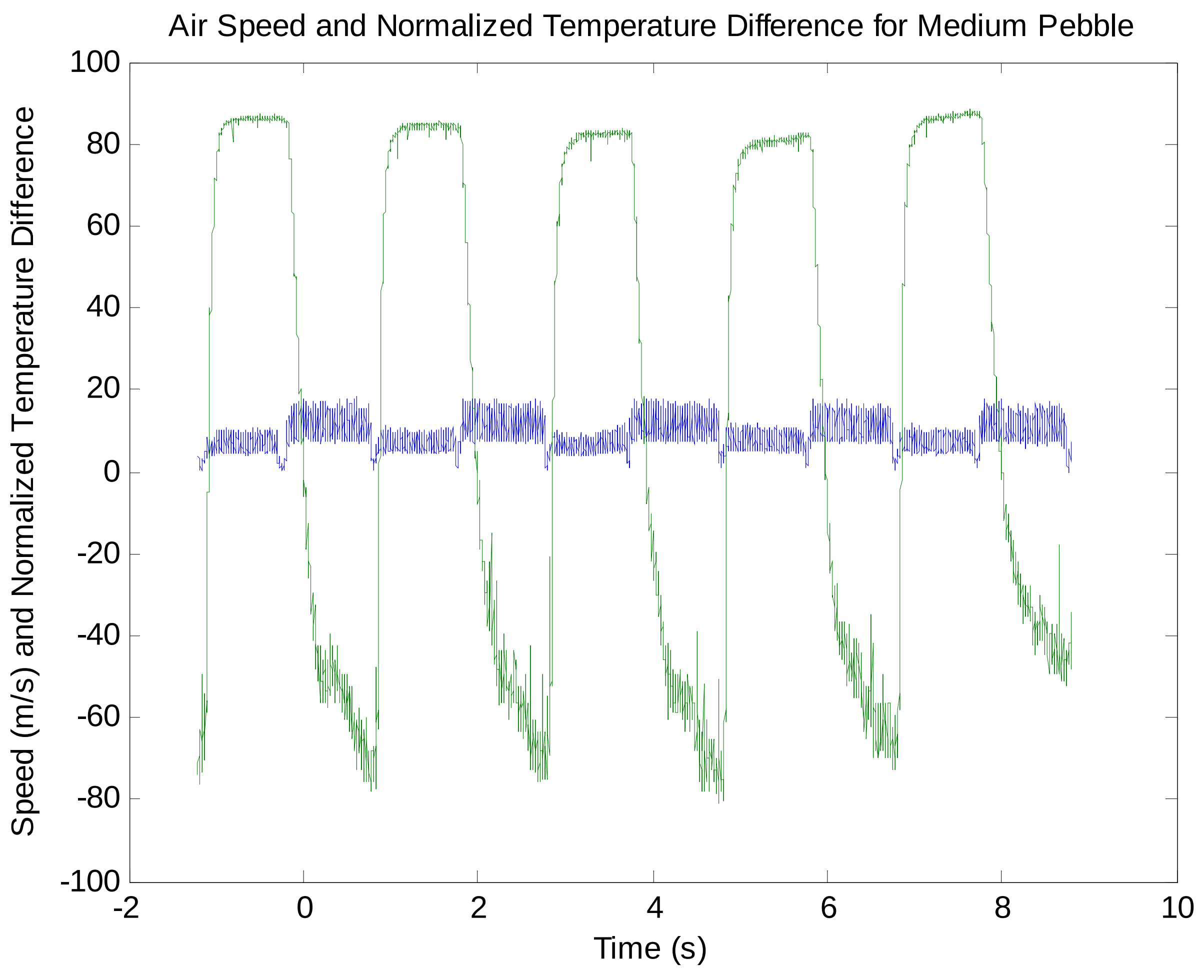
The regenerator was constructed and the material recommended by Paul Agyiri'08 and Lauren '08 with the help of Professors Orthlieb and Everbach as an E14/E90 project collaboration. The goal of this collaboration was to have the E14 students recommend the best regenerative material to use in our system. They setup an experiment which consisted of using two blow-dryers; the one at the hope blowing hot air, while the one at the bottom gave off cold air. They then designed 4 inch, diameter tube with end caps threaded in so as to ease the opening and closing of it. The end caps were fitted around the 4 inch long tube. In addition, steel trays were made to hold the regenerative materials in place. The materials they tested included 4 pads steel wool, 2 pads steel wool, small, medim and large pebbles, with screens and finally, with the regenerator empty. The e14 students determined the temperature of both the hot air at the top, and the cold air at the bottom and how they varied respectively as they moved through the regenerator. In addition, they looked at the velocity profile of the air as it moved through the regenerator to determine the effect different regenerative materials have on the speed of air in the system. Table 4.2.2 below shows the results of their experiments. Based on the data presented here, there is no obvious ideal choice for our regenerative material. Different materials registered different ideal results. For example, the medium sized pebble yielded the highest hot cycle temperature difference, while the 2 pads of steel wool yielded the largest cold cycle temperature difference. The small pebbles on the other hand recorded the biggest hot and cold cycle speeds which, as the data suggests, does not vary much among the different possibilities.

Material	Hot Cycle Normalized Temperature Difference	Cold Cycle Temperature Difference C	Hot Cycle Speed m/s	Cold Cycle Speed m/s
Empty	51.3439 ± 0.1764	5.6513 ± 0.2687	7.0433	9.2747
Screens	66.8115 ± 0.132	15.116 ± 0.169	6.9687	8.4921
Small Pebble	75.56 ± 0.132	-1.1825 ± 0.1753	8.7461	10.7823
Medium Pebble	**78.4945 ± 0.1079**	1.1734 ± 0.1834	7.3498	10.391
Large Pebble	74.1889 ± 0.1006	-0.0549 ± 0.1981	7.3249	9.8273
2 Pads Steel Wool	73.245 ± 0.1115	**17.9396 ± 0.1579**	7.0467	9.7818
4 Pads Steel Wool	70.6185 ± 0.1016	13.186 ± 0.1533	7.6907	9.0476

Table 4.2.2

We made the decision to use the medium size quartzite pebble primarily because of its high temperature difference at the hot side of the engine. The cold size temperature difference in our system was not a big issue since we are of the strong opinion that the cold chamber really is not as cold as we would have wanted it to be. Secondly, the hot and cold cycle speeds recorded are high enough compared to the other materials that we would not have to worry much about kinetic flow losses.

The graph below therefore shows the relationship between air speed and the stage in the cycle for the medium bebble. As is evident from it, the high temperature difference and the approximately constant cold and hot cycle speed makes it an ideal regenerative material for our system.



4.2.6 The Heat Exchanger

A lot of relevant information concerning the heat exchanger has been addressed in the system design section of the report. For preliminary analysis, that is, to enable us test the system, heating with a blow torch, we placed a copper foil, coiled a number of turns to ensure that the air is heated through to the center of the tube. The tube is held in place to the ¾ inch PVC by a material called phenolic, which has the property of being strong as well as serve as a heat bridge between the copper tube and the PVC. The phenolic is glued to the copper tube with high temperature RTV which has the advantage of being able to withstand extremely high temperatures.

For the purposes of testing and empirically knowing the exact amount of heat input into the system, we are going to replace the copper coil within the heat exchanger with nichrome wire. This would be coiled around a rectangular block of carbide silicate material 1.5 * 1.5 * 6 inches. A ½ inch hole was drilled through the center of the block through which a solid rod which would be used to hold the block in place. To do this, holes are drilled through the copper, screws are put through these holes which then press firmly on the rod, to hold it in place.

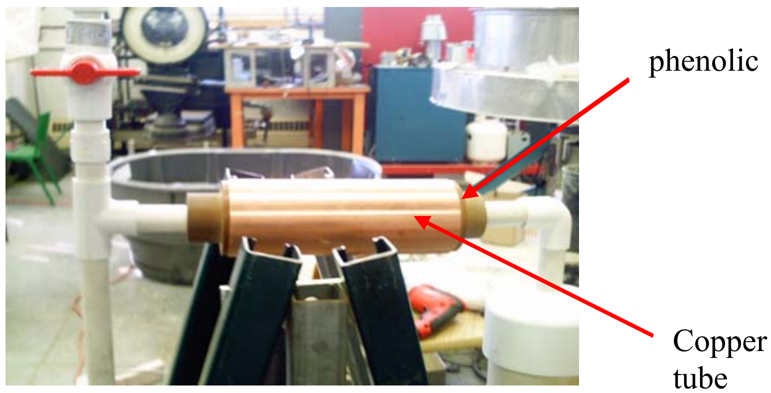


Fig 4.2.1: Picture of the heat exchanger

4.2.5 *Carbide Silicate/Fibre Funnels*

With the huge pipe losses that are associated with air moving from a tube to another tube of different diameter, there was the need to minimize the losses and make the transition less sudden and also to prevent turbulent flow. The phenolic, described above, was machined to have a "funnel like" interior. Also, the carbide silicate material, which provides strong inter-connecting fibre, was machined to funnel-like shapes to fit the regenerator tube ends as well as at the entrance to the 2 inch tuning column.

4.2.6 Unistrut

Given the size of the system, as well as the fact that it will not stay put without a support frame, we designed a Unistrut support frame to hold the displacer portion of the system in place. The frame has two triangular truss-like members and a rectangular base. The triangular members are separate by about 8 inches, giving enough room to hold the machine in place. At the base, the rectangular cross section also provides enough room for the lower portion of the engine. The entire frame is supported at the base by two straight pieces that run across the ground as can be seen in Fig 4.2.2. One inherent drawback of the current unistrut design is the face that it does not lend itself to being able to move the machine around a whole lot which is an issue that could be addressed in the future.

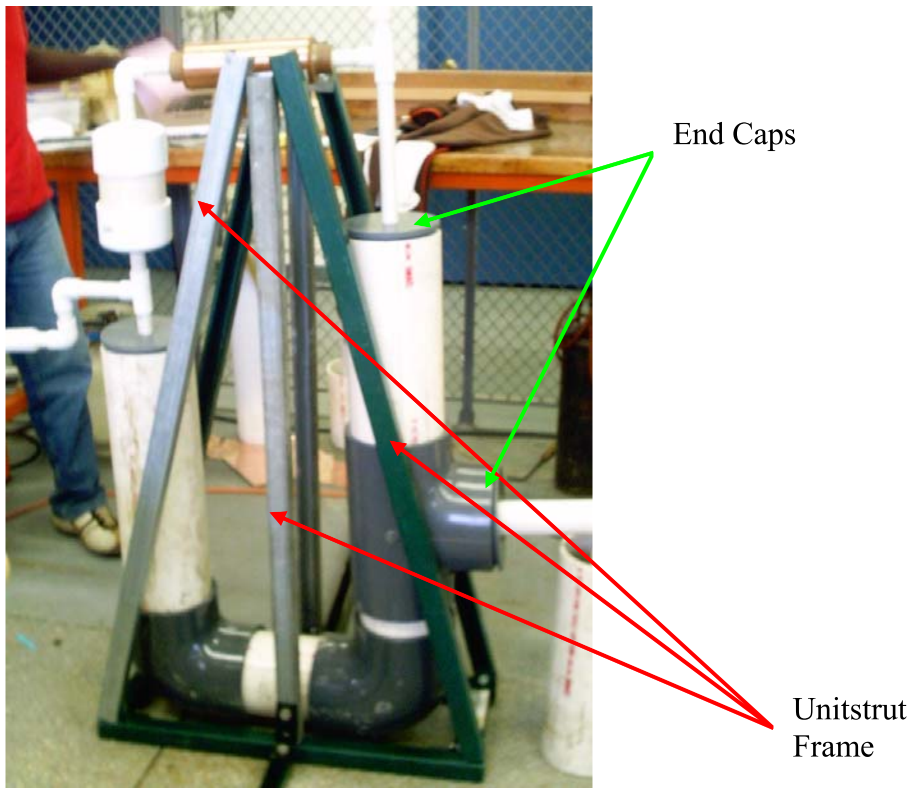


Fig. 4.2.2: Picture with Unistrut frame

4.2.7 Insulating Float

A 4 inch Styrofoam insulating float was made with a height on 4 inches as well as a gap of 1/8 inches between it and the cylinder wall. The gap width was kept at a minimum to reduce the shuttle losses in that space.

4.2.8 Modifications

4.2.8a Heater

The initial heater used during the first run of the system was replaced with another heater that was of similar design. The initial heater was taken from a blow dryer and contained a built in cut-off switch as a safety precaution in case of over heating. The cut-off switch did not allow for the heat exchanger to reach the necessary temperatures of 300-350^{o}C needed for operation. The blow dryer heater was replaced with a heater that was composed of a nichrome coil wrapped around a ceramic base. The new heater was powered by a veriac whose maximum input power was 664 watts.

4.2.8b Volume Changes

To accommodate for the change in the heating device the volume of the heat exchanger was increased by 514 cm^3 to a total volume space of 1440 cm^3. The shifting of the heat exchanger to the vertical position necessitated an increase in the ¾ connection volume by 160 cm^3 to a total volume space of 370 cm^3.

4.2.8c Transitions (entrance and exit losses)

Because their was significant frictional losses from the fluids entering and exiting different size pipe diameters, smoother transitions were machined out of ceramic material and placed at the entrance and exit locations to reduce losses due to friction.

Figure 4.2.8c Machined ceramic material used to smooth the transitions of the fluid in the pipe.

4.3 Data Acquisition System:

To be able to quantify and measure the system's performance as well as the various parameters, it has been equipped with pressure transducers, thermocouples and very soon, will incorporate anemometers. These will be used in conjunction with some data acquisition system in order to observe the engine's operation.

4.3.1 Pressure Transducers

A standard differential pressure transducer is to be used to monitor the engine pressure. The transducer is attached to the engine via the end cap in the dead volume block, at the hot end column. Another one will be fixed at the gas-filled couple where the pressure variations will lead to water being pumped. The output voltage of the transducer is proportional to the pressure difference between its two ports. The transducer will be powered by about 1.5mA power supply. Using a voltage-regulated power supply is recommended, since the transducer produces accurate data as a result of the fact that the voltage can always be set to the desired one. The pressure sensor pictured below was used to measure the pressure variations in the hot cylinder. The calibration that was used in converting the analog voltages into Pascal values was: 1mV = 1820 Pa.

Fig. 4.3.1: 316L Pressure Transducer

4.3.2 Thermocouples

The thermocouples are to be used to monitor the engines temperature. These highly sensitive thermocouples are attached to the system via the heat exchanger. They are also put into the dead spaces in the hot chamber. They rely on the voltage differential between the wires to provide a temperature reading. The calibration has been programmed into the reader to the extent that the output voltage, i.e., the voltage difference between the two wires translates directly to a temperature value.

4.3.3 Proximity Sensor

A distance measuring sensor was installed inside the hot cylinder to output analog voltages as the liquid oscillates. It has a dynamic range of 10 to 80cm. The output was amplified using a 741 Operational amplifier and has a gain of 10 v/v. A low-pass circuit was built to filter out high frequency noise. The proximity sensor gives us a sense of the amplitude of operation.

5. Results and Discussion

5.1 Initial Trial Run

While using the heating device from the hairdryer for the initial test run of the system, the maximum pressure experienced by the system was observed to be 8kPa. The pressure value was due to the manual attempt of jumpstarting the system, a process that is required, before free-oscillation is sustainable, for the operation of many stirling engines. Free oscillation is the stage during operation after the forces used to jumpstart the system has stopped. The initial test run did not sustain oscillation for a number of reasons, of which include leakage, friction, and heat transfer problems.

Figure 5.1 Picture of system; the red arrows show where leakage occurred; the white arrow show where frictional losses were observed.

The leakage of the moving fluid, in our case air, was observed to have occurred at the top and bottom of the heat exchanger and the top of the hot cylinder. The red arrows in figure 5.1 indicate the locations where air was observed to leak. The manner in which the leakage was detected was through hearing whistling sounds at the stated locations during the operation of the system. The leakage of air could be a result of the following:

1. large temperature changes in the heat exchanger – may have opened sealed holes in exchanger
2. pressure variations in the system – the forces due to the pressure could have created small openings at the connections
3. inadequate method and application of sealing material – because of the difficulty in determining where the leakages were some openings may have been omitted

Not only did the leakages contribute to the system's inability to sustain oscillation, the frictional losses were a second set of problems. The frictional losses in the system are mostly due to fluid flow. The system contains many entrances and exit losses at the transitions between:

1. displacer and the tuning column
2. the displacer and ¾ connections
3. the ¾ connections and the heat exchanger and regenerator

The locations of where the frictional losses took place in the system could be viewed in figure 5.1.

5.1.1 Pressure Variations

Furthermore, the heater used during the initial testing was taken from a hair dryer device. This heater consisted of two sets of coils that intermittently cut on and off during operation. The operation of the heater proved to be a problem because not enough input power for operation was achieved. Upon reaching a certain threshold temperature, an integrated safety switch would become operational and trigger the device to shot off. Again, the system requires a certain amount of power throughout the operation and the heater device did not prove to suit the application.

During the initial test run the peak pressure during free oscillation was observed to be 2.5kPa.

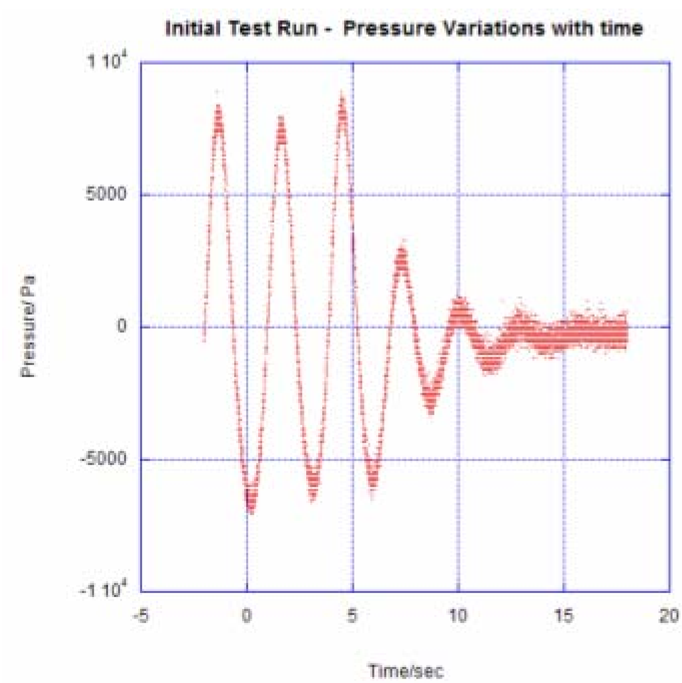


Figure 5.1.1 Using heat device from hair dryer the maximum pressure recorded was 8kPa and the maximum pressure during free oscillation was 2.5kPa; the decay time constant was 5 seconds.

The three initial peak pressure values in figure 5.1.1 are 8kPa and correspond to the forces used to jumpstart the system. During free oscillation the time constant was calculated to be approximately 5 seconds. The oscillations eventually decayed exponentially to zero soon after trying to jumpstart the system.

5.1.2 Temperature Variations in Hot Cylinder

When analyzing the temperature variations in the hot cylinder the maximum recorded temperature was found to be 37^{o}C. Similar to the peak pressure, the max temperature was recorded when the force used to jumpstart the system, which caused the max pressure of 8kPa, was performed.

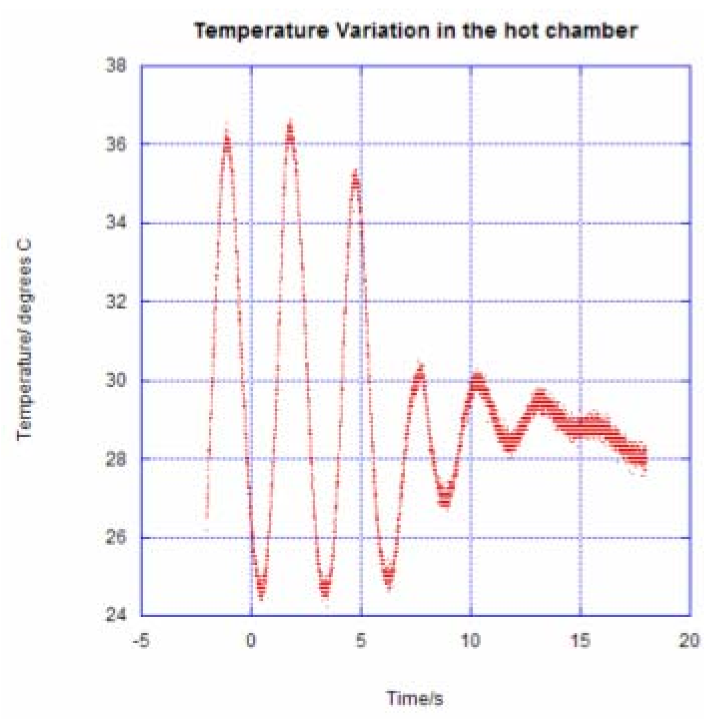


Figure 5.1.2 Graph of temperature variations inside hot cylinder; max temperature during free-oscillation was observed to be 30^{o}C.

The temperature ranged between 27^{o}C and 30^{o}C. The low temperature range of the hot cylinder shows the inefficient heat transfer between the heat exchanger and the hot cylinder. Although the thermo-couple in the heat exchanger did not work during the initial run we have assumed the temperature in the heat exchanger to have greatly exceeded 37^{o}C and thus the temperature range of the hot cylinder should have reached a maximum temperature far greater than the recorded temperature shown in figure 5.1.2. As stated earlier, the heater device inside the heat exchanger cut off when it reached a certain threshold temperature and may have contributed to the low heat transfer between the exchanger and the hot cylinder. The cutting off of the heater made the heat transfer highly inefficient.

5.1.3 Reducing Losses

To reduce the losses in the system as well as improve of the components that were not operational, we made a number of changes to the system.

1. To reduce the losses due to leakage of air a silicon rubber, room temperature vulcanite (RTV), was applied to the locations where leakages were observed to have occurred during the first operation of the system.
2. To reduce the frictional losses in the system ceramic material were used to smooth out the entrance and exits in the system making the transition of the fluids to be smoother.
3. Because the heater was inefficient it was replaced by another heater that was composed of a nichrome wire wrapped around a ceramic base and mounted in the heat exchanger.

5.2 Final Run

5.2.1 Pressure Variation

The changes to the system improved the operation performance and thus the results of the final testing illustrate the improvements. During the final run the pressure variations vs. time show peak amplitude of 7kPa, near 8kPa. This max pressure value is due to the input force needed to jumpstart the oscillations. But unlike the initial run, the free oscillation pressure variation was sustained for a significant period of time, see figure 5.2.1.

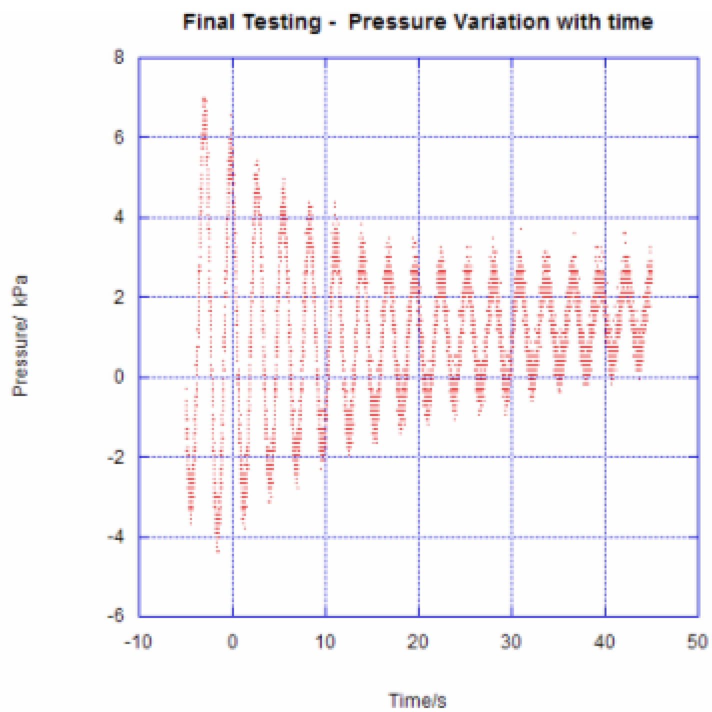


Figure 5.2.1 The maximum pressure during free-oscillation was found to be 2.5kPa; the oscillations decayed exponentially with a decay time constant of 40 seconds

The max pressure value during free-oscillation was recorded to be 2.5kPa and the decay time constant was observed to be 40secs. The reinforced sealing of the system using RTV reduced the amount of air leakage and thus allowing the pressure variations to be sustained for a longer period of time.

5.2.2 Power Output

To calculate the output power the maximum pressure during free-oscillation was taken and turned into a force by multiplying it by the area of the displacer:

$$F = P_{\max} A_d \qquad \text{Equation 5.2.1}$$

Since the pressure at any given time during the operation of the system is the pressure of the entire system the force from this pressure is the same force exerted on the liquid column in the tuning line. The velocity of the tuning line was recorded to be .63 m/s and multiplying this velocity by the force due to the pressure yields the output power of the system, see equation 5.2.1:

$$W_o = FV_t \qquad \text{Equation 5.2.2}$$

The maximum output power was calculated to be 23 watts. The oscillations of the system during the final run were sustained much longer than the oscillations of the initial test run.

5.2.3 Temperature Variation in Heat Exchanger

The temperature variations inside the heat exchanger illustrates that air is moving into and out of the component at different temperatures, see figure 5.2.3. During the final test run the temperatures of the heat exchanger ranged from 125 ^{o}C – 400 ^{o}C. The thermo-couple in the heat exchanger recorded the maximum temperature of 400 ^{o}C during the process of jumpstarting the oscillations.

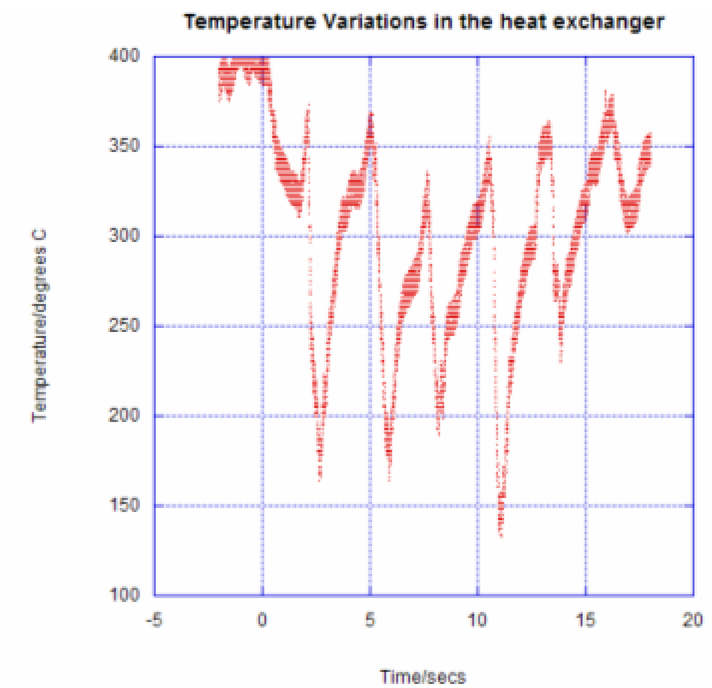


Figure 5.2.3 Heat exchanger temperatures varied between 125^{o}C and 400^{o}C.

During the period of free-oscillation the maximum temperature recorded was 375^{o}C and the minimum temperature recorded was observed to be 125^{o}C. The wide swing in the temperature inside the heat exchanger shows that the heater was effective in delivering the desired amount of power and thus the temperature range between 300 – 350^{o}C, target temperature range, was achieved.

While the system was oscillating the high temperatures experienced inside the heat exchanger caused significant deformation of the outer PVC casing. The thermal resistance of PVC was found to be 212^{o}C. To add further thermal protection for the PVC material a ceramic paper lining was laid on the inside wall of the heat exchanger. As the temperature of the heat exchanger approached the thermal resistance of the PVC material it began to become soft. This lining of ceramic paper helped in protecting the PVC casing from the extreme temperatures, but as the test progressed the PVC casing, nonetheless, began to deform.

The deformation of the heat exchanger may have affected the integrity of the system and may have contributed to the opening of seals that were reinforced with RTV and taping. Although the findings of figure 5.2.3 suggests that air was moving into and out of the exchanger at different temperatures, openings in the exchanger due to the high temperatures that the heater produced and the low thermal resistance of PVC could have contributed negatively to the pressure variations during operation.

5.2.4 Temperature Variation in Hot Cylinder

Similar to the initial test temperature measurements of the hot cylinder, the maximum temperature recorded during free-oscillation for the hot cylinder was found to be 37^{o}C. The low temperature variations of the hot cylinder, see figure 5.2.4, indicates that the heat transfer between the heat exchanger and the hot cylinder was poor. Although the exchanger reached the temperatures needed for sustainable oscillations the transfer of that heat to the hot cylinder was not achieved.

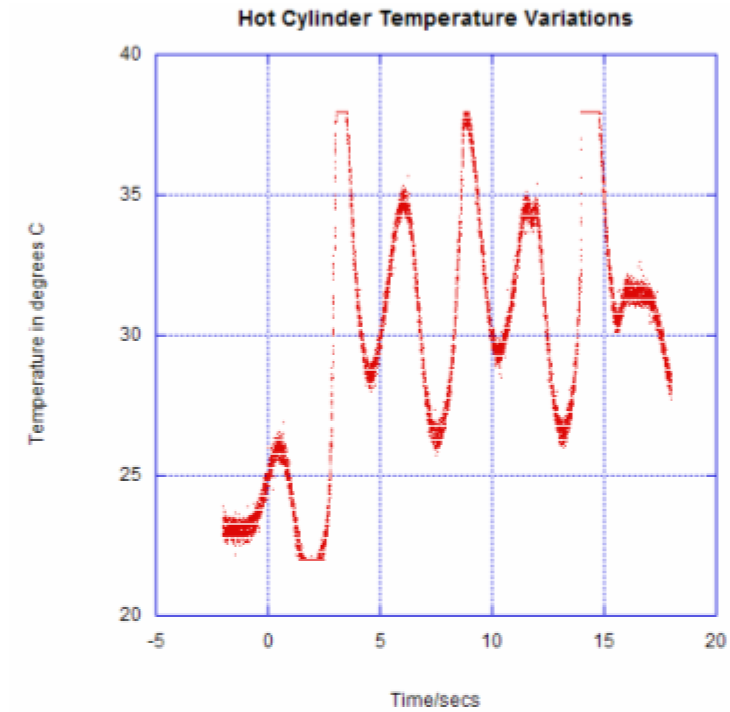


Figure 5.2.4 Temperature varied between 23°C and 37°C in the hot cylinder.

As air moved in and out of the hot cylinder temperature variations were recorded. The occurrence of temperature differences during free-oscillation in the hot cylinder shows that air at room temperature and warm air were entering the component. Because the temperatures entering the cylinder were not close to the temperatures in the heat exchanger, the air in the heat exchanger did not reach the maximum temperatures that the heater in the exchanger reached. In other words, the air circulating in the system did not reach the maximum temperatures of 300-350°C.

5.2.5 Efficiency Calculation

With an input power of 664 watts and an output power of 23 watts the efficiency of the system was computed to be 3.5% using the equation 5.2.5 below:

$$\eta = \frac{power - out}{power - in} \quad \text{Equation 5.2.5}$$

This efficiency corresponds to the efficiencies of stirling engines commonly found in literature, which ranges from 3 - 6 %.

The poor heat transfer between the heat exchanger and the hot cylinder may be due to a number of reasons, of which is the most probable cause for the inefficient heat transfer is the low dead space volume. By increasing the dead space volume the mass of air moving in the system could be increased; thereby, allowing more air to be heated by the heater at any given time in the system. The increased air volume would thus allow for the temperature increase in the hot cylinder. Thus, the heat transfer between the heat exchanger and the system would be improved. The dead space volume could be increased by increasing the diameter of the connections, from ¾ inch PVC piping to 1 or even 2 inch PVC piping.

Additionally, the transitional elements used to smooth the transition from the different diameter piping contributed to the improvement of the system operation. The transition elements, made from ceramic material, were placed at the entrance and exit between the displacer and tuning column and between the ¾ connections and the heat exchanger. More transition elements could have been placed between the displacer and the ¾ connection piping, which may further cut down the frictional losses in the system.

A foam float was placed inside the hot cylinder above the water level to minimize evaporation of the water. The foam helped in keeping the water from entering the connections and thus into the heat exchanger, a situation that would negatively effect the system operation because the water level in the system would decrease and increase the volume of the hot cylinder. The increase hot cylinder volume would decrease the system efficiency because the transfer of heat into the volume space was not ideal, as shown by the low temperature range of the hot cylinder, see figure 5.2.4.

Furthermore, the cold water bath on the cold side did not provide a proper heat sink for the space. Because PVC is a bad thermal conductor the heat entering the cold chamber was not removed and thus did not provide a more extreme temperature difference in the system. The greater the temperature differences in the liquid stirling engine the greater the efficiency of the system during operation.

5.2.6 Tuning Column

The data collected from the tuning column can be viewed in table 5.2.6. The flow in the column was observed to be laminar with a fluid velocity of .63 m/s.

Table 5.2.6 Tuning Column Parameters

Tuning Column Parameters	
Velocity	.63 m/s
Frequency	.4 Hz
Amplitude	.25 m
Reynolds Number	14200
Critical Reynolds Number	56000
Flow Type	Laminar

6. Conclusions

Given our project objectives we feel that the opportunity to work on this endeavor greatly enhanced our appreciation and awareness of the resources and technologies that we take for granted daily. Our goal of developing a liquid piston stirling engine that was capable of producing and output power of 5 watts or 3.5 % efficiency was somewhat realized. The design and materials used to construct the system were kept to a mechanical simplistic model to allow for better integration in developing societies. Although certain aspects of the system proved to be difficult problems the attempts to overcome them brought about many success and setbacks. Our interest and analysis of the liquid piston stirling engine may inspire future work on the system which would advance our last objective, which was to raise greater awareness of fluidyne engines as a low cost energy source for applications found in developing societies.

We have learned a great deal on the operation of the stirling engine, more specifically the liquid piston stirling engine. Hopefully, the experience gained working on this project will affect our futures in regards to advancing technologies that would benefit the most underprivileged our communities and societies. We also enjoyed working along side our advisor Prof. Carr Everbach, and technician Grant Smith. In addition, we learned a great deal from Prof. Fred Orthlieb, who was very instrumental in helping us acquire many of the materials and parts needed.

7. Further Work

To enhance the performance of the engine and improve on its efficiency, further improvements could be made to the existing engine. These improvements will compliment the successes of this project to make the

1. *Improvement of the current heat exchanger design*

   As explained in the results/discussion section of this report, the current heater does *not* protect against heat losses to the surroundings at. The vast temperature difference between the heat exchanger and the hot cylinder confirms this observation. Furthermore, during engine operation heat losses through the insulation – a quarter inch thick ceramic paper - which goes to deform the PVC tube can be clearly observed. This could be seen at the end of testing; the PVC tube had changed shape. The reason for this heat loss is that the insulation used is not thick enough to adequately prevent heat transfer to the PVC tube.

   There are a variety of ways to counter the inefficiency between the heat exchanger and the hot cylinder. To begin with, the focus of this project was not on heat exchanger design. The project's main focus was to design and develop a working liquid piston Stirling engine. It has however become apparent that the heat exchanger design requires a considerable amount of time and effort. One idea would be to replace the current PVC tube that contains the heater with an inexpensive material that is able to withstand the high temperatures. A second solution would be to use a much thicker and better insulation material which will prevent heat exchange between the hot air inside and its surroundings. Lastly the heat energy transfer between the heat exchanger and the hot cylinder should be improved. One way to achieve this is by possibly placing the heater insider the hot chamber. This will ensure that the air in the hot cylinder is heated directly. Alternatively, heat transfer could be improved by placing thermal fins to facilitate conduction.

2. *Experimentation with pressurization*

Equation (2.4.9) on p.20 of the report indicates that the net work per cycle of the engine is proportional to the mean pressure P_m. This means that a greater mean pressure will increase net work per cycle, and thus the engine power output. One simple way to increase the mean pressure would be to pressurize the engine. This could be done by applying an air compressor to a pressure port on the dead volume space to increase the pressure. If p_c is the pressure at the outlet to the compressor (inlet to the pressure port), and p_{mean_old} is the original mean pressure in the engine, then the compressor would act to offset pressure fluctuations in the engine by an amount equal to

$$p_{mean\_new} = p_{mean\_old} + p_c \, , \qquad (0.1)$$

where all pressures are *gage* pressures. The pressure increase in the engine would increase as well, increasing the work per cycle.

The main drawback of pressurization is leakage of the working gas (air in our case) due the greater effect of imperfect sealing at higher pressures. Sealing imperfections not evident currently may appear if a compressor is used to raise the mean pressure. In addition, caution should be exercised in pressurizing since both the PVC tubes could crack.

3. *Replacement of the proximity sensor with one of appropriate dynamic range*

The distance sensor was not that useful to us in this project since the amplitudes of oscillation were outside its dynamic range. It was therefore not possible to get an accurate figure for the amplitudes of oscillation inside the cylinder. A proximity sensor with a larger dynamic range will be much more useful in this regard.

4. *Addition of thermocouples inside the cold cylinder as well as velocity sensors in connecting tubes*

To better characterize the engine, additional sensors should be incorporated into the system. In addition to the thermocouples that have been placed in the heat exchanger and in the hot cylinder, one should be placed inside the cold cylinder to monitor the temperatures there. Secondly, to get an accurate sense of the speed of air flow in the system during each cycle, velocity sensors should be implemented inside the hot and cold chambers. This will be useful in providing the speed of the air flow in the system which is necessary for fluid dynamics' analysis. Knowledge of the speed of air flow will also be useful in determining kinetic flow losses in the engine.

5. *Use bigger diameter PVC tubes for connections*

One of the major decisions we had to make was what size PVC tubes to use in our connections. On the one hand a bigger size PVC for the connections would ensure that a greater mass of air was flowing between the hot and cold chambers. This would generate large pressure variations which means that, on the hot side for instance, the high pressure would lead to greater forces on the oscillating liquid and therefore greater amplitudes in oscillations. On the other-hand, smaller PVC tubes for the connections would minimize the dead space and generate a favourable compression ratio. Our design choice was the latter. However, as it was observed during testing that the oscillations could not be sustained permanently coupled with the low efficiencies, the ¾ inch PVC tubes may have to be replaced with bigger size diameter tubes, 2 inch for example. Even though they contribute to the dead volume in the design, the bigger sized tubes contain a greater mass of air, needed to provide the force to drive the liquid pistons.

6. *Incorporate the pumping line into the system*

The pumping arm should be incorporated into the design. The original goal was to be able to pump water over a height of about 10 feet. To successfully incorporate the pumping line and measure performance, the engine design must be improved upon to function more efficiently.

7. *Fulfill primary project goal*

The end product for this project should be a solar-powered water pump, with the water that has been pumped, heated with focused sunlight. It is our hope that this objective will be pursued in the near future to ensure that all the goals/objectives of this project are met.

8. References

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