THE UNIVERSITY OF CALGARY

Kolin-Type Stirling Refrigeration

by

Joseph Mathews

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ABSTRACT

The Stirling engine is a closed-cycle regenerative thermal machine that operates by the alternating compression and expansion of a fluid between hot and cold temperature reservoirs. Similar to all other thermodynamic engine cycles, operation of the Stirling engine in reverse results in a refrigeration machine. The Kolin Stirling machine is a special variant of the Stirling machine that is able to operate with very low temperature differences as compared to other types of Stirling machines. It is a simply-constructed machine that, unlike other Stirling machines, is able to operate closer to the ideal Stirling cycle due to its mechanical configuration. This thesis discusses the feasibility of using the Kolin Stirling machine as a self-contained refrigerator. Unfortunately, results from basic tests conducted on a Kolin engine running in reverse do not encourage the use of Kolin-type machines as refrigeration devices without further development of the relevant technology.

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To my parents,

For all their love and advice.

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Symbols List

A - area of Styrofoam shell for use in Fourier's equation

k - thermal conductivity

N - speed

p - pressure

p_{max} - maximum pressure of working fluid

p_{mean} - mean pressure of working fluid

P - power

Q - heat energy

Q_E - heat energy lifted from working space

S - entropy

T - temperature

T_e - temperature of expansion space (equal to T_{max} in ideal engine cycle, T_{min} in ideal refrigeration cycle)

T_f - final recorded temperature

T_c - temperature of compression space (equal to T_{min} in ideal engine cycle, T_{max} in ideal refrigeration cycle)

T_{max} - maximum temperature of cycle

T_{min} - minimum temperature of cycle

T_o - initial recorded temperature (equivalent to ambient temperature)

v - volume

V_d - displacer swept volume

$\dot{V}_p$ - piston swept volume

V_D - total dead space volume

V_T - total swept volume (alpha configuration)

W - work energy

α - phase angle - phase difference between piston and displacer motions

Δx - thickness of material for Fourier's equation

η - efficiency

κ - swept volume ratio ($= V_p/V_d$)

τ - temperature ratio ($= T_{min}/T_{max}$)

χ - dead-space ratio ($= V_D/V_d$)

"All that is gold does not glitter,

Not all those who wander are lost."

-- J.R.R. Tolkien

The Fellowship of the Ring

CHAPTER 1

1. INTRODUCTION

1.1 THE STIRLING ENGINE

The Stirling engine is a closed-cycle regenerative thermal machine that operates by the alternative compression and expansion of a fluid between hot and cold temperature reservoirs. In the ideal cycle, there are four distinct parts to the cycle: constant temperature compression, constant volume heating, constant temperature expansion, and constant volume cooling. This ideal cycle in terms of efficiency, is thermodynamically equivalent to the Carnot-engine cycle, which, although physically impractical to construct, is representative of the maximum thermal efficiency theoretically possible.

Actual Stirling engines have been in operation since the Stirling cycle was invented by Robert Stirling in 1816. Such engines do not operate on the ideal Stirling cycle however. Due to mechanical considerations and physical characteristics, these machines operate on a non-ideal cycle, resulting in a thermodynamic efficiency far below that of the ideal cycle.

As is the case with all thermodynamic engine cycles, running the Stirling engine cycle in reverse results in a refrigeration cycle. An actual, operating, Stirling refrigeration machine similarly does not operate on the ideal Stirling refrigeration cycle, but rather on a cycle with much less efficiency.

1.2 THE KOLIN STIRLING MACHINE

The Kolin Stirling machine, which was invented by Dr. Ivo Kolin of Zagreb University, in Zagreb, Croatia, during the 1970's and 80's, was an attempt at building a Stirling machine able to operate on a cycle that more closely resembles the ideal Stirling cycle. This simply-built machine, as shown in figure 1.1, had two seemingly advantageous characteristics over other Stirling machines of the time — flat plate heat exchangers, which reduced overall dead-space within the machine, and discontinuous displacer motion, which resulted in the constant-volume heating and cooling portions of the operating cycle more closely resembling those of the ideal cycle.

Due to the increase in efficiency that these changes brought to the engine, Kolin was able to operate this machine at temperature differences smaller than hitherto possible. He proposed both engine and refrigeration machines working on similar principles (Kolin,

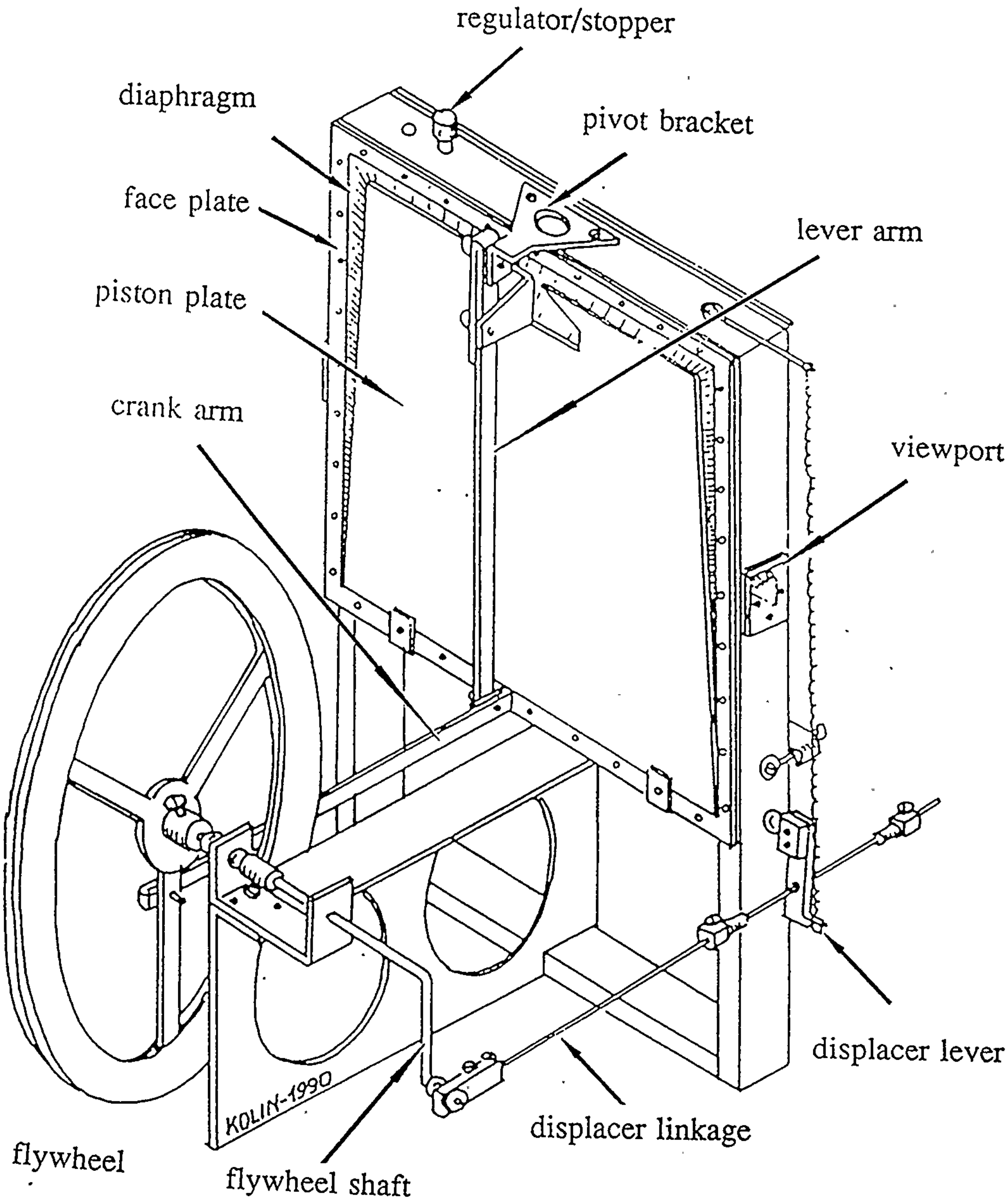

Figure 1.1 The Kolin Stirling Machine (Kolin, 1990)

1984; Kolin 1986; and Kolin 1988) but concentrated mostly on the development of working engine models.

1.3 DECLARATION OF OBJECTIVES

1.3.1 Thesis Objectives

The objective of this thesis is to report on the possibility of developing a simple, self-contained refrigerator, consisting of power, refrigeration, and control units. The suitability of the Kolin-Stirling machine as the power and refrigeration units is determined, and possible control systems are discussed. The results of the experimental portion are also presented and discussed.

1.3.2 Experiment Objectives

Although the overall goal is to develop a self-contained refrigerator, the experimental portion of the project concerns the feasibility of converting the Kolin-Stirling engine to a refrigeration machine. Basic tests were performed, and from the results, modifications in design for improved usability are suggested.

1.4 WHAT FOLLOWS

In the following chapters, a more detailed look into the principles and characteristics of Stirling machines, with particular emphasis on the Kolin Stirling machine, is presented, along with a perspective of other work in the Stirling refrigeration field. The overall self-contained refrigerator concept is discussed, and the results of the experiments on the refrigeration unit are presented and discussed.

1.4.1 Chapter 2

In chapter 2, following a brief presentation on the history of Stirling machines, a review of the previous work done in Stirling refrigeration systems is presented. The need for a self-contained, simple refrigerator is discussed through a presentation of some possible applications of the Kolin Refrigerator.

1.4.2 Chapter 3

Chapter 3 presents a more detailed description of the Kolin Stirling machine including some associated principles of thermodynamics. The ideal Stirling cycle is presented as well as representations of actual cycles. The theory of the flat-plate, discontinuous motion Stirling machine is presented as well as optimization concerns.

1.4.3 Chapter 4

In Chapter 4, the overall concept of the Kolin Refrigerator is presented. A brief introduction to the proposed system is given, with particular emphasis on the three main components: the power unit, the refrigeration unit, and the control system.

1.4.4 Chapter 5

Chapter 5 describes the experiments conducted on the utility of the Kolin Stirling machine as the refrigeration unit. The set-up of the experiments is described as well as the different variables tested, changes made to the system, and method of operation.

1.4.5 Chapter 6

In Chapter 6, the results of the experiments are presented along with a discussion of these results and suggestions for modifications and improvement to the Kolin-Stirling refrigeration unit.

1.4.6 Chapter 7

Finally, the conclusions are presented in chapter 7.

CHAPTER 2

2. A BACKGROUND TO STIRLING MACHINES

2.1 BRIEF HISTORY OF STIRLING ENGINES

The Stirling engine was invented in Scotland in 1816, by Robert Stirling, who continued to develop the engine over many years with his brother, James (Walker, 1980). The engine was the first to use the regenerative heat exchanger concept, in which heat is absorbed from the working fluid when passing through the regenerator in one direction, stored for a short duration, and then returned to the fluid when passing through in the opposite direction.

These "hot-air" engines, as they were called due to the use of air as the working fluid (as opposed to steam in steam engines), were primarily built for small scale uses throughout the 1800's although larger scale engines were also built. Major uses of the engine at the turn of the century included operation of ventilating fans and water pumps.

Work on these engines faded during the early 1900's when steam engines started to dominate large scale systems, internal combustion engines mid-scale systems, and electric motors small scale systems. Research in Stirling engines arose again during the mid 1930's, when the Philips Company of Eindhoven, the Netherlands, began to develop a small scale engine for use in portable electric power generators for radio-sets.

Since then, and particularly since the 1950's, work on Stirling engines has progressed in both small and large scale systems throughout much of Europe and North America.

2.2 STIRLING REFRIGERATION HISTORY

Thermodynamically reversing an engine or power cycle has been recognized as a refrigeration cycle since the early 19th century from the work of Sadi Carnot (Reay and Macmichael, 1988). Lord Kelvin first proposed and advocated the use of heat pumps in the mid 19th century, noting that a refrigeration machine could also be used effectively as a heat pump (Kestin, 1966). A refrigeration process resulting from reversing a Stirling cycle was first recognized by John Herschel in 1834, and an actual machine was described by Alexander Kirk in 1876 (Walker, 1983).

The Philips Company at Eindhoven began developing Stirling refrigerators in the 1940s for operation at cryogenic temperatures. Most of the work on Stirling refrigerators since then has been on machines operating at cryogenic-temperatures, which Walker (1983) has described in detail.

Only more recently, particularly in the last ten-fifteen years, has there been effort placed on the development of Stirling refrigerators operating at "near-atmospheric-ambient" temperatures.

2.3 REVIEW OF STIRLING REFRIGERATION WORK

Though not a lot of work has been done on ambient temperature refrigerators *per se*, a great deal of work as been put forward in the field of Stirling Heat pump systems in the last 10 years. The heat pump is essentially the same thermodynamically as the refrigerator, except for the temperatures of the heat source and sink — the heat source is at the ambient (atmospheric) temperature in the heat pump as opposed to the heat sink in the refrigerator. Applications of heat pumps range from hot water heaters (Pritchard *et al.*, 1991) to waste heat recovery in industrial plants (Chan *et al.*, 1984) to residential and commercial building heating and cooling (Ross *et al.*, 1987). Particular emphasis has been placed on natural gas fired heat pumps (Monahan, 1988), particularly due to the abundance of natural gas supplies in North America (Brodrick and Patel, 1990). Stirling based heat pumps are expected to mature in the 1990's (Spauschus, 1988).

Ross *et al.* (1987) noted that a thermally activated heat pump, *ie.*, a heat pump driven by a heat-engine, would have advantages over electrically-driven ones. These advantages include higher efficiency when in the heating mode and reduced effect of lower temperature heat sources. For Stirling-Stirling heat pumps, additional advantages would include quiet operation, simpler configurations (with free-piston engines), ability to burn a variety of fuels, and better adaptation to changing loads without complicated control mechanisms.

In addition to systems designed specifically for ambient temperature refrigeration, some work has been done on converting Stirling cryocoolers to higher temperature operation (Chen, F.C. *et al.* 1988). With some modifications, such a system was predicted to be viable compared to vapour-compression technology.

The three main areas of current Stirling refrigeration/heat pump work are described below.

2.3.1 Free Piston Heat Pumps

One promising Stirling heat pump design is the free-piston heat pump which is similar to the Ringbom[1] engine. Ackermann *et al.* (1986) have developed a natural gas heat pump for use in residential environments, using up to 50% less natural gas than high-efficiency (90%) furnaces. The heat pump uses a hydraulic coupling between the engine and the compressor, and metallic diaphragms for hermetic seals. Displacer control for optimization has resulted in the heat pump being tested successfully over ambient temperatures between -17.8 and 35 °C. This system has been independently evaluated (Ackermann and Privon, 1988) and compared favourably to electrically powered heat pumps.

---

[1]A Ringbom engine is a beta configuration Stirling engine having a mechanically linked compression piston and a free displacer. The displacer's motion results from a combination of its weight and pressure differences across its top and bottom surfaces.

Penswick and Urieli (1984) describe Sunpower's earlier developments of free-piston, free-displacer Stirling engine powered - Stirling heat pumps, with essentially only three moving parts. These easily started, stable operating systems have been demonstrated in various forms with capacities between 10 and 50 kW.

2.3.2 Magnetically Linked Heat Pumps

Another important area in Stirling refrigeration work (and particularly in the last few years) has been magnetically linked Stirling heat pumps. Magnetically coupled systems have many of the advantages of free piston systems, particularly in the area of avoiding many of the seal problems associated with other types of Stirling machines. In a magnetically linked heat pump, it is possible to hermetically seal separately, the working fluids within the engine and refrigeration units with the use of magnetic couplings between the units. Chen and Beale (1988 and 1989) describe the development of such systems.

2.3.3 Vuilleumier Heat Pumps

A Vuilleumier machine is similar to a directly coupled combination Stirling power-refrigeration machine. The cycle, which is named after its developer, Rudolf Vuilleumier operates with the enclosed working fluid passing through three distinct temperatures. Rather than having two separate working fluids as in a Stirling-Stirling configuration, the Vuilleumier passes the working fluid through both the power and refrigeration portions of the cycle.

Much work has been done recently (within the last five years) on Vuilleumier heat pumps. Colasurdo and Pastrone (1991), and Carlsen and Andersen (1989) have provided detailed simulation analysis of the cycle including an accounting for some major losses.

Carlsen (1989), Okamoto *et al.* (1991), and Kuhl and Schulz (1990) have reported on the experimental results of actual Vuilleumier heat pumps. The major conclusions from these test were that heat output was only about 25% of the output of similar Stirling engine driven heat pumps, resulting in a much larger unit as compared to a Stirling based heat pump.

Kolin (1988) has developed a variation of his Low Temperature Difference Stirling engine to operate as a Vuilleumier engine. Results of the performance of the engine are not however available.

2.3.4 Other Stirling Refrigeration Technology

Most other work in Stirling refrigeration technology has occurred in cryogenic-temperature range machines. Large scale Stirling refrigerators capable of cryogenic temperatures were developed by the Philips company in Eindhoven, the Netherlands, in 1954, and have been used since then primarily for gas liquefaction. Much effort has been placed on the development of small-scale refrigerators for use in infra-red night vision and missile guidance systems (Walker *et al.*, 1988). Walker *et al.* (1988) also discuss other possible uses of such refrigerators, including use in superconductivity and cold semiconductor electronics.

2.4 LOW-ΔT REFRIGERATION: POSSIBLE APPLICATIONS

The Stirling refrigeration and heat pump systems described above are for the most part rather complex and expensive. In addition to the conventional refrigeration applications that these systems would be suitable for, a simple, inexpensive system would have many other possible applications. A refrigeration system based on the Kolin-Stirling machine could be such a system — wherever a small-scale refrigeration system is required, a Kolin Stirling refrigerator could possibly be used. The many advantages of the Kolin Stirling refrigerator as compared to other refrigeration systems, namely its low power requirements, portability, wide choice of power sources, and use of air as the refrigerant, further defines applications.

2.4.1 Relief Agencies

One potentially major application for a Kolin-Stirling refrigerator would be for relief agencies working in third world countries. Typically in these situations where vaccines and other medicines need to be kept cool to maintain their potency (Cowdery *et al.*, 1976), there is no reliable access to electricity without the use of an electric generator. Portability is important — at times a relief agency has to move to remote locations in difficult terrain, to where the afflicted people are. This was the case during the Kurdish refugee crisis after the Gulf War in the early months of 1991.

To operate an electric generator in remote locations requires not only the hauling of the generator to the site but the fuel also. This could become cumbersome depending on the location of the relief site — particularly in mountainous terrain.

The fact that the Kolin Stirling refrigerator does not need a specific power source for the refrigerator means that it can be taken almost anywhere without much difficulty in providing fuel. Whatever fuel is available at the location can be used as the heat source. This portability would be an asset in terms of being able to set up equipment quickly upon arrival, as well as being able to be moved hastily upon need.

For the most part, fuel that is required under relief conditions are usually plentiful in supply. This also includes:

- biomass combustion — grain husks, scrap wood, food/medicine containers, etc.,
- waste heat — from sterilization equipment (especially in medical camps), cooking fires, etc., or
- solar energy — which for the most part is widely available in the third world countries in which the refrigerator would used.

A small, pressurized source of, say, butane or propane could be used as the fuel source during transportation — switching to the site-available fuel upon reaching the destination.

Another factor in the Kolin Stirling refrigerator being useful for relief agencies is that of costs. Along with its ability to use available heat sources, the machine itself would not be expensive to purchase due to its low material and construction costs. Also, the simplicity of the machine means that maintenance and repair costs could be kept low, without major technical assistance being required.

2.4.2 Domestic Refrigerators

Another major application that the Kolin-Stirling engine would be suitable for is as a domestic or household refrigerator in developing countries. This is particularly true in countries where reliable access to electricity does not exist.

In many countries, electricity is cut off from households during the daytime so that the factories can be fully powered and operational. This means that the perishable foods kept in an electric fridge in the houses have to be quickly consumed, effectively defeating the purpose of the refrigerator. Because the Kolin-Stirling engine does not have to rely on electricity, other fuels can be used during the daytime to keep the refrigerator functioning.

The Kolin-Stirling refrigerator would be an ideal refrigerator especially for low-income households. Presently, refrigerators are classified as a luxury item in many countries, and only the "well-to-do" have one. Again, because fuel for the Kolin-Stirling refrigerator can be waste fuel, operating the refrigerator would not have to be expensive either.

The fact that the working fluid in the Kolin-Stirling refrigerator is air would have environmental benefits on a large scale. For example, providing conventional refrigerators to many of the billions of people in the world without one now would mean the use of large quantities of chloro-fluoro-carbons. This chemical is suspected to be a major cause of atmospheric ozone depletion when released into the atmosphere, primarily during construction and disposal of the refrigerators. (Bateman and Bivens, 1988; Ross 1989)

2.4.3 Natural-Gas Fired Air-Conditioning

Refrigeration applications requiring more sophistication than those described above may also be able to use the Kolin system. Many home in North America are using natural gas as a heating fuel. Natural gas in North America is both plentiful and inexpensive when compared to other fuels (Brodrick and Patel, 1990). There is a large consumption of fuel during the winter heating season, but not nearly as great during the summer. In many areas, air conditioners using relatively expensive electrical power are used during the summer months. A Kolin Stirling refrigerator, powered by a Kolin Stirling engine using natural gas combustion as a heat source, could be used as an air conditioner, or as a heat pump during the winter months.

2.4.4 Automobile Air-Conditioning

In an automobile, concerns over the use of CFC's are greater due to the greater likelihood of refrigerant leakage through glands and cracked hosing, in addition to losses during construction and disposal. Replacing these air conditioners with Kolin Stirling refrigerators would help to alleviate such concerns.

Excess heat generation from the automobile engine could be the fuel source for the system. Alternatively, an electric motor could be used to power the refrigeration unit, similar to that of an existing automobile air-conditioner, if providing a heat sink within the already-hot engine compartment is difficult.

2.4.5 Personal Cooling

A more esoteric application for the Kolin Stirling refrigerator would be for personal cooling (Walker, 1990). This would be particularly useful for those who have to work in full-cover environmental suits (eg. hazardous waste clean-up personnel). A small battery or butane powered fuel source could be used to run the refrigerator attached to the person's suit.

CHAPTER 3

3. THE KOLIN STIRLING ENGINE

3.1 INTRODUCTION

The Kolin Stirling engine is a low-ΔT Stirling engine developed by Dr. Ivo Kolin during the late 1970's and 1980's. Named the "Flat Plate Stirling Engine" for the use of flat plate heat exchangers, the engine has gone through several design changes during its development (Kolin, 1984; Kolin 1986; Kolin, 1990). The design changes have been made mostly to simplify the mechanisms involved in its operation.

The Kolin machine is a beta-type Stirling machine as opposed to an alpha-type (the type used most commonly for analysis of the ideal Stirling cycle) and the gamma-type (the most common configuration). The difference between the types are with respect to expansion and compression cylinder configurations. The three types are show diagrammatically in figure 3.1. The alpha-type machine has two pistons in two distinct working volumes, the beta-type has one piston and one displacer (similar motion as the piston, but does not appreciably change the volume of working fluid), both in one working volume, and the gamma-type has one piston and one displacer in two distinct volumes.

The main point of attraction to the Kolin Stirling engine is the use of flat plate heat exchangers and discontinuous displacer motion resulting in an ability to operate at very low temperature differences when compared to other Stirling engines of the time.

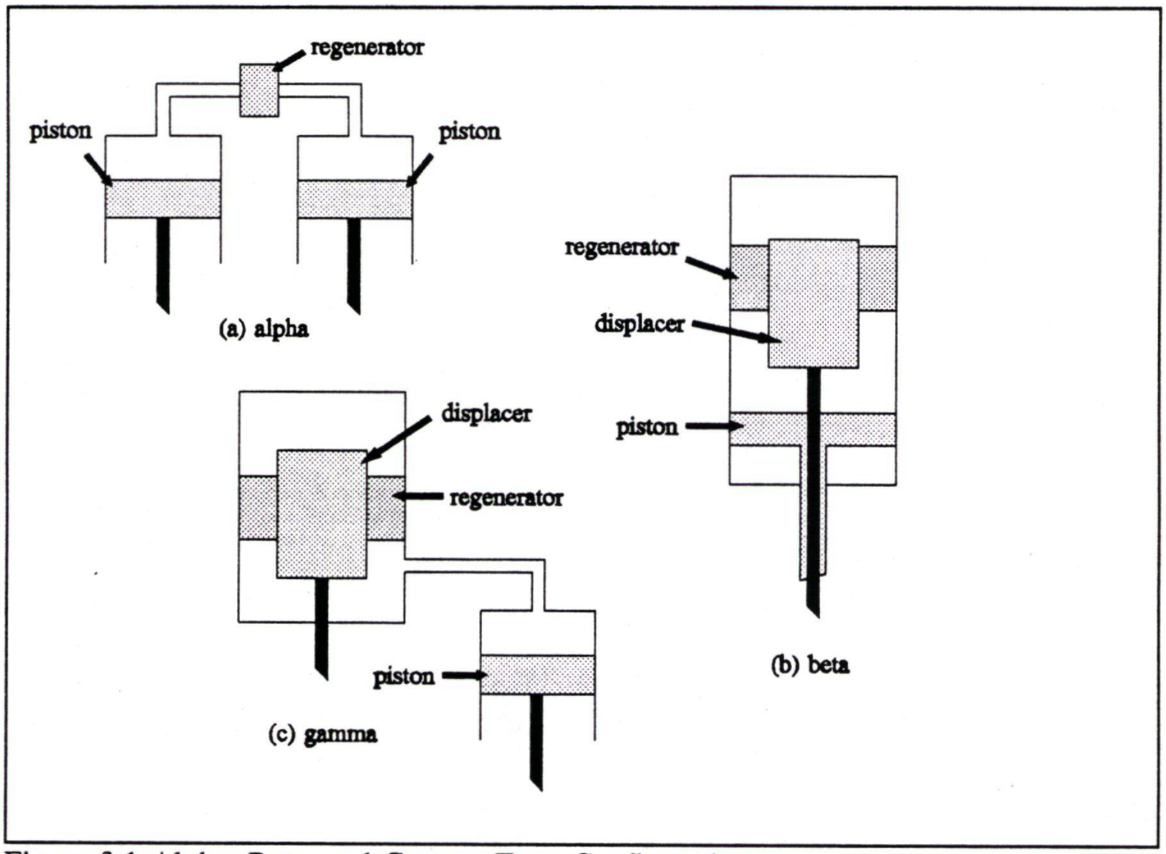

Figure 3.1 Alpha, Beta, and Gamma Type Configurations

Unlike other Stirling engines the components for the Kolin Stirling engine are very simple, easily constructed parts. Along with its ability to operate on low temperature differences, the simple nature of the system opens up a variety of application possibilities that before were not usually associated with Stirling engines.

In this chapter a closer look at the theory of discontinuous motion as demonstrated in the Kolin Stirling engine is discussed and is followed by a detailed description of the Kolin Stirling engine which was modified for use in the experimental tests of the refrigeration system. A description of some of the many possible applications of Kolin Stirling refrigerators follows.

3.2 BASIC THERMODYNAMICS

3.2.1 Thermal Efficiency and the Carnot Cycle

Thermal efficiency is defined as the ratio of work output to energy supplied:

$$\eta = \frac{W}{Q} \tag{3.1}$$

Work output of an engine's thermodynamic cycle is represented, for reversible processes, by the enclosed area on a p-v diagram, while energy supplied is represented by the enclosed area on a T-S diagram.

Applying the first law of thermodynamics, $\Sigma W = \Sigma Q$, and the definition of entropy for a reversible process (Wood, 1982)

$$dS=\left(\frac{dQ}{T}\right) \tag{3.2}$$

to equation 3.1 allows the thermal efficiency to presented as:

$$\eta = \frac{T_e - T_c}{T_e} \tag{3.3}$$

This is commonly referred to as the Carnot efficiency of the cycle. It states that the maximum theoretical thermal efficiency is dependent only upon the maximum and minimum temperatures of the cycle.

The Carnot cycle, as shown in figure 3.2, is a special cycle thermodynamically. As Walker (1980) states clearly, from equation 3.3 and the T-S diagram, no possible sequence of processes could result in a larger ratio of the area a-b-c-d to c-d-e-f, and so the ratio of these areas represents the maximum efficiency possible for a machine operating between T_{max} and T_{min}.

In practice, an engine based on the Carnot cycle is not able to be built, primarily because of the very small amount of work that is able to be produced without the use of very high pressures and very long piston strokes. An engine capable of handling high pressures and having long piston strokes would be so big that the work produced would not be able to overcome friction losses (Walker, 1980).

3.2.2 The Stirling Cycle

As mentioned in Chapter 1, the ideal Stirling cycle's four stages of operation are constant temperature compression, constant volume compression (through heating),

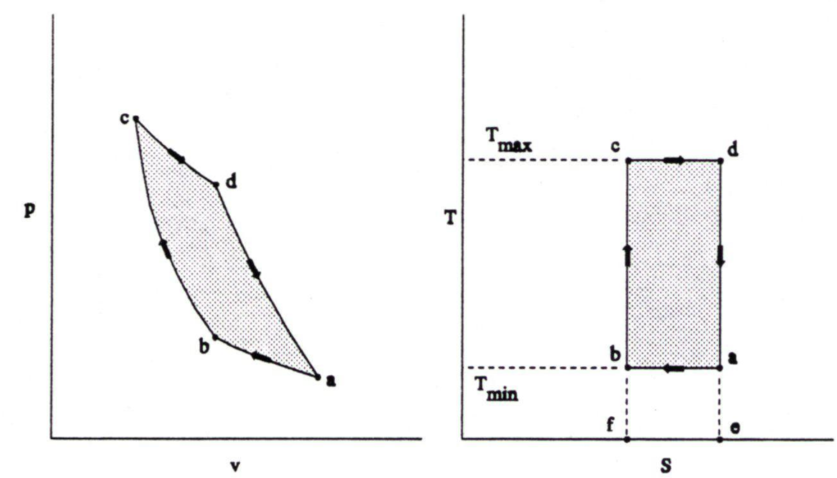

Figure 3.2 The Carnot Cycle (Walker, 1980)

constant temperature expansion, and constant volume expansion (through cooling). A representation of an ideal Stirling cycle is shown in figure 3.3 on both a pressure-volume and a temperature-entropy diagram. Figure 3.4 shows each process with the motions of the piston and displacer in an idealised Stirling engine of alpha configuration. Figure 3.5 shows the processes in a beta configuration machine. In the ideal cycle, the piston and displacer are assumed to move in a discontinuous motion between the positions shown in figure 3.4.

In part (a) of figures 3.4 and 3.5, the working fluid is entirely in the compression space, and is at T_{min} and minimum pressure. As the right-side piston moves to position (b), pressure increase due to compression, but temperature remains constant by rejecting heat to an external sink. The motion (b) to (c) involves both pistons moving simultaneously, so that the volume of the working fluid remains constant. As the working

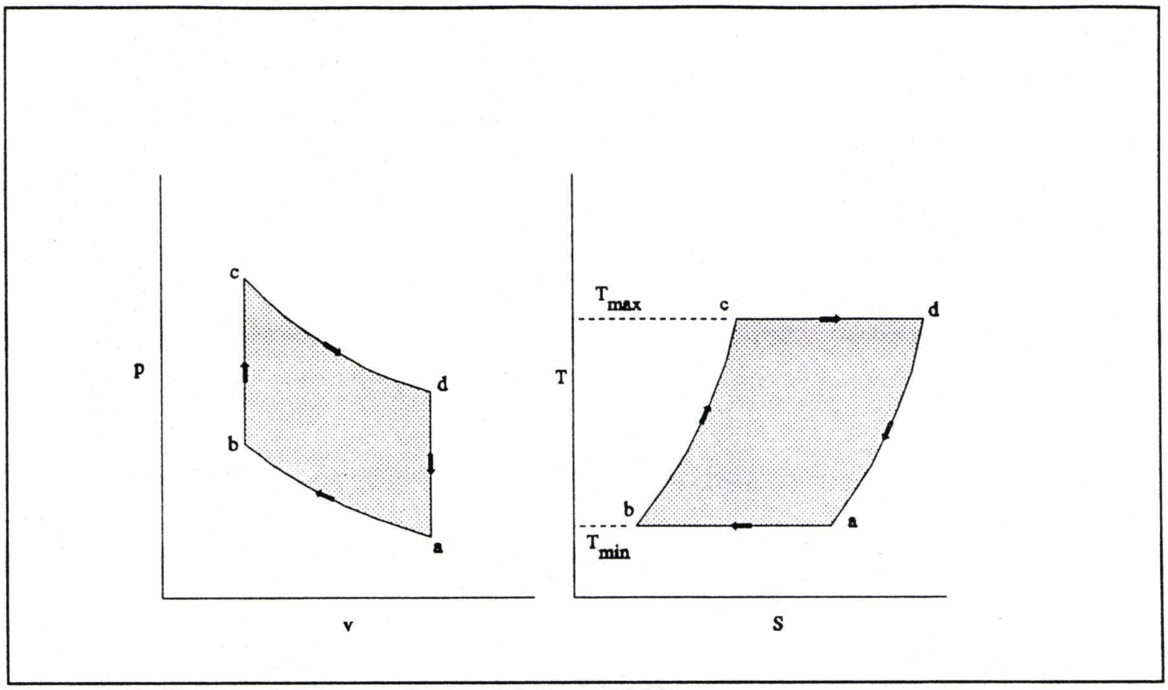

Figure 3.3 The Ideal Stirling Cycle (Walker, 1980)

fluid passes through the regenerator, heat energy is supplied to the fluid from the regenerator, increasing pressure and temperature to T_{max}. The left-side piston then continues to move to position (d), decreasing the pressure of the fluid. Temperature is kept constant by adding heat from an external source. Finally, the motion (d) to (a) involves both pistons moving simultaneously. The volume of the working fluid remains constant, and as the fluid passes through the regenerator, heat is rejected to the regenerator.

Comparing the Stirling cycle with the Carnot cycle, as in figure 3.6, shows that the Stirling cycle is able to operate at the Carnot efficiency because the maximum and minimum temperatures are each constant. However, a larger work output is produced because the amount of heat supplied to and rejected from the cycle is larger for a given T_{max} and T_{min}.

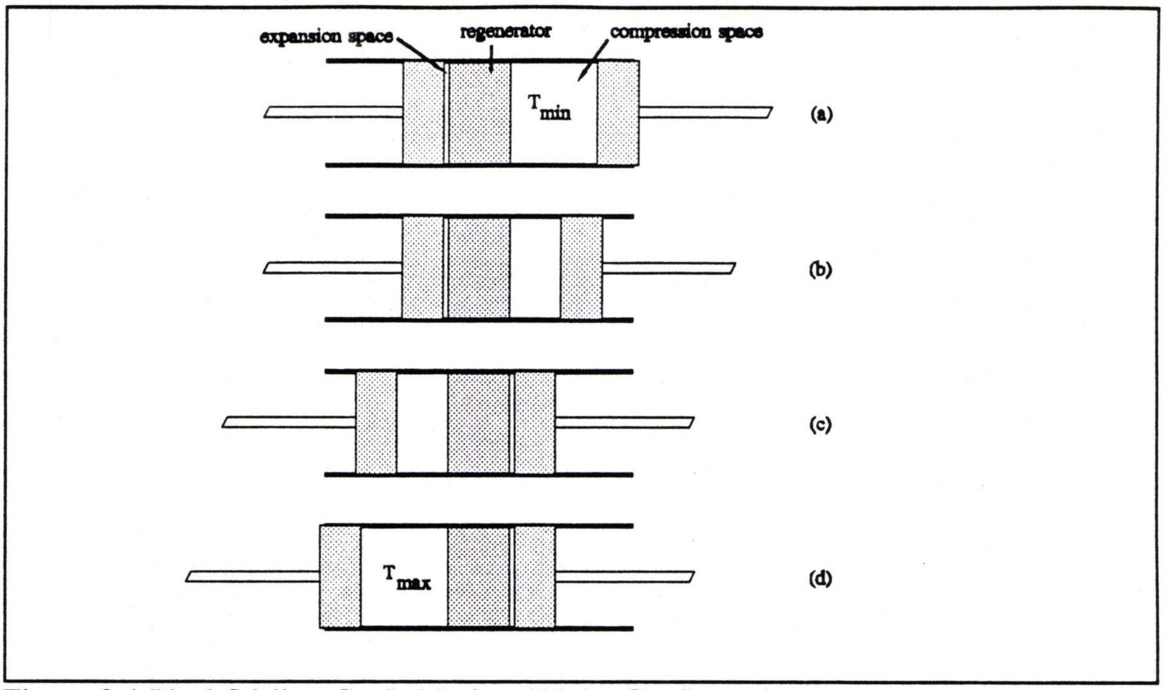

Figure 3.4 Ideal Stirling Cycle Motion (Alpha Configuration)

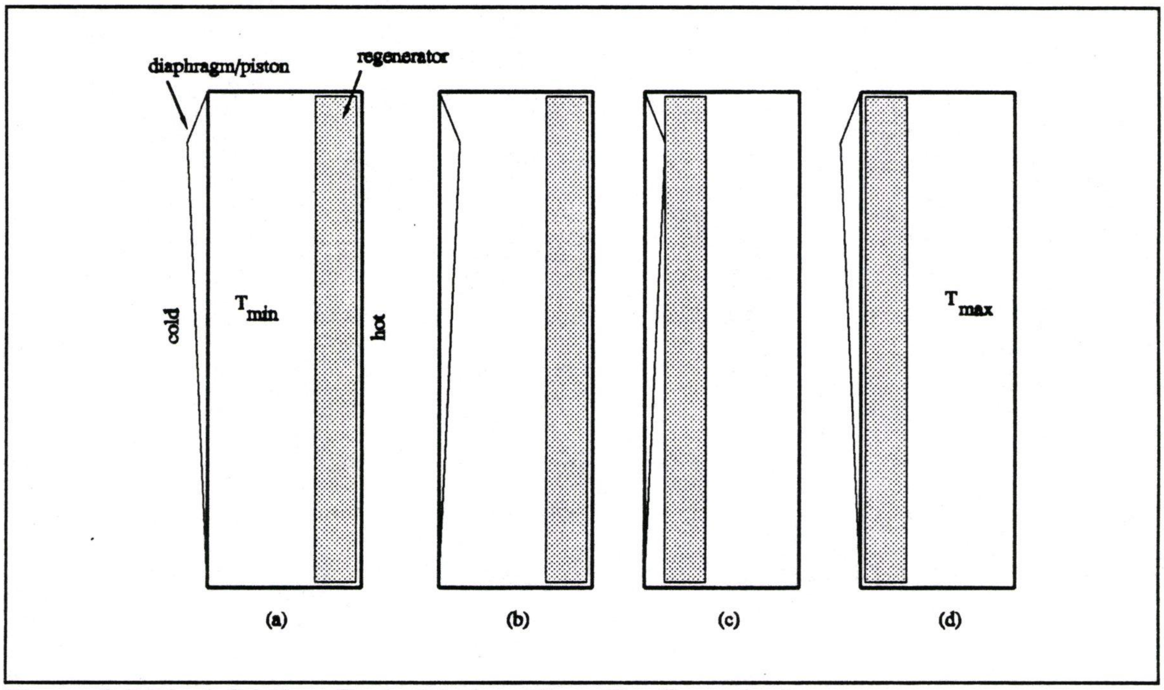

Figure 3.5 Ideal Stirling Cycle Motion (Beta Configuration)

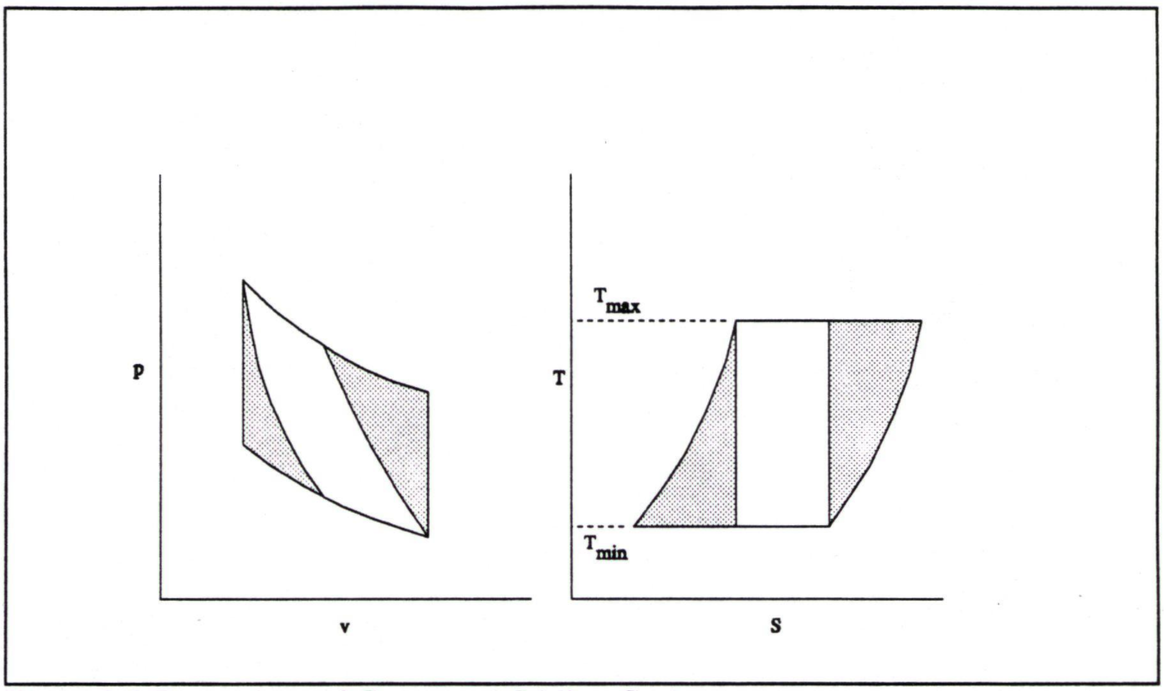

Figure 3.6 Comparison of Carnot and Stirling Cycles

Similar to the ideal Carnot cycle, the ideal Stirling cycle, also assumes infinite heat transfer rates during heating and cooling, zero external heat transfer during constant volume compression and expansion, ideal regeneration effects, and no mechanical and aerodynamical losses.

3.2.3 Refrigeration Cycles

As mentioned earlier, reversing an engine, or power, cycle results in a refrigeration cycle. An ideal Stirling refrigeration cycle is shown in figure 3.7. The processes in the refrigeration cycle are the same as the power cycle, except for the temperatures at which energy is supplied and rejected. Energy is supplied, during isothermal expansion, at a temperature lower than the temperature at which energy is rejected during isothermal compression.

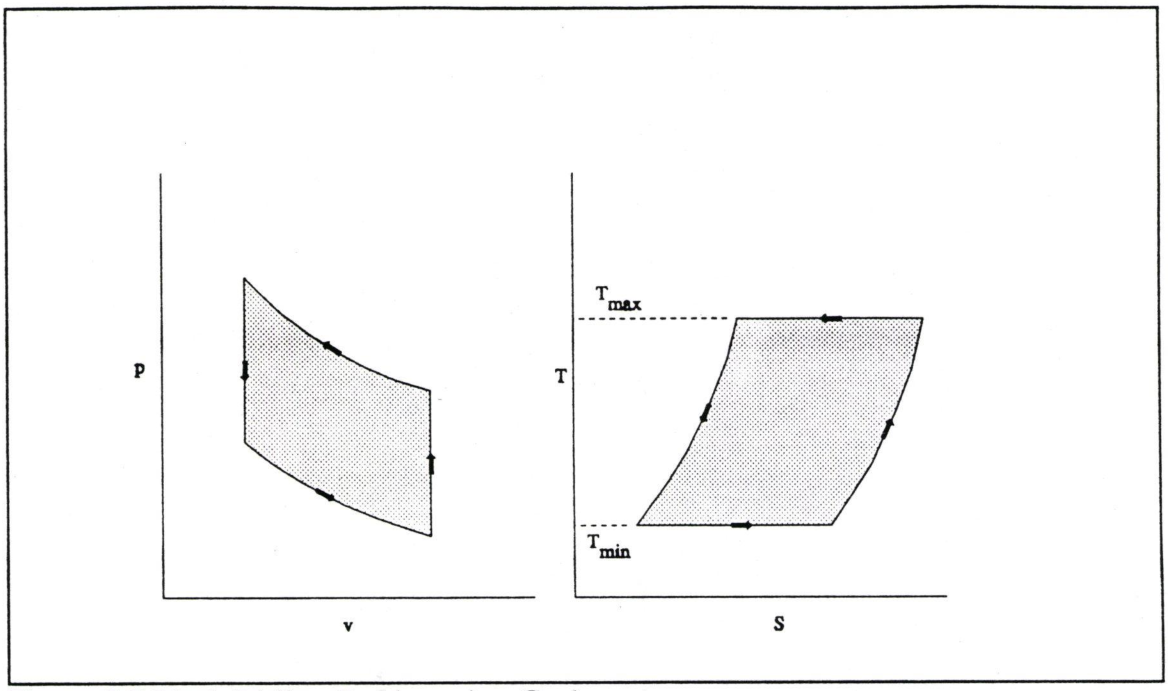

Figure 3.7 Ideal Stirling Refrigeration Cycle

Rather than using the thermal efficiency term as defined previously, refrigeration cycles can be normally compared using a coefficient of performance (COP) term, where

$$COP = \frac{T_e}{T_c - T_e} \tag{3.4}$$

which compares the amount of heat extracted from the refrigerated space (the numerator) to the input power required (the denominator). Similar to power cycles, the COP of an ideal Stirling refrigeration cycle is the same as that for a Carnot cycle, however the cooling capacity of the Stirling cycle is greater for a given temperature difference.

3.2.4 Actual Stirling Cycles

In real Stirling machines, the cycle is not the same as the Ideal Stirling cycle, but is rather closer to that shown in figure 3.8. This diagram is still "idealized" since the

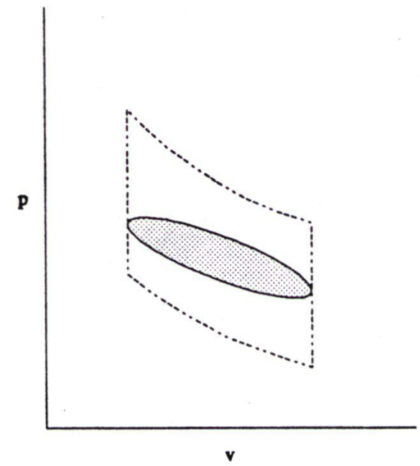

Figure 3.8 Actual Stirling Cycle

working fluid in an actual engine is not entirely in one area of the machine at any given time as this diagram assumes (Walker, 1980). The effect of these factors is shown as a reduction in the enclosed area on the p-v and T-S diagrams as compared to the ideal cycle. The cycle therefore is thermodynamically less efficient. In the case of a refrigeration system, this would mean more power is required to obtain a given amount of refrigeration effect in a real cycle than in an ideal cycle.

There are many factors that contribute to this less efficient cycle, some of the major ones being:

- dead volume — void regions within the actual machine in which the working fluid remains. Because all the fluid does not move to the appropriate place (at the appropriate time), the actual pressure fluctuation of the fluid in the useful region of the machine is reduced;

- adiabatic losses — losses due to poor heat transfer properties of the working fluid, particularly during the short time given for heat transfer to occur within the cylinders. These losses can dominate the performance of a Stirling system where small temperature differences are involved (West, 1986). Increasing heat transfer rates with the use of more complicated heat exchangers dramatically increases the dead space volume;
- transient heat transfer and shuttle loss — the tendency for heat to transfer between the expansion and compression chambers themselves (*ie.*, through conduction);
- imperfect regeneration — another thermal loss, due to only a finite rate of heat transfer possible between the regenerator and the working fluid during short time that the fluid passes through the regenerator;
- pressure loss due to flow — primarily due to the resistance of the working fluid passing through the regenerator and heat exchangers;
- leakage loss — tendency to lose working fluid to the atmosphere, particularly when operating at elevated pressures;
- mechanical friction losses — due to the friction effect in the piston rings, seals, bearings, etc.; and

- non-discontinuous piston and displacer motion — actual motions of the piston and displacer are closer to sinusoidal due to mechanical considerations (though not exactly sinusoidal due to dynamical considerations (Walker, 1980)). This results in the working fluid being distributed within the machine at any given time, rather than being all in one place.

3.2.5 Design Parameters

Walker (1980) suggested 7 independent parameters for Stirling engine design, resulting from the Schmidt analysis. The Schmidt analysis is an idealized Stirling engine analysis that provides for harmonic motion of the piston and displacer. These are:

- temperature ratio , τ
- swept-volume ratio, κ
- dead-volume ratio, χ
- phase angle, α
- working fluid pressure, p_{max}
- engine speed, N
- engine size, or swept volume, V_T.

For a prime-mover the cycle-power per cycle as given by the Schmidt-cycle analysis (Reader and Hooper, 1983):

$$P = (p_{max}V_T)\pi\frac{(1-\tau)}{(\kappa+1)}\left(\frac{1-\delta}{1+\delta}\right)^{\frac{1}{2}}\frac{\delta\sin\theta}{[1+(1-\delta^2)^{½}]} \tag{3.5}$$

where

$$\delta = \frac{(\tau^2+\kappa^2+2\tau\kappa\cos\alpha)^{½}}{(\tau+\kappa+2S)} \tag{3.6}$$

$$\theta = \tan^{-1}\left(\frac{\kappa\sin\alpha}{\tau+\kappa\cos\alpha}\right) \tag{3.7}$$

and

$$S = (2\chi\tau)/(\tau+1) \tag{3.8}$$

The power required to drive a refrigerator is given by the same equation with an appropriate change in sign to indicate input of power. The amount of heat lifted in the cold expansion space per cycle for a refrigerator is given by:

$$Q_E = (p_{max}V_T)\frac{\pi}{(\kappa+1)}\left(\frac{1-\delta}{1+\delta}\right)^{\frac{1}{2}}\frac{\delta\sin\theta}{[1+(1-\delta^2)]^{½}} \tag{3.9}$$

Of the seven parameters, pressure, speed (since equation is per cycle), and size are linear variables when looking at the net cycle-power per cycle when referring to an engine, and heat lifted per cycle when referring to a refrigerator. The remaining four parameters do not produce linear functions, and therefore must be optimized for most efficient operation.

3.3 FLAT-PLATE, DISCONTINUOUS DISPLACER THEORY

Kolin (1984) indicated that a more efficient Stirling machine could be achieved by operation closer to the ideal Stirling cycle, by reducing dead space and with the use of discontinuous motion. Beginning in 1972 (Kolin, 1984), Kolin began to develop an engine that closer approximated the ideal Stirling cycle. His experimentation lead to the "Low-Temperature Difference Stirling Engine", first demonstrated at the Inter-University Centre in Dubrovnik, Yugoslavia, in 1983 (Kolin, 1990). This engine included some significant differences as compared to other Stirling engines; in particular the use of flat-plate heat exchangers, a diaphragm-type piston, and a semi-discontinuous displacer/regenerator motion.

3.3.1 Dead-Space Reduction

Kolin argued that the use of flat plate heat exchangers considerably reduced the amount of dead space in the engine. The flat regenerator/displacer evenly contacts the flat-plate heat exchangers over its entire area thereby helping to eliminate dead space and resulting in a larger cycle trace on a p-v diagram, as shown in figure 3.9. The use of a diaphragm-type piston, with one of the heat-exchangers acting as the piston, also enhanced the reduction of dead space as the space associated with connections to a separate compression space were no longer required.

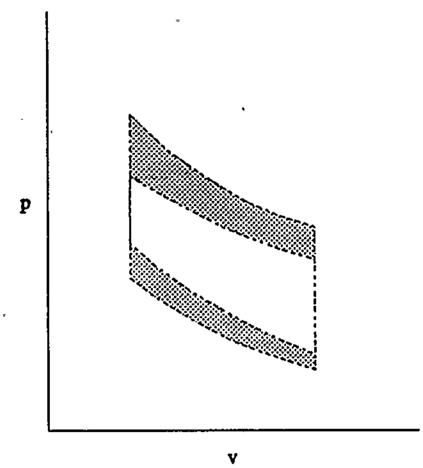

Figure 3.9 Effect of Dead Space on a Stirling Cycle (Reader and Hooper, 1983)

3.3.2 Discontinuous Motion

The other significant change was the use of semi-discontinuous displacer motion. Although not instantaneous, the effective displacer motion is rapid movement during the "constant volume" processes, and dwell times at the "constant temperature" processes. Kolin approximates this as an isothermal processes, having the effect on the cycle as shown in figure 3.10.

3.4 THE KOLIN STIRLING ENGINE DETAILS

For research into refrigeration applications, Kolin donated a 1989 model of the flat-plate, discontinuous displacer motion Stirling engine. It is essentially the same model as that demonstrated in 1988 (Kolin, 1990). It is shown diagrammatically in figure 1.1

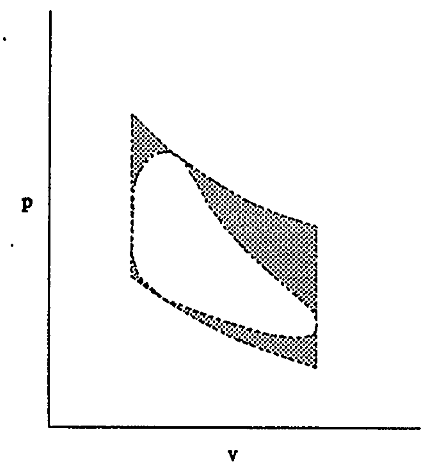

Figure 3.10 Effect of Sinusoidal Motion on a Stirling Cycle (Reader and Hooper, 1983)

and is described in detail below.

The frame of the engine is made of a semi-soft wood (of unknown type, though similar to pine), having a grain size of approximately 1 mm. The members of the frame are connected together with wood screws and glue, with the glue acting also as a sealant at the joints.

The working space is enclosed by the frame on four sides (*ie.*, the top, bottom, and left and right sides) and has inside dimensions of 185 mm (height), 185 mm (width), and 38 mm (breadth), giving a total enclosed space of .0013 m^3 (1.3 L).

The top of the frame has a 4 mm diameter hole used to regulate the initial amount of air enclosed in the work space, and is plugged with a wooden stopper during operation. The left side frame member (when viewed from the diaphragm side) has a 1.6 mm diameter hole through which the displacer rocker arm passes. Also, on both sides of the frame, two view-ports are provided through a 16 x 16 mm opening. These openings are covered with 6 mm thick plexiglass plate, sealed with a 2 mm thick rubber-type gasket, and connected to the wood frame with four wood screws.

The back of the working space consists of a 2 mm thick aluminum plate attached to the wood frame with wood screws, spaced approximately every 22 mm. A 2 mm thick rubber gasket is used as a seal around the plate/frame interface. This back-plate makes up one of the flat-plate heat exchangers on the Kolin Stirling engine.

The working space is enclosed at the front by a the piston arrangement, which is composed of a 2 mm thick aluminum plate attached to a 1.5 mm thick rubber diaphragm. The aluminum plate acts as the second flat-plate heat exchanger.

The aluminum plate attached to the diaphragm is cut into a trapezoidal shape, allowing it to move into the working space during the compression stage. The working space is sealed from the atmosphere by using the edge of the rubber diaphragm as a gasket between the wood-frame and a 2 mm thick aluminum face plate. The face plate and rubber diaphragm are held in place by wood screws spaced approximately 22 mm apart. The aluminum piston plate is joined to the rubber diaphragm by glue (a metal-to-rubber type contact cement), and is restricted from moving away from the wood frame at the bottom during compression and expansion strokes by two metal tabs. The top of the trapezoidal plate has a metal tab protruding outwards to which the diaphragm-piston is attached to the crank-lever arm.

Directly above the protruding tab on the piston plate, an aluminum bracket is attached to the wooden frame. The lever arm attaches to this bracket with the use of a small metal bolt (a portion of which has a bearing surface). This point of attachment acts as the pivot point for the lever arm, and allows the piston plate to move in a motion with the arcing motion of the lever arm. The lever arm thus moves in a vertical plane, perpendicular to the heat-exchanger planes.

The lever arm is attached to another aluminum bar, the crank arm, which is nominally in a horizontal position. This bar is coupled to the flywheel.

The aluminum flywheel has an outside diameter of 230 mm, an inside diameter of 170 mm, and is 8 mm thick. The flywheel is connected to the flywheel shaft by three 74 x 10 x 4 mm thick radial spokes. The horizontal aluminum bar is attached to one of the spokes. Compression ratio is set by the connection point of the crank arm to the flywheel spoke.

The flywheel is supported by an aluminum frame which is attached directly to the wooden frame. The flywheel shaft has an L shaped bend beyond the support structure, to which the displacer linkage is attached.

The displacer linkage, a 1.63 mm diameter steel rod, is passed through a 3 mm diameter hole in the displacer lever. Rubber stoppers are attached to the displacer linkage in such positions that the displacer lever moves only at the end points of the displacer linkage's back-and-forth motion. The displacer linkage is connected to the L-shaped portion of the flywheel shaft. The lack of a hard (*ie.*, rigid) connection between the displacer lever and the displacer linkage results in the discontinuous displacer motion.

The phase angle between the piston and displacer motion can be adjusted by rigidly attaching the flywheel to the shaft with set screws after appropriately positioning the flywheel-crank arm connection with respect to the L-shaped bend in the shaft. On this engine, there is no shaft available for output power.

The engine had a folded-aluminum displacer/regenerator, as shown in figure 3.11. The total thickness of the regenerator was 6 mm, and was 173 mm high and 175 mm wide. The thin aluminum used in the regenerator was .005 mm thick. The overall dead space within the regenerator was approximately 0.15 L.

The regenerator/displacer was supported by a wire-rocker arm, with its ends supported in holes made into the frame. As mentioned earlier, the left-hand side hole passes entirely through the frame through which the rocker arm passes outside the enclosed space and is attached to the displacer lever.

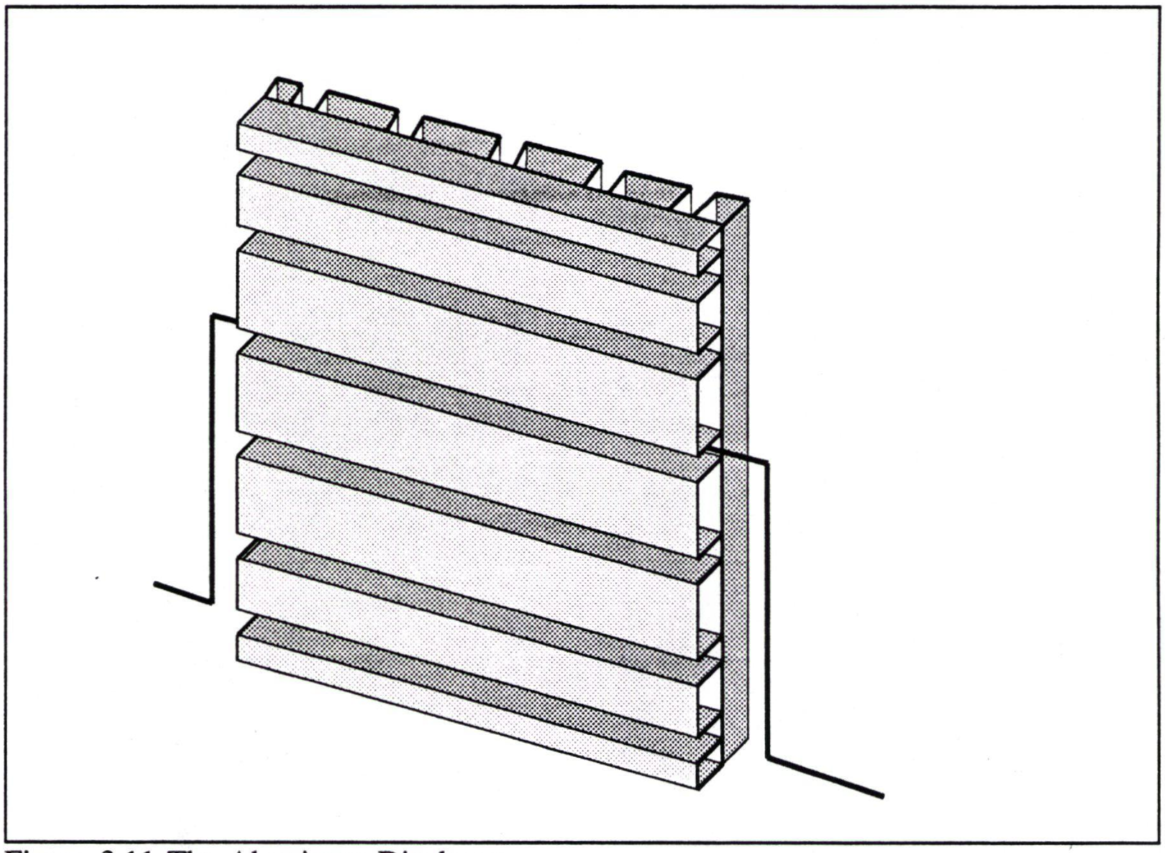

Figure 3.11 The Aluminum Displacer

3.5 OPTIMIZATION PARAMETERS

The Schmidt equations that were developed for Stirling engine analysis are based on the alpha-type configuration. Modifications need to be made in the analysis for beta and gamma type configurations (Senft, 1991). The modified equations for the design parameters of a beta system are as follows:

$$P = \frac{\pi(1-\tau)p_{mean}V_d\kappa\sin\alpha}{Y+(Y^2-X^2)^{1/2}} \tag{3.10}$$

where

$$X = \left[(\tau-1)^2+2(\tau-1)\kappa\cos\alpha+\kappa^2\right]^{\frac{1}{2}} \tag{3.11}$$

$$Y = \tau+\frac{4\chi\tau}{1+\tau}+D \tag{3.12}$$

and

$$D = (1+\kappa^2-2\kappa\cos\alpha)^{1/2} \tag{3.13}$$

Because the Schmidt equations are developed for a sinusoidal motion of both the piston and displacer elements, the analysis for a Kolin machine will be conservative due to the discontinuous motion of the displacer not being taken into account.

A dimensionless form of equation 3.10 can be written as follows:

$$\frac{P}{V_d\, p_{mean}} = \frac{\pi(1-\tau)\kappa\sin\alpha}{Y+(Y^2-X^2)^{1/2}} \tag{3.14}$$

The term on the left is referred to as the power parameter.

CHAPTER 4

4. THE KOLIN REFRIGERATOR

4.1 INTRODUCTION

The Kolin Refrigerator is a proposed self-contained compact refrigerator, using the principles developed by Kolin in the development of the Flat-Plate Stirling Engine.

There are three main components to the proposed refrigerator: the power unit, the refrigeration unit, and the control system. In this chapter, these three components will be discussed. The need for such system will also be discussed through the presentation of some of its proposed applications.

The overall unit would be made as compact as possible with the refrigeration and power units positioned side-by-side. These units may in fact share the diaphragm and piston plate as shown in figure 4.1.

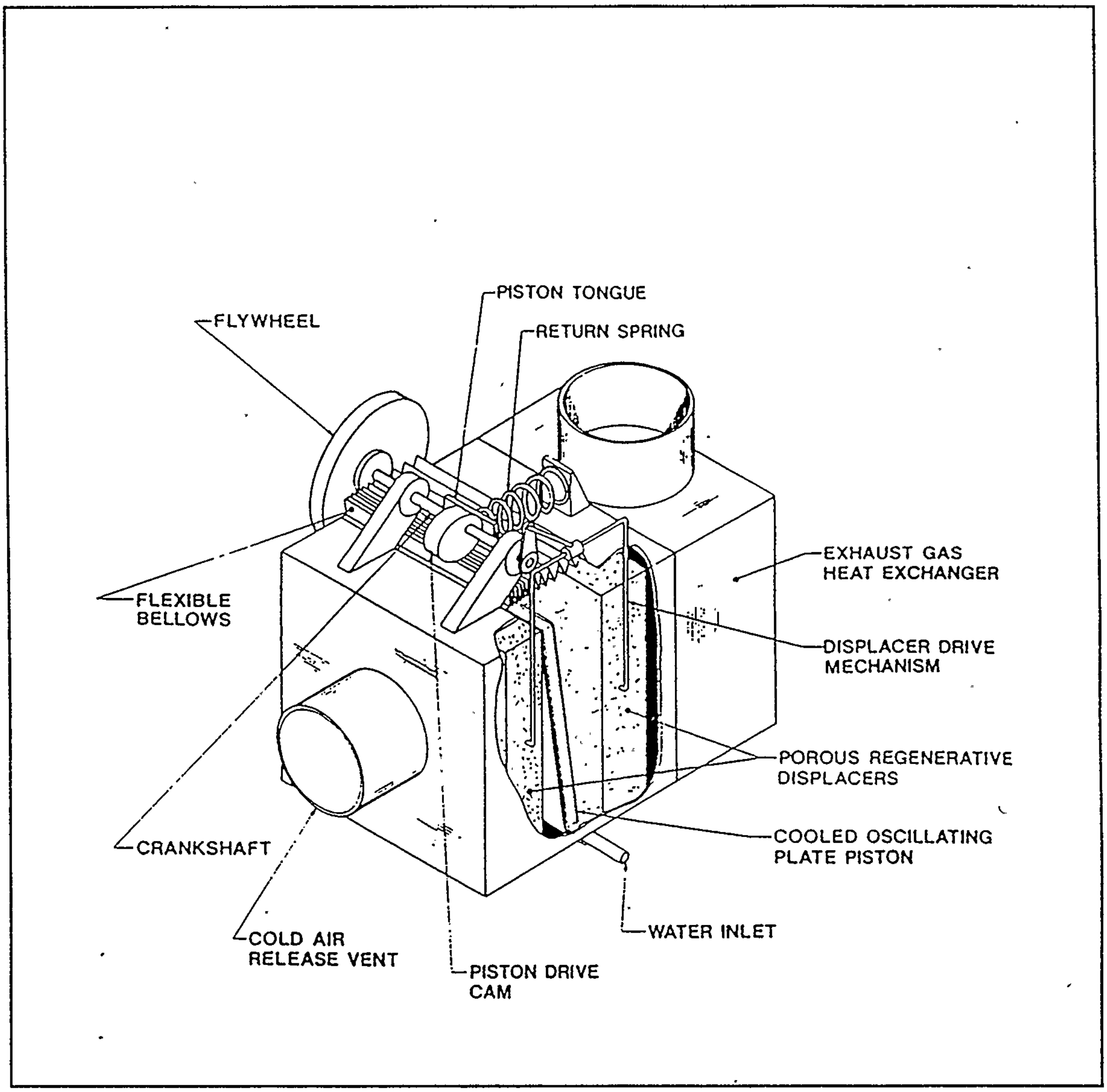

Figure 4.1 A Compact Kolin Refrigerator Arrangement (Walker, 1989)

4.2 THE POWER UNIT

The power unit for the refrigerator is a Kolin engine as described earlier. Parametric analysis with appropriate engine values yield useful data, as shown in figures 4.2, 4.3, 4.4, and 4.5. From these figures, it can be seen that the Kolin engine is at an optimized state for the designed configuration values.

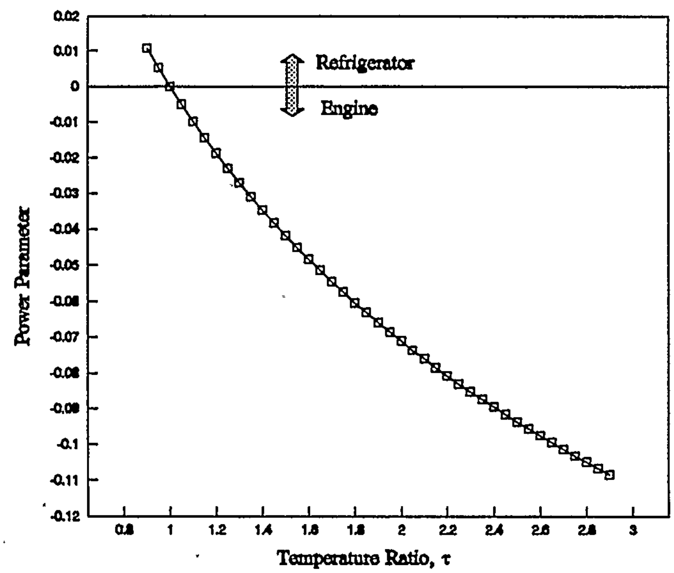

Figure 4.2 Temperature Ratio (τ) Variation Effect on a Kolin Machine

Using the values from the donated Kolin engine described in chapter 3, a power output prediction can be calculated from the modified Schmidt equations. The donated Kolin engine, with aluminum displacer, had the following configuration:

- dead space volume, $V_D = 0.156$ L;
- displacer swept volume, $V_d = 0.980$ L;
- piston swept volume, $V_p = 0.227$ L; and
- phase angle, $\alpha = 90$ degrees.

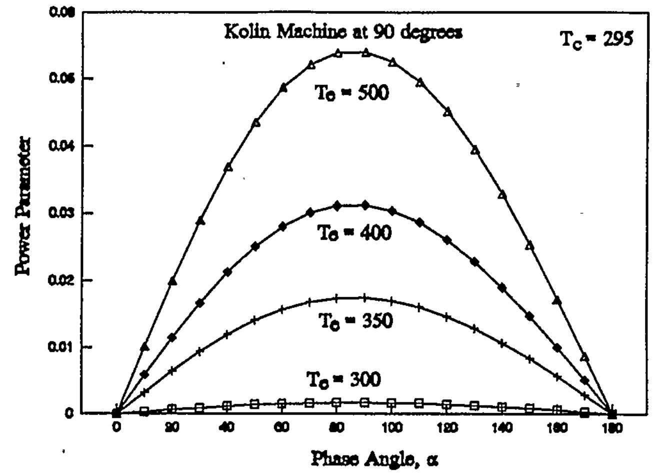

Figure 4.3 Phase Angle (α) Variation Effect on a Kolin Engine

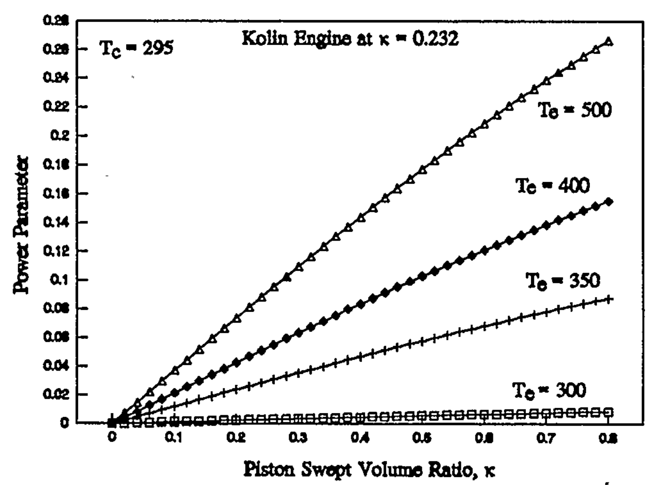

Figure 4.4 Swept Volume Ratio (κ) Variation Effect on a Kolin Engine

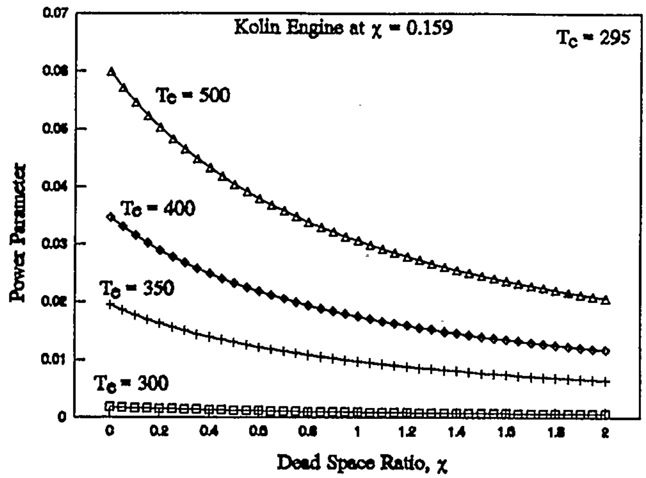

Figure 4.5 Dead Volume Ratio (χ) Variation Effect on a Kolin Engine

Assuming the engine operates at a mean pressure equivalent to atmospheric (p_{mean} = 101 kPa), cold heat exchanger temperature at 300 K, and hot heat exchanger temperature at 400 K, equation (3.10) gives the modified Schmidt value for power, per unit cycle, to be P = 7.3 W·s. At an operating speed of 100 revolutions per minute, power output is 12.2 W. This of course is an ideal value due to the assumptions inherent in the Schmidt analysis.

4.3 THE REFRIGERATION UNIT

Similarly, parametric analysis for a Kolin Refrigerator yields figures 4.6, 4.7, and 4.8 (as well as figure 4.2, which is applicable for both refrigerators and engines). Again, for the given configuration values, the machine is designed for operation at the optimal state.

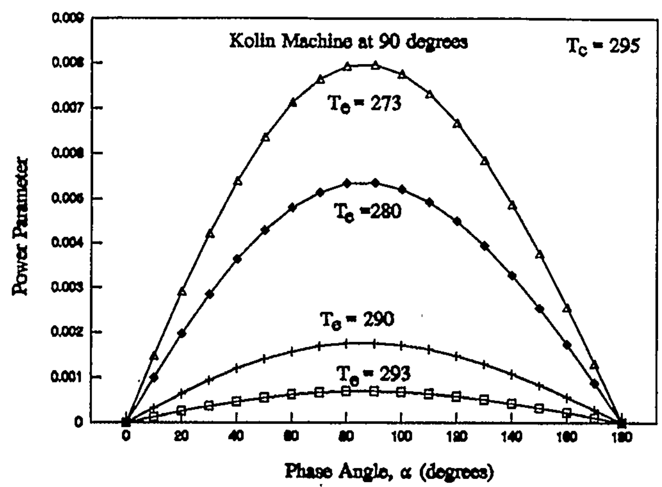

Figure 4.6 Phase Angle (α) Variation Effect on a Kolin Refrigeration Unit

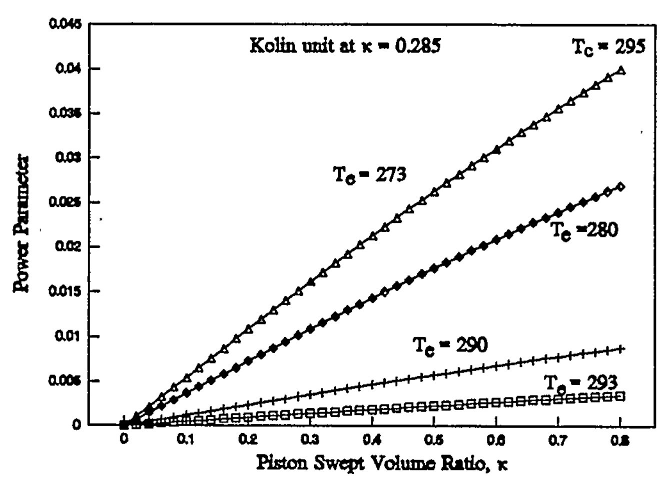

Figure 4.7 Swept Volume Ratio (κ) Variation Effect on a Kolin Refrigeration Unit

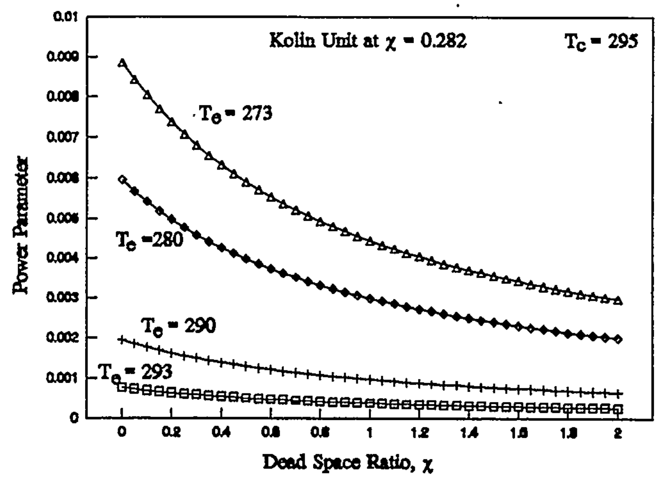

Figure 4.8 Dead Volume Ratio (κ) Variation Effect on a Kolin Refrigeration Unit

Again, using the values from the donated Kolin engine described in chapter 3, a power input prediction can be calculated from the modified Schmidt equations. Replacing the aluminum displacer with a nylon foam displacer (with a 60% porosity) for better regeneration effect, the system has the following configuration:

- dead volume, V_D = 0.224 L;
- displacer swept volume, V_d = 0.796 L;
- piston swept volume, V_p = 0.227 L; and
- phase angle, α = 90 degrees.

Assuming the refrigeration unit operates at a mean pressure equivalent to atmospheric (p_{mean} = 101 kPa), "hot" heat exchanger (*ie.*, heat sink) temperature at 300 K, and with an input power of 1.83 W·s per unit cycle (25% of the above produced power), re-arranging equation (3.10) gives the cold side heat exchanger temperature to be 277 K, a temperature drop of 23 °C. Again, these values are for the ideal situation.

Although the Kolin-Stirling engine has been found to be operational (Kolin, 1984, 1986, 1990), tests on the usability of the Kolin machine as a refrigeration unit has yet to be done. Before development of the proposed Kolin Refrigerator can continue, basic tests on the machine are required. Chapter 5 describes the experiments conducted towards these goals.

4.4 THE CONTROL SYSTEM

Keeping with the nature of the proposed power and refrigeration units, a simple, low-cost control system is required for the Kolin Refrigerator. The variable of interest to be controlled would be the cold-side temperature of the refrigeration unit (*ie.*, the refrigerated space temperature), although controls on the temperature of the hot-side of the power unit would also be required to some extent.

Due to the absence of electrical power, a mechanical-type system would be the easiest to implement. The actuators used to initiate control could be a thermal-expansion material — either a bi-metallic strip or a shape-memory-alloy[2] type structure, being the simplest.

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[2]Shape memory alloys are materials which undergo changes in crystal structure at specific transformation temperatures between -200 °C and 100 °C (depending on alloy composition), allowing the metal to "remember" previous shape configurations (Stevens, 1991).

Walker (1980) and Reader and Hooper (1983) discuss many of the common control systems used with Stirling engines. The main types discussed are: mean-pressure level modulation, short-circuiting between adjacent working spaces, phase-angle variation, pressure-amplitude variation, and stroke variation. Of these systems, the pressure-amplitude variation, short-circuiting, and stroke variation methods seem most promising for the Kolin Refrigerator.

4.4.1 Mean-Pressure Modulation and Phase Angle Variation

Mean-pressure level modulation is the most widely used and best known control system with respect to Stirling machines (Walker, 1980). It was used in the early "hot-air" engines, and also adopted by the Philips company during the 1940's (Reader and Hooper, 1983). It is based on the fact that power output is directly proportional to the pressure of the working fluid. The system works simplest by venting the working fluid to atmosphere to reduce the power output (in engine applications) and resupplying it for increased output. Such a system would be difficult to implement in the Kolin system without a high-pressure source of the working fluid, or a means of pressurizing the working fluid on demand.

Phase angle difference control works by changing the phase between the volume variations of the expansion and compression spaces. Walker (1980) states that power output is a sinusoidal function of phase angle The effect of phase angle variation on the Kolin engine is shown in figure 4.3. Phase angle variation would be difficult to implement in the Kolin system as the phase angle is controlled by the position of the piston linkage and displacer linkage connection to the flywheel and flywheel shaft respectively.

4.4.2 Short-Circuiting

Although the short circuiting method as described by Walker (1980) is between the working space and a buffer space, in the Kolin Refrigerator the short-circuit can exist between the refrigeration unit and the heater unit. When the temperature of the cold-side of the refrigeration unit reaches the set point, the actuator device could open a valve to the connection between the two units, allowing the colder, refrigeration unit fluid to mix with the hotter power unit's fluid, thereby reducing the pressure in the power unit. This in turn reduces the power output of the unit. As the refrigerated space warms up, the valve would close, and the system would be in operation again.

4.4.3 Pressure Amplitude Variation

Pressure amplitude variation, also called dead volume variation, works in a similar manner to the short-circuiting method, except that the actuating device opens a valve to a connection between either working space to a region of increased dead space. The increase in dead space effectively reduces the pressure change, and hence the efficiency of the system. Reader and Hooper (1983) mention that this system is being considered by various organizations, particularly in Stirling cryocooler development.

4.4.4 Stroke Variation

The stroke-variation system operates in a directly-mechanical manner, and may be implemented easiest with memory-metal type devices. Stroke variation may be seen to control a few of the design parameters discussed in chapter 3. Controlling the stroke of the diaphragm/piston varies V_T, which is directly proportional to power output.

Displacer control would be difficult in the Kolin system, due to the soft connection between the displacer lever arm and the displacer linkage. If achievable though, displacer stroke control could be viewed as controlling dead space (*ie.*, increasing clearance space inside the working space), or as a variation of the swept volume ratio.

This system more so than the others would depend more upon the physical set-up of the refrigeration and power unit connections. Temperature control however could only be varied by the selection of the memory metal.

CHAPTER 5

5. KOLIN REFRIGERATION EXPERIMENTS

5.1 INTRODUCTION

The experimental portion of the research consisted of testing a Kolin-Stirling engine as a refrigeration machine. The tests were conducted on a modified low-temperature-difference Stirling engine provided by Kolin, as described in the previous chapter.

In this chapter the experimental apparatus and method of experimentation are discussed as well as the modifications to the original engine.

5.2 MODIFICATIONS TO THE KOLIN MACHINE FOR TESTING

5.2.1 Flywheel Shaft

The original flat-plate Stirling engine is useful only as a model since there is no output shaft for useful work. That is, both ends of the flywheel shaft are used as cranks for piston and displacer motions. For use as a refrigeration machine, a means of supplying external power is required.

The flywheel shaft was modified so as to allow mechanical power to be used to operate the Kolin Stirling machine for the refrigeration experiments. This modification was essentially the replacement of the flywheel shaft with a shaft having a U bend at the displacer motion portion, *ie.*, a continuation of the shaft beyond the displacer linkage. The free end of the shaft beyond the U bend was therefore used to provide mechanical power input.

5.2.2 Reduction in Dead Space.

As mentioned in Chapter 3, dead space reduction is important in Stirling engines, and more so for low-temperature-difference machines. To reduce dead space in the Kolin Stirling Machine meant the elimination of the only remaining superfluous space, *ie.*, the space associated with the view ports. These two view ports provided .0074 L of dead space. This dead space was eliminated by filling the view ports with styrofoam squares, and sealing with duct tape.

5.2.3 Other Modifications

Other modifications were carried out on the original Kolin Stirling engine, but were directly related to the experimental variables. These modifications are discussed in the next section.

5.3 EXPERIMENTAL VARIABLES

The variables for the experiments on the Kolin Stirling refrigeration machine were grouped into two categories, operational and material. Operational variables were those variables not relating to changes in the refrigeration machine's material components. Material variables were those variables in which components in the refrigeration machine were changed.

5.3.1 Material Variables

The material variables in the experiments were mostly concerning two parts of the machine, the regenerator/displacer material and the flat-plate heat exchanger material. The other material variable changed was the method of positioning and sealing the piston to the rubber diaphragm.

5.3.1.1 Regenerator/Displacer Materials

Regeneration effect, and hence regenerator material, is more critical to Stirling refrigeration machines than Stirling power machines (West, 1986). Therefore testing different regenerator materials with the Kolin Stirling refrigerator was deemed important.

The experiments carried out for the Kolin Stirling machine tested four different regenerator materials: solid extruded styrofoam, extruded styrofoam with holes, nylon foam batting, and folded thin aluminum (the regenerator accompanying the original donated Kolin Stirling engine). These materials were tested due to their simplicity and low cost, in keeping with the expected application of the Kolin Stirling refrigerator.

The fact that the whole displacer/regenerator apparatus moves in the Kolin machine restricted the type of regenerators tested. Loose packed (*ie.*, non-rigid), and other-wise heavy or cumbersome types of regenerators had to be avoided. This meant that many of the more effective types of regenerators (Isshikki *et al.*, 1987) could not be tested.

5.3.1.2 Heat Exchanger Materials

Two heat exchanger materials were tested: the original flat-plate heat exchangers, 2 mm thick aluminum, and those made of 3 mm thick copper plates.

5.3.1.3 Heat Exchanger/Diaphragm Connection

Although not exactly a material variable, two different methods of joining the "hot" side heat exchanger to the rubber diaphragm were tested. The first method, was simply to glue the flat-plate to the diaphragm, in a manner similar to that of the original Kolin Stirling engine. However, an aluminum support plate was attached at the top of the plate on the inside of the working space, sandwiching the top portion of the diaphragm, so as to prevent the rubber diaphragm from peeling away from the heat-exchanger during operation. This method of connection is referred to as the "Standard connection" in Chapter 6, and is shown in figure 5.1.

The second method of connection was to sandwich the diaphragm between the heat-exchanger plate and face plates along the entire periphery of the plate, allowing the middle portion of the diaphragm to be removed. This was in an attempt to improve heat

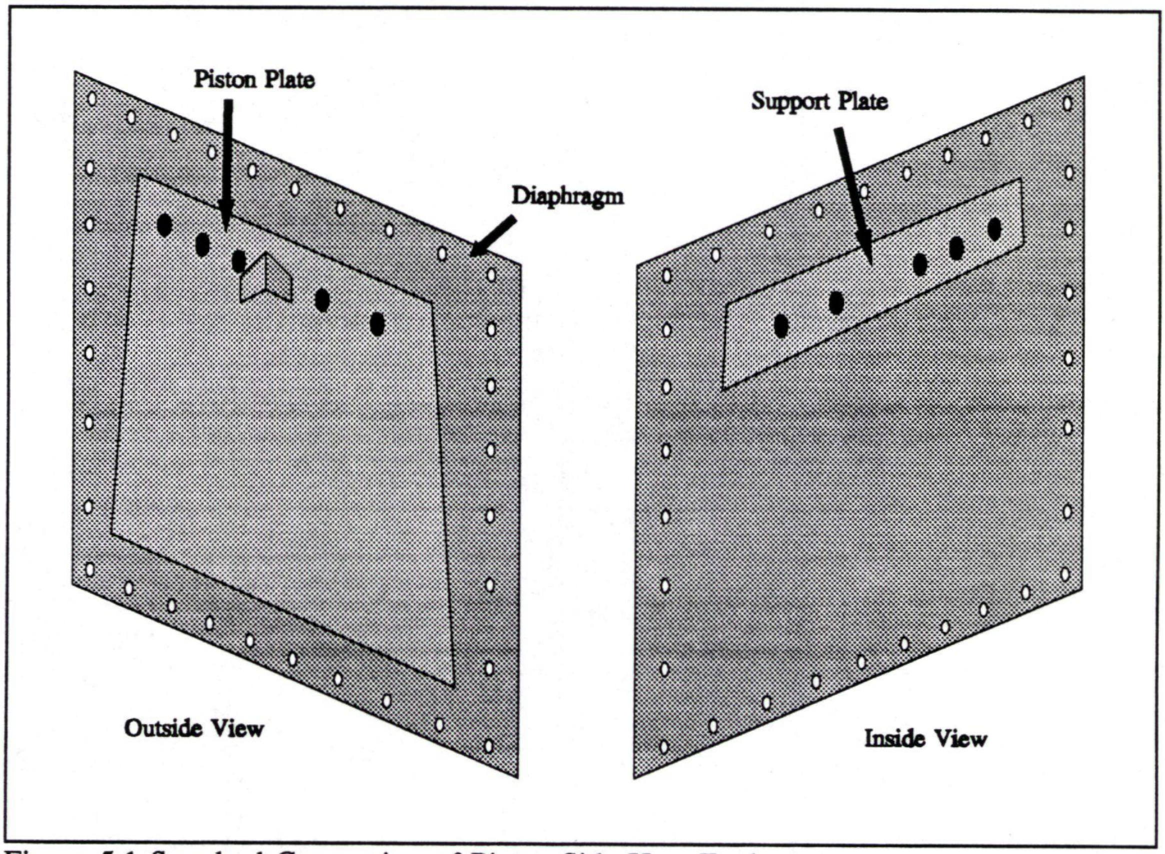

Figure 5.1 Standard Connection of Piston-Side Heat Exchanger

transfer between the air in the working space and the "hot" side heat exchanger, without the rubber diaphragm acting as an insulator. This method of connection is referred to as the "Sandwiched connection" in chapter 6, and is shown in figure 5.2.

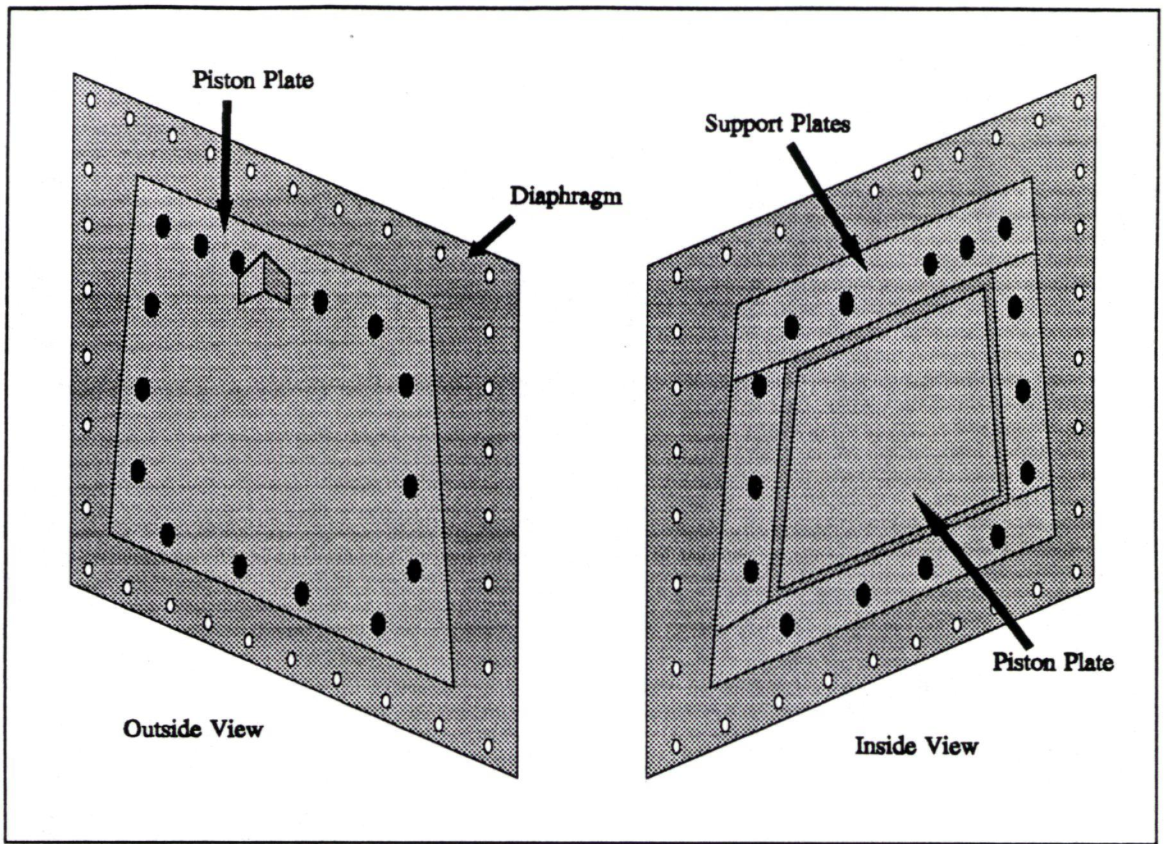

Figure 5.2 Sandwiched Connection of Piston-Side Heat Exchanger

5.3.2 Operational Variables

Two operational variables were tested, operating speed and working pressure. As mentioned in chapter 3, power input required for a refrigerator is directly proportional to speed and pressure.

5.3.2.1 Operating Speed

Due to the relatively fragile nature of the Kolin Stirling machine, relatively slow operational speeds were required. The tests were conducted over two speeds, 75 rpm and 100 rpm, as measured by the flywheel rotation. Speeds much greater than 100 rpm were deemed to be destructive to the piston/diaphragm arrangement as well as the displacer mechanism.

5.3.2.2 Working Pressure

For the pressure tests, an initial higher pressure was created with the use of a foot-type bicycle pump. The pump was connected to a modified cold heat-exchanger plate. This copper flat plate had a bicycle inner-tube spigot valve attached in the upper-left corner. The spigot valve, with accompanying rubber flange, was sealed over a 6.25 mm diameter hole in the plate, with silicone rubber, and clamped between the heat-exchanger plate and an aluminum covering plate with the use of six equally spaced set screws. The total area of the heat-exchanger plate that was covered was 35 cm^2, effectively reducing the total cold heat exchanger area to 308 cm^2.

The cold-side heat exchanger was selected as the place for the spigot valve due to problems in sealing a connection on the wood frame, and the lack of total "hot" side heat exchanger area (due to the rubber diaphragm/piston arrangement).

5.4 TESTING APPARATUS

For the experimental tests, the primary measurement of interest was the temperature drop achieved for each test. Other measurements required were for the pressure trials and measuring input (electrical) power. Instruments for these three variables were required, as well as the test-bed apparatus (*ie.*, mechanical power source, etc.).

5.4.1 Mechanical Power Source

For the experiments, the Kolin Stirling engine was powered mechanically by a 1200 RPM DC motor connected to a 6:1 reduction gearbox. Both the DC motor and gearbox were from previously used surplus supplies, and were slightly modified for use in the experiments. The major modification was to the gear box, in which the output gear shaft was reversed to allow for easier connection to the Kolin Stirling engine flywheel shaft.

The DC motor and gearbox were connected initially with a Tygon-tubing connector. This was replaced for the pressure tests with a cylindrical aluminum connector, using set screws for grasping the gearbox shaft and a cotter-type pin for the DC motor shaft (using an existing hole in the motor shaft). This connection change was made due to the twisting in the Tygon-tubing connector when the pressure tests resulted in an irregular flywheel rotation.

Power for the DC motor was supplied using an HP 6291A metered power supply. Motor speed was controlled by adjusting the voltage control. Voltage differential and current flow was measured using a Universal Avometer Model 8 Mark III (set to measure voltage differential) and a Fluke Digital Multimeter (8050A) set to measure current flow, placed in parallel and series, respectively, with the power supply-DC motor circuit.

Mechanical connection between the gear box and the free end of the flywheel shaft was made initially by a Tygon-tube soft coupling, but was replaced by a cylindrical aluminum connector due to premature wearing and slippage between the tubing and shafts. Anchoring of the aluminum connector to the shafts was done using set-screws.

5.4.2 Temperature Measurement

Temperature measurement of the "cold" side of the Kolin Stirling refrigeration machine was done using a type T thermocouple. The thermocouple bead, about 2 mm in diameter, was attached to a semi-circular cavity made in the flat-plate heat exchanger. The thermocouple was kept in place with a small piece of duct tape.

The thermocouple was attached to a Doric Trendicator 410A temperature meter calibrated for type T thermocouples, and set to measure (*ie.*, display) degrees Celsius.

A styrofoam shell, enclosing a total volume of 2.6 L (60 x 170 x 170 mm inside dimensions), was used to insulate the cold heat exchanger from the surrounding air. The shell, made of 6.3 mm expanded styrofoam sheets, was used to reduce any air flow between the cold heat exchanger and the surrounding air.

5.4.3 Pressure Measurement

The pressure sensor used to measure the increase in working space pressure was constructed by the Mechanical Engineering Dept. (although not specifically for this experiment), and was calibrated to measure in pounds per square inch. Due to a relatively slow response time with respect to the cycle speed, it was decided to use the pressure sensor only for measuring initial and final pressures of the working space, taking precautions to ensure the piston position was the same for both measurements.

The pressure sensor was connected to the working space via a Tygon tube connected to a 25.5 mm pipe stem which was inserted and sealed to a 6.3 mm diameter hole in the wood frame. Because the pressure sensor was used only for initial and final measurements, and also to reduce dead space, the tube was clamped off as close as possible to the wood frame access hole.

5.5 METHOD OF CHANGES TO SYSTEM

Testing the various material variables (as opposed to operational variables) meant that the system had to be essentially disassembled for each variable change. This meant that one or both of the flat-plate heat exchangers had to be removed from the wood frame before the change could be made.

Each time a heat exchanger was removed from the wood frame, all remaining sealants were removed from both the plate and the wood frame. After the material variable was changed, the heat exchanger plates was reattached and resealed to the wood frame.

5.6 METHOD OF EXPERIMENTAL OPERATION

In general, individual experiments were carried out over a period of days, with each trial tested on a single day. This was due to various reasons, but primarily two: changing variables usually meant that sealants had to cure or set for a time period lasting at least over night, and secondly so that the effects of previous trials were kept to a minimum.

For each trial, prior to testing, the initial temperature (T_o) of the cold heat exchanger plate was measured and recorded (which was the same as the ambient temperature due to the almost 22 hour time period between trials). Final temperature (T_f) of the cold heat exchanger was recorded after steady state conditions had been achieved.

At the beginning of the trial, the motor was switched on and was adjusted to the appropriate speed. Cycle speed was measured with the use of a stop watch, which was assumed accurate enough due to the relatively slow speeds of operation. As mentioned earlier, the DC motor speed was controlled by adjusting the voltage input.

Each variable was tested twice, and each test was carried out for two hours, plus or minus 30 minutes, when possible.

For the trials where higher initial pressures were required, pressure measurements were recorded before each trial began, and immediately after.

CHAPTER 6

6. RESULTS AND DISCUSSION

6.1 INTRODUCTION

The experimental tests done on the modified Kolin Stirling engine provided only a few results with an actual temperature drop measured at the "cold" heat exchanger. The other "no-temperature-drop" results from the remaining tests were not however without use, as they pointed to some of the many problems associated with the Kolin-Stirling machine being used as a refrigeration unit. These problems will be presented in the discussion section of this chapter, following a presentation of the results from the experimental tests.

6.2 RESULTS FROM EXPERIMENTAL TESTS

The results from the experiment are grouped into four sections based on heat exchanger materials, method of connection between the diaphragm and hot-side heat exchanger, and operating pressure: standard connection with aluminum heat exchanger, standard connection with copper heat exchanger, sandwiched connection with copper heat exchanger, and standard connection with copper heat exchanger operating at elevated pressure..

Ideal COP is the Carnot COP as determined by equation 2-4. Actual COP values were calculated from the assumption that at steady state conditions, the heat extracted by the Kolin machine is equal to the heat transfer through the styrofoam shell. The styrofoam shell has a thermal conductivity of k=0.13 W/m·°C (Perry and Chilton, 1973). Using Fourier's law (Holman, 1986):

$$Q_E = -\frac{kA}{\Delta x}(T_o - T_f) \qquad (6.1)$$

Experimental (exp.) COP is then found by dividing the heat transfer found using equation 6.1 by the input power, which is assumed to be 25% of the electrical power input to the motor. (Only 25% so as to take into account electrical and mechanical losses prior to the Kolin machine.)

6.2.1 Standard Connection/Aluminum Heat Exchanger

Four regenerator/displacer materials were tested, at operating speeds of 75 rpm and 100 rpm. The results from these test are presented in table 6.1. Temperatures are in °C.

Table 6.1

Regenerator material	75 rpm					100 rpm				
	T_o	T_f	ΔT	ideal COP	exp. COP	T_o	T_f	ΔT	ideal COP	exp. COP
folded aluminum	22.5	22.5	0	n/a	n/a	22.5	22.5	0	n/a	n/a
solid styrofoam	22.3	20.8	1.5	196	2.2	22.4	20.8	1.6	184	1.6
styrofoam with holes	21.0	19.3	1.7	172	2.5	21.0	19.6	1.4	209	1.4
nylon foam	21.7	19.4	2.3	127	3.4	21.6	19.3	2.3	127	2.3

No major problems were encountered during these tests. Of minor concern was the loosening of some connections during the trials. In this instances, the electric motor was stopped, the connections tightened, and the tests resumed. Total down time was approximately one minute. Electrical power input for the 75 rpm tests was 3.9 W (8.87 V x 0.44 A), and for the 100 rpm tests was 5.6 W (12.25 V x 0.46 A). These values did not change appreciably (*ie.*, ±0.1 W) between tests.

6.2.2 Standard Connection/Copper Heat Exchanger

Three regenerator/displacer materials were tested, at operating speed of 75 rpm and 100 rpm. The folded aluminum displacer was not tested due to the lack of positive results in the previous tests. The results from these tests are presented in table 6.2.

Table 6.2

material	75 rpm					100 rpm				
	T_o	T_f	ΔT	ideal COP	exp. COP	T_o	T_f	ΔT	ideal COP	exp. COP
solid styrofoam	22.5	21.0	1.5	196	2.2	22.7	21.2	1.5	196	1.5
styrofoam with holes	22.2	20.7	1.5	196	2.2	21.9	20.5	1.4	209	1.4
nylon foam	21.5	19.2	2.3	127	3.4	22.0	19.8	2.2	133	2.2

Again, no major problems were encountered during these tests, except for some loosening of connections. The soft-connection between the displacer linkage and displacer lever arm needed to be lubricated — some wearing of the bearing surface of the lever arm was noticed. Power input values did not vary appreciably from the previous tests (section 6.2.1).

6.2.3 Sandwiched Connection/Copper Heat Exchanger

Three regenerator/displacer materials were tested, at operating speeds of 75 rpm and 100 rpm. The results from the tests are presented in table 6.3.

The sandwiched connection caused some problems, particularly with seal integrity. The resulting sealant and tighter clamping used to correct the problem seemed to result

Table 6.3

material	75 rpm			100 rpm		
	T_o	T_f	ΔT	T_o	T_f	ΔT
solid styrofoam	22.1	22.1	0	22.3	22.2	0.1
styrofoam with holes	22.2	22.3	-0.1	22.1	22.1	0
nylon foam	21.9	21.9	0	21.7	21.8	-0.1

in a stiffer exposed region of diaphragm. Again, power input values did not vary significantly from the previous tests (section 6.2.1 and section 6.2.2).

6.2.4 Standard Connection at Elevated Pressure

Three regenerator/displacer materials were to be tested at speeds of 75 rpm and 100 rpm, with the system pressurized to both ~7 kPa (~1 psi) and ~13 kPa (~2 psi) above atmosphere. However, no tests could be carried out as the system stopped during the initial compression stroke. The exposed portion of the rubber diaphragm "bubbled", or deflected outwards, before the compression stage commenced. The rapid stoppage of motion resulted in the displacer linkage and the aluminum crank and lever arms being slightly bent. Pressure within the system quickly lowered to atmospheric conditions, even though no leakage occurred around the working area. Application of soap solution made evident of air leakage through the wood, along the grain.

6.3 DISCUSSIONS

As can be seen from the results of the basic tests done on the Kolin machine run in reverse, there is a significant difference between the expectations and the actual ability of the machine as a refrigeration unit. Not only are the temperature reductions very small, the amount of heat lifted from the refrigerated space is much less than what is expected for the amount of power supplied to the unit, as is shown by comparison of the actual COP value to the idealized COP value.

The lack of "positive" results during the operation of the Kolin Stirling refrigerator pointed out three main faults of the system which are assumed to be, at least partially, the cause of the inoperative nature of the machine. The faults of the Kolin Stirling refrigerator were in respect to the rubber diaphragm, difficulty in sealing, and damage to the frame through wear-and-tear. Other faults were also noted, and are also described below.

6.3.1 The Rubber Diaphragm

The rubber diaphragm was used in the Kolin Stirling refrigerator so that one of the flat-plate heat exchangers (in this case the hot-side heat exchanger) could simultaneously act as the compression/expansion piston in a bellows-like manner. However, it was found that the use of the rubber diaphragm created some material problems.

A major concern with using rubber as the diaphragm material is that it is prone to cracking and tearing. This is particularly evident at the corners of the piston plate, especially during the compression stroke. Cracking and tearing of the diaphragm material is obviously detrimental to the operation of the Kolin Stirling refrigerator, as no pressure variation would take place within the enclosed space, and hence no refrigeration effect would result. Because the corners of the piston plate act as stress raisers in the rubber diagram during compression, tears in the diaphragm appeared there first. However, the dry, warm atmospheric-air promoted cracking of the diaphragm in the other exposed regions of the diaphragm as well, and so tears appeared in other areas of the diaphragm.

Another problem with the diaphragm is the tendency to "bubble" when there is a pressure difference between the enclosed space and the atmosphere. The size of the region of rubber that is exposed is a compromise between two factors: the amount of compression/expansion volume change required, and minimization of the "bubbling" effect. The amount of exposed rubber surface on the original Kolin Stirling engine was adequate to allow enough compression/expansion displacement without incurring much bubbling effect. However, that same amount of space allows too much bubbling when the machine is run (or a run is attempted) with an initial higher pressure within the enclosed space.

6.3.2 Seals

As is the case with other Stirling engine/refrigeration technology, seals seam to play an important part in the problems associated with the Kolin Stirling engine. The primary seal difficulty with the Kolin Stirling refrigerator is with respect to sealing the heat exchangers to the wooden frame. A secondary difficulty is the displacer/regenerator arm when it passes through the wood frame.

The sealing problems of the heat exchanger plates are difficult to deal with, because of the need to open the Kolin Stirling refrigerator to change system parameters (*ie.*, displacer materials), during the experimental testing of the machine. The cold side heat exchanger provided only a few problems, because it was a static seal problem. That is, once sealed, the plate/gasket does not move.

The seal situation at the hot-side (*ie.*, the piston side) created more difficulties, primarily because of the dynamic nature of this seal. Because the rubber diaphragm was itself used as the sealing gasket, the motion of the diaphragm-piston plate arrangement was in-effect detrimental to seal integrity over time. Although the gasket was sealed to the wood frame using liquid latex rubber and clamped to the frame with the aluminum face-plate, the continuing motion eventually created seal failures.

A less critical seal problem was the point at which the displacer rocker arm passes through the wood frame, connecting the displacer/regenerator on the inside of the enclosed space and the flywheel camshaft on the outside. The close-tolerance fit that is used as the sealing mechanism at this point is adequate for short time periods of low overall-pressure-difference (eg, for the non-pressurized test cases), but for larger periods of higher overall pressure difference (such as for the pressurized trials) such a system is not adequate. Continual air leakage means that the overall efficiency of the system would gradually decline, if, in fact the system had been runnable.

6.3.3 Frame Damage

A third major concern is with the use of wood as the major frame members. Although it is easy to attach the heat exchangers and diaphragm to the wood, continuously opening and resealing the unit caused wear and tear damage. This eventually lead to loss of seal-integrity as the air within the enclosed space could leak out through the wood. This was particularly noted in the wood grain direction, as shown in figure 6.1. Even if sealed on the outside, this path of leakage would greatly increase the dead space, and could be one of the causes for inoperative nature during the sandwiched connection trials. This method of leakage was only detected during the pressurized trials.

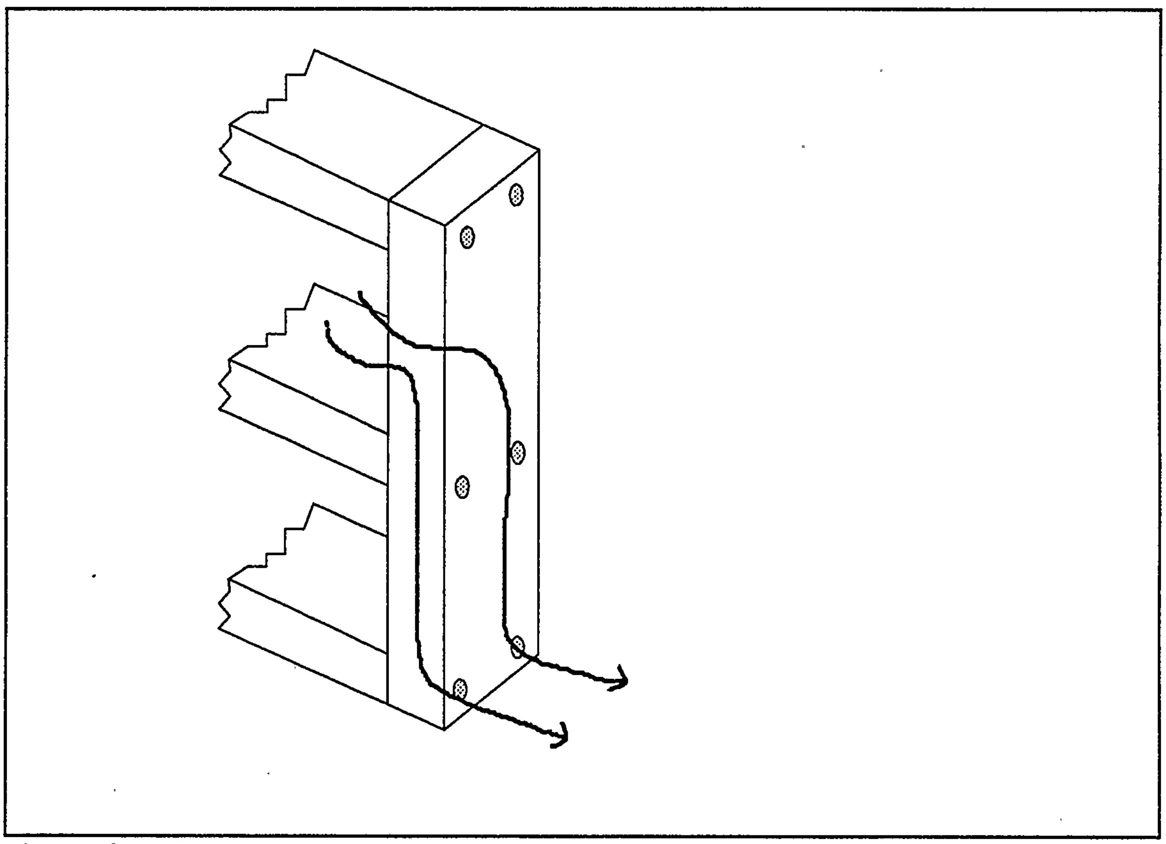

Figure 6.1 Air Leakage Through Frame

6.3.4 Other Problems

There are other problems noted from the experimental trials, particularly concerning the bearing surfaces on the linkage connections. The linkages are primarily connected using screw-type devices having a small bearing surface. Over time, these small surfaces wear away at the linkage surfaces, causing rough motion. This rough motion is probably a major cause of mechanical losses. Frequent oiling of the surfaces helped, but did not prevent such damage.

The linkages themselves are prone to some bending stresses in the weak direction. Although most of the force action is along the plane of motion, some movement perpendicular to the plane of motion, possibly caused by wearing of the bearing surfaces, eventually caused bending in the linkages.

Finally, the displacer linkage caused some problem in its motion, particularly in its loose connection at the displacer rocker arm lever. Because this connection cannot be a hard-connection, so as to allow discontinuous displacer motion, the displacer linkage slides on a bearing surface on the rocker arm lever. Coupled with the linkage's see-saw motion, this bearing surface wears down, and sometimes causes the linkage to catch on the lever — causing the machine to stop operation, as well as damaging the linkage itself. Again, frequent oiling of the bearing surface helped, but did not eliminate this problem.

6.4 RECOMMENDATIONS

The recommendations listed below are an attempt to resolve some of the problems with the Kolin Stirling refrigerator that were evident from the experimental tests. These primarily include changes to the piston/hot-heat-exchanger configuration and the pressurization problems, but include other changes as well.

6.4.1 Piston/Heat Exchanger Configuration

The first recommendation would be to move from a square design to a more conventional cylindrical design. This by itself would eliminate many of the diaphragm problems, particularly stress raisers in the corners of the diaphragm. However, such a design would result in problems of compression alignment, *ie.*, non-uniform compression piston movement, and is actually a backward step in the evolution of the Kolin machine (Kolin, 1990). The actual compression mechanism would also be more complex, as a simple hinge-type lever (or bellows-type) mechanism as used on the Kolin Stirling refrigerator would not be feasible.

A second recommendation would be to move from a rubber diaphragm piston to a flexible metal diaphragm system, similar to that used by McBride and Cooke-Yarborough (1984). Precise stress calculations would be required so as not to induce fatigue conditions in the metal diaphragm.

It may also be possible to use an articulated metal diaphragm such as that developed by Cooke-Yarborough (1985). This could allow for a discontinuous piston compression/expansion motion similar to that of the displacer. The resulting motion —

dwell periods at maximum compression and expansion, and quick travel times between the two — would theoretically result in an actual cycle even closer to that of the Kolin-Stirling cycle.

Another recommendation would be to remove the diaphragm-type piston entirely, moving to a conventional positive-displacement-type cylinder-piston arrangement. Although this again seems to be a backward step as compared to the evolution of the Kolin Stirling engine, having a separately contained compression/expansion devise would have some advantages. It would allow for more effective heat transfer, without the rubber diaphragm acting as an insulator between the hot heat exchanger plate and the working space. Also, diaphragm problems would not be evident, and sealing problems would be of a more conventional nature. On the down-side, there would be more dead space associated with the connections between the work space and the piston cylinder, and the seal problems (although now of a more conventional piston-cylinder nature) would need more attention. Depending on the location of the compression/expansion device, less mechanisms would be required than what is presently on the Kolin Stirling refrigerator.

6.4.2 Pressurization Problems

Apart from the seals problems associated with pressurization, the effects of the "bubbling" could be eliminated by using the conventional cylinder-piston type arrangement. However, other pressure-differential problems, as well diaphragm-type piston problems, could be reduced by adjoining both an engine unit and a refrigeration unit side-by-side. Because both units could be pressurized to similar pressures, the bubbling effect would be reduced as the total pressure difference across the diaphragm would be much smaller. This type of system would be similar to that proposed by Walker (1989).

6.4.3 Frame Members

For other general sealing problems, coating the wood frame with a silicone-type sealant could help to reduce leakage problems, but would not eliminate the problems of leakage through the wood due to damage/overuse. Although the general availability and inexpensiveness of the wood is a major reason for using it as the base material on the Kolin Stirling refrigerator, a plastic-type material, with good insulating properties would probably be more practical, at least during the experimental testing stage of development.

6.4.4 Other Recommendations

To reduce momentum fluctuations, a heavier flywheel could be used, but would result in a greater cumbersomeness. The reduced pressure difference across the diaphragm of a directly coupled Kolin Stirling refrigerator/engine system as discussed earlier would reduce the momentum fluctuation considerably.

Once the basic system is shown to work, heat-exchanger effectiveness could be increased by providing a greater surface area on the outer side. The most practical methods would probably be to attach fins to the flat plate heat-exchangers, or use a pre-formed plate with extruded fins.

Bearing surfaces for the displacer and piston linkages need to be lubricated quite often. A more permanent bearing surface or ball-bearing-type mechanisms would help to reduce some of the operating inefficiencies.

CHAPTER 7

7. CONCLUSIONS

The Kolin Stirling machine is a relatively new technology, and still has many wrinkles to be ironed out.

From the results of the experiments conducted, it is fairly obvious that the Kolin Stirling refrigerator is not suitable as a refrigeration unit in its present state. As compared to other Stirling refrigeration systems being developed, the Kolin system needs considerable amount of work — more testing and variations in the design are required before a more practical unit is likely to materialise.

Even after the problems in the refrigeration unit have been resolved, there still remain a number of problems that need to be addressed on the overall Kolin Refrigerator concept. The power and refrigeration units need to be designed to fit together, and operational tests need to be conducted. The control system needs to be arranged and tested on the overall system as well.

There is a definite need for low-cost, simple refrigeration units, particularly for use in developing countries. This need should be taken as an incentive for development of the Kolin Stirling refrigerator to continue.

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