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Alternative technologies for refrigeration and air conditioning applications

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Alternative technologies for refrigeration and air conditioning applications

by

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CHAPTER 1. PROJECT OVERVIEW

Introduction

Presently, the most widely used method of accomplishing cooling for for domestic, commercial, and mobile refrigeration and air conditioning applications is the vapor-compression cycle. Vapor- compression refrigeration dates back to 1834 when Jacob Perkins patented a closed cycle ice machine using ether as the refrigerant. The development of commercial vapor compression machinery continued in America, Europe and Australia between 1850 and 1870 with the principal application being ice making. In the 1890s, smaller compressors were developed, bringing about refrigeration units suitable for household use.

Refrigerants for early vapor compression systems included ammonia, carbon dioxide, ethylamine, methylamine, ethyl chloride, methyl chloride, and sulphur dioxide. The industrial and heavy commercial refrigeration industry used (and was satisfied with) ammonia. The light commercial and domestic sector used sulphur dioxide, isobutane, methylamine, ethyl chloride, and methyl chloride. When leaked, even in small concentrations, sulphur dioxide created a very irritating atmosphere capable of waking a person from a deep sleep. Isobutane was flammable though not very toxic. Ethyl and methyl chloride were toxic in concentrations as low as 2% by volume.

In the late 1920s, Frigidaire Corp., a division of General Motors, was the largest

manufacturer of light commercial and household refrigerators. Frigidaire used sulphur dioxide as a refrigerant. The nuisance caused by the escape of sulphur dioxide prompted Frigidaire to ask the GM research laboratory to develop a new refrigerant for them. In 1930, dichloromonofluoromethane (CFC-21) and dichlorodifluoromethane (CFC-12) were developed by a team led by Thomas Midgley, Jr. By the end of the decade the use of vapor-compression systems with CFC-12 as the refrigerant was common [1].

Vapor-compression technology has been developed to its present level of maturity because of CFC and HCFC refrigerants. These refrigerant compounds have excellent thermodynamic properties for cooling cycles. They are inexpensive, stable, non-toxic; and until 1974, thought to be environmentally safe.

In 1974, Rowland and Molina [2] published a paper hypothesizing the potential destruction of upper atmosphere ozone due to the release of chlorofluoromethanes. This naturally occurring ozone in the upper atmosphere serves to shield the earth's surface from ultraviolet radiation emitted from the sun. Depletion of ozone would result in additional transmittance of ultraviolet (UV) band electromagnetic radiation to the environment. Overexposure to UV radiation has been linked to skin cancer and other medical problems.

In 1987, an international conference, the "Montreal Protocol for Substances that Deplete the Ozone Layer," sponsored by the United Nations Environmental Programme (UNEP), was convened to identify which substances were harmful to the ozone layer. The resulting agreement, referred to as the Montreal Protocol, established a world wide time table for the reduction of CFCs, and an eventual ban on their production.

A subsequent UNEP sponsored conference was convened in London in 1990. Scientific findings regarding ozone depletion were reassessed. Based upon these findings the phase-out for CFCs was scheduled for the year 2000. In addition, the phase-out of hydrogenated chlorofluorocarbons (HCFCs) was discussed. To facilitate the phase-out in the United States, federal legislation was enacted. The Clean Air Act Amendment [3] was passed in November, 1990.

The Clean Air Act Amendment calls for a total production phase-out of HCFCs by the year 2030.

Based upon findings that the ozone layer was being depleted more rapidly than it was thought, President Bush announced an accelerated schedule for the reduction of ozone-depleting compounds in the United States, with the complete phase-out of CFCs by the year 1995.

Refrigeration equipment utilizing the vapor-compression cycle is capable of cooling performance which has been considered acceptable in areas where a ready supply of low-cost electricity is available. Vapor-compression machinery also has the advantages of a low first cost and high reliability as compared to other existing refrigeration methods. This is partly due to its high level of development.

Recently, two global problems have caused the engineering community to explore alternatives to vapor-compression refrigeration:

1. Global environmental changes brought about by ozone depletion in the upper atmosphere and global warming.

2. The continuing need and an increased desire for refrigeration in parts of the world where electricity is not readily available or economical.

Ozone depletion has brought about the phase-out of CFC (and eventually HCFC) refrigerants. New refrigerants such as HFC-134a are being developed as replacements.

Global warming is caused by the release of greenhouse gases into the atmosphere. It has been estimated that one-half to two-thirds of the enhancement of the greenhouse effect is expected to come from increasing concentrations of atmospheric CO_2. Since the early 19th century, there has been a 25% increase in atmospheric CO_2 with 10% of the increase occurring since 1958. Three-fourths of the total CO_2 emissions come from the combustion of fossil fuels [4]. The release of HCFC and CFC refrigerants also contributes to the greenhouse effect. These gases can have a long atmospheric lifetime and a much higher global warming potential (GWP) than CO_2.

Refrigeration systems can make two potential contributions to the greenhouse effect:

1. The direct GWP contribution results from the release of refrigerants with a high GWP into the atmosphere. The leakage can originate from shaft seals, pipe joints, and refrigerant hoses, or it can be caused by system failure, recharging, and intentional venting during repair and salvage operations.

2. The indirect GWP results from the creation of CO_2 during the combustion of fossil fuels to produce work to drive mechanical systems or to convert the fossil fuel energy to thermal energy to drive heat driven systems.

This research project takes into consideration both the direct and indirect GWP of a refrigeration system.

Project Objective

The objective of this project was to identify, analyze, and assess technologies which could serve as alternatives to vapor-compression for the purpose of accomplishing refrigeration.

Project Description

The project was conducted in three phases:

1. Identification and classification of refrigeration technologies.

2. Thermodynamic analysis of some of the more promising cycles.

3. Technical assessment of the alternative technologies.

The U.S. patents and the technical literature were used as sources for identifying the different means for accomplishing refrigeration. Once a representative group of refrigeration method concepts had been identified, a method of classifying them for thermodynamic analysis was developed.

Some of the alternative refrigeration cycles were analyzed in detail. A thermodynamic model was developed for each of these cycles and computer subroutines were written for each model. In some cases, thermodynamic property subroutines were also developed to approximate the properties of the working material used in the cycle. An interactive main program was written to allow the user to choose which cycles they wished to consider and to vary specific parameters on a case by case basis. The program was used to provide an estimate of the both the coefficient of performance (COP) and the Second Law efficiency for the cycles.

The final segment of this project was a technology assessment of refrigeration concepts. Criteria which were common to all refrigeration systems were identified. These criteria were rated on a scale of 1 (low) to 5 (high) for each technology and application category. A computer program was written to rank the refrigeration technologies from best to worst for each of the application areas.

CHAPTER 2. IDENTIFICATION OF REFRIGERATION TECHNOLOGIES

Introduction

The first phase of this project involved the identification of refrigeration technologies for the purposes of further analysis and technical assessment. A survey of U.S. Patents and the literature was conducted to discover what refrigeration methods were known to the technical community.

The starting date for the U.S. Patent survey was 1918 and continued through 1992. The 1918 date was chosen to predate the commercial introduction of household refrigerators using the vapor compression cycle, CFC refrigerants ($\approx$ 1930), and air conditioning.

A literature survey was conducted in parallel with the patent survey. The purposes of this survey were to:

- Identify additional refrigeration methods which may not have been found during the survey of U.S. patents.
- Provide additional information regarding the theory underlying a particular refrigeration method found during the patent survey.

- Provide additional information regarding hardware used to accomplish a refrigeration method found during the patent survey.
- Determine the environmental consequences related to the use of working materials commonly found in each refrigeration method found.
- Discover what alternative working materials were known for the refrigeration methods.
- Identify the potential applications for which the refrigeration method was intended.
- Identify the potential temperature lift for a single stage of a system using a particular refrigeration method.
- Determine the actual COP range which had been observed by experiment for a particular technology.
- Identify the source of major performance losses for a particular refrigeration method and the manner in which these losses could be reduced.

Sources for the literature survey included technical journals, thermodynamic texts, refrigeration trade publications, and information supplied by the Environmental Protection Agency Air and Energy Engineering Research Laboratory.

U.S. Patent Survey

Patents are assigned a number in the chronological order in which they are granted. An initial challenge in locating patents for this project was to determine which patents (by number) would be possible candidates for review.

Patents are grouped into different technological categories through a system of classes and subclasses. These classes and subclasses can be found in the *Index to the U.S. Patent Classification System.* [5] This index is an alphabetical list of technology subject headings, by indecia number, to the classification system.

The *Manual of Classification* [6] lists the numbers and brief descriptive titles (more than 100,000 in all) of the classes and subclasses. If a more definitive description of a class or subclass is required, it can be found in the *Classification Definitions.*

There are continual changes in the classification system. New classes are established as a result of advancements in science and technology. Conversely, old classes and subclasses are abolished when rendered obsolete by technological advancement [5].

Search Method for Patents Granted Prior to 1950

The initial search for refrigeration patents was done manually at the Iowa State University Library. An orderly search procedure similar to one recommended by Ardis [7] was developed. The procedure for manually locating refrigeration patents was as follows:

1. Determine the function or effect of the art or instrument to be investigated; in this case, refrigerators, coolers, and air conditioning.

2. Scan the *Index to the U.S. Patent Classification System* or the *Manual of Classification* to determine the classes which appear to describe the invention sought.

3. After determining the applicable classes, review the subclasses in the Index or Manual of Classification to determine which ones are applicable.

4. Locate the *U.S Patent Index* for the year being surveyed. Patents are listed by number within the class and subclasses. Therefore, a list can be constructed for all classes and subclasses of interest for a particular year.

5. Locate the volume of the *Official Gazette of the United States Patent Office.* [8] which contains the abstract of the patent being sought.

6. Review the patent abstract to determine the nature of the patent.

7. Accept or reject the patent based upon the information in the abstract. If the patent was accepted, a copy was ordered from the Commissioner of Patents and Trademarks. If the patent was rejected, no further action was taken.

Search Method for Patents Issued from 1950 to Present

Databases are available which contain a complete listing of all U.S. patent titles and abstracts granted from 1950 to present. The patents relating to a particular technology are located in the database by supplying the computer with a list of the appropriate class and subclass numbers.

Class and subclass numbers which had been identified during the manual patent search (for the time period from 1918 to 1950) were used as a starting point. Since classes and subclasses within the system change with time, the 1991 *Index to the U.S. Patent Classification System* was used as a reference to determine if any additional classes or subclasses which might be applicable had been added to the system. Table 2 contains a listing of the refrigeration and air conditioning classes and subclasses used for the database search.

The procedure for the database survey was:

Table 2.1: U.S. patent classes and subclasses surveyed.

Class Name	Subclass Name	Class No.	Subclass No.
Automobile	Cooler	62	243
Cooler	Air	62	404
Cooler	Air Conditioning Equipment	D23	351
Cooler	Cooling and Heating Apparatus	165	58
Cooler	Liquid	62	389
Refrigerators	Cabinet structure, Combined	312	236
Refrigerators	Car	62	239
Refrigerators	Compositions to produce, Chemical	62	532
Refrigerators	Compositions to produce, Processes combined	62	114
Refrigerators	Compositions to produce, Processes combined, Sorption type	62	114
Refrigerators	Compositions to produce, Refrigerants, Brines	252	71
Refrigerators	Compositions to produce, Refrigerants, Evaporative	252	67
Refrigerators	Design of machine	D15	81
Refrigerators	Occupant type, Vehicle	62	244

1. The database was queried using the list of patent classes and subclasses.

2. A list of patent titles (but not patent numbers) was returned along with a number which was proprietary to Dialog Database Service.

3. These patent titles were reviewed to determine the probable nature of the patent. If the patent title implicated that the patent dealt exclusively with a hardware item or other feature which was not applicable to this project, it was discarded.

4. Abstracts were purchased from Dialog Database Service for patent titles which appeared to have merit with respect to the project.

5. The abstracts were reviewed to determine the nature of the patent, as done in the "manual" search.

6. The patent was accepted or rejected based upon the information in the abstract. If the patent was accepted, a copy was ordered from the Commissioner of Patents and Trademarks. If the patent was rejected, no further action was taken.

Survey Results

In all, approximately 2140 patent titles and abstracts were surveyed. Roughly, 800 of these were from the 1918 to 1950 time period, the remainder were taken from the post-1950 group.

Many patents were rejected since they dealt with a minor hardware item, such as the door latch on a refrigerator. Some were rejected since they were granted for

a technologies which were no longer usable; ice boxes, for example. Several patents were abandon because they dealt specifically with an unacceptable refrigerant such as methyl chloride.

Many of the refrigeration concepts set forth in the remaining patents were similar. Therefore, *representative* patents were selected from the remaining group. Seventy eight patents were selected as being representative of the refrigeration technologies found during the patent survey. Three additional refrigeration concepts were found during the literature search.

Appendix A contains a list of the refrigeration concepts found during the patent and literature survey.

Once a representative sample of refrigeration technologies was found, a method of classifying them into similar thermodynamic cycles was developed. The classification procedure is presented in Chapter 3.

CHAPTER 3. CLASSIFICATION OF REFRIGERATION TECHNOLOGIES AND REFRIGERATION APPLICATIONS

Introduction

Two classification systems were developed for this project: One to classify refrigeration technologies which had been identified during the U.S. Patent and literature search; the second to define the types of applications in which the refrigeration technologies would be used.

Refrigeration Technology Classification System

During the review of the U.S. patents found during the patent survey, it was determined that the technologies tended to fall into specific groups. Within these groups the technologies had many common features. These common features were used to categorize the refrigeration technologies for the thermodynamic analysis and technical assessment phases of the project.

Figure 3.1 is a diagram illustrating the classification method developed to categorize the refrigeration technologies found during the patent and literature surveys. The number in parenthesis indicates the number of representative technologies which were placed in a particular category.

The first tier in Figure 3.1 represents the energy source used to drive the re-

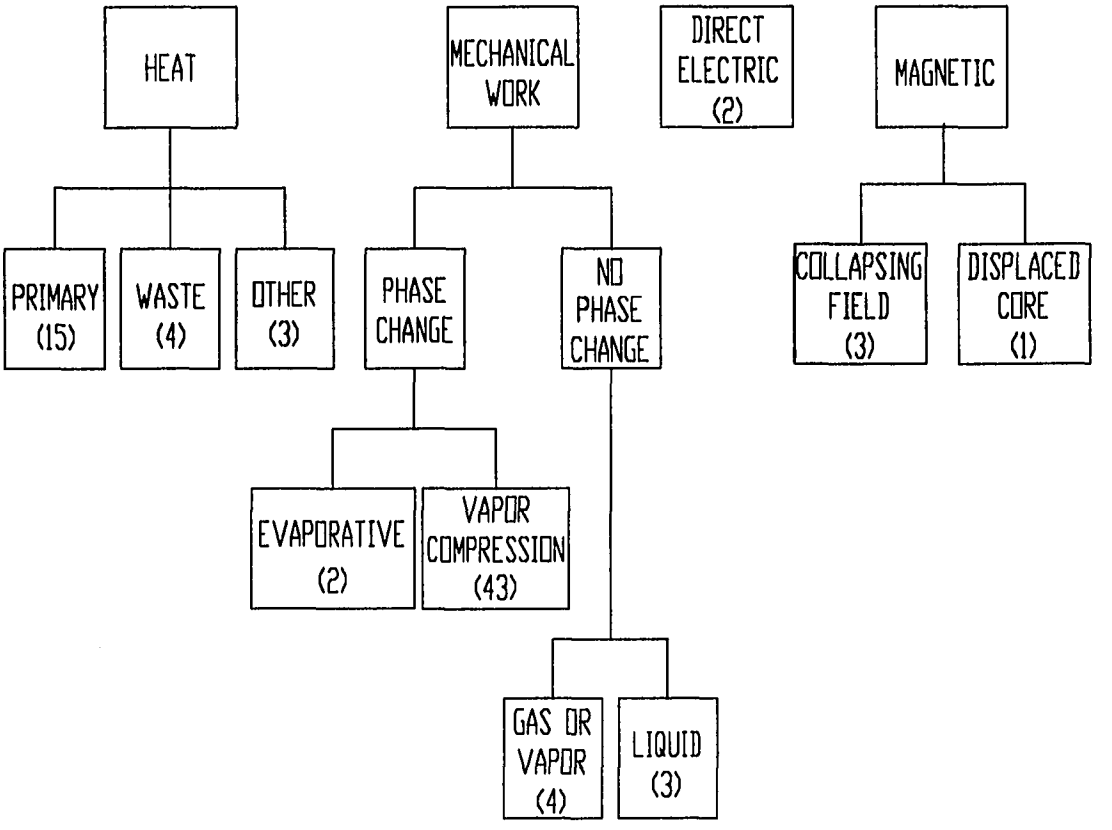

Figure 3.1: Refrigeration Technology Classification Diagram.

frigeration system. Heat, mechanical work, the direct conversion of electricity into refrigeration, and the indirect conversion of electricity into refrigeration (through electromagnetic fields) are the four sources of energy used to drive the systems.

A change in phase of the working fluid occurred in the thermodynamic cycles for all of the heat driven systems. Most of the technologies driven by mechanical work involved a phase change process during the cycle. Phase change during the cycle is important because the heat acceptance and rejection from the system can approach an isothermal process.

The mechanically driven refrigeration technologies which did not have a phase change process used either a gas or liquid as the working material.

The working materials used in all of the direct electric conversion and magnetic systems were solids which did not undergo a phase transformation during the refrigeration cycle.

The thermodynamic cycle used to accomplish refrigeration was established during the review and selection of representative patents.

Table 3.1 summarizes the thermodynamic cycles used to accomplish refrigeration in each technology category.

Table 3.1: Thermodynamic cycles used to accomplish refrigeration identified during the survey of technologies.

Ref. Technology Category	**Thermodynamic Cycle or Process**
Heat Driven	Absorption Adsorption Ejector
Mechanical Work, Phase Change	Vapor-Compression Evaporative
Mechanical Work, No Phase Change	Reversed Brayton Reversed Stirling Pulse Tube & Thermoacoustic Reversed Malone
Direct Electric	Thermoelectric
Magnetic	Magnetic

Classifying Refrigeration Applications

During the refrigeration technology identification and classification phases, no consideration had been given as to the application for which the refrigeration system would be used. The next step was to define general application areas. These application areas were selected:

1. Domestic air conditioning.
2. Commercial air conditioning.
3. Mobile air conditioning.
4. Domestic refrigeration.
5. Commercial refrigeration.

The temperatures of the thermodynamic source (from which heat is accepted) and sink (to which heat is rejected) were established for each of the applications. A search of standards and other technical literature was conducted to determine a practical set of source and sink temperatures for each application areas.

Standards to determine the performance of domestic air conditioners and refrigerators have been promulgated by the Association of Home Appliance Manufacturers (AHAM). These performance standards have been adopted by the American National Standards Institute (ANSI) to help bring about uniformity in the domestic refrigeration industry. Guidelines for testing the performance of commercial air conditioners and refrigerators are established by the Air-Conditioning and Refrigeration Institute (ARI) and the American Society of Heating, Refrigerating, and Air-Conditioning

Engineers (ASHRAE). ASHRAE has also established a standard for the environmental conditions in buildings. As with the AHAM standards, ANSI has adopted the ASHRAE standards to bring about conformity in the refrigeration industry.

Standards for establishing the performance of domestic and commercial air conditioners [10, 11, 12] all specified the temperature to which heat is rejected (sink temperature) during performance tests to be, 35 C (95 F). All three standards specified a room air temperature (source temperature) of 26.7 C (80 F) for performance testing. This temperature appeared to be too high for actual domestic and commercial air conditioning applications. Therefore, the ANSI/ASHRAE Standard for Thermal Environmental Conditions for Human Occupancy [13] was consulted. The optimum temperature for people during light, primarily sedentary activity at 50% relative humidity and mean air speed $\leq$ 0.15 $\frac{m}{s}$ was given as 24.5 C (76 F) with an acceptable temperature range of $\pm$ 1.5 C.

Multerer and Burton [14] established an interior temperature for automobiles as 24 C for a study of alternative automotive air conditioning systems.

For domestic refrigerators, AHAM lists three ambient temperatures for testing refrigerators, freezers, and refrigerator-freezers [15]:

- 21.1 C (70 F)
- 32.2 C (90 F)
- and 43.3 C (110 F)

AHAM also recommended an average freezer compartment temperature of -17.8 C (0 F).

ARI Standard 420 specifies an ambient temperature of 35 C for performance tests. Four groups were established by the ARI for performance the testing of commercial refrigerators. Each group corresponds to a cooling space temperature for different commercial refrigeration applications. These temperatures are given in Table 3.2.

Based upon this survey a set of source and sink temperatures has been established for each of the five application categories. Table 3.2 is a summary of the five refrigeration application categories and the source and sink temperatures to be used for comparing refrigeration technologies in each category.

Table 3.2: Thermal source and sink temperatures for the five refrigeration application categories.

Ref. Application Category	Source Temp. (C)	Sink Temp. (C)
Domestic Air Conditioning	25.0	35.0
Commercial Air Conditioning	25.0	35.0
Mobile Air Conditioning	25.0	35.0
Domestic Refrigeration	−18.0	35.0
Commercial Refrigeration		
ARI Group I	2.8	35.0
ARI Group II	1.7	35.0
ARI Group III	−2.2	35.0
ARI Group IV	−23.3	35.0

CHAPTER 4. COMPARING THE PERFORMANCE OF REFRIGERATION SYSTEMS

Introduction

In this chapter, methods of comparing the performance of refrigeration systems will be discussed.

The refrigeration technologies found during the survey use mechanical work, heat transfer, and electricity to drive them. The cycle efficiency was used to compare the relative performance of the different technologies.

The Clausius statement of the second law of thermodynamics is: " It is impossible to construct a device that operates in a cycle and produces no effect other than the transfer of heat from a colder to a hotter body." For refrigeration systems, the Clausius statement implies that a system that accomplishing the transfer of heat from a cooler *source* to a hotter *sink* requires the input of additional work or energy to cause the temperature lift.

Coefficient of Performance

The performance of refrigeration and air conditioning systems is the ratio of the amount of heat accepted from the cooling space to the amount of heat or work

required to drive the refrigeration system. This ratio is known as the coefficient of performance (COP).

Figure 4.1 illustrates a refrigeration system communicating with two thermal reservoirs: The source, at temperature T_L, and the sink, at temperature T_H. The refrigeration system is driven by mechanical work. The COP is defined as:

$$COP_w = \frac{Q_L}{W_{in}} \tag{4.1}$$

where,

COP_w = The coefficient of performance for a work driven system.

Q_L = The amount of heat accepted from the source reservoir.

W_{in} = The amount of work input from an external system.

Heat driven refrigeration systems can be considered as two systems: a refrigeration system driven by a heat engine. Figure 4.2 is a schematic of the two systems. Three thermal reservoirs at three different temperatures are required.

The heat engine accepts heat from a high-temperature reservoir at temperature $T_{Gen.}$ and rejects heat to the sink reservoir at temperature T_H producing work. The refrigeration system accepts heat from the source reservoir at temperature T_L and rejects heat to the sink reservoir. The net work from the heat engine is used to drive the refrigeration system.

The COP for the refrigeration system is given by Equation 4.1.

The thermal efficiency of the heat engine can be defined as,

$$\eta_{th} = \frac{W}{Q_{Gen.}} \tag{4.2}$$

where,

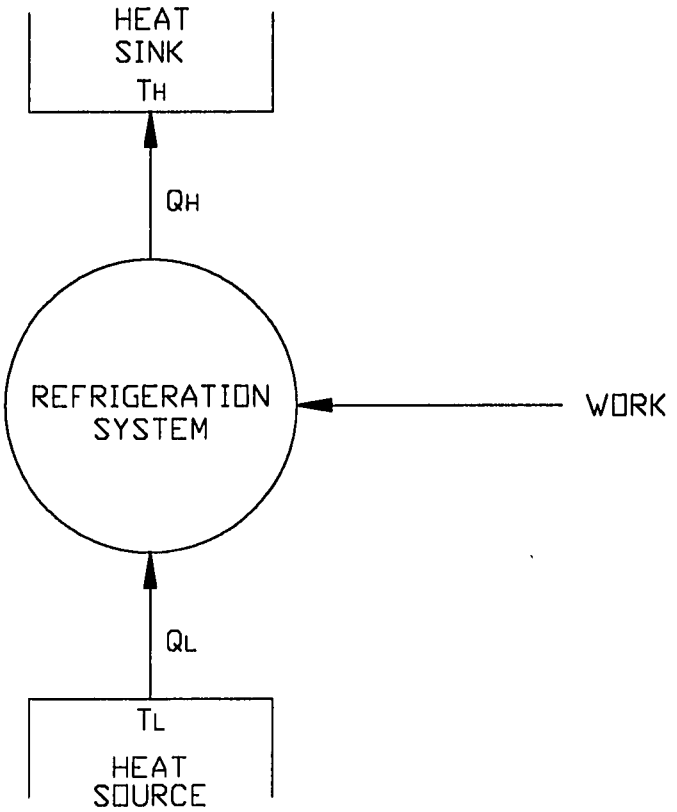

Figure 4.1: Schematic of a refrigeration system driven by mechanical work.

W = The net work output of the heat engine.

$Q_{Gen.}$ = The heat transferred to the heat engine.

The COP for heat driven systems is then,

$$COP_h = \eta_{th} \cdot COP_w \tag{4.3}$$

$$= \frac{Q_L}{Q_{Gen.}} \tag{4.4}$$

Refrigeration systems with high COPs are desirable because they are less expen-

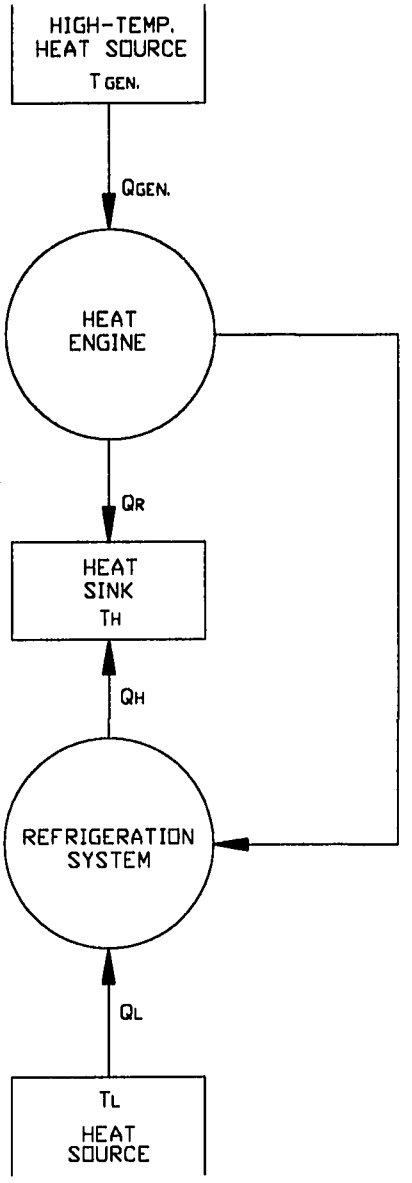

Figure 4.2: Schematic of a refrigeration system driven by heat transfer.

sive to operate, and they have a lower indirect GWP since less fuel must be burned to operate them.

Ideal COP

An ideal refrigeration system would transfer heat reversibly between the source and sink. The COP for a reversible refrigeration system would be the highest theoretically possible. Using the definition of the Kelvin temperature scale, it can be shown that the ideal COP for work driven refrigeration cycles can be expressed as ratios of the absolute temperatures of the source and sink reservoirs [17]. The ideal COP (COP_{iw}) for a work driven system is [18],

$$COP_{iw} = \frac{T_L}{T_H - T_L}. \tag{4.5}$$

For heat driven systems, the ideal COP (COP_{ih}) would be that of a reversible heat engine driving a reversible refrigerator. It can be expressed as ratios of the absolute temperatures of the high-temperature reservoir, sink, and source. Using Equation 4.2 and the definition of the thermal efficiency for an ideal heat engine, an expression involving absolute temperatures can be written for the ideal heat driven refrigeration system:

$$COP_{ih} = \left(\frac{T_{Gen.} - T_H}{T_{Gen.}}\right)\left(\frac{T_L}{T_H - T_L}\right). \tag{4.6}$$

While reversible operation (and thus the ideal COP) are not possible for actual refrigeration systems, it can be shown as a corollary to the second law of thermodynamics that any two reversible refrigeration cycles accepting and rejecting heat at two particular temperature levels must have the same COP. Therefore, the ideal

COP can be used as a standard of comparison for the performance of modeled and actual refrigeration systems.

Modeled COP

The COP of refrigeration systems can be estimated through modeling of the thermodynamic cycle with which the system operates. The models attempt to account for some of the irreversibilities which occur in actual systems.

There are different levels of sophistication in thermodynamic models, ranging from simply multiplying the ideal COP by an efficiency through multi-dimensional transient models in which the energy, momentum, and continuity equations are simultaneously solved for elemental control volumes or mass elements throughout the system. As the level of sophistication of the model increases, so does the amount of information which must be known or assumed about the system.

For this project, models were constructed in which the thermodynamic state was determined at the end of each process composing the cycle. Where possible, component efficiencies were accounted for. The properties defining the thermodynamic state were determined by using property routines.

Actual COP

The actual COP is that of an actual refrigeration system as is determined through experiments conducted using laboratory refrigeration systems or production systems. For work driven systems the COP is calculated using Equation 4.1, while for heat driven systems it is calculated using Equation 4.3.

Cycle Efficiency

The cycle efficiency, η_{Cycle}, will be used to examine the efficiency of a refrigeration technology operating at different source temperatures and operating conditions. It will also be used to compare the relative efficiencies of different technologies at the same source temperature. The cycle efficiency is defined as,

$$\eta_{Cycle} = \frac{COP}{COP_{Carnot}}. \tag{4.7}$$

CHAPTER 5. REVERSED BRAYTON REFRIGERATION

Introduction

Refrigeration can be accomplished by employing a gas cycle rather than a vapor cycle in which the working fluid undergoes changes from the liquid phase to the vapor phase and vice versa. Gas refrigeration cycles include the reversed Brayton, Stirling, and pulse-type cycles including the pulse tube developed by Gifford and Longsworth [50] and thermoacoustic devices which have been studied by Hoffler [57]. In this chapter the reversed Brayton cycle, with and without a regenerator, will be examined.

The refrigeration effect per unit mass of fluid circulated in a vapor-compression cycle is equivalent to a large fraction of the enthalpy of vaporization. In contrast, the refrigeration effect in a gas cycle is the product of the temperature rise of the gas passing through the low-temperature heat exchanger and the constant-pressure specific heat of the gas. Therefore, as compared to the vapor-compression cycle, a larger mass flow rate is required in the gas cycle to produce the same amount of heat removal from a space.

Gas-cycle refrigeration can be designed and operated either as an open or closed system. In open systems the gas, commonly air, is expanded into the space to be cooled which is at atmospheric pressure, and then exhausted or re-compressed. Open

systems often require dehumidification of the air prior to expansion to prevent ice formation at the low-temperature points of the system. Open-cycle air systems have become a common method of space conditioning in aircraft. The principal advantages of the open-cycle air system in aircraft applications are:

1. Pressurization of the cabin may be required.
2. Ventilation air is required in the aircraft cabin.
3. Compressed air is available and is a small fraction of the air compressed in the aircraft engine compressor section.
4. Cool ambient air is available for cooling the compressed air.

In closed-cycle gas refrigeration systems, the refrigerant gas is contained in the piping and component parts of the system at all times. Furthermore, the low-temperature heat exchanger is maintained at pressures above atmospheric. Historically, the term "dense-air system" was derived from the higher pressures maintained in the closed system as compared to the open system [19].

History

Open- and closed-cycle gas refrigeration systems using air as the refrigerant were some of the earliest mechanical refrigeration means dating back to 1834 [23]. The first commercial air cycle machine was an open-cycle machine introduced by Franz Windhausen in 1889 [20]. The primary application for the Windhausen system was cold-storage and space conditioning aboard ships. Air-cycle refrigeration machinery was favored by the shipping industry because ammonia or carbonic acid used in

other refrigeration cycles of that era was unavailable in many ports of call. Air-cycle refrigeration was also used in other commercial applications as land-based cold-storage and theater cooling. Another advantage of the air system was a completely safe and inexpensive refrigerant [19].

The principal objections to the Windhausen open-cycle design were directly related to moisture in the air which created the need for increased maintenance of the machinery and frost contamination of the cold-storage cargo. The Allen dense-air system, incorporating a closed-air system operating at a low pressure of 60 to 70 psig and a pressure ratio of three or four, was adopted to solve the moisture-related problems.

The introduction of CFC refrigerants removed the safety and refrigerant cost advantages of air-cycle refrigeration machines and vapor-compression systems were favored due to their higher efficiencies and compactness. The vapor-compression system was inherently more adaptable to different cooling applications.

U.S. Patent Search

A U.S. patent search was conducted to discover different gas-cycle technologies for refrigeration applications. One patent was discovered for an air reversed-Brayton cycle machine.

U. S. Patent number 1,295,724 was issued February 25, 1919 to Julius Frankenberg for an "Air-Refrigerating Machine" [21]. This machine was a unitized compressor, expander, and high-temperature system. It incorporated rotary compressor and expander sections connected by a common shaft. A water-cooled heat exchanger was mounted above the compressor/expander unit to cool the air between the compressor

and expander stages. No claim for a low-temperature heat exchanger or regenerator was made in the patent.

Literature Review

The technical literature was reviewed to determine what present research has been conducted to develop reversed Brayton or modified reversed Brayton refrigeration cycles.

Kauffeld et al. [22] investigated the reversed Brayton cycle as a replacement for vapor-compression in refrigeration and air conditioning applications. An analysis of fifteen variations of the reversed Brayton cycle was conducted. The variations included:

- open cycle,
- regeneration,
- and two-stage compression with intercooling.

Calculated coefficients of performance from 0.6 to 1.16 were reported assuming an ambient temperature of 30 C, a room entry temperature of 5 C, and isentropic efficiencies of the expansion and compression devices of 80%.

Open-cycle test apparatus with single- and two-stage compression were constructed and evaluated. Measured COPs of up to 0.45 were reported. Problems with moisture removal, oil odor, and noise were also reported.

Henatsch and Zeller [23] thermodynamically modeled the Joule (reversed Brayton) process and a modified Joule-Ericsson process including the effects of regeneration. The model included adiabatic two-stage compression with intercooling.

As part of the study, an earlier investigation comparing the isentropic efficiencies to volumetric flow rates of commercially available turbines, radial flow compressors and dry-type screw compressors by Henatsch was incorporated. Efficiencies on the order of 88% were noted for large displacement turbines and radial flow compressors and 80% for large radial flow compressors.

For a non-regenerative open cycle, coefficients of performance ranging from 0.61 to 0.77 were noted for mass flow rates of 0.10 and 0.35 $\frac{kg}{s}$ respectively, an ambient temperature of 42 C, and a temperature ratio of 1.1.

Thermodynamic Model

Introduction

Thermodynamic models for the non-regenerative and regenerative reversed Brayton cycles were constructed and programmed in FORTRAN for analysis on an IBM compatible personal computer. A subroutine to calculate the thermodynamic properties of the air was also developed.

Non-Regenerative Reversed Brayton Cycle

The ideal thermodynamic model of the reversed Brayton cycle includes two isentropic and two isobaric processes [25]. Since the actual compression and expansion processes are irreversible, provisions were made in the model to allow for and vary the degree of irreversibility using isentropic compressor and expander efficiencies.

Figures 5.1 and 5.2 are the schematic and temperature vs. entropy diagrams for a non-regenerative reversed Brayton cycle.

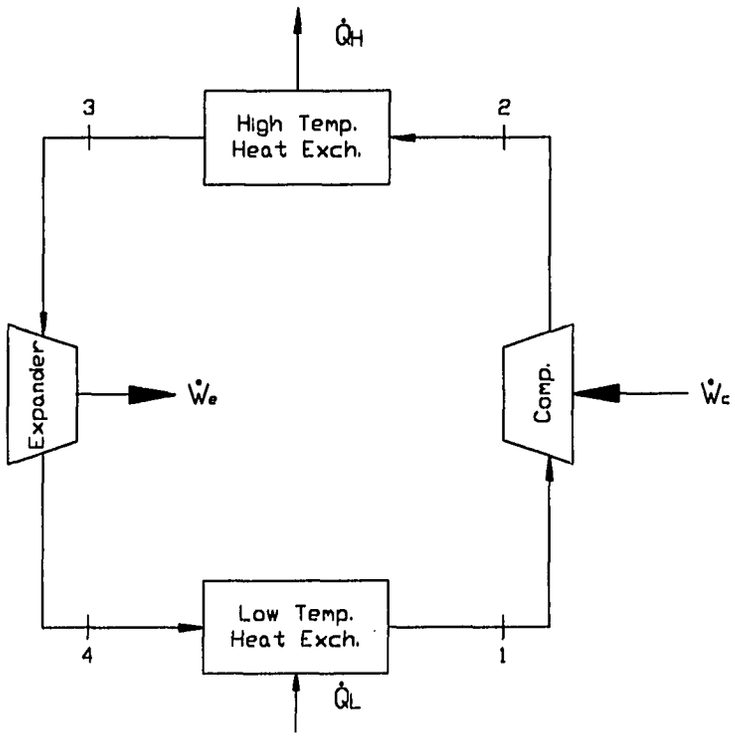

Figure 5.1: Schematic of a non-regenerative reversed Brayton cycle.

The gas exiting the low-temperature heat exchanger undergoes a compression process from state 1 to state 2. The reversible process is illustrated as a solid line from state 1 to state 2s, the irreversible process is illustrated as a dashed line from state 1 to state 2 since the specific path during the irreversible compression process is unknown. Heat is rejected at constant pressure to the environment from the compressed gas through a high-pressure heat exchanger (state 2 to state 3). The gas is then expanded through an expander (commonly a turbine) from state 3 to state 4. The unknown path of the irreversible expansion process is again illustrated as a dashed line in Figure 5.2. Heat is removed at constant pressure from the space to be

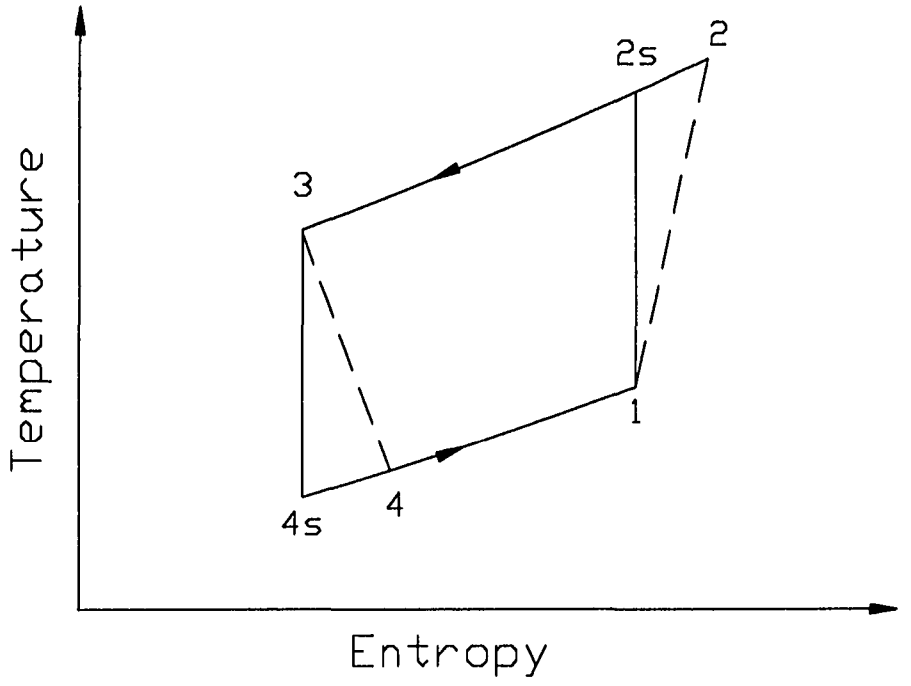

Figure 5.2: Temperature vs. entropy diagram for a non-regenerative reversed Brayton cycle.

cooled through the low-pressure heat exchanger (state 4 to state 1), completing the cycle.

The following assumptions were made to simplify the model:

1. Steady state operation.
2. Adiabatic compression.
3. Adiabatic expansion.
4. Negligible changes in potential and kinetic energy of the fluid.

5. Negligible pressure drop through the heat exchangers and related piping (no fluid friction).
6. Ideal gas with variable specific heats.
7. The heat exchangers were of types and sizes to permit the exiting gas to approach the source and sink temperatures by a fixed temperature difference.

Therefore, the temperatures of the air at states 1 and 3 can be written as,

$$T_1 = T_{low} - \delta T \tag{5.1}$$

$$T_3 = T_{high} + \delta T. \tag{5.2}$$

The isentropic compressor (η_c) and expander (η_e) efficiencies were defined as [24],

$$\eta_c = \frac{\left(\frac{\dot{W}_c}{\dot{m}}\right)_s}{\frac{\dot{W}_c}{\dot{m}}} \tag{5.3}$$

$$\eta_e = \frac{\frac{\dot{W}_c}{\dot{m}}}{\left(\frac{\dot{W}_c}{\dot{m}}\right)_s}. \tag{5.4}$$

Applying an energy balance to each of the four system components in the cycle, an expression can be derived for the coefficient of performance (COP):

$$COP = \frac{\frac{\dot{Q}_{in}}{\dot{m}}}{\frac{\dot{W}_{Total}}{\dot{m}}} \tag{5.5}$$

$$= \frac{\dot{Q}_{in}}{\left(\dot{W}_c - \dot{W}_e\right)} \tag{5.6}$$

$$= \frac{(h_1 - h_4)}{[(h_2 - h_1) - (h_3 - h_4)]}. \tag{5.7}$$

Regenerative Reversed Brayton Cycle

The regenerative reversed Brayton cycle includes an added heat exchanger to cool the gas entering the expander below the ambient temperature [24]. Figures 5.3 and 5.4 are the schematic- and temperature vs. entropy diagrams for a regenerative reversed Brayton cycle. Gas exiting the low-pressure heat exchanger (state B)

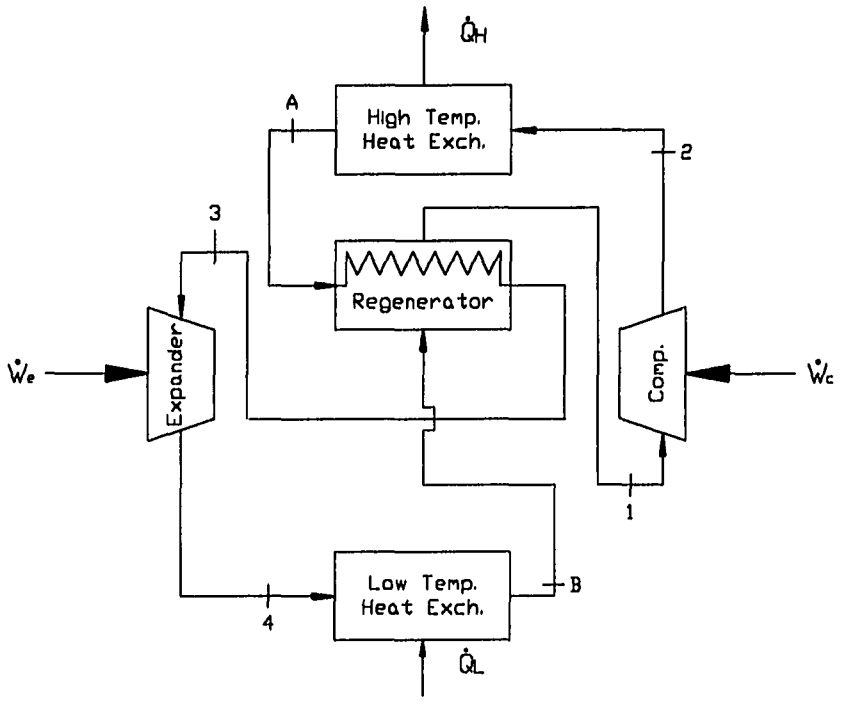

Figure 5.3: Schematic of the regenerative reversed Brayton cycle.

enters the regenerator and cools the gas exiting the high-pressure heat exchanger (state A). The remainder of the cycle is similar in operation to the non-regenerative cycle.

The following assumptions were made to simplify the model:

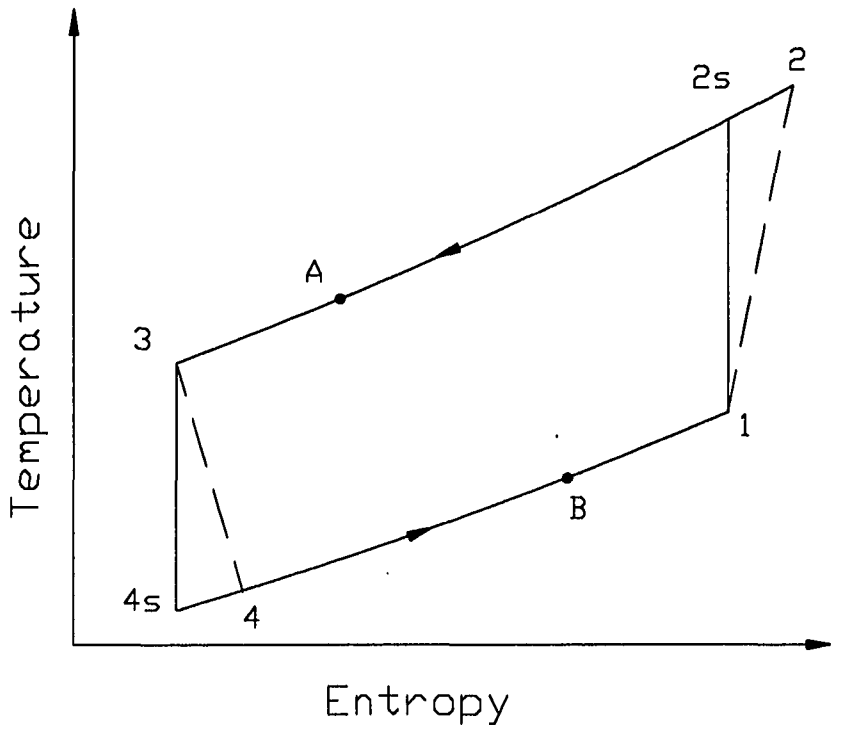

Figure 5.4: Temperature vs. entropy diagram for the regenerative reversed Brayton cycle.

1. steady state operation.
2. adiabatic compression.
3. adiabatic expansion.
4. negligible changes in potential and kinetic energy of the fluid
5. negligible pressure drop through the heat exchangers and related piping (no fluid friction).
6. the working fluid was an ideal gas with variable specific heats.
7. and the regenerator operates adiabatically.

As with the non-regenerative cycle, it was assumed that the heat exchangers were of types and sizes to permit the exiting gas to approach the source and sink

temperatures by a fixed temperature difference. Therefore, the temperatures at which heat is transferred to and from the working fluid in the system can be expressed as,

$$T_B = T_{low} - \delta T \tag{5.8}$$

$$T_A = T_{high} + \delta T. \tag{5.9}$$

The regenerator effectiveness was defined as the ratio of the amount of heat transferred from the high-pressure gas in the regenerator to the amount of heat which would be transferred if reversible regeneration occurred. The regenerator effectiveness can be expressed as,

$$\eta_r = \frac{\left(h_A - h_3\right)}{\left(h_A - h_B\right)}. \tag{5.10}$$

Applying an energy balance to each of the system components in the cycle, an expression can be derived for the COP for the regenerative reversed Brayton cycle,

$$COP = \frac{\frac{\dot{Q}_in}{m}}{\frac{\dot{W}_{Total}}{m}} \tag{5.11}$$

$$= \frac{\dot{Q}_{in}}{\left(\dot{W}_c - \dot{W}_e\right)} \tag{5.12}$$

$$= \frac{\left(h_B - h_4\right)}{[(h_2 - h_1) - (h_3 - h_4)]}. \tag{5.13}$$

Thermodynamic Properties

A FORTRAN subroutine was written to determine the specific properties of the working gas. Given two properties to fix a thermodynamic state, the subroutine finds the remaining properties for that state and inputs them into the cycle model.

It was assumed that the gas behaved ideally in the temperature and pressure range over which the cycles operated. The constant pressure specific heat ($C_p(T)$) was variable over the temperature range of -73 C to 827 C.

A functional relationship was found for $C_p(T)$ data published by Reynolds [26]. The enthalpy is found by integrating the constant pressure specific heat function directly,

$$h(T) - h_0 = \int_{T_0}^{T} C_p(T)dT. \tag{5.14}$$

The internal energy is found from the definition of enthalpy and the ideal gas equation of state,

$$u(T) = h(T) - RT. \tag{5.15}$$

The entropy can also be found from the specific heat function and the pressure ratio,

$$s(T) - s_0 = \int_{T_0}^{T} \frac{C_p(T)}{T} dT - R \ln\left(\frac{p}{p_0}\right). \tag{5.16}$$

The reference state for determining the properties was established as -273.15 C and 101.325 kPa. The thermodynamic property code is given in Appendix B.

Results

The coefficients of performance and cycle efficiencies were calculated for three cases:

1. The ideal case in which the heat was transferred reversibly (no heat exchanger δT). The compression and expansion were isentropic. For regenerative cycles, the regenerator effectiveness was 1.0

2. The actual case which used estimated compressor and expander isentropic efficiencies and regenerator effectiveness. A minimum approach temperature of 5 C was interposed between the sink and high-temperature heat exchanger and the source and low-temperature heat exchanger to account for irreversibilities due to heat transfer.

3. The "best possible" case in which higher compressor and expander isentropic efficiencies and a larger regenerator effectiveness were chosen. These values were estimates of the upper limit for component efficiencies in the future, given further technological development in the turbomachinery and heat exchanger industries.

Table 5.1 summarizes the parameters for the actual, best possible and ideal cases.

Table 5.1: Parameter values for the actual, best possible and ideal reversed Brayton and regenerative reversed Brayton model case study.

Case	$\eta_{Comp.}$	$\eta_{Exp.}$	$\eta_{Regen.}$	HX ΔT (C)
Ideal	1.0	1.0	1.0	0
Best Possible	0.95	0.95	0.95	5
Actual	0.85	0.85	0.85	5

Each case was modeled at several different pressure ratios to establish trends for the COP and cycle efficiency.

Figure 5.6 is a graph of the COP versus source temperature for an ideal reversed Brayton refrigeration system using air as the working gas. The model parameters are given in Table 5.1 for the ideal case. The COP was highest for the pressure ratio of 2.5; increasing the pressure ratio resulted in lower COPs at the same source temperature.

The COP remained constant over the entire range of source temperatures from −24 C to 28 C. Figure 5.6 is a graph of the cycle efficiency versus source temperature for the ideal reversed Brayton system. For each pressure ratio, the highest cycle efficiencies

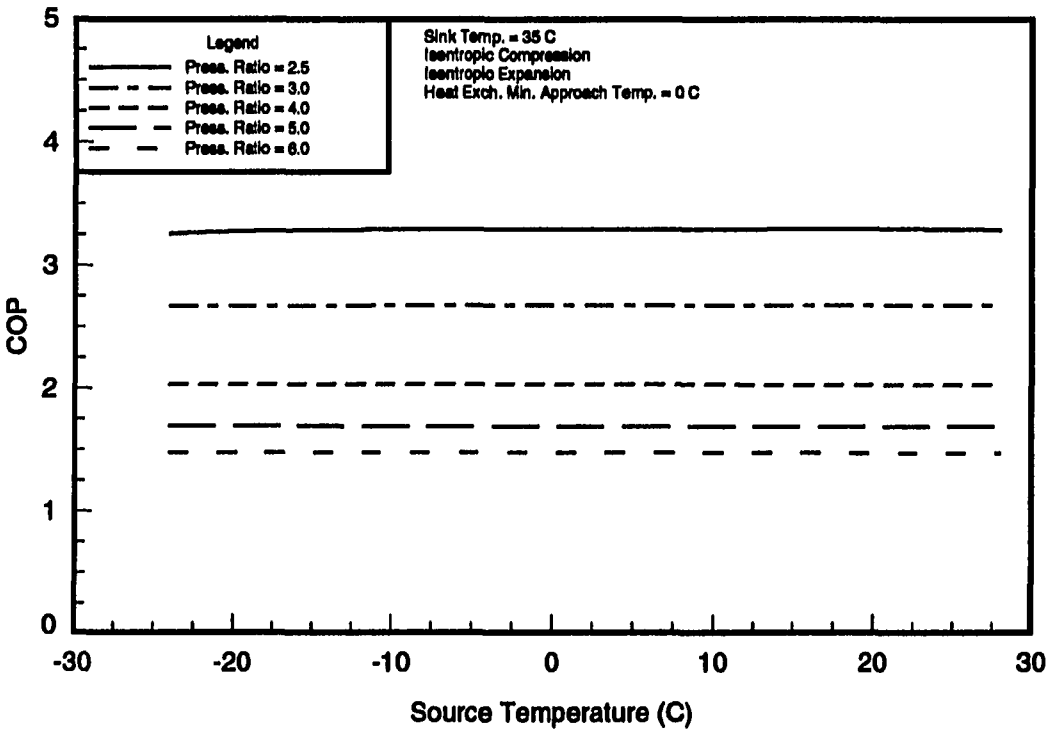

Figure 5.5: COP vs. source temperature for an ideal reversed Brayton refrigeration system.

occurred at low source temperatures and decreased linearly with increasing source temperature.

The COP versus source temperature graphs for the "best possible" reversed Brayton cycle case are presented in Figure 5.7. At source temperatures above 20 C, the highest COPs were produced by cycles operating at pressure ratios below 3.0. The lowest source temperature achieved by the cycle operating at a pressure ratio of

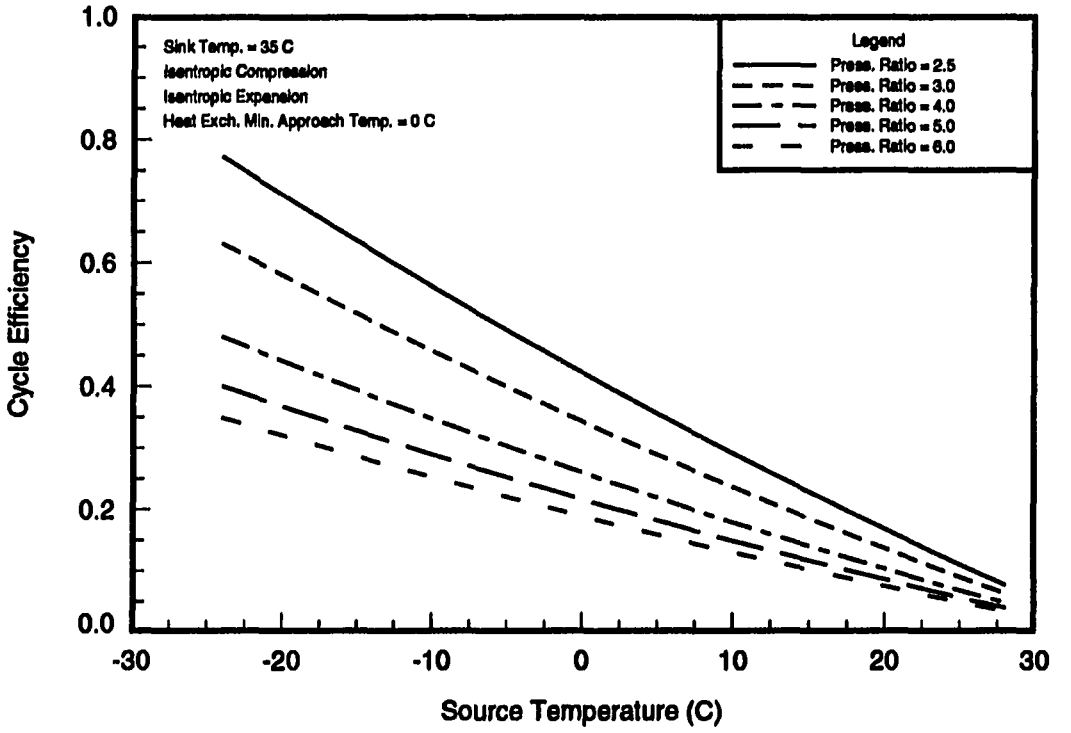

Figure 5.6: Cycle efficiency vs. source temperature for an ideal reversed Brayton refrigeration system.

1.5 was 16 C. A pressure ratio of 3.0 was required to accomplish the temperature lift over the entire range of source temperatures. As the pressure ratio was increased, the slope of the lines decreased until a relatively constant COP of approximately 1.0 was noted over the entire source temperature range, at a pressure ratio of 6.0. The cycle efficiencies for this case are shown in Figure 5.8. For pressure ratios of 3.0 and below, the cycle efficiency vs. source temperature line was parabolic. A maximum cycle efficiency which occurred at a specific source temperature was observed. This indicates that different optimum pressure ratios exist for reversed Brayton systems operating at a fixed sink temperature, but different source temperatures. For a given

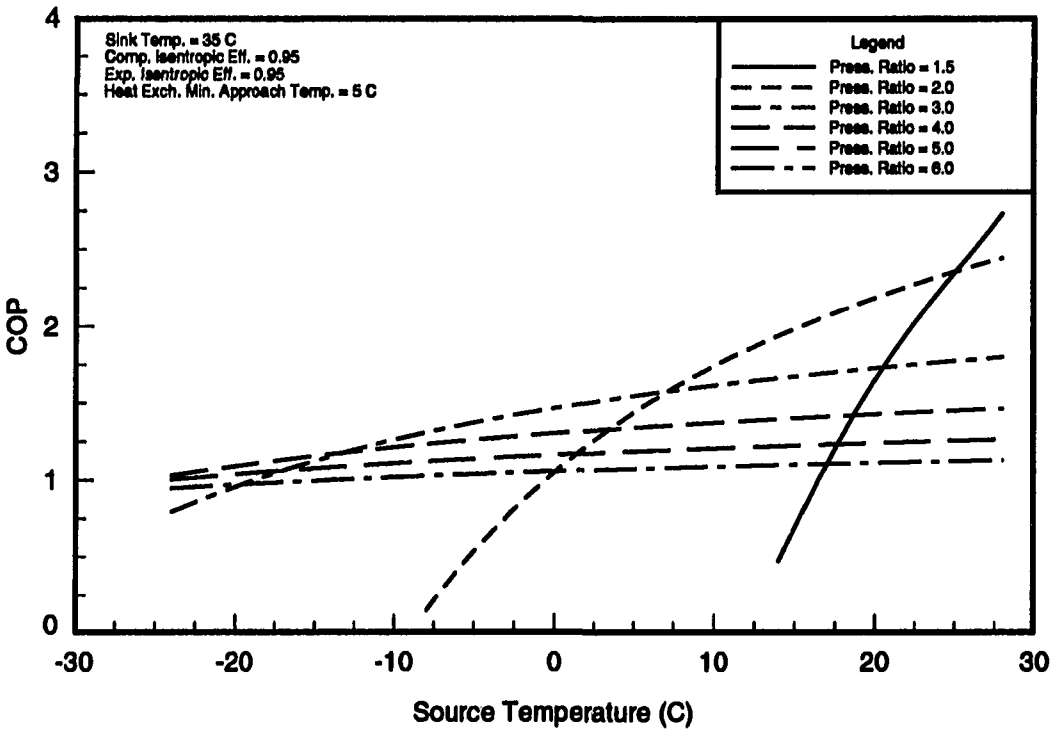

Figure 5.7: COP vs. source temperature for a reversed Brayton refrigeration system operating with parameters given in the "best possible" case.

pressure ratio, the range of source temperatures at which the system will operate at near-maximum cycle efficiency is narrow.

Figure 5.9 is a graph of COP as compared to source temperature for the actual reversed Brayton cycle model case. For low pressure ratios, the COP decreased rapidly with decreasing source temperature. As the pressure ratio was increased, the slope of the COP versus source temperature line decreased. At source temperatures below 0 C, the COP increased with the pressure ratio for a given source temperature.. Above 0 C, the COP increased with *decreasing* pressure ratio at a fixed source

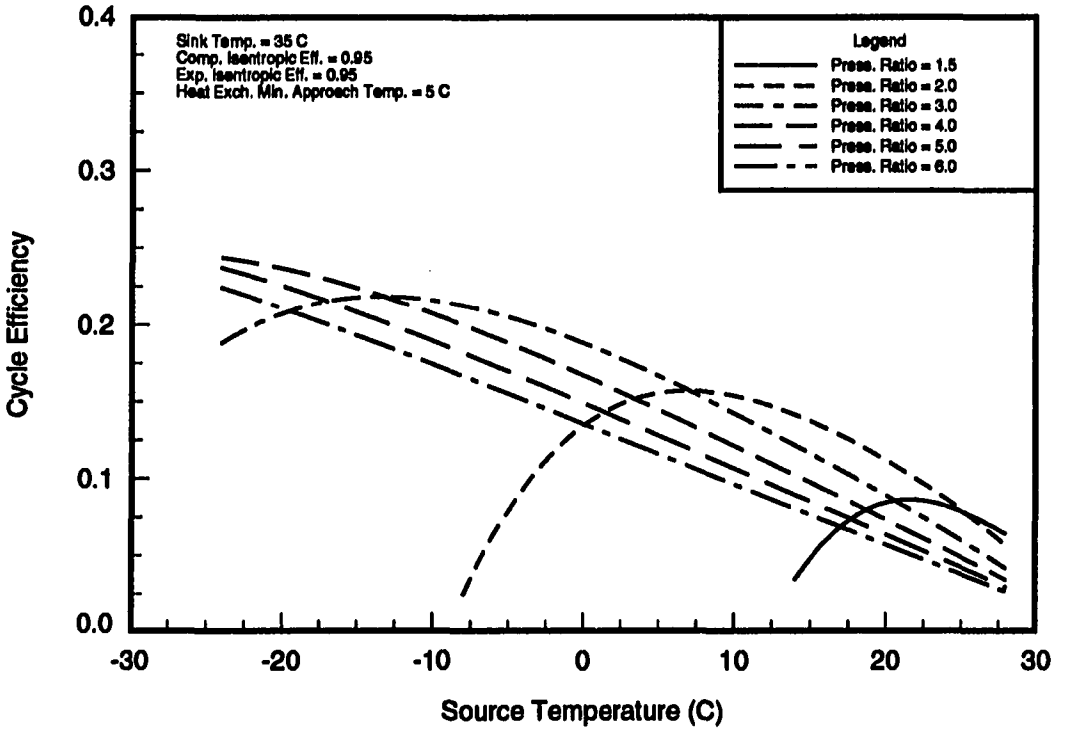

Figure 5.8: Cycle efficiency vs. source temperature for a reversed Brayton refrigeration system operating with parameters given in the "best possible" case.

temperature.

The trend of parabolic cycle efficiency versus source temperature lines was evident at higher pressure ratios than for the previous (best possible) case. Decreasing the isentropic compressor and isentropic expander efficiencies from 0.95 to 0.85 resulted in a lower maximum cycle efficiency for a given pressure ratio. Also, the maximum efficiency occurred at a higher source temperature than for the ideal or beat possible cases.

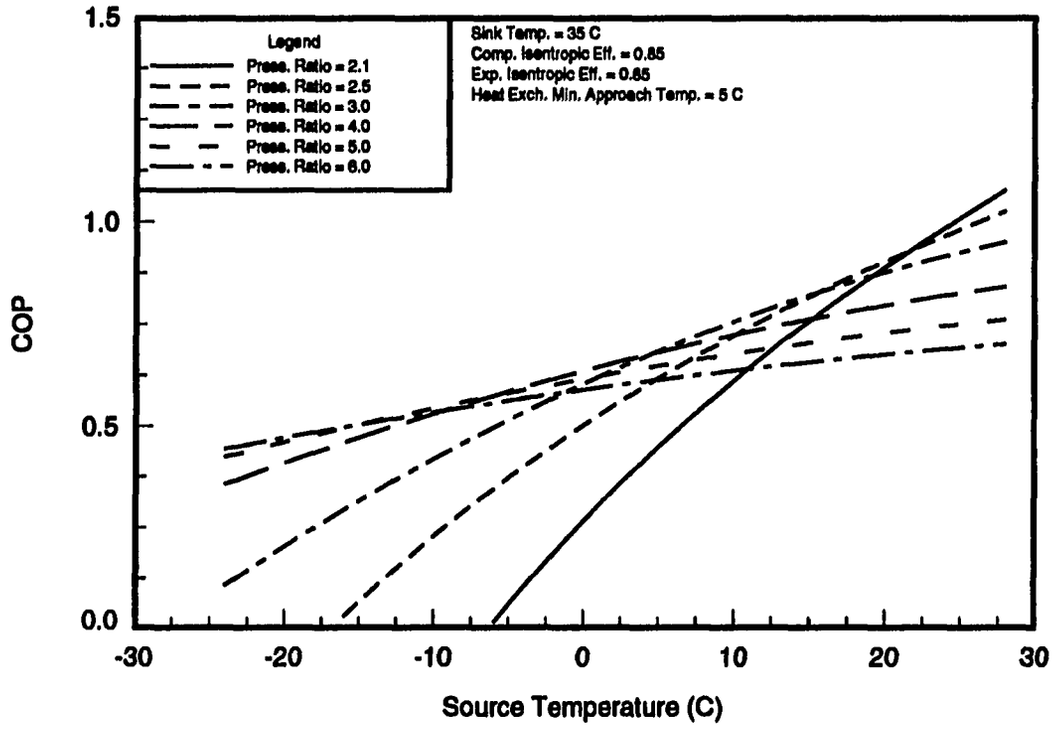

Figure 5.9: COP vs. source temperature for a reversed Brayton refrigeration system operating with parameters given in the actual case.

Figures 5.11 and 5.12 are graphs of the COP versus source temperature and cycle efficiency versus source temperature for an ideal regenerative reversed Brayton cycle.

The COP is higher at the upper end of the source temperature range than it was for the ideal non-regenerative cycle. At −24 C, the cycle efficiency was approximately the same as it was for the ideal reversed Brayton cycle without regeneration (Figure 5.6). At higher source temperatures, the cycle efficiencies were higher. Also, the highest efficiencies occurred at lower pressure ratios than for the ideal non-regenerative cycle.

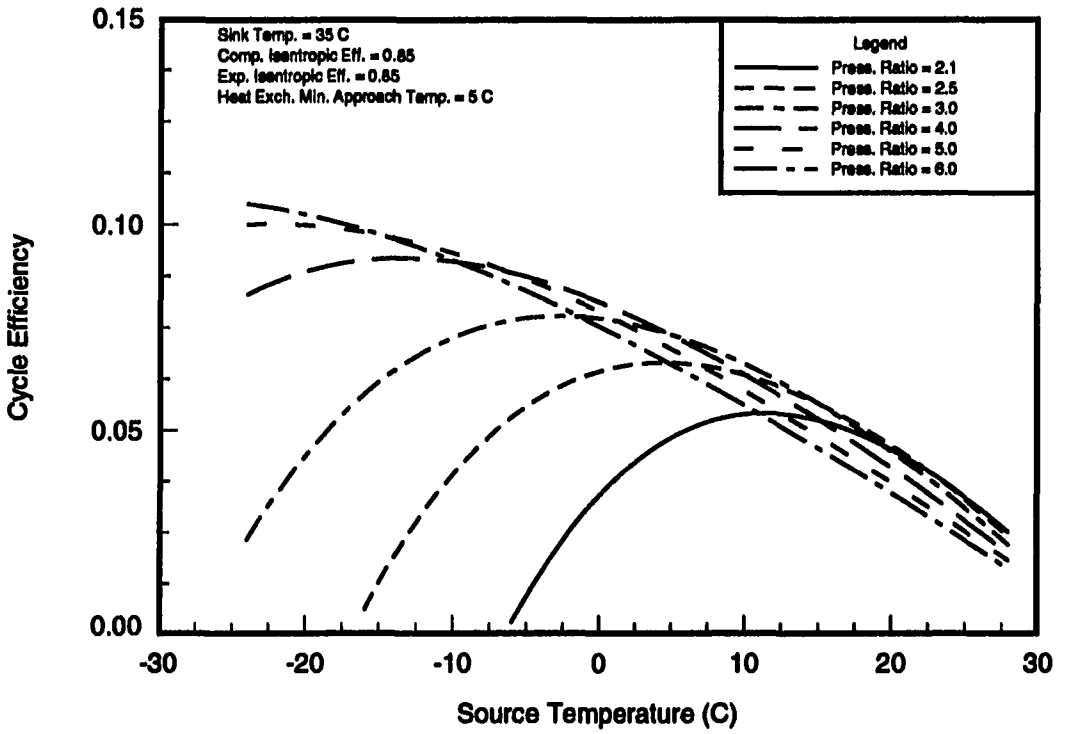

Figure 5.10: Cycle efficiency vs. source temperature for a reversed Brayton refrigeration system operating with parameters given in the actual case.

The COP and cycle efficiency versus source temperature for the "best possible" regenerative reversed Brayton cycle are shown in Figures 5.13 and 5.14. In contrast to the non-regenerative "best possible" case, the temperature lift could be accomplished over the entire source temperature range. The COP increased with decreasing source temperature. The cycle efficiencies increase with decreasing source temperature. For all pressure ratios, the maximum cycle efficiency occurred at a source temperature below −24 C.

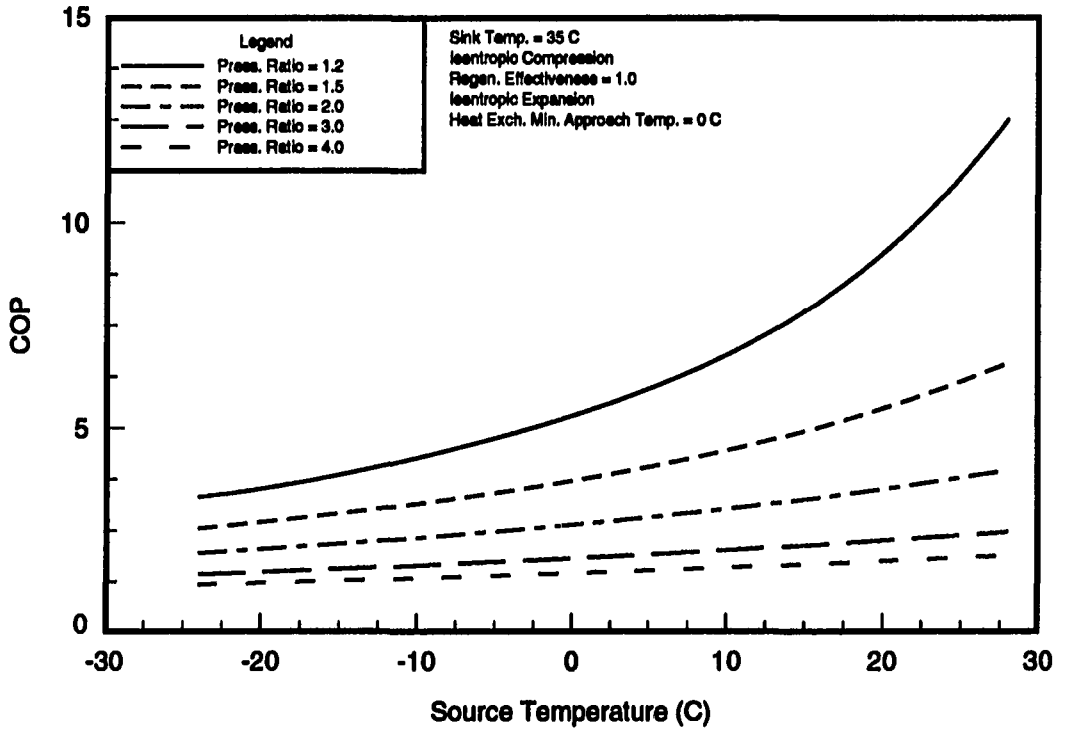

Figure 5.11: COP vs. source temperature for an ideal regenerative reversed Brayton refrigeration system.

The COP and cycle efficiency versus source temperature for the regenerative reversed Brayton cycle "actual case" are presented in Figures 5.15 and 5.16. The COP decreased with increasing source temperature at a faster rate when operated at low pressure ratios rather than high pressure ratios. The temperature lift could be accomplished over the entire source temperature range, even at low pressure ratios. The cycle efficiency increased with decreasing source temperature. For a pressure ratio of 1.5, the maximum efficiency occurred at approximately -10 C. The maximum cycle efficiency occurred below -24 C for pressure ratios of 2.0 and above.

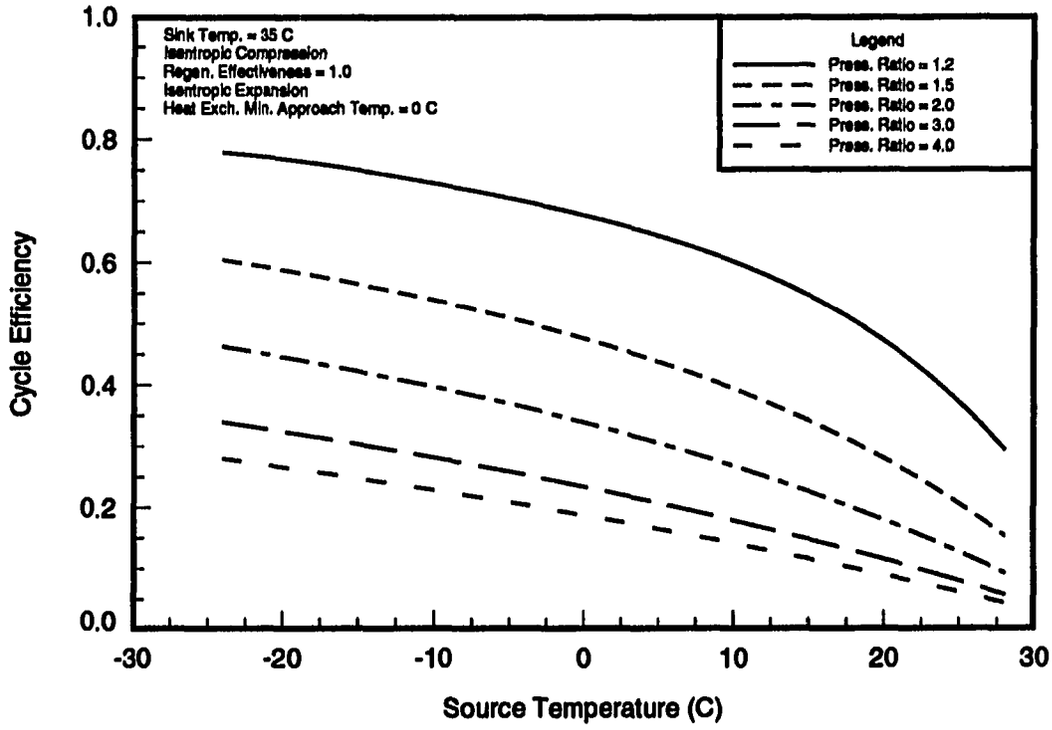

Figure 5.12: Cycle efficiency vs. source temperature for an ideal regenerative reversed Brayton refrigeration system.

The regenerative cycle provided better performance than the non-regenerative cycle in the source temperature range considered in this study. The primary performance benefit of the regenerator occurred when the cooling application required large temperature lifts. This was particularly evident when the source temperature was below 0 C, as in the case of refrigeration applications. The maximum cycle efficiency occurred below the lowest source temperature, −24 C, which was the lowest temperature considered in this project. The reversed Brayton cycle, particularly with regeneration, appears to be best suited for low-temperature applications requiring operation at source temperatures below −24 C.

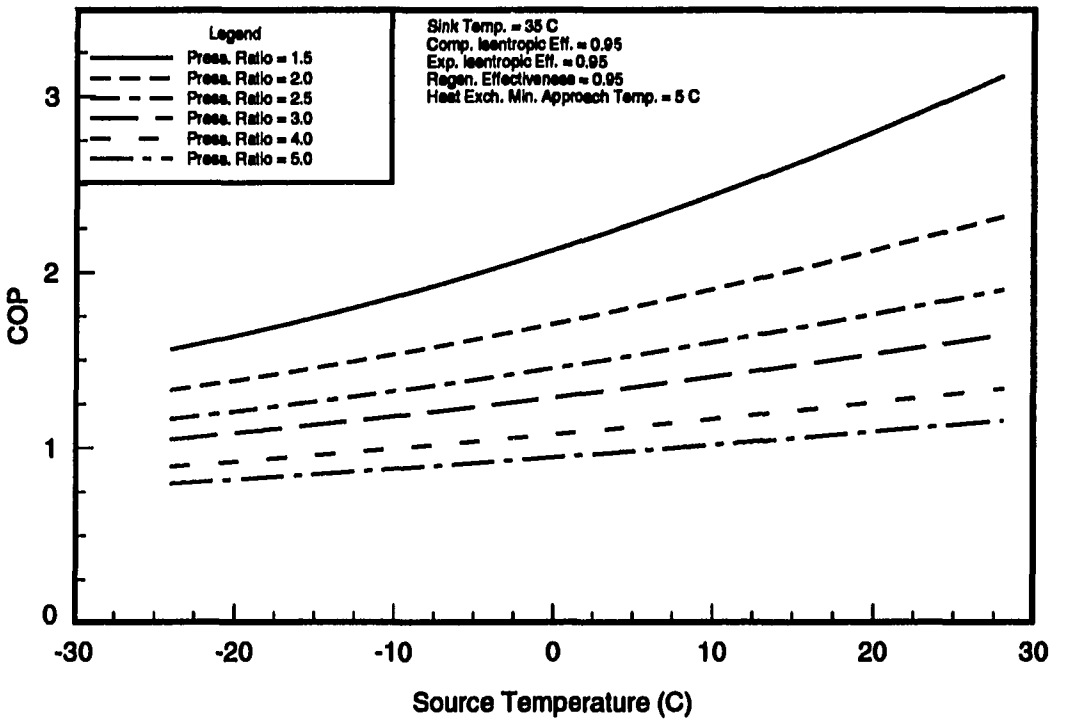

Figure 5.13: COP vs. source temperature for a regenerative reversed Brayton refrigeration system operating with parameters given in the "best possible" case.

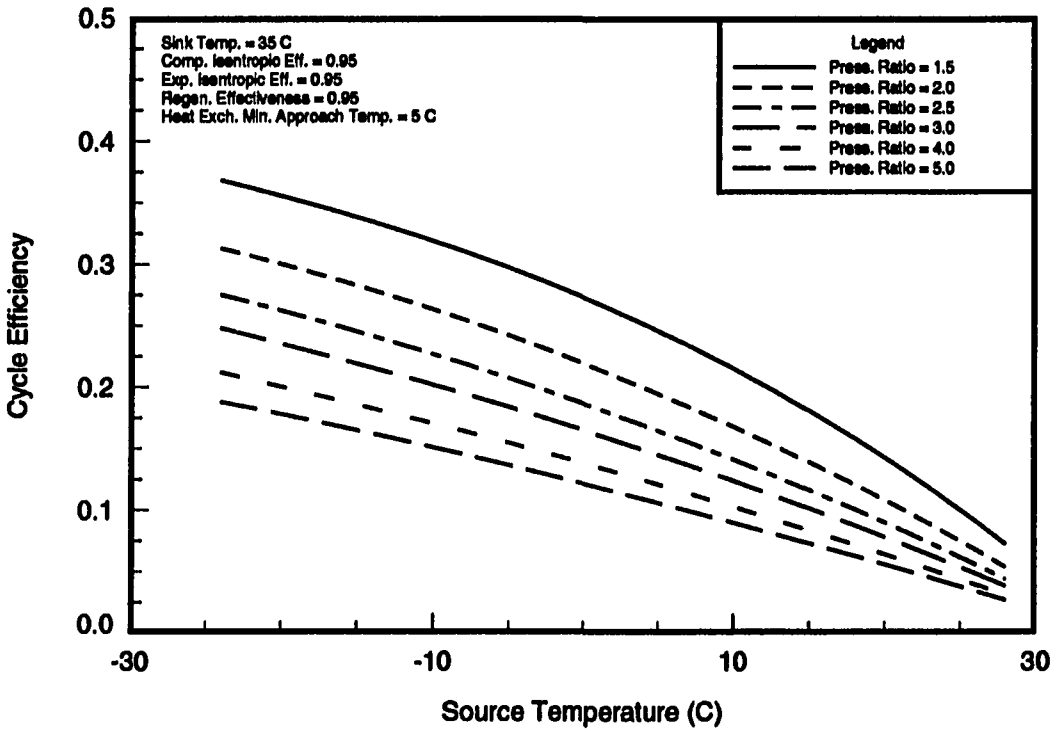

Figure 5.14: Cycle efficiency vs. source temperature for a regenerative reversed Brayton refrigeration system operating with parameters given in the "best possible" case.

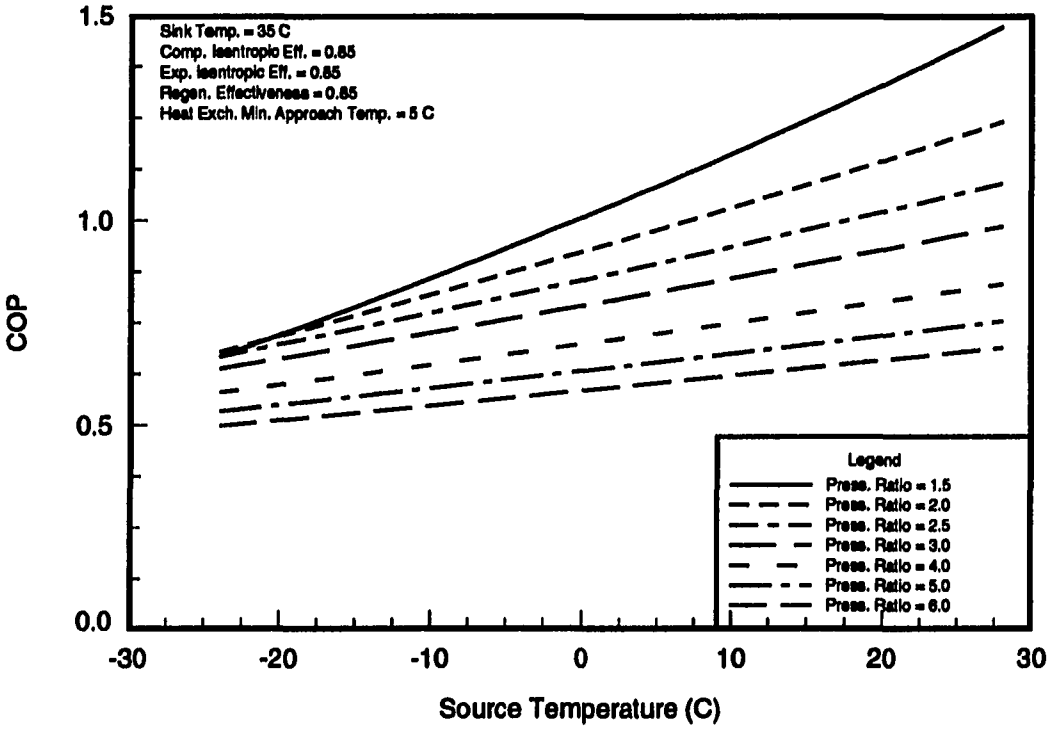

Figure 5.15: COP vs. source temperature for a regenerative reversed Brayton refrigeration system operating with parameters given in the actual case.

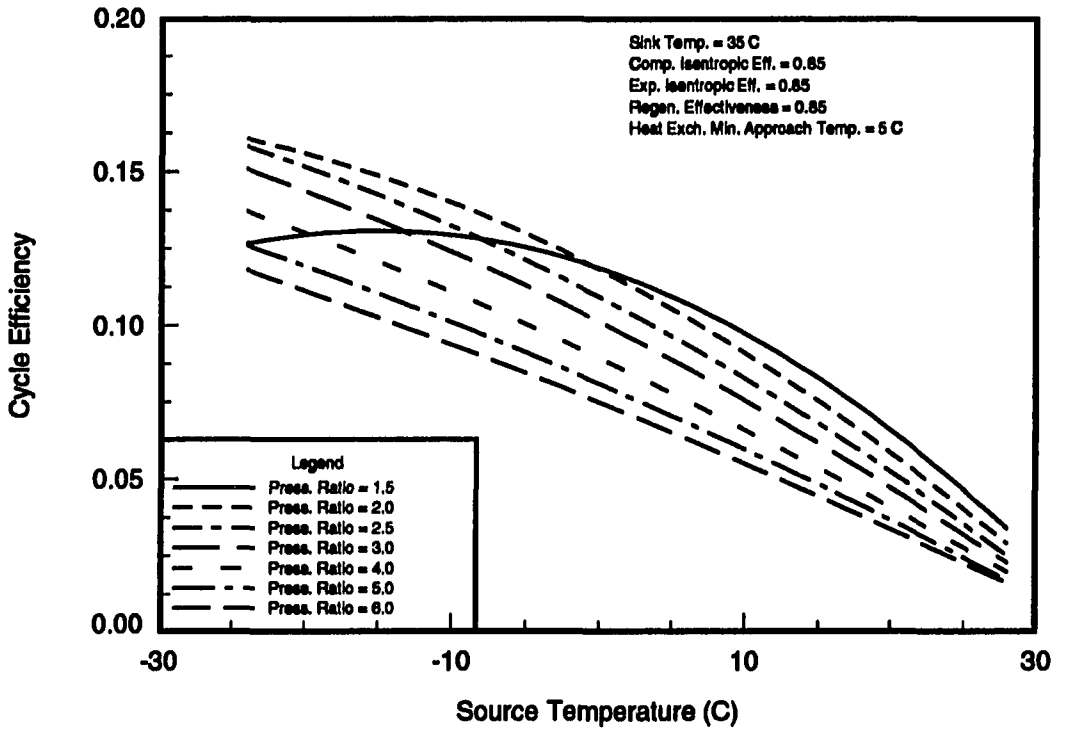

Figure 5.16: Cycle efficiency vs. source temperature for a regenerative reversed Brayton refrigeration system operating with parameters given in the actual case.

CHAPTER 6. STIRLING REFRIGERATION

Introduction

The Stirling cycle was first used as an external combustion heat engine for the conversion of thermal energy to mechanical work. The ideal Stirling cycle is composed of two isothermal processes, expansion and compression, and two isometric processes during regenerative heat transfer. If the cycle is reversed, it can be used as a refrigerator.

Stirling refrigerators have been employed as cryocoolers in chemical and industrial applications. As a cryocooler the heat source temperature for a Stirling refrigerator is typically between −193 C and −93 C.

Concern about the use of CFC refrigerants has brought about renewed interest in Stirling refrigerators for applications near room temperature.

History

Robert Stirling, a Scottish minister, first developed an external combustion engine using air as the working fluid in 1817 [27]. Although the Stirling cycle makes use of regenerative heat transfer, the thermodynamic significance of regeneration was not understood until 1854 when the concept of regeneration was explained by Rankine [28].

In the 1940s, N. V. Philips Company's research laboratory began a project to design an air-process engine that made use of modern heat transfer methods, fluid flow concepts, and materials [29]. In the 1950s, N. V. Philips made use of this technology to develop a Stirling refrigeration machine. The design objective was to produce refrigeration in a single stage between the temperatures of −180 C and room temperature [30].

U.S. Patent Search

A U.S. patent search was conducted to discover Stirling-type refrigeration technologies.

Patent number 1,508,522 was granted to Ivar Lundgaard on September 16, 1924 for "an air refrigerating machine of the closed-cycle type." This patent was for a modification of a concept previously patented by the same inventor (U.S. patent number 1,240,862).

Lundgaard's machine was composed of expansion and compression cylinders, a regenerator, and high- and low-temperature heat exchangers. The reciprocating motion of the pistons was accomplished by a rotating camshaft which displaced roller followers. The followers were connected to the expansion and compression pistons. A spring return mechanism was attached to each follower to keep the follower in contact with the cam [33].

Literature Review

Rinia and Du Pré [27] first modeled the Stirling cycle as an idealized cycle but with harmonic piston motion. They also defined the regenerator "efficiency" for the

Stirling cycle as being the percentage of heat contained in the air in-flow that is stored in the regenerator and transferred back to the air as it returns to the regenerator. The part of the heat not stored during regeneration is carried off by the cooler and lost for the cycle.

Köhler and Jonkers [30] reviewed the idealized Stirling cycle for refrigerators in detail. They also applied the harmonic piston motion analysis to a Stirling refrigeration cycle. In a second paper [31] Köhler and Jonkers discussed the deviations of the actual cycle from the ideal cycle. These deviations include losses which result in increased shaft power, reduced the refrigerating capacity, regeneration losses, and heat exchanger losses.

Chen et al. [32] tested an off-the-shelf Stirling cryocooler and investigated the implications of the experimental results to household refrigeration applications. It was found that the optimum expansion temperature was between −173 C and −123 C. The optimum operating temperature range was defined in terms of the expansion head (low-temperature heat exchanger) temperature. At an expansion head temperature of −23 C, the cycle efficiency was 7%. It was concluded that the COP would need to be tripled if the Stirling technology was to be a viable alternative to vapor-compression for household refrigeration applications.

Bauwens and Mitchell [34] published experimental and numerical data intended to verify a one-dimensional transient thermodynamic model of a Stirling refrigerator. It was assumed that the working fluid was a perfect gas with constant specific heat. The solution of the equations used in the model is time dependent due to the periodic piston motion. A comparison of the experimental and model performance data indicated that the agreement between prototype and model was, at best, one order

of magnitude.

Carlsen et al. [35] constructed a computer model which accounted for cylinder volume, phase angle, temperature ratio, and dead volume. The number of transfer units (NTU) was an independent variable accounting for heat exchange in the cylinder volumes. It was concluded that the thermal performance of the actual system would be reduced due to losses associated with heat transfer between the piston, cylinder and the gas. In terms of heat transfer, the cylinder has an inherently low NTU-number (below 5) which limits performance. For a temperature ratio of 1.18, which is the approximate value for applications near room temperature, the cycle efficiency was less than 0.7 for a system which was assumed to have:

1. Reversible regeneration.
2. Perfect mixing of gas in the cylinder volumes.
3. No frictional losses in the machinery.
4. No fluid friction losses.
5. Ideal gas.

It was concluded that it would be very difficult to design a Stirling refrigerator with a competitive COP as compared to the vapor-compression cycle when the additional losses associated with non-ideal conditions are considered.

Carrington and Sun [36] concluded that regenerator heat transfer losses increase sharply at lower cold-end (expansion) temperatures. On the other hand, frictional losses dominate at the compression end.

Fabian [37] reported experimental results for prototype free-piston Stirling coolers intended for domestic refrigeration. His results are summarized in Table 6.1.

Berchowitz [38] reported a cycle efficiency of 0.3 for a free-piston Stirling cooler operating between −26 C and 41 C. The COP was calculated as the ratio of measured heat removal to the electrical power input.

Table 6.1: Experimental results reported by Fabien [42] for prototype free-piston Stirling coolers intended for domestic refrigerators.

Unit Number	T_{source} (C)	T_{sink} (C)	η_{Cycle}
1	−33	18	0.217
2	−57	16	0.240
3	−75	42	0.226

Stirling Cycle Models

Introduction

Models for the Stirling refrigeration cycle vary in complexity from the ideal thermodynamic model to transient numerical models which take into account fluid flow and heat transfer through the system.

Idealized Stirling Refrigeration Cycle Model

A schematic of an idealized model of a Stirling refrigeration cycle is illustrated in Figure 6.1. The state points on the schematic correspond to the state points given on the P-V diagram and T-S diagrams for the cycle, Figure 6.2 and Figure 6.3.

The assumptions made for the ideal model are:

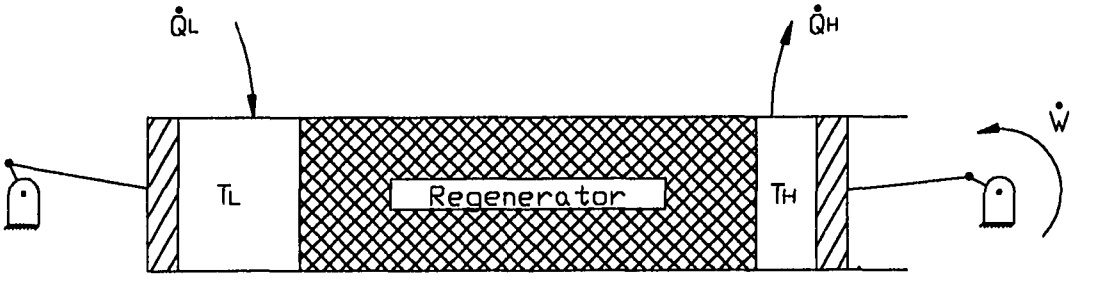

Figure 6.1: Schematic of a Stirling refrigeration cycle.

1. Ideal gas.
2. The rate at which heat is accepted and rejected from the system is unchanging with time.
3. Perfect heat transfer to the heat source and sink.
4. 100% regenerator effectiveness.
5. Discontinuous motion of the pistons.
6. Negligible changes in potential and kinetic energy of the fluid.
7. Negligible pressure drop in the heat exchangers and related piping.
8. No clearance volume in expansion or compression cylinders.

Since an ideal gas has been assumed, internal energy is a function of temperature only in the closed system; $u = u(T)$.

During the following development, it will be assumed that,

- heat *into* the system is positive,
- and work done *by* the system is positive.

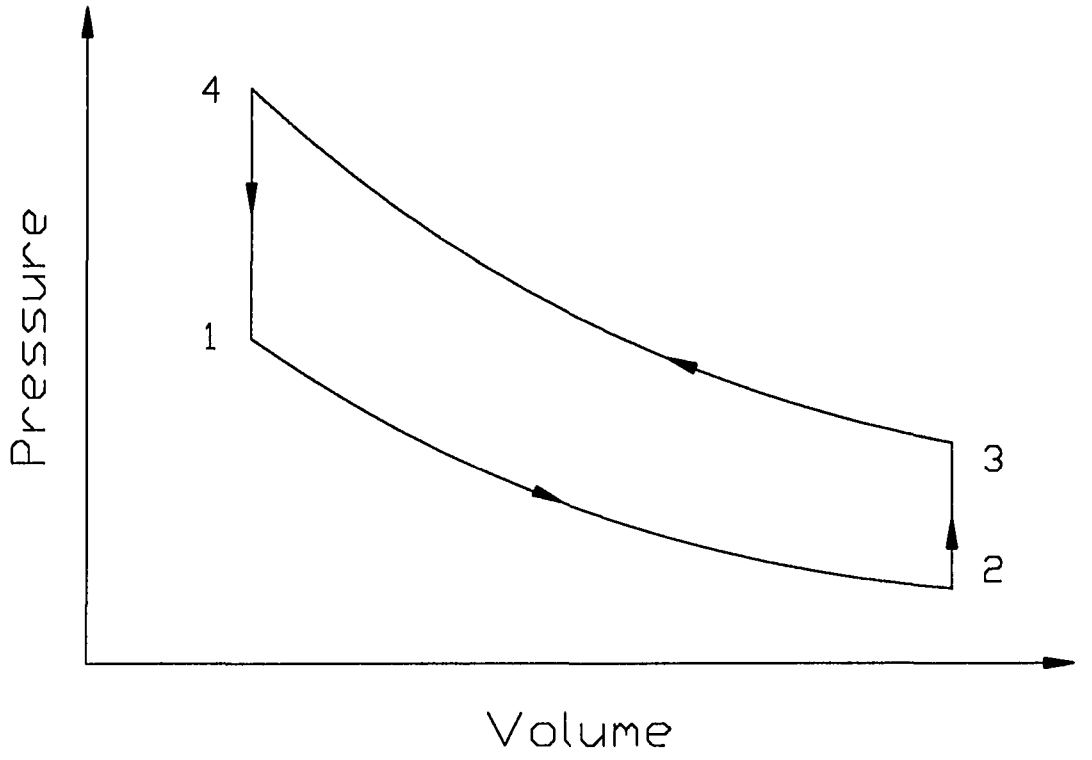

Figure 6.2: Pressure vs. volume diagram for a Stirling refrigeration cycle.

The process from state 1 to state 2 is assumed to be isothermal expansion, therefore, there is no change in internal energy. An energy balance on the expansion cylinder yields,

$$Q_{in} - W_{exp} = m(u_{out} - u_{in}) \tag{6.1}$$

$$Q_{in} = W_{exp}. \tag{6.2}$$

The expansion work is given by,

$$W_{exp} = \int p\,dv \tag{6.3}$$

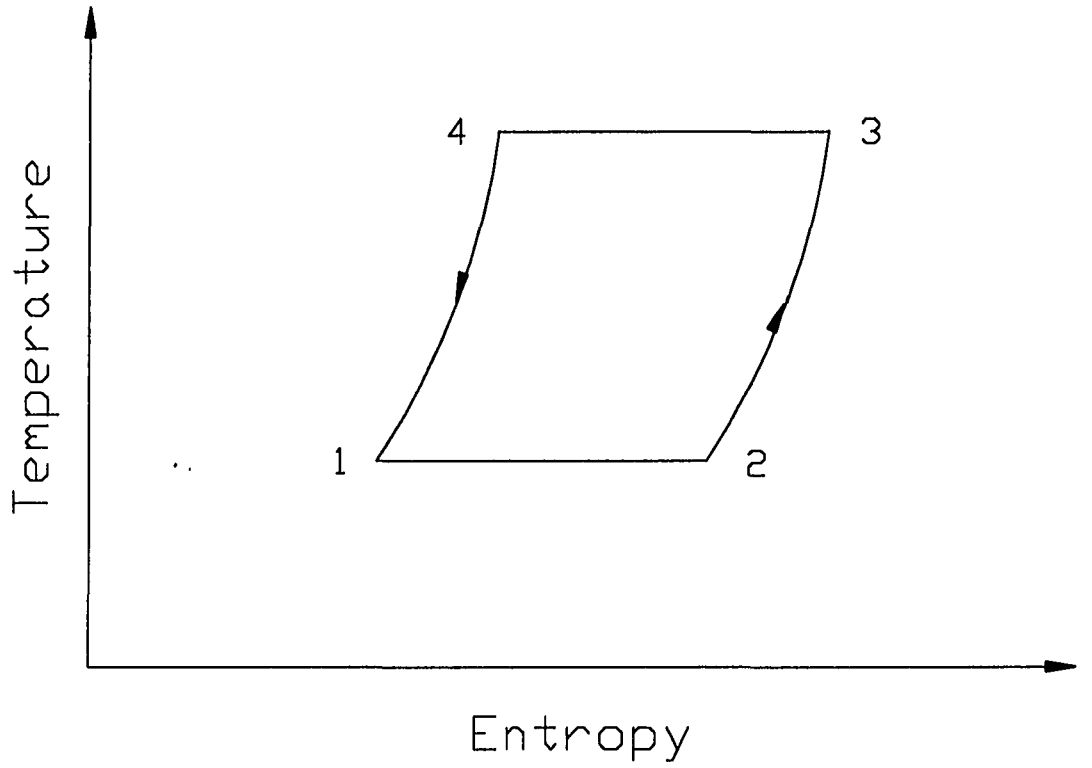

Figure 6.3: Temperature vs. entropy diagram for a Stirling refrigeration cycle.

Substituting the ideal gas equation of state,

$$W_{exp} = \int_1^2 \left(mRT_{exp}\right) \frac{dV}{V} \tag{6.4}$$

$$= mRT_{exp} \ln\left(\frac{V_2}{V_1}\right) \tag{6.5}$$

$$= mRT_1 \ln\left(\frac{V_2}{V_1}\right) > 0 \tag{6.6}$$

where, $T_1 = T_2$.

Similarly, the work of compression from state 3 to state 4 is given by,

$$W_{comp} = -mRT_{comp} \ln\left(\frac{V_3}{V_4}\right)$$

$$= mRT_3 \ln\left(\frac{V_4}{V_3}\right) \tag{6.7}$$

where, $T_3 = T_4$ and $\frac{V_4}{V_3} = \frac{V_1}{V_2}$.

During regenerative heat transfer from state 2 to state 3 and from state 4 to state 1 an energy balance yields,

$$Q_{regeneration\ 2\rightarrow 3} = |Q_{regeneration\ 4\rightarrow 1}|. \tag{6.8}$$

The COP for the idealized Stirling refrigeration cycle is then,

$$COP_{ideal\ Stirling} = \frac{Q_{in}}{W_{net}} \tag{6.9}$$

$$= \frac{T_{exp}}{T_{comp} - T_{exp}} \tag{6.10}$$

$$= COP_{Carnot}. \tag{6.11}$$

Ideal Stirling Refrigeration Model with Harmonic Piston Motion

For practical application of the Stirling refrigeration cycle, the piston motion would be continuous. An approximate cycle can be realized by harmonic movement of the compression and expansion pistons with a phase displacement between them. If it is assumed that the assumptions for the idealized model apply, a thermodynamic analysis of the cycle can be conducted if the following parameters are known:

V_{exp}	Expansion space volume, excluding clearance vol.
T_{exp}	Absolute Temperature of the expansion fluid.
V_{comp}	Volume of the compression space.
T_{comp}	Absolute Temperature of the compression fluid.
V_0	Maximum volume of the expansion space.

wV_0	Maximum volume of the compression space.
w	Ratio of the *max.* values of V_{comp} and V_{exp}.
V_s	Vol. of all non-displaced spaces in the system including the regenerator.
T_s	Average absolute temperature of all non-displaced spaces.
α	Crankshaft angle (radian).
φ	Phase angle between V_{exp} and V_{comp} (radian).
s	Relative reduced dead space, $\left[\left(\frac{V_s}{V_0}\right)\left(\frac{T_{comp}}{T_s}\right)\right]$.
τ	Temperature ratio, $\frac{T_{comp}}{T_{exp}}$.
θ	Phase angle of the pressure w.r.t. the exp. cylinder volume (radian).
δ	Dummy variable relating $\tau, w, \varphi, and\, s$.

The volumes of the expansion and compression spaces can be written as functions of the crank and phase angles,

$$V_{exp} = \frac{V_0}{2}(1 + \cos\alpha) \tag{6.12}$$

$$V_{comp} = \frac{wV_0}{2}[1 + \cos(\alpha - \varphi)]. \tag{6.13}$$

The pressure can be written as a function of the crank and phase angles using the ideal gas law,

$$P = P_{max}\frac{1 - \delta}{1 + \delta\cos\alpha - \theta} \tag{6.14}$$

where,

$$\delta = \frac{\sqrt{\tau^2 + w^2 + 2\tau w\cos\varphi}}{\tau + w + 2s}$$

$$\tan\theta = \frac{w\sin\varphi}{\tau + w\cos\varphi}. \tag{6.15}$$

If the polar coordinate system is defined such that P_{min} occurs at $\alpha - \theta = 0$ and P_{max} occurs at $\alpha - \theta = \pi$, the pressure ratio can be defined as,

$$\frac{P_{max}}{P_{min}} = \frac{1 + \delta}{1 - \delta}. \tag{6.16}$$

An expression can be obtained for the mean pressure,$\bar{P}$, by integrating the pressure with respect to the crank angle, α:

$$\bar{P} = P_{max}\sqrt{\frac{1-\delta}{1+\delta}}. \tag{6.17}$$

The quantity of heat absorbed by the fluid when the pressure and volume both vary sinusiodally is found from,

$$Q = \oint P\,dV. \tag{6.18}$$

For the expansion process, the solution is,

$$Q_{exp} = \pi \bar{P} V_0 \frac{\delta}{1+\sqrt{1-\delta^2}} \sin\theta \tag{6.19}$$

and for the compression process,

$$\begin{aligned} Q_{comp} &= \pi \bar{P} w V_0 \frac{\delta}{1+\sqrt{1-\delta^2}} \sin\theta - \varphi && (6.20) \\ &= \tau \cdot Q_{exp}. && (6.21) \end{aligned}$$

The COP for the idealized Stirling refrigeration cycle with harmonic piston movement is then,

$$\begin{aligned} COP_{idealStirling,harmonic} &= \frac{Q_{in}}{W_{net}} && (6.22) \\ &= \frac{Q_{exp}}{|Q_{comp} - Q_{exp}|} && (6.23) \\ &= \frac{T_{exp}}{T_{comp} - T_{exp}} && (6.24) \\ &= \frac{1}{\tau - 1} && (6.25) \\ &= COP_{Carnot} && (6.26) \end{aligned}$$

where τ is defined as,

$$\tau \equiv \frac{|Q_{comp}|}{Q_{exp}} = \frac{T_{comp}}{T_{exp}}. \tag{6.27}$$

The constraint of harmonic piston movement does not affect the reversibility of the cycle, so the COP remains the same as for the basic ideal reversed Stirling model. The resulting cycle is no longer composed of two isothermal and two isometric processes, but rather, the pressure and volume vary continuously throughout the cycle. Figure 6.4 illustrates the pressure vs. volume relationship for the expansion space, compression space, and regenerator for a Stirling refrigerator with harmonic piston motion. By comparing Figure 6.4 to the P-V diagram for the ideal case, Figure 6.2), it can be seen that the regeneration processes are no longer isometric. Furthermore, the compression and expansion processes are no longer isothermal.

Discussion It has been shown that the COP for an ideal Stirling refrigerator is the reversed Carnot cycle COP regardless of piston motion. The continuity of piston motion does not affect the reversibility of the system. Irreversibilities in the Stirling cycle result from:

1. Mechanical losses (friction).
2. Non-isothermal operation.
3. Heat losses through machine members.
4. Heat exchange via finite temperature differences between the system and the environment.
5. Imperfect regeneration.
6. Fluid frictional losses in the cylinders, heat exchangers and regenerator.
7. Fluid leakage.

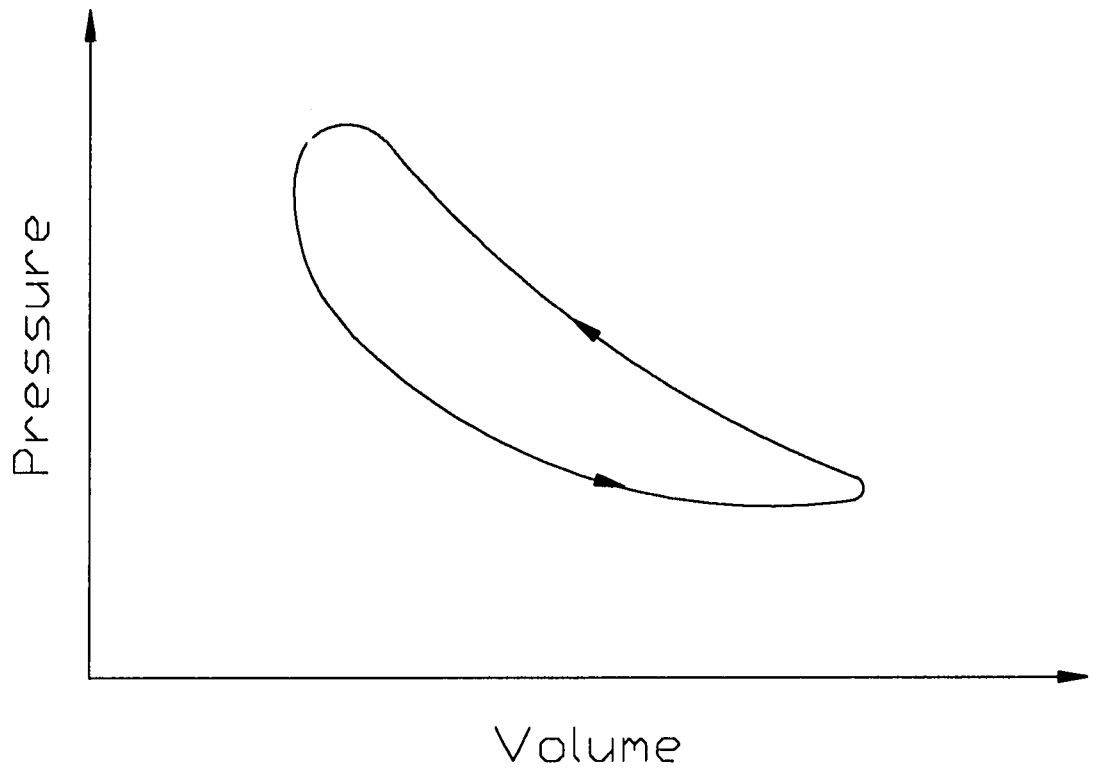

Figure 6.4: Pressure vs. volume diagrams for a Stirling refrigerator with harmonic piston motion.

Other Models

A demonstration version of the Mitchell/Stirling MS*2 computer program was reviewed. This is a one-dimensional transient model which simultaneously solves the equations for the conservation of mass, momentum, and energy within the system boundary. The model employs empirical correlations to calculate the heat transfer coefficients at the wall. The geometry of the solution domain is determined by the length, heat transfer area and net cross-section in the spaces, and the periodic motion of the pistons [34].

The MS*2 program requires the specification of all of the physical dimensions related to the geometry of a basic Stirling engine or refrigerator. These include:

1. Expansion cylinder bore and stroke.
2. Compression cylinder bore and stroke.
3. Clearance volumes for the expansion and compression cylinders.
4. Initial pressure.
5. Compression ratio.
6. Phase angle.
7. High and low temperature heat exchangers:

   (a) Tube numbers
   (b) Length
   (c) Inner and outer diameters
   (d) Material properties

8. Regenerator:

   (a) Length
   (b) Diameter
   (c) Material properties
   (d) Mesh size
   (e) Fill factor

9. Working fluid.
10. Shaft speed (rpm).

The requirement to establish specific design parameters, including materials and dimensions, makes the MS*2 program difficult to use as a tool for comparing the Stirling refrigeration system with other different refrigeration technologies.

Kelly et al. [40] concluded that the actual thermodynamic cycle which occurs in Stirling-type refrigeration systems more closely resembles a modified regenerative reversed Brayton cycle. The thermodynamic processes by which this cycle is accomplished are:

1. Compression to a temperature greater than that of the thermal sink.
2. Isobaric heat rejection to the thermal sink.
3. Isometric regeneration.
4. Expansion to a temperature below that of the thermal source.
5. Isobaric heat acceptance from the thermal sink.
6. Isometric regeneration, thus completing the thermodynamic cycle.

Results

Figure 6.5 is a graph of the Stirling refrigeration cycle COP versus source temperature. The only irreversiblities present in the system result from irreversible heat transfer during the heat acceptance and rejection processes. A minimum approach temperature of 10 C was used for each heat transfer process. The cycle efficiency versus source temperature for this system is shown in Figure 6.6. The cycle efficiency is low at the upper end of the source temperature range and increases to approximately 0.7 at −24 C.

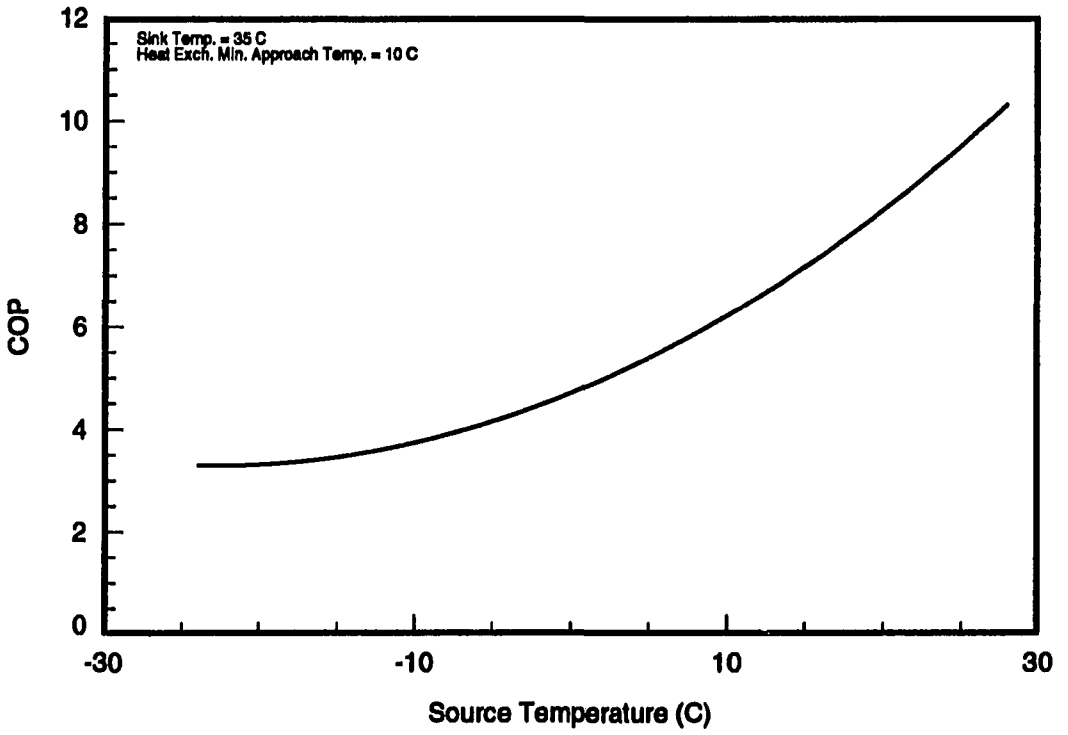

Figure 6.5: COP vs. source temperature for a Stirling refrigerator with irreversible heat exchange processes.

Since other irreversibilities are known to be present in Stirling refrigerators, the Kelly model [40] was used to calculate the COP and cycle efficiencies for a Stirling-type refrigeration system.

The sink temperature was 35 C. The isentropic compression and expansion efficiencies were both 0.85. The effectiveness of the regeneration process was set at 0.85. A 10 C minimum approach temperature was used to account for the high- and low-temperature heat exchanger irreversiblities.

Figure 6.7 is a graph of the COP versus source temperature calculated using the model developed by Kelly et al. Figure 6.8 is a graph of the corresponding cycle

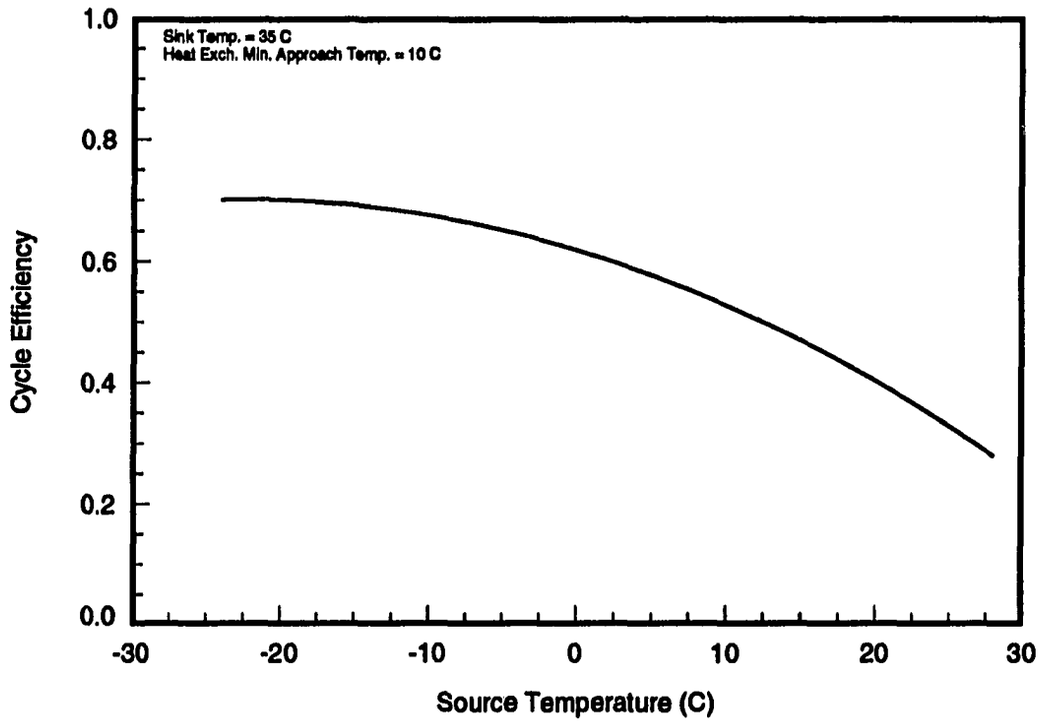

Figure 6.6: Cycle efficiency vs. source temperature for a Stirling refrigerator with irreversible heat exchange processes.

efficiencies for the same set of parameters. The cycle efficiency is 0.28 at $T_{source} = -24$ C and decreases to 0.13 at $T_{source} = 28$ C.

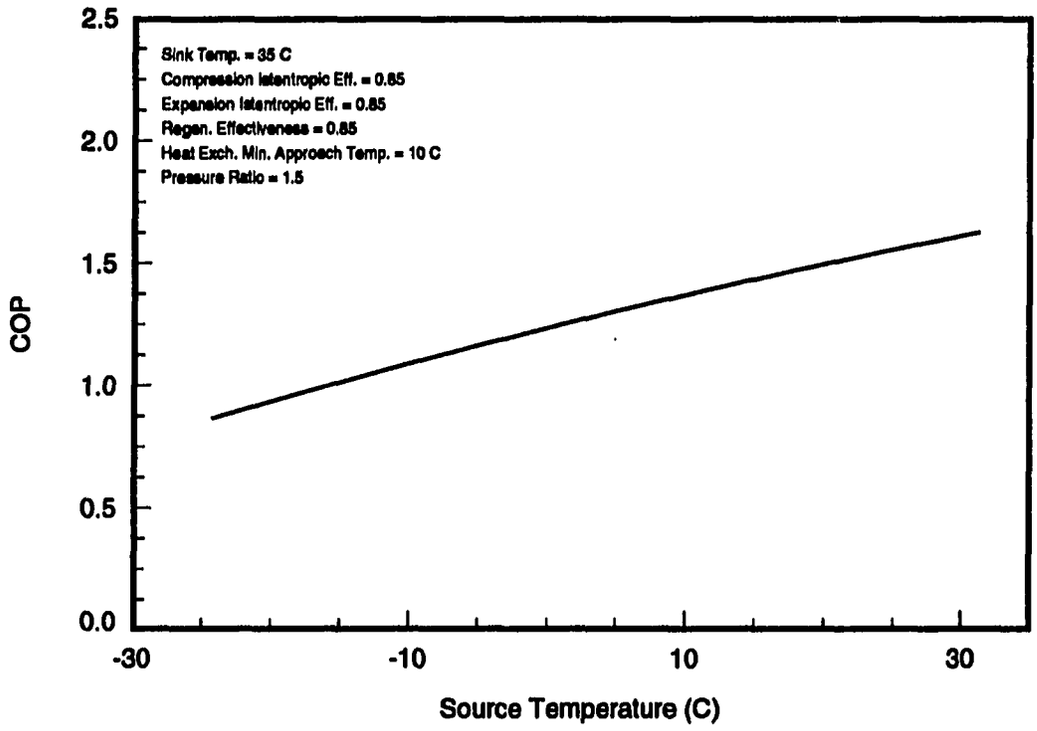

Figure 6.7: COP vs. source temperature for a Stirling-type refrigerator calculated using the Kelly model.

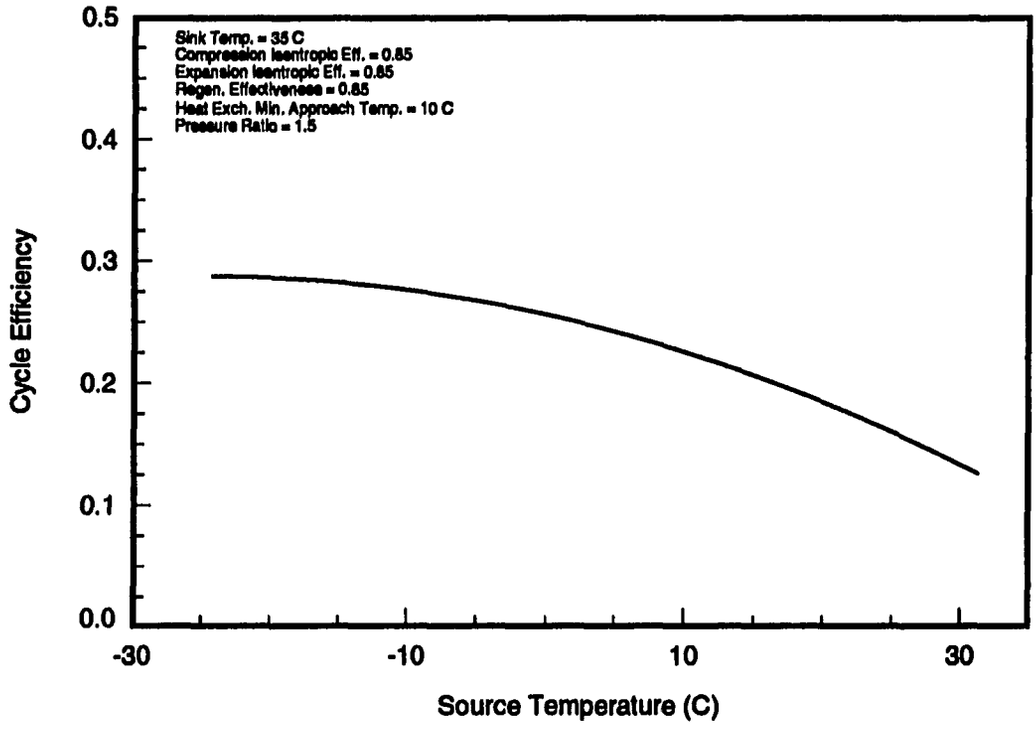

Figure 6.8: Cycle efficiency vs. source temperature for a Stirling-type refrigerator using the Kelly model.

CHAPTER 7. PULSE TUBE AND THERMOACOUSTIC REFRIGERATION

Introduction

Two refrigeration concepts utilizing a gas-filled column in which the pressure is varied cyclically to produce cooling will be discussed in this chapter. The pulse-tube system uses a compressor to pressurize the gas; the thermoacoustic system relies upon a diaphragm displaced by an electromagnetic coil for the same task.

Pulse Tube

History

The pulse-tube concept was developed by Gifford and Longsworth in the mid-1960s [50]. Mikulin [51] and Radebaugh [52] improved the COP of the pulse tube by adding an orifice and expansion chamber to the high-temperature end of the tube.

Theory of Operation

Operating Principle of an Ideal Pulse Tube An ideal pulse tube system is composed of a compressor, regenerator,high temperature heat exchanger, and the

tube. Figure 7.1 depicts two pulse-tube schematics illustrating different methods of periodically compressing the working gas.

Periodic compression can be accomplished by a steady flow compressor and rotary control valve, as shown on the left side of Figure 7.1 or by a reciprocating compressor, as shown on the right.

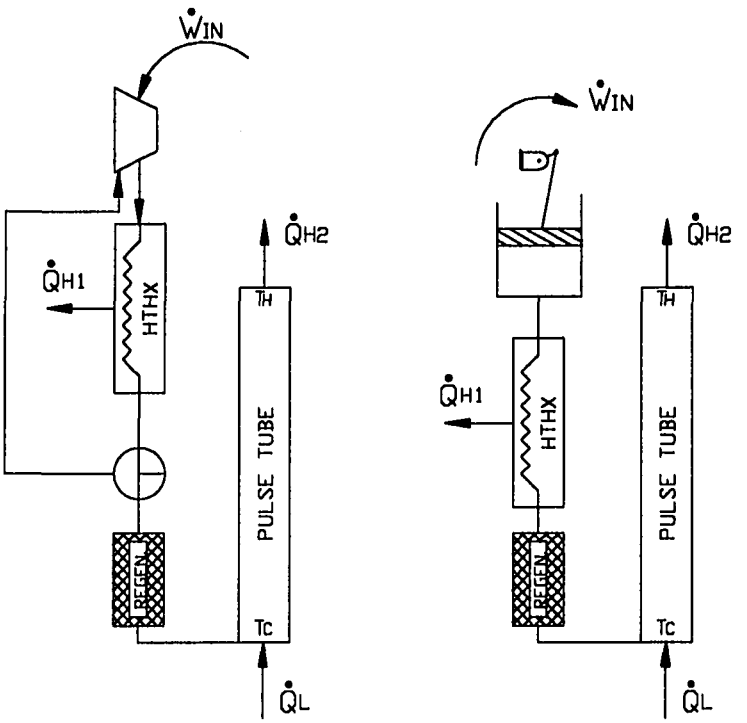

Figure 7.1: Two pulse tube refrigerator concepts.

The heat generated due to compression of the gas is rejected by a high temperature heat exchanger; the gas then passes through a regenerator.

The tube wall is constructed of a material with a low thermal conductivity, commonly stainless steel. The ends of the tube are capped with a highly conductive material like copper. The heat is accepted by the low temperature heat exchanger

located at the entrance end of the tube, and rejected by the high temperature heat exchanger at the opposite end.

The principle of operation of the ideal cycle may be understood by considering a thin cylindrical control mass segment as shown in Figure 7.2. The following assumptions are made:

1. Inviscid flow.
2. The change in pressure from low to high and vice versa are both step functions with respect to time as shown in Figure 7.3.
3. Perfect regeneration.
4. The heat is transferred to and from the mass segment through the tube wall.
5. Ideal gas.
6. Isentropic compression.
7. All ducts are adiabatic.

The mass is initially contained in volume, V_1, of tube inside diameter and thickness, h_1. As the pressure and temperature of the gas in the element are changed during the cycle, the volume will also change. Since the tube wall is rigid the diameter of the element will not change, however, the *thickness* of the element must change. The gas is at a thermodynamic state having properties P_L and T_1, and an initial height in the tube, x_1. When the pressure in the tube is increased to P_H, the volume is compressed to V_2, elevated to position x_2, and the temperature is raised to T_2. A temperature gradient ranging from T_L to T_H exists along the

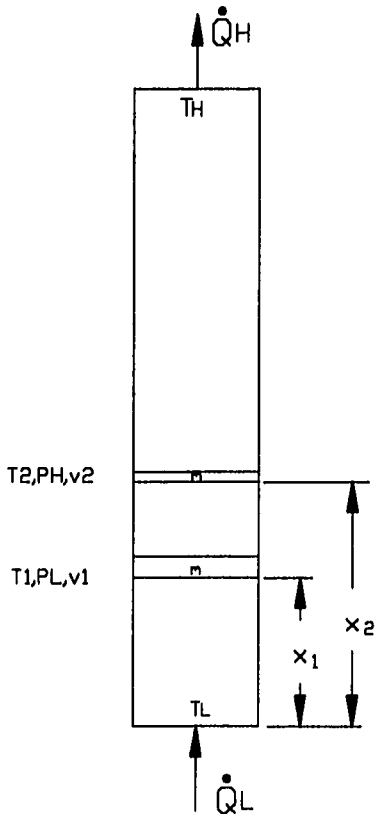

Figure 7.2: Cycle executed by a control mass element in a pulse tube.

length of the tube. The local temperature of the tube wall at position x_2 is lower than the temperature of the mass segment, T_2. Therefore, heat will be transferred from the mass segment to the wall, and the thermodynamic properties will be: T_3, P_H, and v_3. When the pressure is lowered to P_L, the mass segment undergoes an isentropic expansion to state 4 with properties T_4, v_4, and P_L, at a slightly higher elevation than the original position. As the gas is re-heated, it will be returned to the original thermodynamic state (state 1) and position in the tube; the cycle has been completed.

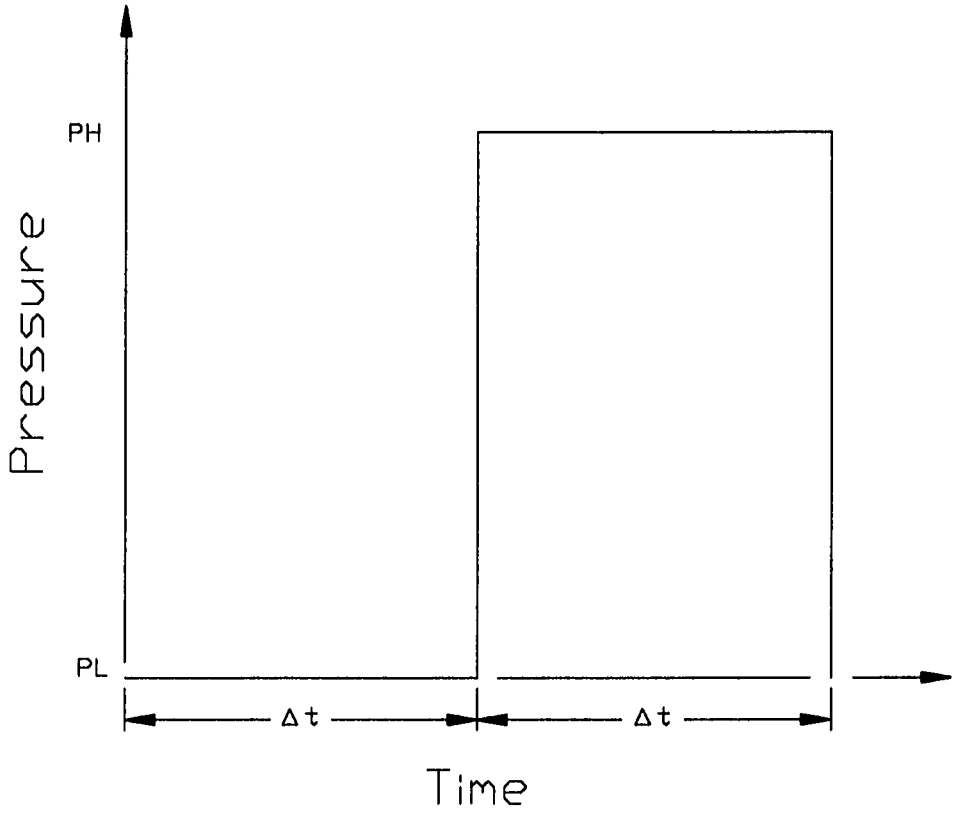

Figure 7.3: Step function pressure change in an ideal pulse tube.

Ideally, the control mass of gas in the tube undergoes an isentropic compression from states 1 to 2, isobaric cooling from states 2 to 3, an isentropic expansion from states 3 to 4, and isobaric heating from 4 to 1. The ideal cycle formed by the series of processes undergone by the control mass is the reversed Brayton cycle. Furthermore, the cycle for the entire mass of gas in the tube is an interconnected series of Brayton cycles as shown in Figures 7.4 and 7.5. The tube wall acts as a continuous regenerator [41].

The regenerator shown in Figure 7.1 serves, in part, to isolate the cold end of the

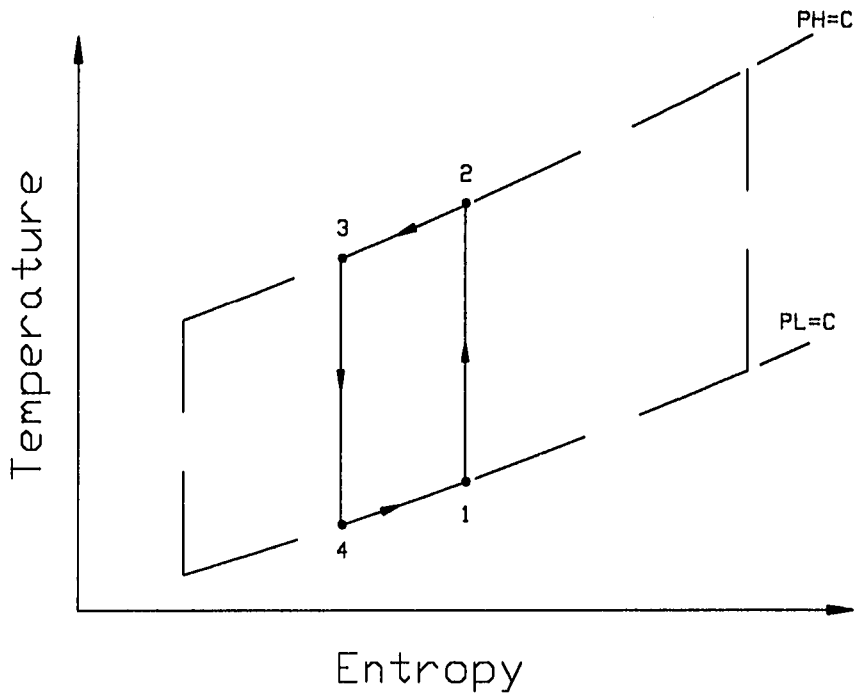

Figure 7.4: T-S diagram for the pulse tube refrigeration cycle.

tube from the pressure source which is at a higher temperature. Swift [53] concluded that the regenerator serves an additional function which produces a Stirling-type refrigeration effect. He reasoned that since both heat and mass are transferred through the regenerator, a Stirling-type cycle must exist within the regenerator. This cycle produces an additional refrigeration effect. Since the additional cycle in the regenerator involves two constant pressure (rather than constant volume) processes and two isentropic processes, it is more nearly a second Brayton cycle serving to remove additional heat from the low-temperature heat exchanger.

The total refrigeration effect in the ideal pulse tube refrigerator is the sum of these two heat removal mechanisms. Ideally, if the effectiveness of both the tube and

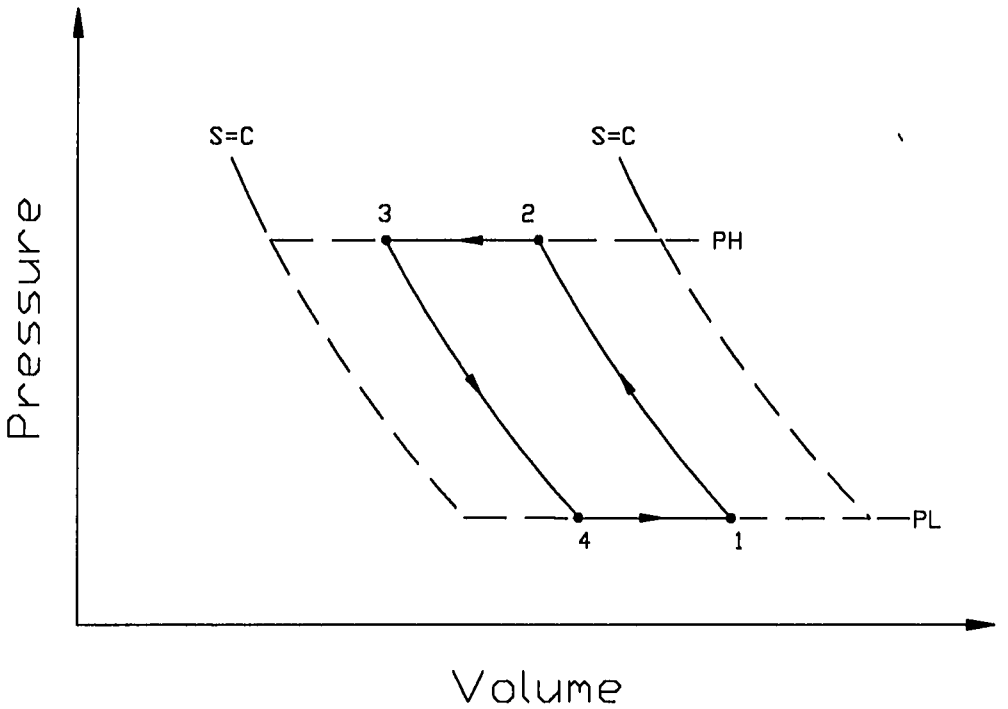

Figure 7.5: P-v diagram for the pulse tube refrigeration cycle.

the regenerator were the same, the cycle for the entire system could be viewed as one continuous Brayton cycle as depicted in Figures 7.4 and 7.5.

Losses Associated with the Actual Cycle The principal losses in the actual cycle are attributable to:

1. Non-isentropic compression.

2. Compression is non-instantaneous.

3. Frictional losses in the piping and valves.

4. The work of re-expansion is dissipated due to wall friction and therefore not recoverable.

5. Imperfect regeneration in the regenerator and tube.

6. The boundary layer at the wall serves as an impediment to heat transfer.

7. Axial conduction of heat in both the tube and gas column.

Theoretical Model

A simple approximation of the maximum theoretical performance of "pulse" refrigeration can be made by considering the entire system to be operating as a reversed regenerative Brayton cycle. As illustrated in Figures 7.4 and 7.5, small control mass elements of the working gas are oscillating within the tube (pulse tube) or along the stack (thermoacoustic) and executing a Brayton cycle. This model assumes that the end thermodynamic states of one control mass element are the same as opposite end states for the adjacent elements on each side.

Thermoacoustic Refrigerator

History

The first documented experiments involving thermoacoustic heat engines were by Higgins in 1777. His research eventually lead to pulse combustion used in the German V-1 rocket during World War II and the Lennox pulse furnace in 1982 [52]. Other thermoacoustic engines include the Rijke tube, an acoustic laboratory demonstration

used in physics, and the Sondhauss tube which generated mechanical work in the form of resonant vibration due to the expansion of the heated air.

The thermoacoustic refrigeration concept, in which the gas in a closed tube is periodically compressed and expanded using an electromagnetic coil and diaphragm to produce cooling, is less well known. Wheatley [56] first published a paper regarding the concept in 1983. He was granted U.S. patents in 1983 and 1984. Later, Hofler conducted experimental research in which the performance of a thermoacoustic refrigerator was measured [53, 57].

U.S. Patents

Two United States Patents were found which covered thermoacoustic refrigeration. Patent number 4,398,398 entitled "Acoustical heat pumping engine," was granted in August 1983 to John Wheatley. A second patent, number 4,489,553, entitled "Intrinsically irreversible heat engine," was also granted to him in December 1984.

Thermoacoustic Refrigerator Theory of Operation

The thermoacoustic refrigerator uses a modified loudspeaker as an acoustic generator to compress the working gas. The loudspeaker on the left generates a standing wave in the gas-filled tube. The speaker frequency and the tube length are chosen such that there will be a resonant frequency in the tube. A stack of closely spaced plates are located near the right end of the tube which is sealed with a rigid cap. The spacing of the plates is chosen to be several thermal penetration depths (δ_k). The thermal penetration depth represents the distance over which heat will be transferred

during one acoustic period, $T = \frac{1}{f}$, is:

$$\delta_k = \left(\frac{k}{\pi f \rho c_p}\right)^{0.5} \tag{7.1}$$

where,

k = the thermal conductivity of the gas.

f = the frequency of the loudspeaker (Hz).

ρ = the density of the gas.

c_p = the constant pressure specific heat of the gas.

According to Garrett [57], the heat transfer between the gas and the plates in the stack occurs primarily in a narrow region near the plate surfaces which is defined by the thermal penetration depth, δ_k. Therefore, the stack plate spacing is a critical parameter in thermoacoustic refrigerators. Heat is alternately accepted and rejected as the gas translates back and forth due to acoustic oscillation. As with the pulse tube, the gas is adiabatically compressed and expanded. Therefore, the thermoacoustic and pulse tube processes are the same. Heat is exchanged with the tube wall in the pulse tube and with the stack in the thermoacoustic refrigerator. A temperature gradient exists along the stack. The acoustic work is used to transport heat *up* the gradient.

An experimental thermoacoustic refrigerator, the space thermoacoustic refrigerator (STAR) developed by the Naval Postgraduate School, uses helium as the working gas. The gas is maintained at a mean pressure of 10 atmospheres in the stack and resonator. The present stack configuration uses polyester film with monofilament fishing line as a spacer. A copper-fin heat exchanger is located at either end of the

stack [57].

Results

A numerical model was constructed using the regenerative reversed Brayton cycle as an approximation of the pulse tube cycle. A thermodynamic property routine for helium gas was written to determine the properties of the working gas.

Isentropic compressor and expander efficiencies were assumed to be 0.80. The regenerator effectiveness was assumed to be 0.80. The sink temperature was 35 C, and the source temperature was varied from -24 C to 28 C. A 10 C minimum approach temperature was assumed to exist between the heat exchangers and the source and sink.

Figure 7.6 is a graph of the COP versus the source temperature for six different pressure ratios. At pressure ratios of 3.5 and below, the pulse tube cycle was not capable of producing the required temperature lift over the entire range of source temperatures. The COP remained relatively constant over the entire source temperature range for each pressure ratio. The lower pressure ratios provided the highest COPs.

Figure 7.7 is a graph of the cycle efficiency versus source temperature for the same operating conditions and pressure ratios. The cycle efficiencies were highest at the low source temperatures. Even so, 0.10 was the highest efficiency noted for this parameter set.

Hofler measured a coefficient of performance of 12 % of Carnot for a thermoacoustic refrigerator operating at a source/sink temperature ratio ($\frac{T_L}{T_H}$) of 0.82. The cooling load was 3 watts from an electrical resistance heater [53]. The COP was

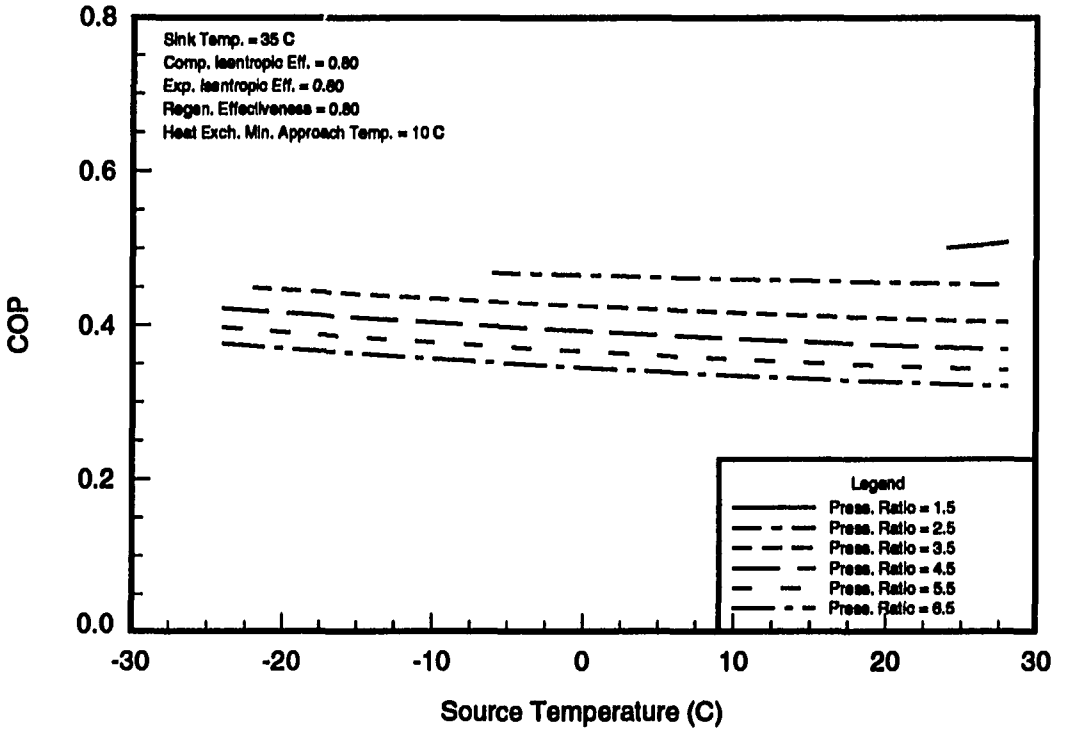

Figure 7.6: Coefficient of performance vs. source temperature for the pulse tube cycle using helium gas.

defined as,

$$COP = \frac{\dot{Q}_{elec}}{\dot{W}_{acoustic}}. \tag{7.2}$$

where,

$\dot{Q}_{elec}$ = The electric power dissipated by the heater.

$\dot{W}_{acoustic}$ = The measured acoustic power delivered by the loudspeaker.

Garrett and Hofler [57] reported measured COPs of up to 16 % of Carnot. The test conditions and temperatures were not reported. Garrett also reported experimental data for the STAR project (Figure 7.8).

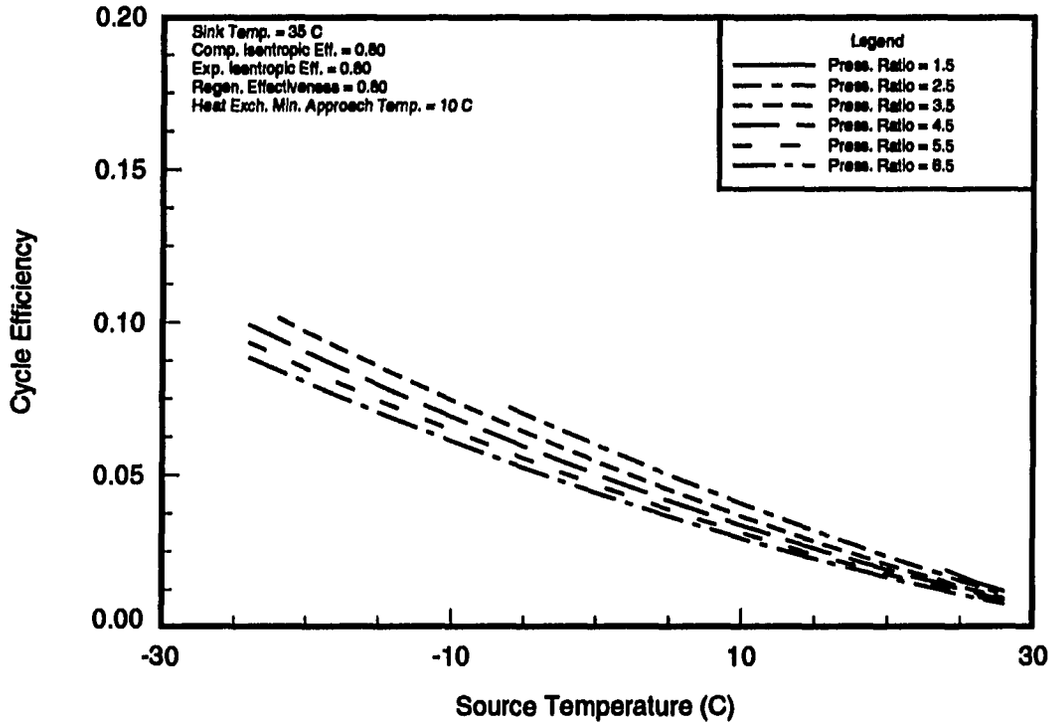

Figure 7.7: Cycle efficiency vs. source temperature for the pulse tube cycle using helium gas.

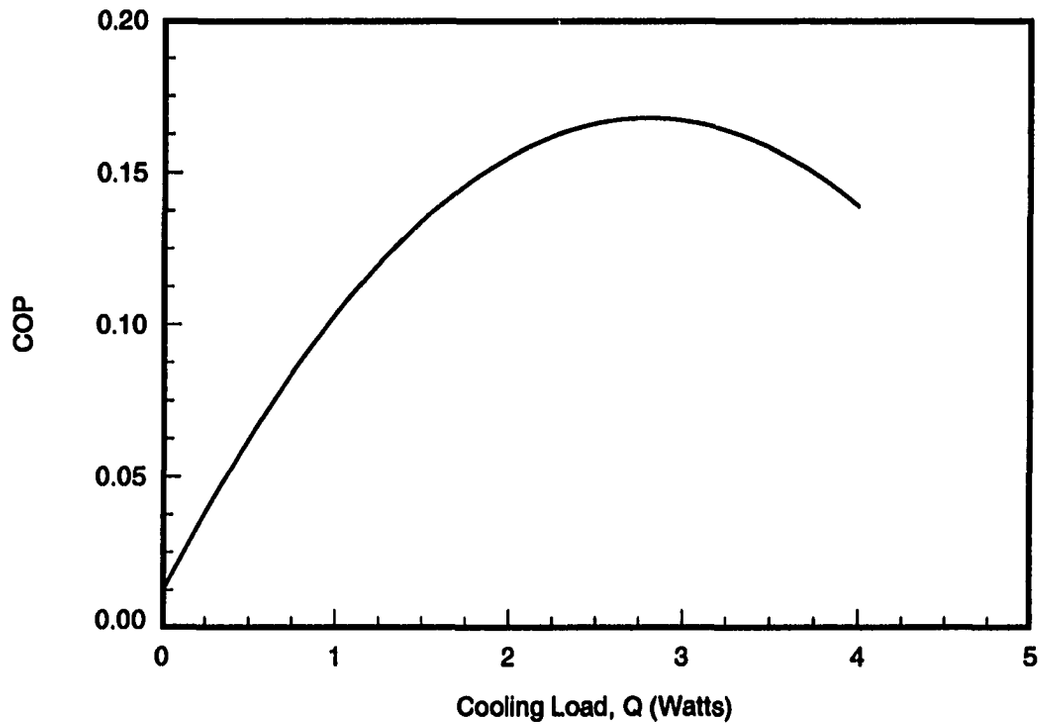

Figure 7.8: Measured coefficient of performance for the STAR refrigerator [62].

CHAPTER 8. THERMOELECTRIC REFRIGERATION

Introduction

Refrigeration can be accomplished by the direct conversion of electrical to thermal energy. Thermoelectric refrigeration technology has been used commercially to cool electronic equipment and for small portable refrigerators used for recreational activities [41].

Thermoelectric devices use two dissimilar semi-conducting materials, one P-type and one N-type. Since these materials are generally poor conductors of heat [42], they are often joined by a conducting material such as copper to form the junction between the two.

A thermodynamic analysis of the thermoelectric refrigeration system requires the consideration of the following physical phenomena:

1. Joulean power loss.
2. Seebeck effect.
3. Peltier effect.
4. Thomson effect.
5. Heat transfer.

Literature Search

A literature search was conducted to find information regarding thermoelectric energy conversion methods and the physical effects which the determine the principle of operation.

The Seebeck, Peltier, and Thomson (later Lord Kelvin) effects are discussed in electrical engineering and materials science texts; for example, Smith [42], and Van Vlack [43]. A relationship between these effects was first predicted by Thomson. Thomson assumed the system to be internally reversible. Results of his experiments were verified by Jaumont [44]. The inter-dependence of the Seebeck, Peltier, and Thomson effects was later demonstrated to be the same in a system which was assumed to be irreversible [45].

An analysis of the thermodynamics of direct conversion of thermal to electrical energy was conducted by Ioffe [46] and Angrist [47]. Ioffe also conducted a study of the thermodynamics of thermoelectric refrigeration [48].

Theory

Introduction

The thermal and thermoelectric effects which determine the operation of a thermoelectric refrigeration system will be discussed briefly, followed by a discussion of the development of the idealized thermoelectric refrigeration model. Two expressions for the coefficient of performance (COP) will be derived. The first is a general expression of the COP for the thermoelectric refrigerator; the second is an expression for the maximum possible COP for a given pair of P- and N-type materials.

Joulean Power Loss

When an electric current, I, flows against an electrical resistance, R, there is an irreversible conversion of electrical to thermal energy,

$$\dot{Q}_j = I^2 R_j \ \ (watts). \tag{8.1}$$

where,

I = the current (*amperes*).
R_j = the electrical resistance of the junction (*ohms*).

The electrical resistance, R, is determined by the resistivity, ρ, and the length to area ratio of the conducting material.

Seebeck Effect

If the junctions of two dissimilar electrical conductors are at different temperatures, T_H and T_L, an electrical potential will be created at each junction. This voltage is equal to the sum of the voltages at each junction.

The Seebeck Coefficient, α, is defined as the constant of proportionality relating the change in potential to the temperature change,

$$\alpha = \frac{dV}{dT} \ \left(\frac{volts}{degree}\right). \tag{8.2}$$

The Seebeck voltage for one material can then be written as,

$$V_i = \int_{T_L}^{T_H} \alpha_i dT. \tag{8.3}$$

For two materials in a circuit, the Seebeck voltage is the sum of the voltages in the circuit,

$$V_i = \oint \alpha_i dT \tag{8.4}$$

$$= \int_{T_L}^{T_H} \alpha_P dT + \int_{T_L}^{T_H} \alpha_N dT \tag{8.5}$$

$$= \int_{T_L}^{T_H} \left(\alpha_P - \alpha_N\right) dT. \tag{8.6}$$

Peltier Effect

The Peltier effect occurs at the junction of two dissimilar materials because energy must be conserved. When direct current flows through the junction, electrical energy is brought to the junction by charge carriers in the first material at rate $\dot{Q}_1$ and carried from the junction through the second material at rate $\dot{Q}_2$. The Peltier coefficient for each material is defined as,

$$\pi_i = \frac{\dot{Q}_i}{I} \left(\frac{watts}{ampere}\right) = volts. \tag{8.7}$$

The junction has a finite electrical resistance, R_j, which results in the irreversible dissipation of Joulean power. The net heat transfer rate at the junction can be found by applying an energy balance to the junction while operating at steady state,

$$\dot{Q}_1 - \dot{Q}_2 - \dot{Q}_j + I^2 R_j = 0$$

$$\dot{Q}_j = \dot{Q}_1 - \dot{Q}_2 + I^2 R_j$$

$$\dot{Q}_j = I\left(\pi_1 - \pi_2\right) + I^2 R_j \tag{8.8}$$

$$= I\pi_{1\ 2} + I^2 R_j. \tag{8.9}$$

The Peltier effect can provide a net heating or cooling effect, depending upon the relative magnitudes of the Peltier coefficients of the two junction materials, or by a reversal in the direction of current flow in the circuit. The cooling effect cannot be as large as the heating effect since the Joulean power loss term is always positive.

Thomson Effect

When a temperature gradient exists in a homogeneous material through which electric current is flowing, a voltage gradient also exists due to thermal agitation of the charge carriers. This voltage gradient is additive to the customary voltage drop resulting from the resistance of the material. The Thomson coefficient is defined as,

$$\tau = \frac{\Delta V_\tau}{\Delta T} \quad \left(\frac{volts}{K}\right). \tag{8.10}$$

A radial heat loss from the material occurs. This heat loss is the sum of the Thomson power loss and the Joulean power loss in the material,

$$\dot{Q}_r = I\tau\Delta T + I^2 R. \tag{8.11}$$

Thermoelectric Refrigeration Model Development

Figure 8.1 illustrates a thermoelectric refrigerator in communication with a thermal source and sink. Heat is accepted from the source at T_L and rejected to the sink at T_H. Direct current electric power is supplied to the system from an external source.

Figure 8.2 is a schematic of the thermoelectric refrigeration circuit. It is assumed that conductors C_1, C_2, and C_3 are isothermal at temperatures T_H and T_L, respectively. Furthermore, these conductors are assumed to have negligible thermal

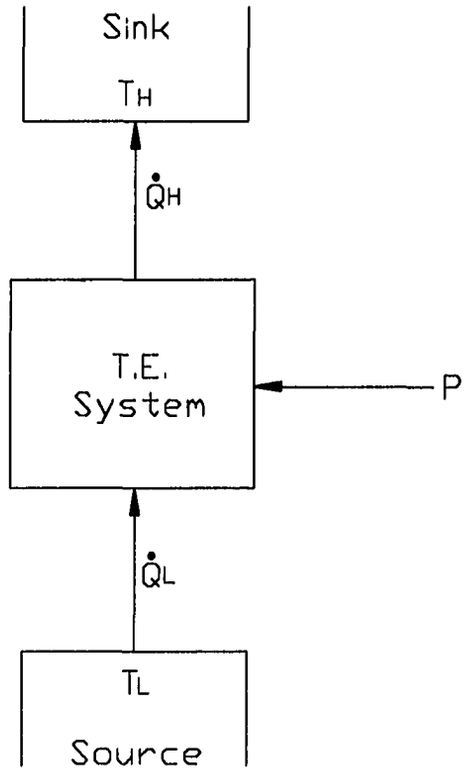

Figure 8.1: Schematic of a thermoelectric refrigerator and its surroundings.

and electrical resistances as compared to the P- and N-type semiconductors. Both semiconductors are assumed to have a uniform rectangular cross-section; the electrical resistivity (ρ_i), thermal conductivity (k_i), and Seebeck coefficient (α_i) for each material are assumed to be constant.

Heat is conducted through both legs of the refrigerator in parallel whereas the current flows in series. The combined coefficient of heat transfer, K, can be expressed

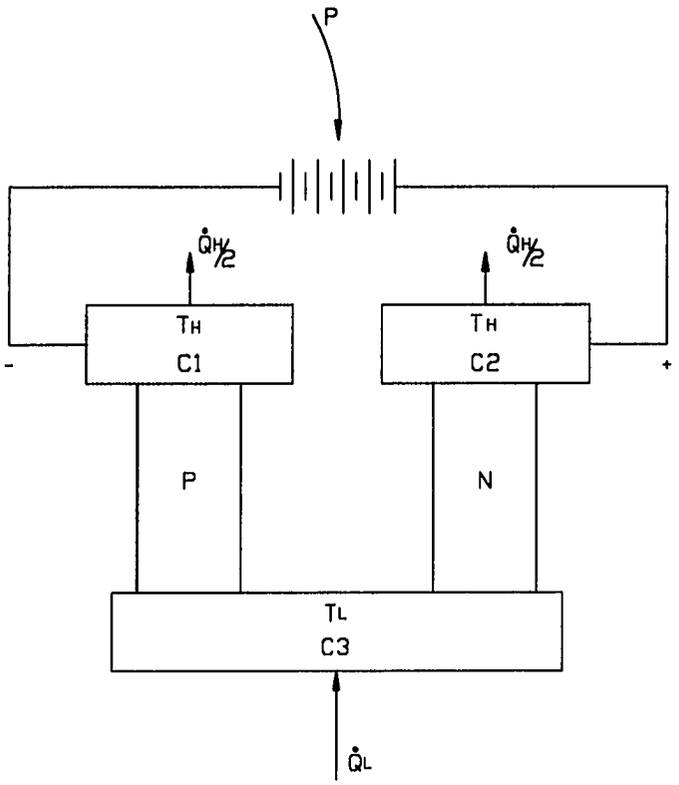

Figure 8.2: Schematic of a thermoelectric refrigeration circuit.

as,

$$K = k_N\left(\frac{A_N}{l_N}\right) + k_P\left(\frac{A_P}{l_P}\right). \tag{8.12}$$

Assuming the electrical resistances of the material junctions and other circuit members are negligible in comparison to those of the two semiconductors, the total resistance can be calculated,

$$R_t = \rho_N\left(\frac{l_N}{A_N}\right) + \rho_P\left(\frac{l_P}{A_P}\right). \tag{8.13}$$

The rate at which heat can be accepted, $\dot{Q}_L$, and rejected, $\dot{Q}_H$, by the device can

be expressed in terms of the Peltier effect, heat conduction rate, and Joulean power loss rate. The Peltier effect at the junctions can be expressed as:

$$\dot{Q}_{P\ cold\ junction} = I\pi_{PN} = I\alpha_{PN}T_L. \tag{8.14}$$

$$\dot{Q}_{P\ hot\ junction} = -I\pi_{PN} = -I\alpha_{PN}T_H. \tag{8.15}$$

The relationship between the Peltier and Seebeck coefficients in the above equations can be derived using the definition of the Peltier coefficient and Kelvin's second relation,

$$T\frac{V_{PN}}{dT} = T\alpha_{PN} = \pi_{PN}. \tag{8.16}$$

The sign change in these equations is due to the change in direction of current flow in the cold and hot junctions. The cooling rate can now be expressed as,

$$\dot{Q}_L = I\alpha T_L - K\Delta T - \frac{1}{2}I^2R_t. \tag{8.17}$$

Similarly, the rate of heat rejection can be written as,

$$\dot{Q}_H = I\alpha T_H - K\Delta T + \frac{1}{2}I^2R_t. \tag{8.18}$$

The Joulean power loss is distributed equally between the hot and cold ends of the semiconductors.

The electrical power required, P, is then the difference in the heat rejection and acceptance rates,

$$P = \dot{Q}_H - \dot{Q}_L = I\alpha\Delta T + I^2R_t. \tag{8.19}$$

Coefficient of Performance

An equation for the COP for a thermoelectric refrigerator can be derived from the definition of the COP, Equation 8.17, and Equation 8.19,

$$\begin{aligned} COP &= \frac{\dot{Q}_L}{P} \\ &= \frac{I\alpha T_L - K\Delta T - \frac{1}{2}I^2R_t}{I\alpha\Delta T + \frac{1}{2}I^2R_t}. \end{aligned} \tag{8.20}$$

Although the coefficient of performance of the ideal thermoelectric device has been found, a more useful form for determining the maximum COP can be expressed in terms of the Carnot COP and a figure of merit, Z [41].

To find the maximum COP, the numerator and denominator in equation 8.20 are multiplied by $\frac{R_t}{\alpha^2}$ to obtain,

$$COP = \frac{\left(\frac{IR_t}{\alpha}\right)T_L - \left(\frac{R_tK}{\alpha^2}\right)\Delta T - \frac{1}{2}\left(\frac{I^2{R_t}^2}{\alpha^2}\right)}{\left(\frac{IR_t}{\alpha}\right)\Delta T + \left(\frac{I^2{R_t}^2}{\alpha^2}\right)}. \tag{8.21}$$

The figure of merit, Z, is defined as,

$$Z = \frac{\alpha^2}{R_tK}. \tag{8.22}$$

The Seebeck coefficient,α, depends upon the P- and N-type materials used in the system. The heat transfer coefficient, K, and the resistance, R_t, are functions of the material types and the geometry of the device. For a particular pair of materials, the maximum Z occurs when the denominator, R_tK, is minimized with respect to the area-to-length ratios of the semi-conducting materials.

To minimize the denominator (R_tK) in Equation 8.22, Equations 8.12 and 8.13 are combined. Equations 8.12 and 8.13 are written in terms of the area-to-length

ratios of the two junction materials. Letting the variable β represent the area/length ratio, multiplying Equation 8.13 by Equation 8.12, and taking the derivative with respect to $\frac{\beta_N}{\beta_P}$ yields the following expression:

$$(R_t K)_m in = \left[\left(\rho_N k_N\right)^{0.5} + \left(\rho_P k_P\right)^{0.5}\right]^2. \tag{8.23}$$

Therefore, the maximum Z for a particular semiconductor pair is,

$$Z_{max} = \frac{\alpha_N P^2}{(R_t K)_{min}}. \tag{8.24}$$

The maximum COP for an ideal thermoelectric refrigeration system with a particular pair of semi-conducting materials can now be found by taking the derivative of Equation 8.21 with respect to the term $\left(\frac{IR_t}{\alpha}\right)$, setting the result equal to zero, and solving for $\left(\frac{IR_t}{\alpha}\right)$. Inserting this expression into equation 8.21 yields an expression for the maximum COP for an ideal thermoelectric refrigeration cycle:

$$COP_{max} = \left(\frac{T_L}{T_H - T_L}\right)\left[\frac{\left(1 + ZT_{avg}\right)^{0.5} - \frac{T_H}{T_L}}{\left(1 + ZT_{avg}\right)^{0.5} + 1}\right]. \tag{8.25}$$

where,

$$T_{avg} = \frac{\left(T_H + T_L\right)}{2}.$$

The first term on the right side of equation 8.25 is the Carnot COP, the bracketed term is a measure of the internal irreversibility of the thermoelectric refrigeration system. Equation 8.25 accounts for the major sources of irreversibility in the system; however, it does not take into account Joulean heating in the conductors and junctions, heat transfer due to radiation, or losses associated with heat exchange with the source and sink.

Results

A FORTRAN subroutine was written to calculate the COP of a thermoelectric refrigeration system using the equations developed in the previous section. This routine was used to calculate the COP and cycle efficiency for a thermoelectric system rejecting heat to a thermal sink at 35 C for source temperatures ranging from −24 C to 28C. Four values of Z were considered. A 5 C minimum approach temperature was assumed to exist between the thermal source and sink and the hot and cold surfaces of the thermoelectric refrigeration system.

Figure 8.3 is a graph of the COP versus the source temperature. The source temperatures range from −24 C to 28 C. Four different values of Z were considered. The COP increases with the source temperature with the greatest increase occurring above 10 C. The COP also increases with increasing values of Z.

Figure 8.4 is a graph of the cycle efficiency, η_C, for the same source temperature range. The maximum cycle efficiency was approximately 0.075 at 14 C for $Z = 0.003$. Increasing Z has two effects: increased cycle efficiency and a shift of the maximum cycle efficiency to a lower source temperature. It should be noted that the highest value of Z for semiconductor pairs is presently 0.003.

Table 8.1 provides a comparison of the COPs for selected refrigeration and air-conditioning applications at $Z = 0.003$, and a 5 degree C temperature difference between the heat exchangers and the air.

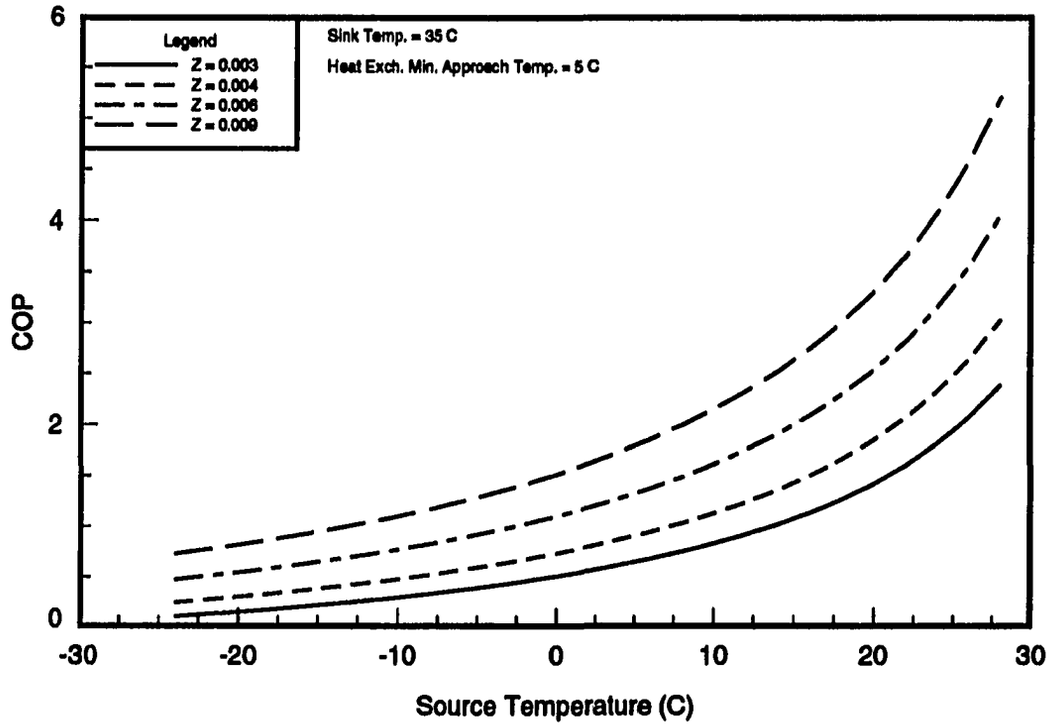

Figure 8.3: COP vs. source temperature for an ideal thermoelectric refrigerator.

Conclusions

The thermoelectric refrigeration system presently offers the advantages of simple and compact design, high reliability, and the utilization of non-ozone depleting working materials. However, the COP of current thermoelectric systems is low as compared to vapor-compression systems.

Equation 8.25 indicates that improvement of the COP hinges upon finding semiconducting material pairs which have a larger figure of merit, Z. Presently the highest value of Z is approximately 3.25×10^{-3} $\left(\frac{1}{K}\right)$ if radiant heat transfer is neglected. If the radiant load is considered, the maximum effective value of Z has been found

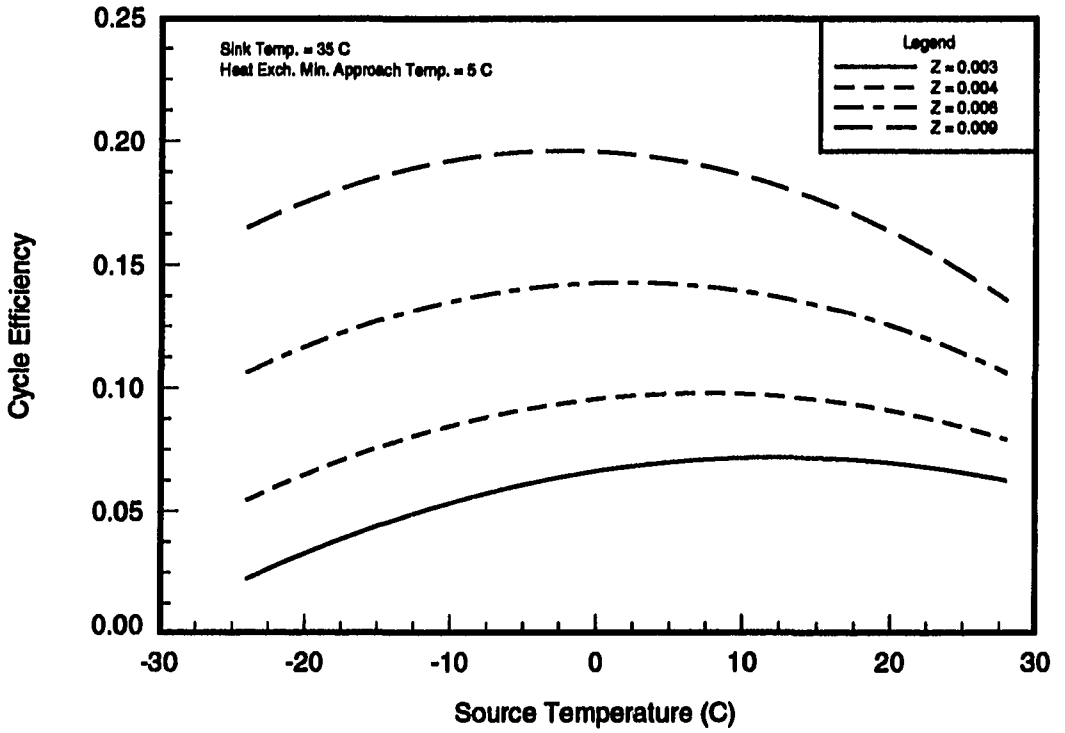

Figure 8.4: Cycle efficiency vs. source temperature for an ideal thermoelectric refrigerator.

to be 2.74×10^{-3} $\left(\frac{1}{K}\right)$ [49]. Research continues in the quest for semiconductor pairs which will provide a higher value of Z [47]. Until these materials are developed, thermoelectric refrigeration is not a viable alternative to vapor-compression for refrigeration and air conditioning applications.

Table 8.1: Thermoelectric cycle coefficients of performance for different refrigeration and air conditioning applications.

Application	T_{source} (C)	T_{sink} (C)	COP
Domestic Refrigeration	−17.8	35.0	0.18
Commercial Refrigeration, ARI Grp. 1	2.8	35.0	0.58
Commercial Refrigeration, ARI Grp. 2	1.7	35.0	0.56
Commercial Refrigeration, ARI Grp. 3	−2.2	35.0	0.50
Commercial Refrigeration, ARI Grp. 4	−23.3	35.0	0.11
Domestic Air Conditioning	25.0	35.0	1.95
Commercial Air Conditioning	25.0	35.0	1.95
Mobile Air Conditioning	25.0	35.0	1.95

CHAPTER 9. MAGNETIC REFRIGERATION

Introduction

The magnetic heat pump transports heat from a cold reservoir to a hot reservoir by means of a material with suitable magnetic properties in the presence of a magnetic field. The cooling process which results from the alternate magnetization and demagnetization of the material is known as the magnetocaloric effect.

The magnetic heat pump has been proposed as an alternative to vapor-compression cycles for refrigeration and air conditioning applications.

Literature Review

Magnetocaloric cooling was first employed in 1933 when Giauque and MacDougal [59] conducted adiabatic demagnetization experiments to produce cooling at source temperatures from −269.65 C to −272.65 C.

Magnetic cryogenic refrigerators were developed by Heer in 1949 [60] and by Rosenblum in 1976 [61]. These were low-power systems for operation below −272.15 C. Both devices operated intermittently in what could be termed a semi-continuous refrigeration process.

G. V. Brown [62] investigated the application of the magnetocaloric effect to refrigeration applications near room temperature in 1976. Brown concluded that

a magnetic refrigerator operating at 1 C was theoretically feasible. He proposed a system in which a gadolinium core would be translated back and forth in a tube filled with a recuperative fluid. Electrical coils located at the ends of the tube would apply the magnetic field.

The feasibility of magnetic refrigerators using a rotating core which passes through regions of high and low magnetic field with regeneration accomplished using a recuperative fluid has been studied by Steyert [66], Kirol and Dakus [67], and Hull and Uherka [68, 69].

Barclay [70] made a comparison of the expected thermodynamic losses in gas and magnetic refrigeration systems. He also provided an overview of the applications of magnetic refrigeration at source temperatures from −273.05 C to near room temperature.

Chen et al. [63] conducted an assessment of several theoretical thermodynamic refrigeration cycles for refrigeration applications requiring source temperatures above the cryogenic temperature range.

Theory

Ideal Magnetic Materials

Some materials exhibit a physical characteristic in which the entropy level of the material changes when the strength of a magnetic field surrounding the material is varied. As the field strength is increased, the entropy level of the material decreases. If the temperature of the material is to remain constant, heat must be rejected from the material specimen to the surroundings. Conversely, if the magnetic field strength

is decreased, heat must be accepted from the surroundings if the material specimen is to remain at a constant temperature. This phenomenon is the magnetocaloric effect.

An ideal magnetic substance is one which behaves according to a magnetic equation of state known as the Curie law,

$$M = \frac{C_C H}{T}. \tag{9.1}$$

where,

M = The magnetization.

H = The external magnetic field.

C_C = The Curie constant.

Modified Helmholtz and Gibbs functions, magnetic Tds equations, functional thermodynamic relationships, and the Curie law can be used to derive the thermodynamic properties internal energy, enthalpy, and entropy for ideal magnetic materials. Furthermore, constant field and constant magnetism specific heats can be derived. These ideal magnetic properties are analogous those derived for gases. Two differences between the thermodynamic properties of ideal magnetic solids and ideal gases are important in the development of magnetic refrigeration cycle theory:

1. Lines of constant pressure and constant volume remain parallel on the T-s diagram for an ideal gas.

2. Lines of constant magnetism remain parallel on the T-s diagram for an ideal magnetic material; however, lines of constant field strength *do not.*

Figure 9.1 is a temperature versus specific entropy diagram for an ideal magnetic substance which obeys the Curie law. Lines of constant field strength and lines

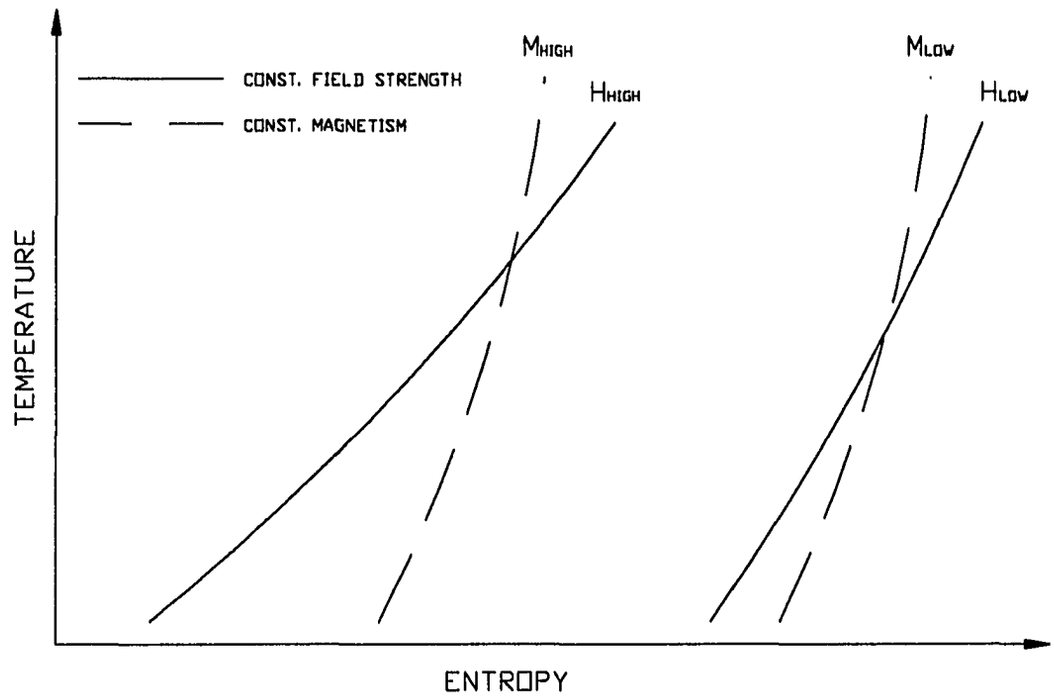

Figure 9.1: Temperature versus specific entropy diagram for an ideal ferromagnetic material.

of constant magnetism are illustrated. These constant property lines are analogous to lines of constant pressure and volume for an ideal gas, with one important exception: Lines of constant field strength are not independent of temperature. The horizontal distance between lines of constant field strength decreases with increasing temperature; i.e., the lines tend to converge as temperature increases and diverge as temperature decreases.

Thermodynamic Properties of Actual Magnetic Materials

Actual magnetic materials exhibit a characteristic temperature at which the material changes from a ferromagnetic material to a paramagnetic material. This temperature is known as the Curie temperature or Curie point. Most materials which exhibit a suitable magnetocaloric effect for magnetic refrigeration have a Curie temperature below 20 C.

One consequence of the transition from ferro- to para-magnetism is illustrated in Figure 9.2. The rate of temperature change with respect to entropy at a constant field strength, $\left.\frac{dT}{ds}\right)_{H=c}$, decreases dramatically above the Curie temperature. This phenomenon is particularly pronounced for the zero field strength line where a sudden change in the slope occurs at the Curie temperature.

If the magnetic working material is a solid and assumed to be incompressible, the infinitesimal work interaction is due solely to the opposition to changes in orientation of the magnetic dipole moments of the material when in the presence of a magnetic field,

$$\delta W = -HdM. \tag{9.2}$$

This change in entropy is brought about by three separate interactions of particles in the material. From the strongest to the weakest, these entropy changes are:

1. Ion spin, ΔS_M.
2. Lattice vibration, ΔS_L.
3. Conduction electron flow, ΔS_E.

The total entropy change, ΔS_T, is the sum of the three individual changes.

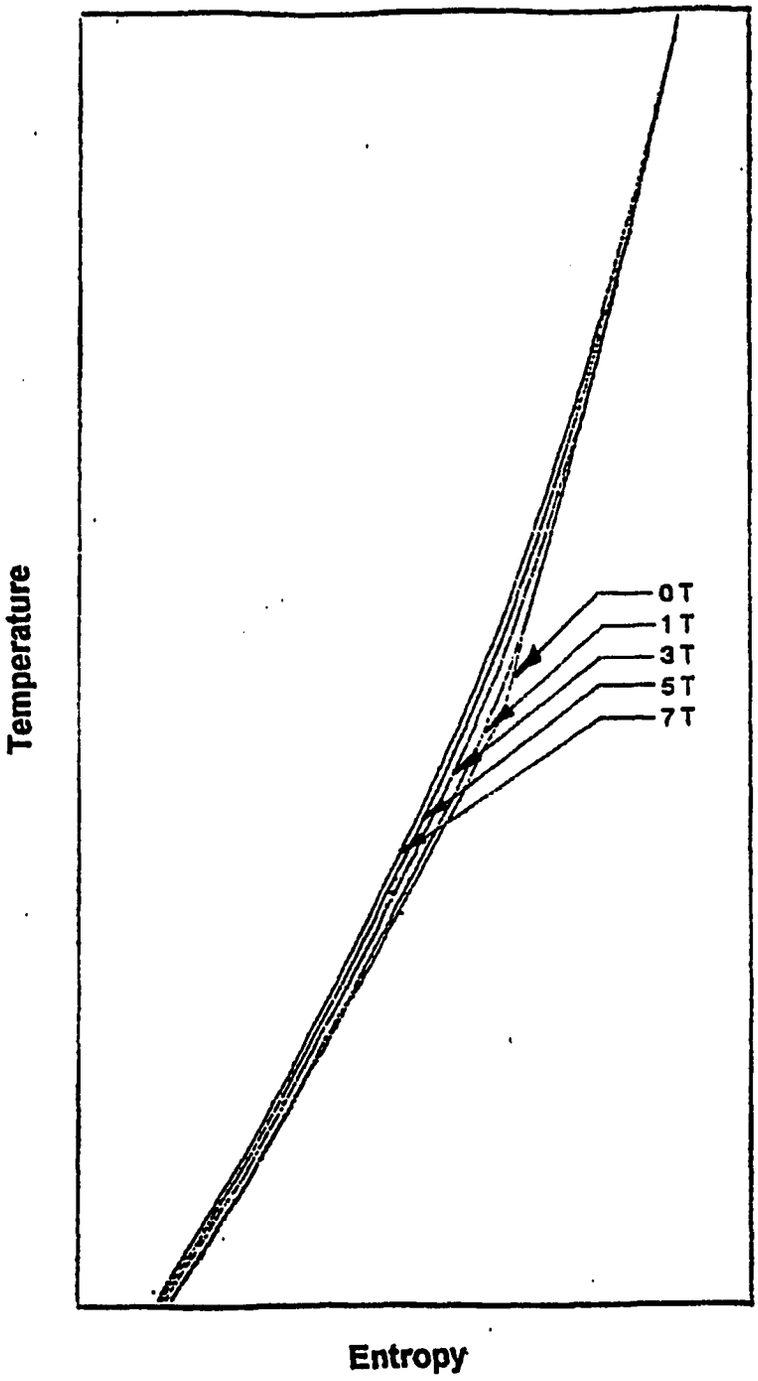

Figure 9.2: Temperature versus entropy at constant field strength diagram for a ferromagnetic material.

Chen et al. [63], present set of equations which can be used to calculate the theoretical change in ΔS_M, ΔS_L, and ΔS_E. The method was validated by demonstrating agreement between calculated values and experimental data obtained by Griffel [64].

Several theoretical refrigeration cycles can be constructed by combining a series of isofield or isomagnetism and isothermal processes. These include the "magnetic" Stirling and Ericsson cycles and a cycle composed of isothermal, isofield, and isentropic processes.

Magnetic Stirling Cycle

Figure 9.3 illustrates a magnetic Stirling cycle for an ideal magnetic material. The material is heated isomagnetically from state 1 to state 2. Heat is rejected isothermally from state 2 to state 3. The material is cooled during a second isomagnetism process from state 3 to state 4. Finally, the cycle is completed by an isothermal heat acceptance process from state 4 to state 1. The heating and cooling during the isomagnetism processes is accomplished through regeneration. Since the two lines of constant magnetism are parallel, this cycle would be capable of perfect regeneration.

It can be shown that the COP for this cycle is the COP_{Carnot}. However, if the system boundaries are chosen to include the hot and cold thermal reservoirs and that a temperature difference will exist at each reservoir and heat exchanger interface, the COP for the magnetic Stirling cycle would be,

$$COP_{CM} = \frac{T_L - \Delta T_L}{\left(T_H + \Delta T_H\right) - \left(T_L - \Delta T_L\right)}. \tag{9.3}$$

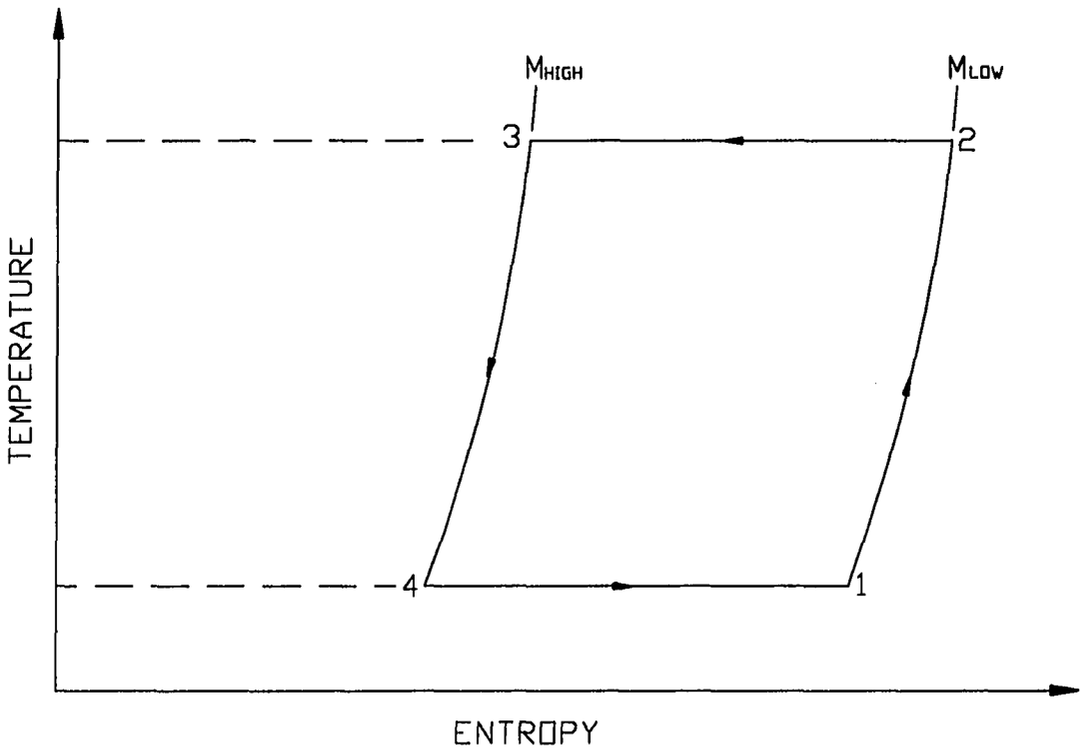

Figure 9.3: Temperature versus specific entropy diagram for a magnetic Stirling cycle.

Magnetic Ericsson Cycle

A magnetic Ericsson cycle is depicted in Figure 9.4. This cycle consists of two isofield and two isothermal processes. Since the lines of constant field strength do not remain parallel over the temperature range, additional heat would be required from an external source for energy to be conserved. This condition is illustrated in Figure 9.4 as a larger regenerative cooling area bounded by a-b-3-4-a in contrast to the regenerative heating area bounded by c-d-2-1-c.

The magnetic polytropic cycle, in which energy is conserved, is illustrated in Fig-

ure 9.5. To maintain ideal regeneration, the field strength during the cooling process from state 3 to state 4 would be continuously lowered as a function of temperature so as to maintain a constant entropy difference throughout the temperature range.

This would be analogous to a polytropic process in an ideal gas. The path of this polytropic process would then be parallel to the low field strength line. The COP for this cycle would also be COP_{Carnot} as it was for the magnetic Stirling cycle. Similarly, if the boundary of the system was chosen to include the hot and cold thermal reservoirs and a temperature difference exists at each reservoir/heat exchanger interface, the COP for the magnetic Stirling cycle would be,

$$COP_{ME} = \frac{T_L - \Delta T_L}{\left(T_H + \Delta T_H\right) - \left(T_L - \Delta T_L\right)}. \tag{9.4}$$

A second method of conserving energy during the cycle is the combined magnetic cycle shown in Figure 9.6. This cycle is composed of an isofield heating process from state 1 to state 2, isothermal heat rejection from state 2 to state 3, and isofield cooling from state 3 to state 4. Once all of the heat has been transferred during the regeneration process an isentropic process would proceed until the low temperature is reached. In other words, when the area bounded by a-b-3-4-a is equal to the area bounded by c-d-2-1-c, an isentropic process commences at state 4 and ends at state 5. Heat would then be accepted isothermally from 5 to 1.

During the combined cycle, all of the heat is transferred during the regeneration process as evidenced by the equal areas bounded by a-b-3-4-5-a and c-d-2-1-c. However, since the heat is now transferred through a finite temperature difference, an inherent internal irreversibility exists and the COP will be less than the Carnot coefficient of performance.

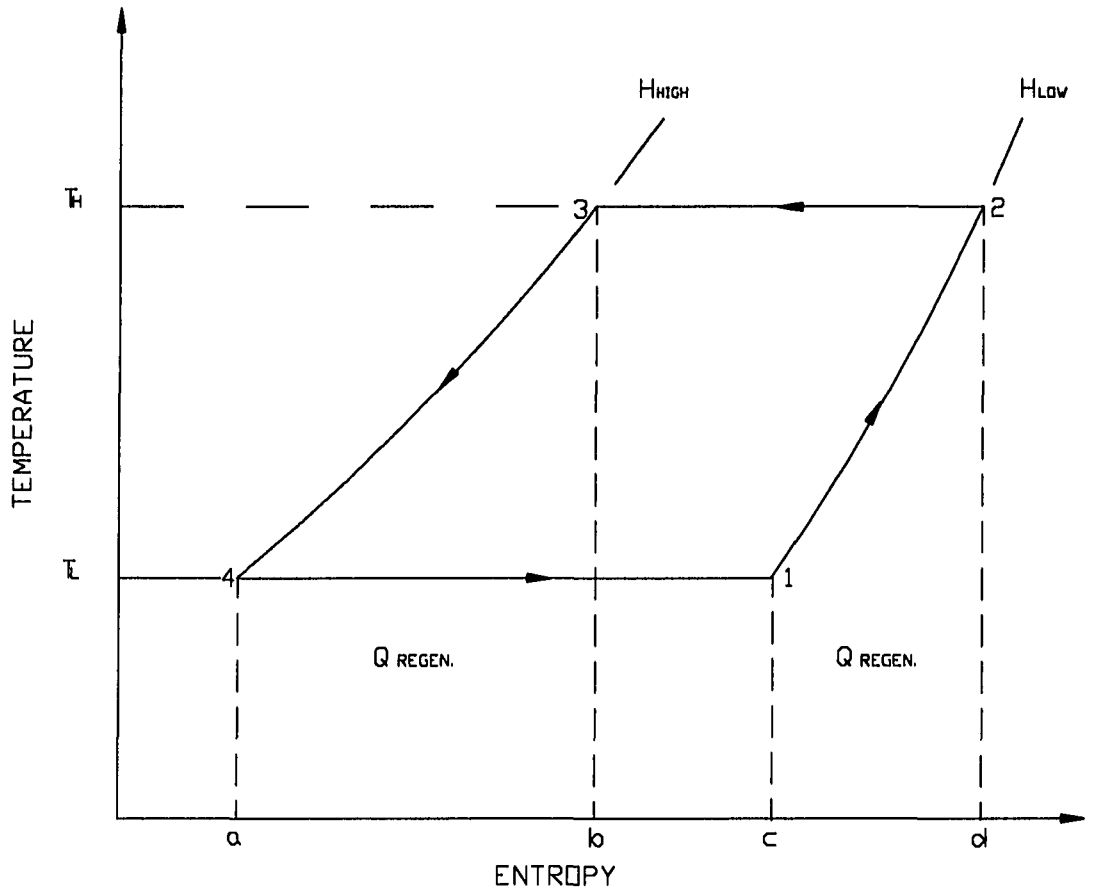

Figure 9.4: Temperature versus specific entropy diagram for a magnetic Ericsson cycle.

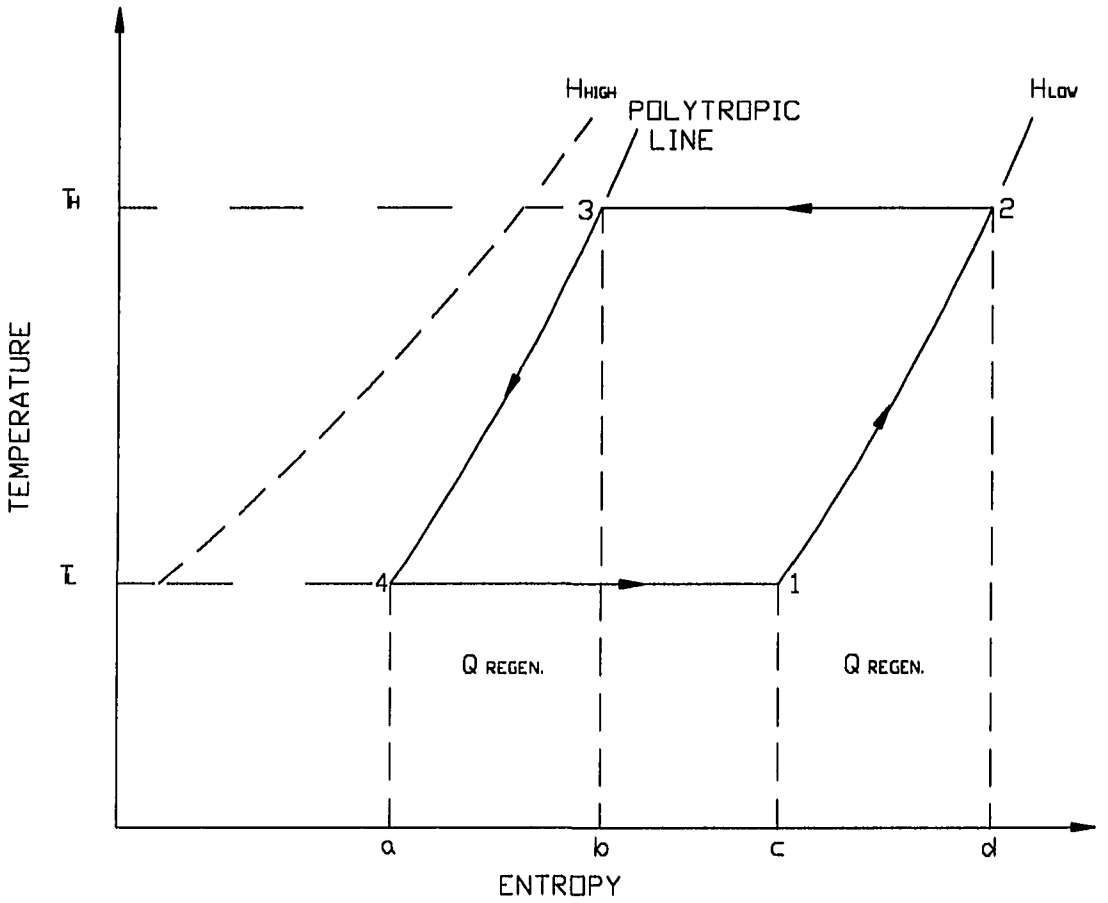

Figure 9.5: Temperature versus specific entropy diagram for a cycle involving a magnetic polytropic process.

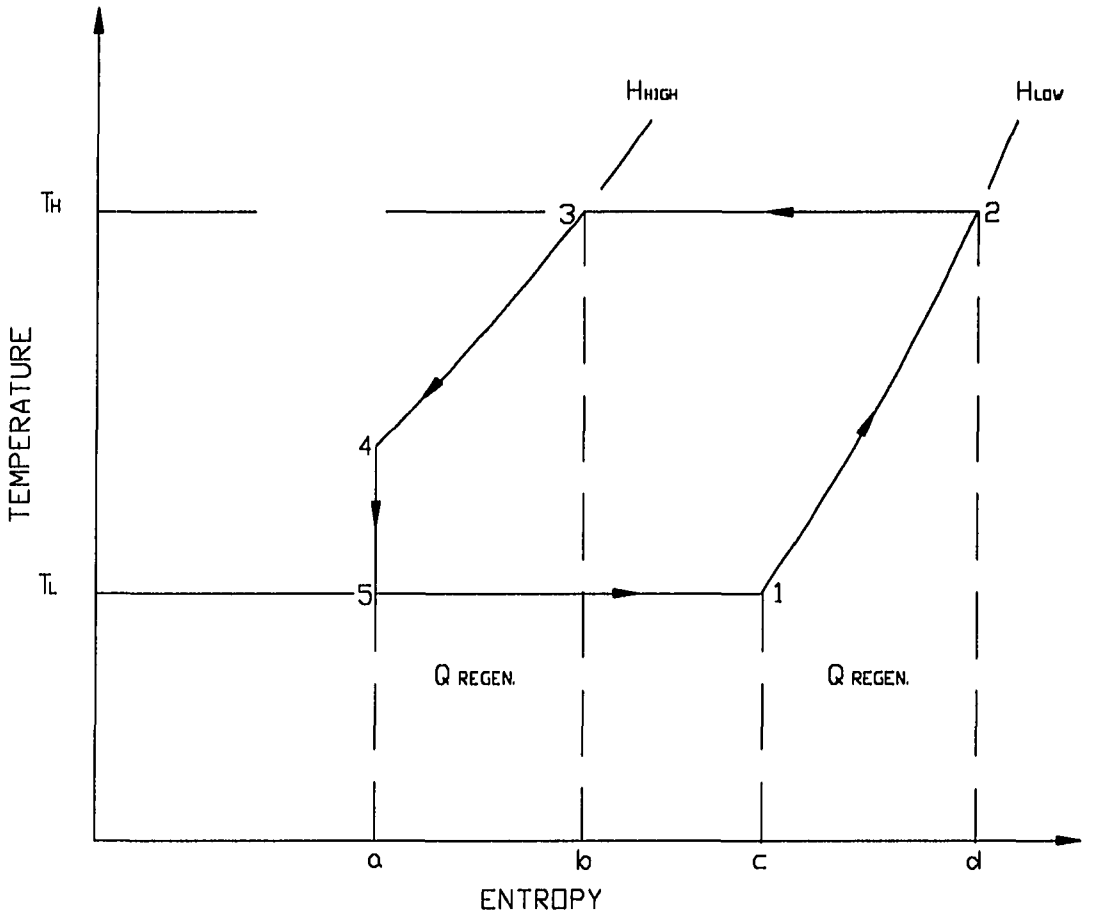

Figure 9.6: Temperature versus specific entropy diagram for a combined cycle.

Numerical Model

The coefficient of performance for the ideal magnetic Stirling and Ericsson cycles is identical to the COP for the ideal gas Stirling model. The ideal Stirling model in the alternative refrigeration cycle modeling program (Appendix B) can be used to estimate the COP for these cycles. A more realistic magnetic refrigeration cycle is the combined or ideal regenerative cycle. A computer subroutine developed for the combined magnetic cycle is discussed in the following sections.

Thermodynamic Properties

A consideration in developing a magnetic heat pump model for refrigeration and air conditioning applications is the behavior of real ferromagnetic materials in contrast to the ideal (Curie law) approximation. The best ferromagnetic materials, like gadolinium, have a Curie point in the temperature range of interest. As noted earlier, there is a sudden change in the temperature-entropy relationship at the Curie point. If the change in entropy was calculated based upon the Curie law equation of state, considerable error between calculated and actual values would exist, particularly at temperatures near the Curie point. Therefore, it was necessary to develop a thermodynamic property routine which was based upon the behavior of actual magnetic materials.

A thermodynamic property routine was written in FORTRAN using total entropy change versus absolute temperature data from Chen et al. [63]. The temperature data were for magnetic field strengths of 0, 1, 3, 5, and 7 Teslas for gadolinium. These data were fitted with a second-degree polynomial in both temperature and field strength. This function can be used to determine the entropy change due to

changes in temperature and field strength. A FORTRAN subroutine, GD.FOR, was written to calculate entropy change using the curve fitted polynomial. This subroutine returns a specific entropy value in $\frac{kJ}{Kg-K}$ for a given absolute temperature and a magnetic field strength in Teslas.

Combined Magnetic Cycle Numerical Model

The combined magnetic cycle model calculates the COP for a magnetic refrigerator operating between two field strengths.

Figures 9.7 and 9.8 illustrate the graphical technique used to calculate the COP for the combined cycle.

The following parameters are specified by the user:

1. Sink temperature.
2. High temperature heat exchanger minimum approach temperature.
3. Low temperature heat exchanger minimum approach temperature.
4. High magnetic field strength.
5. Low magnetic field strength.

The sixth parameter, source temperature, is established in the program. During execution of the numerical model, the source temperature is incremented from −24 C to 28 C.

The T-s diagram is broken into four areas, numbered I to IV. Between states 1 and 2, the area under the low field strength curve in found by numerical integration.

Area I, bounded by c-d-2-1-c represents the heat transferred from the regenerator to the gadolinium core,

$$Q_{Regen.,Right} = \int_{T_L}^{T_H} Tds\Big|_{H=H_L}. \tag{9.5}$$

Area II is bounded by the high temperature isotherm at the top, abscissa on the bottom, an isentropic line passing through state 3 on the left, and an isentropic line trough state 2 on the right.

The heat transferred from the gadolinium core to the regenerator, $Q_{Regen.,Left}$, is represented by area III bounded by a-b-3-4-a. This area is determined by performing a second numerical integration, beginning at state 3, and proceeding to the left along the line of high constant magnetic field. For steady state operation area III must equal area I. The program compares the current value of area III with area I. When the two areas are equal, state 4 is established. It is assumed that an isentropic process occurs between state 4 and state 5. Therefore, state 5 is known since it must lie on the low temperature isotherm and along the isentropic line passing through state 4.

Area IV, the amount of heat accepted from the thermal source, Q_L, can now be determined,

$$Q_L = T_L\,(s_1 - s_5). \tag{9.6}$$

The magnetic work is represented by the area enclosed by the high and low constant field lines and the high and low temperature isotherms,

$$W_M = A_{III} + A_{II} - A_I - A_{IV}. \tag{9.7}$$

The COP is,

$$COP = \frac{Q_L}{W_M}. \tag{9.8}$$

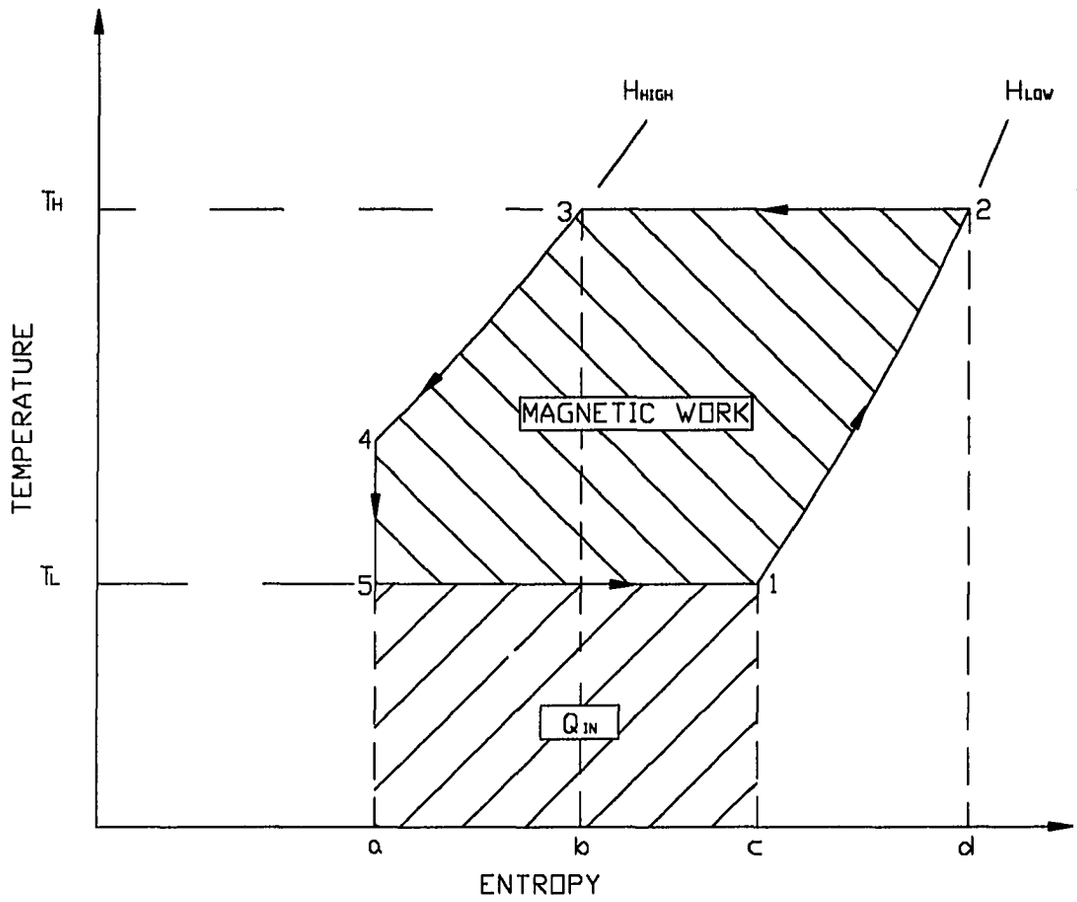

Figure 9.7: Temperature versus specific entropy diagram illustrating the determination of areas for the combined cycle.

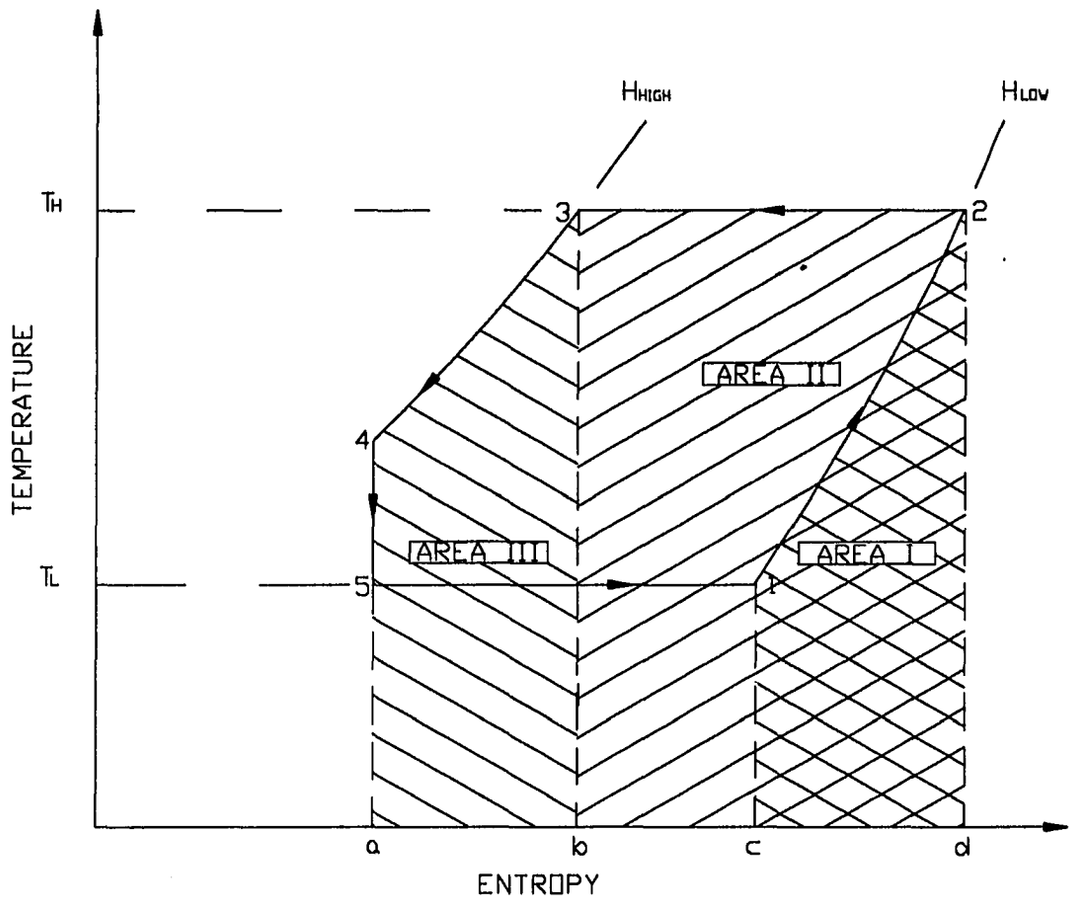

Figure 9.8: Temperature versus specific entropy diagram illustrating the magnetic work and heat acceptance areas for the combined cycle.

Results

From theory, the maximum COP for both the magnetic Stirling and magnetic Ericsson cycles is COP_{Carnot}. As with the gas Stirling model, it was assumed that the only irreversibility in the system was due to the temperature difference between the heat exchangers and the thermal source and sink. The COP for both cycles can then be expressed as,

$$COP = \frac{T_L - \Delta T_L}{\left(T_H + \Delta T_H\right) - \left(T_L - \Delta T_L\right)}. \tag{9.9}$$

The COP for the combined model was calculated using the "graphical" model.

The coefficients of performance for the magnetic Stirling, Ericsson, and combined refrigeration cycles were calculated for a sink temperature of 35 C at different source temperatures from −24 C to 28 C. The low and high field strengths were 0 and 7 Teslas. A minimum approach temperature of 10 C was assumed to exist between both heat exchangers and the thermal source and sink. Ten degrees, rather than five, was chosen because it was assumed that a fluid heat transfer loops would be interposed between the solid core and the source and sink. Thus two additional heat exchangers would be needed in the system with each heat exchanger having a 5 C minimum approach temperature. The loops were assumed to have adiabatic, inviscid flow. The pumping work was neglected. Figure 9.9 is a graph illustrating the maximum *theoretical* COP for the combined magnetic refrigeration cycle. In practice, the actual performance would be a small fraction of these values. The principal reasons are:

1. The need for almost perfect regeneration due to the small temperature lift of the magnetic core material.

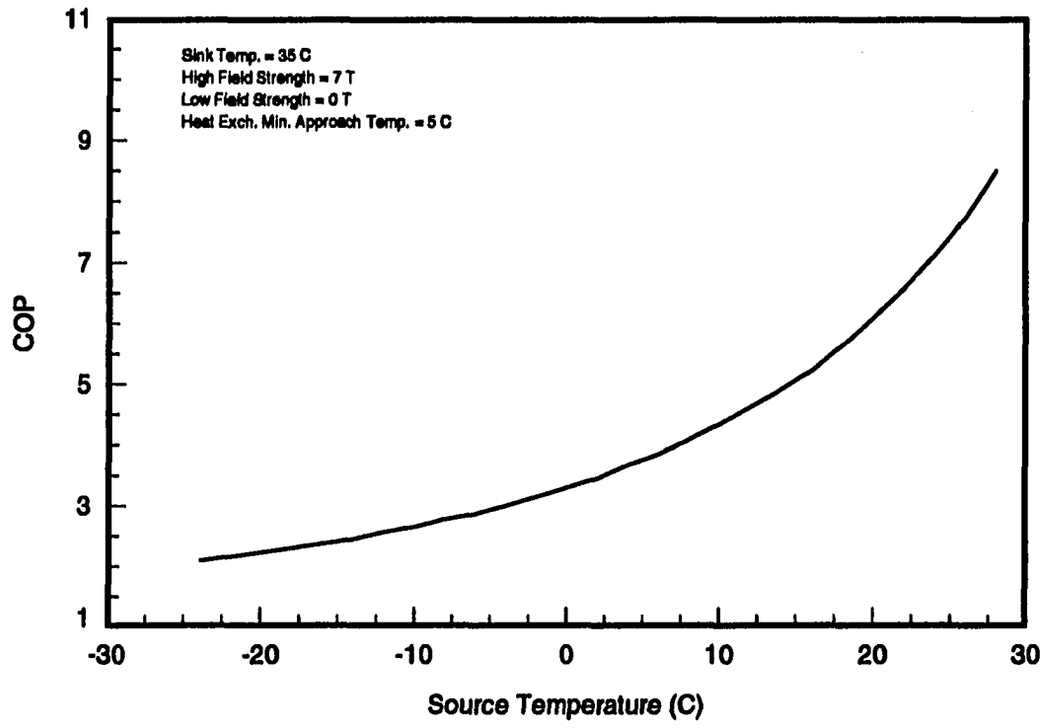

Figure 9.9: COP versus source temperature for an ideal combined magnetic refrigeration cycle.

2. Current electromagnet technology requires cooling at field strengths on the order of 7 Teslas.

Figure 9.10 illustrates the cycle efficiency for an ideal combined magnetic refrigeration cycle over a range of source temperatures. These efficiencies are much higher than can presently be attained in practice. The cycle efficiency increases with decreasing source temperature indicates that the cycle would be better suited to refrigeration, applications, rather than air conditioning.

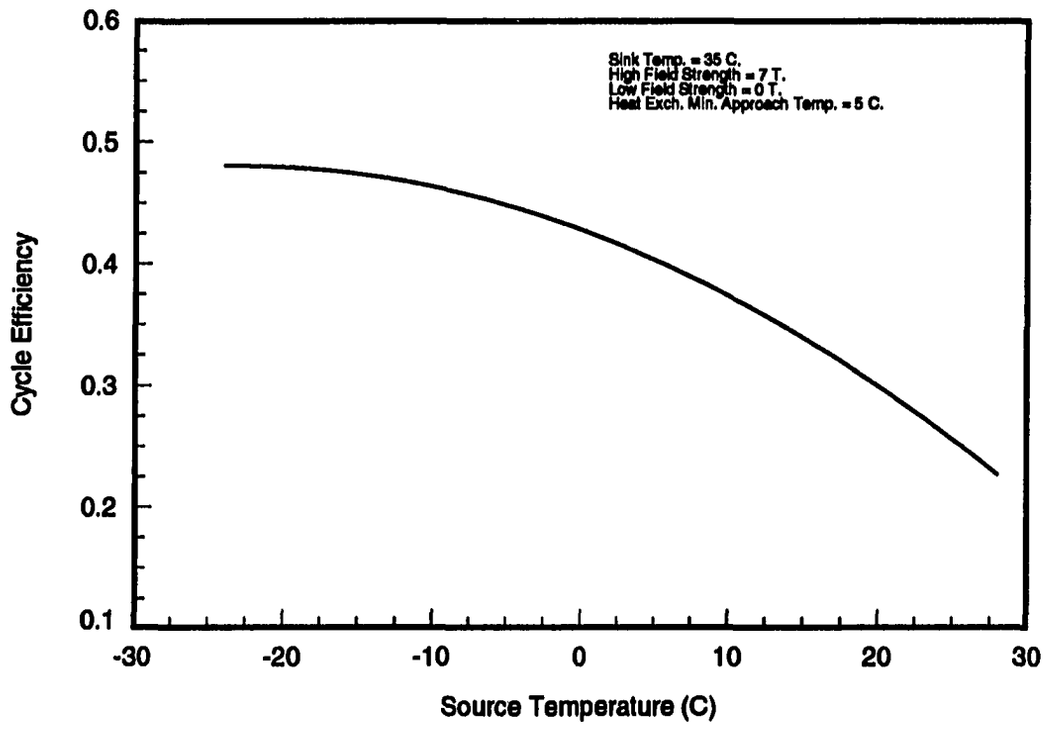

Figure 9.10: Cycle efficiency versus source temperature for an ideal combined magnetic refrigeration cycle.

CHAPTER 10. TECHNICAL ASSESSMENT OF ALTERNATIVE REFRIGERATION TECHNOLOGIES

Introduction

The assessment of alternative refrigeration technologies involved the evaluation of two fundamental criteria categories common to all refrigeration and air conditioning applications. These criteria are environmental acceptability and system cost.

Environmental Acceptability Considerations

Environmental acceptability considerations include:

1. Ozone depletion potential (ODP) of the working material. It was decided that only alternative refrigeration technologies which were capable of using working materials which are *not* ozone depleting would be considered in this study.

2. Global warming potential (GWP) of the refrigeration technology. There are two GWP components which can be contributed by a refrigeration system: direct GWP, and indirect GWP. Direct global warming results from the leakage or release of a working material which is known to have a high GWP. Indirect global warming results from the release of CO_2 into the atmosphere. CO_2 is a combustion product released during the burning of fossil fuels during the

generation of electricity or for heat. *All* refrigeration systems which utilize heat or electricity to drive the system which was derived from fossil fuel combustion contribute to global warming. Refrigeration systems which have a high COP require less heat to produce the same amount of cooling as refrigeration systems with a low COP. The COP of a refrigeration system is inversely proportional to its indirect GWP. Therefore, systems which have a high COP are desirable from a global warming perspective since they have a lower indirect GWP. Over the life of a refrigeration system, the indirect global warming contribution is many times larger than its direct global warming contribution. For the technologies assessed in this project, only the indirect global warming potential is considered through the energy costs, ie. COP, of the system.

3. **Toxicity of the working material.** The toxicity of working materials was considered. Materials which pose a known danger with grave consequences were not considered for this study.

4. **Flammability.** It is recognized that some working materials which have been or could be used in refrigeration systems are flammable. By the same token, it is known that fuels are intentionally burned to provide heating in buildings. If the liability issue is set aside for a moment, the comparative risk of fire resulting from the release of a flammable working gas from a modern closed refrigeration system is small. The risk could be further minimized through the use of devices which provide an alarm, stop the heat or work input, and provide external venting of the escaping flammable material. The incorporation of any or all of these devices into a refrigeration system would translate directly into

a higher cost of the system.

5. **Noise.** Noise emitted by a refrigeration system is an environmental factor which can be dealt with by providing vibration isolation, acoustic insulation, and other techniques to achieve acceptable sound pressure levels. The addition of these devices to a refrigeration system would result in higher first costs.

In conclusion, alternative refrigeration technologies with environmental assessment criteria which are unacceptable from an ODP or toxicity standpoint were not considered in this project. Technologies which could have other environmental hazards, such as noise, which could be minimized or eliminated by design were considered.

Cost Related Technology Assessment Considerations

Cost related technology assessment considerations include:

1. **State of the art.** Some alternative refrigeration technologies are more mature than others. Research and development were considered in two broad areas: basic technology development and system development. For this study, a basic technology was defined as one which is not unique to refrigeration and would have many potential applications in other areas. An example would be the development of materials with high-temperature superconducting (HTSC) properties to reduce electrical resistance losses. HTSC would improve the performance of the thermoelectric and magnetic refrigeration technologies. It would also be important in electric power generation and transmission, electric motor design, and other applications using electric power. Generally, improving a basic technology is extremely expensive and there are no guarantees of success.

For this project, system development costs were defined as the research and development (R & D) costs which would be incurred in advancing the maturity of a refrigeration technology to the point at which marketable systems could be sold. It would include costs for developing the system hardware beyond the prototype stage to its final marketable form, and building the infrastructure necessary to manufacture the system.

2. **Size/weight.** Size and weight considerations are important for many refrigeration applications. Larger, heavier systems with the same cooling capacity contain more raw material which increases the first costs of the system. Increased size and weight create higher first costs for structures in which they are used, or reduce the usable space within the structure. This is particularly true in the transportation industry where it is desirable to maximize the useful load of the vehicle.

3. **System complexity.** Assessment of system complexity includes considerations regarding the number of subsystems, number of moving parts, and exotic materials used in an alternative refrigeration system. The difficulty in manufacturing the system, including likely manufacturing techniques and precision, were also considered. These issues relate directly to the first cost of the refrigeration system.

4. **Useful life.** Useful life of the refrigeration system was defined as the length of time during which the major components would remain functional while operating with a nominal duty cycle and receiving normal maintenance. For example, the useful life of a domestic central air conditioner would be the life

of the compressor unit.

5. **Maintenance.** Maintenance cost considerations include the amount of repair and preventative maintenance required, skill level of maintenance personnel, portion of time an operator would need to attend to the system, likelihood of component failure, and recurring costs (such as the periodic recharging of a system with working material) for normal system operation.

6. **Energy/efficiency.** Two factors are affected by the efficiency of the system: the cost to operate the refrigeration system and the indirect component of the global warming potential. It is assumed for this study that all heat or electricity required to operate the refrigeration systems originates from the combustion of fossil fuels. The energy/efficiency criteria rating is based upon the cycle efficiency (percent of the Carnot COP) at which the refrigeration system could operate for a particular application. This rating is based upon what is technically feasible in the 1990s. As technology advances, the cycle efficiency of some the less mature technologies may improve. Therefore, some of these technologies may become more attractive in the future.

Rating Factors

In the following sections, the technical assessment criteria will be evaluated for the refrigeration technologies. A table which summarizes the technical assessment numerically is included at the end each technology assessment section. The number is a rating factor which is the investigators' best estimate on a scale of 1 (very low) to 5 (very high) of the merit of a particular technology for a technical assessment

criteria. A rating of 5 for a criteria would indicate that that aspect is particularly attractive for a technology. A rating of 1 would indicate that the technology was very unattractive with respect to the criteria being considered. Table 10.1 summarizes the linguistic interpretation of the extreme ratings (1 and 5) for each criteria. The rating numbers for the cycle efficiency criteria are listed in table 10.2.

Table 10.1: Linguistic interpretation of the numerical ratings for technology assessment criteria.

Criteria	Rating of 1	Rating of 5
State of the art	Theory only	Fully matured
Complexity	Very complex	Very simple
Size/Weight	High	Low
Maintenance	High	Low
Useful Life	Short	Long
Energy/Efficiency	0 to 0.12	Above 0.50

Table 10.2: Numerical definition of the energy/efficiency rating scale in terms of cycle efficiency (percent of the Carnot COP).

Energy/Efficiency Rating Number	Cycle Efficiency Range
1	0.00 to 0.12
2	0.13 to 0.24
3	0.25 to 0.36
4	0.37 to 0.49
5	Above 0.50

Examples of extreme technical assessment criteria ratings would be:

- A system which required no maintenance would receive a rating of 5 since there would be a major cost savings benefit and high reliability over the life of the system.
- An air conditioning system that has a cycle efficiency of 1% would receive an energy cost rating of 1 (very low) since it would be costly to operate and have a negative impact on the environment through additional indirect global warming since extra fuel would have to be burned to achieve the same amount of cooling as an efficient system.

Magnetic Refrigeration

Environmental Acceptability of the Technology

Currently, the best working materials for use in magnetic refrigerators operating at cooling temperatures above −43 C are gadolinium and gadolinium salts. Gadolinium is rare-earth metal which reacts slowly with oxygen and water. It is non-toxic and does not pose an ozone-depletion or direct global warming hazard.

Heat transfer systems would be necessary to exchange heat with the core during the cycle. These systems would use a liquid with a low viscosity and high thermal conductivity. This heat transfer technology is well developed. Non-toxic materials which do not pose a threat to the environment are available.

Questions exist within the medical community regarding the health effects of electromagnetic fields upon humans and animals [77, 78, 79]. Shielding to minimize the risk of exposure to the electromagnetic fields generated by magnetic refrigeration

may be possible. Addition of shielding would increase the first cost and weight of magnetic refrigeration systems.

The noise level for magnetic refrigerators is not expected to be higher than for vapor-compression systems. The source of noise would be liquid pumps for the heat transfer loops in fixed core systems and the machinery to move the core in rotary systems.

State of the Art

Although the existence of the magnetocaloric effect has been known since the early 1930s, magnetic refrigeration at temperatures of −23 C and above is a relatively new concept (1970s). Very little experimental work has been done at the higher (−23 C and above) source temperatures. All prior experimental work was at source temperatures in the low cryogenic range (approaching absolute zero).

Theoretically, ideal magnetic refrigeration cycles are capable of high cycle efficiencies (60% *to* 100% of COP_{Carnot}).

Experimentation has verified that small temperature lifts can be accomplished at source temperatures above −23 C. Technical problems in three basic areas must be overcome in order to realize high COPs from magnetic refrigerators:

1. Achieving more effective regeneration.

2. Producing higher field strengths in the magnets.

3. Developing materials capable of a higher temperature lift for a given magnetic field strength.

The practical magnetic field strength limit of current electromagnet technology is approximately 7 Teslas [63, 73]. Cooling of the magnets is required when they are operated at high field strengths to remove heat generated due to electrical resistance in the coil windings and to lower the resistivity of the winding material. Providing cooling for the magnets would create an additional cooling load on the magnetic refrigeration system, thus reducing its cycle efficiency. If materials were developed with high electrical conductivity at room temperature, the magnets could operate at higher field strengths without cooling. The net effects would be to reduce the electrical work input into the magnetic refrigeration system and provide magnets capable of operating at a higher field strength differential. Both improvements would result in higher COPs.

Magnetic refrigeration systems will require a regenerator which is capable of nearly ideal regeneration, due to the small temperature lift of the magnetocaloric effect. Therefore, the regenerator must be capable of heat transfer at very low temperature differences. At the same time pumping losses due to fluid friction must be kept to a minimum [62]. It is expected that regenerators for magnetic refrigeration will be expensive to develop and manufacture for these reasons. Waynert [74], DeGregoria [75], and Brown [62, 73] have investigated regenerator designs for magnetic refrigeration. All three concluded that developing regenerators with high effectiveness is the largest single technical challenge in designing a workable magnetic refrigerator using gadolinium cores.

The COP of all magnetic refrigeration cycles could be improved if the slope of the constant field lines, $\left.\frac{\partial T}{\partial s}\right|_H$ (illustrated on the temperature-entropy diagram in Figure 9.2), was greater for the working material used in the core. At present,

materials which exhibit this property relationship are unknown. The search for working materials with better inherent thermodynamic properties for high-temperature magnetic refrigeration would be expensive, with no guarantee of success.

Manufacturing development costs are also expected to be high. Once basic technology has been developed to raise the actual performance of the magnetic refrigerator to an acceptable level, hardware must be developed to create a marketable system. Manufacturing development costs would include design costs and tooling costs.

It is anticipated that the demand for high-efficiency electromagnets would exist in a variety of technological applications other than magnetic refrigeration. Therefore, a portion of the hardware development costs could be shared by other industries; even so, the manufacturing development costs are expected to be quite high as compared to vapor-compression technology.

Complexity

Magnetic refrigeration system costs due to hardware complexity are related to the cost of the working material and system components. Waynert [74] estimated the cost of gadolinium at slightly over 500 $\frac{\$}{kg}$. Approximately 1 kg of gadolinium is required per kW of cooling. It was projected that the cost would decrease to 100 $\frac{\$}{kg}$ in large quantities. The greatest material cost would result from building electromagnets capable of delivering the high magnetic field strengths necessary to create a significant magnetocaloric effect within the core.

Secondary heat transfer loops consisting of heat exchangers, pumps, piping, and a fluid would be needed to transport heat to and from the magnetic core, and through the regenerator. One heat exchanger would accept heat from the cooling space, the

other would reject heat to the environment. Providing these secondary heat transfer loops would involve an additional investment in heat exchangers, pumping equipment, piping, and fluid. The net effect of the added heat-exchange interface between the magnetic core and the thermal source and sink would be a reduction in the COP. This is due to the additional irreversibilities introduced by the minimum approach temperatures in the heat exchangers, pumping work, and heat losses throughout the piping and fluid storage system.

For mobile applications, an electrical source would have to be provided. If the vehicle were not powered by an external electricity source (such as the 3rd rail in a mass transit rail system) an electric generation or storage system would have to be carried on board. The capacity of this system would relate to the COP and cooling capacity of the refrigeration system as well as the generating or storage capacity of the electrical system. The need for an electrical system will be a major penalty for mobile magnetic refrigeration applications in terms of system complexity, size and weight, maintenance, and useful life.

Maintenance

It is expected that little maintenance would be required for the core, magnets, and solid-state controls used in magnetic refrigeration systems. However, the secondary heat transfer loops and the regenerator may require some periodic maintenance of the pumps, motors, heat exchangers, piping, and heat transfer fluid. This maintenance could include periodic flushing of the system and replacement of the fluid and rebuilding or replacing pumps and motors.

On-site repair of the actual magnetic system would most likely involve replace-

ment of subsystems such as controllers and magnets. These components would be discarded or rebuilt at another location.

Once a magnetic refrigeration system was in service, operation and maintenance levels and expertise would not appear to be markedly different than for vapor-compression systems. Little preventative maintenance is foreseen; when failure does occur, it is expected that components would be replaced, rather than repaired on site. Repair of electromagnets would probably be done on a regional or factory return basis. Control devices, heat exchangers, and other minor components would be disposable.

Since system repair would be a diagnosis and component interchange process, technicians repairing magnetic refrigerators would require skills similar to those needed to repair vapor-compression systems.

Useful Life

The life of magnetic refrigerators (both fixed core and displaced core systems) is expected to be comparable to that of vapor-compression systems. This conclusion is supported by the observation that electromagnet applications are generally capable of years of service prior to failure.

Magnetic refrigerators using the displaced core concept would require machinery to either rotate or reciprocate the core with respect to the magnets; common technology which is also capable of long life. Pumps to circulate heat transfer fluid could be off-the-shelf items of designs which have demonstrated reliability. Given the low reactivity of the core material and heat transfer fluid, corrosion related failures are not expected to be a problem. Finally, heat transfer fluid can be circulated at con-

stant, low pressure. Therefore, failure of the piping, regenerator, and heat exchanger walls due to fatigue will not occur since the pressure would be non-cyclic. Secondly, burst-failures would not be a problem given the low operating pressure.

For mobile applications, the life of an electrical generation or storage system to provide power for the air conditioning system must be considered. It is expected that either a generating or storage system would have a shorter life than the refrigeration system itself, particularly if the power-to-weight ratio of the electric power system was maximized.

Size/Weight

No estimate of the size and weight of magnetic refrigerators relative to vapor-compression systems was found in the literature. However, it is clear that at least one electromagnet capable of generating high magnetic fields would be necessary. Electromagnets would be heavy due to the wire coils around the core. The secondary heat transfer loops would also contribute to the size and weight of magnetic refrigeration systems. Finally, if shielding were required to make the system suitable for use in the proximity of humans and animals, an additional cost penalty would exist.

Heat transfer loops containing a liquid, heat exchangers, and pumps would add to both the size and weight of the system.

Energy/Efficiency

Several studies have been performed to predict the performance of magnetic refrigerators by using theoretical models. Chen et al. [63] projected a COP of 60% of Carnot for a magnetic refrigerator operating with a constant field cycle between the

source and sink temperatures of −13 C and 47 C. In Chapter 9, a theoretical model of a combined cycle magnetic refrigerator was described. The COP was found to be 63% of Carnot assuming a 5 C minimum approach temperature between the source and sink and the system. Neither of these models accounted for magnetic, regeneration, or magnet cooling losses. In contrast, the limited experimentation which has been done with magnetic refrigerators indicates that the actual COP of present designs is very low. Brown [62] obtained a 47 C temperature lift with no load (hence no COP) at room temperature. Steyert [66] measured a COP of 26% of Carnot for a rotary magnetic refrigerator operating with a magnetic reversed Brayton cycle at room temperature, however, the temperature lift was 7 C. The energy/efficiency criteria for magnetic refrigerators was rated as very low (rating of 1) due to the low cycle efficiency of present systems.

Closure

Clearly, considerable technical development remains to be done before the magnetic refrigerator can be seriously investigated through experimentation. Although the theoretical COP of the magnetic refrigeration system is high, it is expected that the COP of an actual system would be much lower. The reasons for the lower actual COP are:

- The effectiveness of the regenerator will not approach 100%.
- The actual system would include the secondary heat transfer loops. Additional work will be required to pump the fluid, each additional heat exchangers will have a minimum approach temperature, and some heat will be transferred into the cold-side loop in the piping and reservoir.

Magnetic refrigeration is not well suited to mobile applications requiring an on-board electrical system to provide electrical power.

The numerical technical assessment ratings for magnetic refrigeration are given in Table 10.3.

Table 10.3: Technology assessment for magnetic refrigeration.

Criteria	Ratings				
	Dom. AC.	Com. AC.	Mob. AC.	Dom. Ref.	Com. Ref.
State of art	1	1	1	1	1
Complexity	2	2	1	2	2
Size/Weight	2	2	1	2	2
Maintenance	3	3	2	3	3
Life	4	4	2	4	4
Cycle Effic.	1	1	1	1	1

Pulse Tube and Thermoacoustic Refrigeration

Environmental Acceptability of Pulse/Thermoacoustic Systems

The working fluid in prototype thermoacoustic refrigerators in the space thermoacoustic refrigerator (STAR) program use a gas mixture of 97.1% helium and 2.9% xenon by mole fraction. Xenon is added to reduce fluid frictional losses by lowering the viscosity of the gas mixture. Helium and xenon are inert gases. They will not react with ozone in the upper atmosphere nor do they present a global warming hazard.

Thermoacoustic refrigerators operate at frequencies in the audible range ($\approx$

$400Hz$), however, experimental results indicate that the noise level is low outside the tube. Garrett likened the sound level to that of an electric fan [58].

State of the Art

Much of the basic research for pulse tube and thermoacoustic refrigeration has been done. Both the pulse tube and thermoacoustic refrigerator concepts have been brought to the prototype stage and demonstrated to be workable by experiment. The remaining system development would involve optimizing system performance and to creating marketable hardware. While the costs associated with this development are not expected to be as high as for less proven concepts like the magnetic refrigerator, they would be higher than the cost of converting vapor-compression refrigeration technology to use non-CFC refrigerants.

Complexity

The materials used in pulse tube and thermoacoustic refrigerators are common engineering materials. The tube is generally stainless steel. The heat exchangers are a metal with high thermal conductivity like copper. The pulse tube refrigerator would require a compressor. The thermoacoustic refrigerator would require an acoustic wave generator (speaker) to pressurize the working gas.

Since the high- and low-temperature heat exchangers in pulse tube and thermoacoustic refrigerators are small, secondary heat transfer systems would be necessary to exchange heat at a reasonable rate with the air on both the hot and cold sides. These systems would utilize forced convection or a circulating liquid. In either case, cost and size would be added to the system.

Manufacturing costs are expected to be approximately the same as for current vapor-compression systems. Although the thermoacoustic refrigerator does not use a compressor, the generating spherical end of the resonator section would require manufacturing methods which could add to the cost of creating the system hardware. On experimental systems, the resonator is constructed of fiberglass to facilitate its manufacture, and to reduce conduction heat transfer through the tube wall.

The pulse tube would require a compressor and prime mover (electric motor) since the pressure ratio between the high and low pressure could be approximately 2 or 3. The compressor may need to be capable of operating at pressures higher than those for vapor-compression systems. This conclusion is drawn from the fact that the static (low) pressure for the experimental thermoacoustic refrigerators is 10 atmospheres [57]. The compressors for direct-acting compression systems would need to be capable of higher speeds than vapor-compression compressors to create a high pulse frequency. Systems using a continuous flow compressor would require a rotary valve. Therefore, the cost of manufacturing a compression system for pulse tube refrigerators would be higher than the cost for vapor-compression compressors.

The materials (stainless steel and copper) will be slightly more expensive (per unit volume) than those used in vapor-compression system compressors since pulse tube refrigerators may need to be capable of higher operating pressures and speeds than those used in vapor-compression systems.

Size/Weight

Pulse refrigeration systems are not as compact as vapor-compression systems. This added volume is due, in part, to secondary heat transfer systems and the added

equipment size necessary to house the working gas which has a high specific volume.

Maintenance

Operation and maintenance levels and expertise do not appear to be markedly different than for vapor-compression systems.

Preventative maintenance would be the periodic recharging. Given the low molecular weight of helium and the high pressure, sealing of the system to preclude the escape of the working fluid would pose a problem. Periodic (possibly frequent) recharging of the system is anticipated.

When failures did occur, it is expected that components would be replaced, rather than repaired on site. Components for both pulse tubes and thermoacoustic refrigerators would probably be disposable rather than remanufactured. If cracks did occur in a tube, resonator, or acoustic diaphragm, it would be indicative that additional fatigue related cracks are likely in the future, indicating the need for tube replacement.

Since system repair would be a diagnosis and component interchange process, technicians repairing pulse refrigerators would require skills similar to those needed to repair vapor-compression systems.

Useful Life

The life of both pulse tube and thermoacoustic refrigerators is expected to be less than that of vapor-compression systems. Both high-pressure compressors and acoustic generator diaphragms have finite life expectancies. The high operating pressures and lack of lubricating oil in the gas would cause compressors to fail sooner

than vapor-compression compressors. Although the remaining components of these systems are largely static, given the pressure ratios and resonant operating frequency, the cyclic stresses in the tube may result in fatigue cracking. Consequently, the tube could have a limited fatigue life.

Energy/Efficiency

Experimental results for thermoacoustic refrigerators indicate that the COP for temperature lifts commonly experienced in refrigeration applications is about 16% of Carnot [57]. It should be noted that this measurement was the ratio of the rate heat was accepted from a heater placed *at the low-temperature heat exchanger* to the acoustic power required to accomplish the heat lift. The system *did not* include heat exchangers communicating with the air. Therefore, it is expected that the COP of a system in communication with the source and sink would be much lower.

From a theoretical standpoint, a reversed regenerative Brayton cycle operating with helium, isothermal regeneration, and isentropic compression and expansion would have a COP ratio of approximately 30% for the same temperature lift as the experimental unit. This would suggest that the potential level of cooling performance for pulse refrigerators is far below that of vapor-compression systems. These investigators have rated the energy cost of the pulse tube and thermoacoustic refrigerator as being high due to the low COP of this technology.

Closure

Although the pulse tube and thermoacoustic refrigeration have been demonstrated to be workable methods of accomplishing heat lift experimentally, a tech-

nical assessment indicates that neither system is a viable replacement for vapor-compression in refrigeration and air conditioning applications, even though both can use environmentally safe working materials. The principal reason these systems are not good alternatives to vapor-compression is an inherently low COP for the cycle when it is operated in the temperature range used for refrigeration and air conditioning. When secondary heat transfer loops are added to facilitate the transfer of heat between the system and air, the COP of the system would be lowered even further.

The numerical technical assessment ratings for pulse tube and thermoacoustic refrigeration are given in Table 10.4.

Table 10.4: Technology assessment for pulse and thermoacoustic refrigeration.

Criteria	Ratings				
	Dom. AC.	Com. AC.	Mob. AC.	Dom. Ref.	Com. Ref.
State of art	2	2	2	2	2
Complexity	3	2	3	3	3
Size/Weight	3	3	2	3	3
Maintenance	3	3	3	3	3
Life	3	3	3	3	3
Energy Effic.	1	1	1	1	1

Thermoelectric Refrigeration

Environmental Acceptability of the Systems

The solid, semi-conductor working materials used in thermoelectric refrigerators do not pose an ozone-depletion or direct global warming hazard.

Thermoelectric refrigerators operate silently; they have been considered for use as an air conditioning system on submarines [76].

State of the Art

Thermoelectric refrigeration is currently marketed as a cooling/heating device for transporting food and beverages. These units are operated on 12 volt DC electrical sources. Generally, the electrical power source is the electrical system of the vehicle. When the polarity is connected in one direction, heat is accepted from the interior of the insulated container and rejected to the environment, providing refrigeration for the contents. When the polarity is reversed, heat is accepted from the environment and rejected within the container, heating the contents. Coleman Industries advertises the maximum temperature difference between the interior and exterior of the container as being 22.2 C (40 F).

In the 1950s, the Johnson Wax building in Milwaukee, WI used a thermoelectric system for heating and cooling. The system was incorporated into the wall panels of the building.

On a smaller scale, thermoelectric cooling has been used to cool electronic circuitry.

The present state of the art for thermoelectric cooling systems is:

- The technology has been developed to the point that marketable systems can be built, but with a low COP.
- The COP of thermoelectric systems is low even when the best working materials are used (those containing tellurium). Tellurium is in short supply, limiting the number of units which could be produced.

- To improve the cooling performance of thermoelectric systems, new working material pairs must be developed.

Both the cooling performance and temperature lift could be increased by improving the figure of merit, Z (as discussed in Chapter 8). Developing semiconductor material pairs with a higher value of Z would require developing mixtures which have a low thermal conductivity, but high electrical conductivity. The solution to finding efficient working materials for thermoelectric refrigeration lies in high temperature super conducting. This area of basic research area would be expensive with no assurances of success.

Complexity

Bismuth and tellurium are the primary elements commonly used in thermoelectric refrigerator semiconductors [41]. Tellurium is a trace element which is extracted from the waste generated when refining of copper, and to a lesser extent, gold. The total annual production of tellurium is approximately 3.65×10^5 kg. About 10% of the production is consumed by the semiconductor industry, largely for the production of thermoelectric modules. This material must be refined to a higher purity, so the net yield is even less. The remaining 90% is consumed by the steel and chemical industries for material processing.

Mathiprakasam et al. [82] estimated that approximately 400 standard thermoelectric modules would be required to provide cooling for an automobile. Therefore, if the total tellurium available for thermoelectric module production were dedicated solely to the production of automotive air conditioners, only 20,000 units could be

constructed per year. If the demand for thermoelectric refrigeration increased, the price would increase, as well.

Thermoelectric refrigerators utilize a direct current (DC) power supply. An added first cost would be incurred in providing a rectifier or generator system to convert alternating current (AC) to DC for applications which do not have DC electrical supplies readily available.

Manufacturing costs are expected to be approximately the same as for current vapor-compression systems. The technology to produce thermoelectric modules is already in place, increased production (assuming working material availability) would require facility expansion.

Present thermoelectric devices rely upon natural convection and/or conduction to exchange heat between the system and the source and sink. Hardware has been proposed for transportation applications which would employ a circulating liquid (water and glycol, for example) to transport heat from the source to the system. Additional development costs would be expected if active thermoelectric systems were to be marketed.

Size/Weight

Thermoelectric refrigeration modules are thin rectangles which can be incorporated into the walls of the building or cabinet to be cooled. The interior walls of buildings are often composed of structural members, electrical wiring, plumbing, and an outer wall covering. If portions of the wall covering were replaced with thermoelectric refrigeration modules, cooling could be provided without intruding into the interior volume of the building. While this may be an advantage from a functional

perspective, it may well be a disadvantage from an aesthetic standpoint since wall decoration would be limited.

The exterior walls of refrigeration cabinets are filled with insulation. If the insulation thickness is not reduced, a portion of the interior volume of would have to be occupied by thermoelectric modules. A second option would be to provide the refrigeration unit outside the cabinet and transport the air (or another fluid) to and from the cabinet. In either case, the space-saving advantage of incorporating the thermoelectric modules into the structure would be lost.

If thermoelectric cooling was used for mobile air conditioning, a substantial weight and size penalty would be realized due to the electrical generation system necessary to supply the thermoelectric modules. For example, if the peak capacity of the mobile air conditioning system was 10.55 kW (3 tons) and the peak temperature lift was 23 C (41.4 F), the COP for an ideal system using semi-conductor material with Z-value of 0.003 would be 0.92. The electrical system would have to be capable of supplying 11.47 kW power. In comparison, a 70 ampere alternator on an automobile can supply slightly under 1 kW of DC electrical power.

Maintenance

Operation and maintenance levels and expertise would appear to be less than for vapor-compression. Little preventative maintenance would be required. Failed components would be replaced rather than repaired on-site and would be recycled to recover the tellurium.

Since system repair would be a diagnosis and component interchange process, technicians repairing thermoelectric refrigerators would require skills similar to those

needed to repair vapor-compression systems.

Useful Life

The life of thermoelectric refrigerators is expected to be long assuming a constant, continuous DC power supply. If pumps are used to circulate heat transfer fluid, their life would have to be considered. It is expected that the life of thermoelectric modules will exceed that of vapor-compression compressors.

The life of an electric generator for mobile applications would probably be shorter than the life of reciprocating or rotary compressors currently used on automobiles. This would be due to the life of the commutator brushes and sealed bearings normally used in automotive electrical systems.

Energy/Efficiency

The energy cost for thermoelectric refrigerators will be high. The New York City Transit Authority commissioned a study of the space conditioning requirements in subway cars.

Braking of subway trains is accomplished by reversing the DC electric drive motors used to propel the cars, causing the motors become electric generators. The electric power generated during braking is dissipated through large electric resistors, causing the train to decelerate. Heat generated in the resistors is rejected to the environment.

It was proposed that the power generated during braking be used to operate a thermoelectric system to provide cooling in the cars. The system was to be an active

system, in which the thermoelectric unit would be used to chill water which in turn would be circulated to heat exchangers in the subway cars.

It was calculated that 66.8 kW (19 tons) of refrigeration were necessary to provide cooling during peak demand and that 46 kW of electrical power would be available from dynamic braking of the train. The COP would then have to be 1.45 if the thermoelectric system were to supply the entire cooling load. The thermoelectric system investigated was capable of providing 20.7 kW (5.9 tons) of cooling, i.e., the COP was 0.45.

A combined vapor-compression thermoelectric system was considered. This system, while theoretically lower in operating cost, was estimated to be two times higher in first cost. It was not considered cost effective over a 20 year life cycle [80, 81].

Mathiprakasam [82] conducted a study of thermoelectric technology for air conditioning in automobiles. He concluded that the COP would be 0.42 for a 1.14 ton (4.0 KW) system. Furthermore, the power required to operate the system would represent approximately 10% of the total power capability of the vehicle engine.

Thermoelectric refrigeration has a relatively low actual COP, particularly for larger temperature lifts, and therefore, the energy cost will be high.

Closure

Although thermoelectric refrigeration has been developed to a marketable stage for recreational applications, its performance is currently too low to be used as a replacement for vapor-compression refrigeration. The limited availability of the element tellurium limits the extent to which thermoelectric refrigeration could be applied even if first costs and energy costs were not a consideration.

The temperature lift of present thermoelectric refrigerators is limited. Current recreational models advertise a temperature lift of about 22 C, although the theoretical lift is higher. For example, a lift of 75 C (from $T_L = -40$ C to $T_H = 35$ C) is theoretically possible for a material pair with a figure of merit (Z) of 0.003; however, the COP ratio would only be 0.01.

If materials can be developed which provide improved figures of merit, better performance and temperature lifts will be possible.

Thermoelectric refrigeration has the benefits of:

1. Silent operation.

2. Low preventative maintenance.

3. Space saving due to the ability to be incorporated into the members of the structure.

4. Simple operation and repair.

The numerical technical assessment ratings for thermoelectric refrigeration are given in Table 10.5.

Table 10.5: Technology assessment for thermoelectric refrigeration.

Criteria	Ratings				
	Dom. AC.	Com. AC.	Mob. AC.	Dom. Ref.	Com. Ref.
State of art	3	3	3	2	2
Complexity	2	2	2	1	1
Size/Weight	5	5	3	5	5
Maintenance	5	5	3	5	5
Life	5	5	4	5	5
Energy Effic.	1	1	1	1	1

Reversed Stirling-Type Refrigeration

Environmental Acceptability of Reversed Stirling-Type Systems

Reversed Stirling-type refrigeration machinery typically uses helium as the working gas. Based upon the assumption that helium or another inert gas would be used in Stirling-type refrigerating machinery, there would be no ODP, GWP, or toxicity problems related with the working material.

As with other cooling systems utilizing secondary heat transfer loops to transport heat between the cooled space and the refrigeration unit, a fluid with a low freezing point would be necessary. Usually, the fluid is a mix of water and glycol, which has a low toxicity. Water/glycol mixtures are the coolant of choice in the automotive industry. If leakage is minimized and used fluid is recycled, the additional use in refrigeration systems should not present an additional environmental hazard.

State of the Art

Reversed Stirling refrigeration has long been used in cryogenic cooling applications where low cooling temperatures and small cooling capacities are required. The source temperature for these applications is generally in the −213 C to −173 C range. When traditional reversed Stirling machinery has been used at higher source temperatures (−23 C, for example), COPs were found to be on the order of 10% to 20% of the Carnot COP [31, 32].

The ideal reversed Stirling thermodynamic cycle model consists of two isothermal and two isometric processes. In practice, isothermal compression and expansion are not possible due to practical limitations of heat exchanger area and flow of the working gas in the closed-cycle reciprocating machinery. Non-isothermal heat transfer is one reason actual systems cannot operate with the Carnot COP.

The COP of reversed Stirling cooling systems is further degraded by thermodynamic losses associated with imperfect regeneration, friction, and unintentional heat transfer throughout the system. The net effect of these losses is an actual COP which is much lower than the ideal. Kelly et al. have proposed a modified reversed regenerative Brayton cycle as being a more realistic model of the actual thermodynamic processes in Stirling-type refrigeration machinery [40].

As discussed in Chapter 6, the causes for the low COP in reversed Stirling refrigeration equipment are work related when operated at source temperatures commonly found in domestic and commercial refrigeration applications. Some of these losses could be minimized by reducing the friction in the rotating and sliding elements of the machinery. Current development of reversed Stirling systems has centered around a free piston design using a linear motor and gas spring to eliminate the rotating com-

ponents traditionally found in reciprocating machinery, thus reducing the frictional power requirement of the system. Both Sunpower and Stirling Technology Company have tested prototype free-piston reversed Stirling systems at source temperatures of −23 C [37, 38, 83]. The cycle efficiency for these systems was found (by experiment) to be between 25% and 32%. Both companies determined the COP by measuring the electrical power input and the heat accepted at the expansion (cold) heat exchanger. Neither experiment took into account the temperature differences or pumping power across the cold-side secondary heat transfer loop. If these factors had been taken into account, the experimental COP would have been lower.

A second source of losses in reversed Stirling refrigerators is in-cylinder heating of the working gas during the compression process and in-cylinder cooling during the expansion process. The heating and cooling are due to heat exchange between the gas and the piston and cylinder walls.

The current state of the art for Stirling-type refrigeration technology is:

- The basic technology used in Stirling-type refrigeration has been demonstrated to be workable. The technology has been developed to the extent that it has been used commercially in cryogenic refrigeration applications.

- To create a refrigeration system which could be used as a replacement for vapor-compression, heat exchangers or heat transfer methods must still be developed which will allow heat to be exchanged between the system and the cooled space, as well as the system and the environment at a reasonable rate. This heat transfer would have to be accomplished without losses which would degrade the coefficient of performance below current levels.

- Bearings must be developed which will reduce the frictional losses in the system and improve the useful life of the system.
- Sealing systems must be developed not only to minimize the escape of the working gas, but to reduce leakage by the piston (blow-by).
- Regenerators must be developed which have high effectiveness, low pressure drop, and minimum volume (dead space).
- A means of providing compression and expansion spaces which approach isothermal heat transfer conditions must be developed.

Complexity

Reciprocating and free-piston Stirling-type refrigerators can make use of standard materials including ferrous and non-ferrous metals for most of the hardware. More exotic materials may be necessary for seals and sliding surfaces if the useful life is to be lengthened and friction is to be reduced. Insulating coatings (such as ceramics) might be used to insulate the piston and cylinder. Neither modification is expected to add appreciably to the cost of the hardware.

The Stirling-type refrigeration system will require at least one secondary heat transfer loop (on the cold side). Realistically, a second loop would be required for the hot side as well in order to reject heat from the system high-temperature heat exchanger at a reasonable rate. Each secondary loop would require an air-to-fluid heat exchanger, piping, a means of circulating the fluid, and a reservoir.

Size/Weight

The basic Stirling-type cooling system without the secondary heat transfer loops is reasonably compact and light weight. The addition of the secondary heat transfer systems will cause the system to be similar in size to a comparable vapor-compression system.

There has been some development of a duplex reversed Stirling heat pump [84]. This system uses heat rather than mechanical work to drive the system. In a conventional reversed Stirling refrigerator, mechanical work is put into the system. Usually, this work is done by an electric motor. In the duplex Stirling system, heat is used to drive a Stirling engine which drives the reversed Stirling refrigerator. An additional size/weight penalty would be encountered for the duplex system when the combustor to convert fuel to heat is consider as being part of the refrigeration system package.

Maintenance

Present Stirling-type refrigeration machinery does not use lubricants. Given the small clearances and small tolerances for the cylinder to piston fits, periodic replacement of the reciprocating system is expected. Complete refrigeration units would be interchanged in the field. Depending upon their size, the units would either be rebuilt (systems with larger cooling capacity) or discarded. The latter case is likely if the systems are hermetically sealed.

Hermetic sealing may be a likely means of construction for small Stirling-type refrigeration units to prevent leakage of the working gas from shaft seals. This would eliminate the need for periodic recharging of the system with gas. Helium is even more difficult to contain than R-12 or R-22. Units which are not completely enclosed

will require periodic recharging of the working gas.

If secondary heat transfer loops are used, some periodic maintenance will need to be performed on these subsystems, as well. As previously discussed for other alternative technologies, these loops may require repair of the circulating devices, flushing of the system and replacement of the coolant, cleaning of the air-side heat exchanger surfaces, and repair of leaks.

In summary, a moderate amount of maintenance is expected to be required for Stirling-type refrigeration systems used in all applications; in comparison, vapor-compression requires little or no maintenance.

Useful Life

Presently, the useful life of Stirling-type refrigerators is expected to be well below that of vapor-compression systems. The primary cause for the reduced life of the system is lubrication. Reciprocating compressors used in vapor-compression systems use oil to reduce friction, assist in sealing the piston in the cylinder, and aid in transferring heat from the compression space. The compression and expansion spaces of current Stirling-type machinery are operated without a lubricant. If life is to be increased, low-friction, wear-resistant material combinations or an effective piston ring (seal) combination which allows lubrication of the piston and wall, but precludes contamination of the compression/expansion space, must be developed.

Energy/Efficiency

Although the coefficient of performance for the ideal reversed Stirling cycle is COP_{Carnot}, experience has shown that actual systems have low COPs at source

temperatures of −23 C and above. Köhler and Jonkers [31] reported an efficiency ratio of approximately 8% for an unmodified Philips cryocooler operating between the temperatures of 27 C and −23 C. Chen reported a cycle efficiency of 7% for an off-the-shelf cryocooler operating at similar temperatures. Sunpower and Stirling Technologies report cycle efficiencies ranging from 25% to 32% for free-piston Stirling coolers [37, 38, 83].

In all cases, the COP was determined by measuring the cooling rate and the electrical power input. The temperatures were those of the high- and low-temperature heat exchangers. The effect of secondary heat transfer loops was not considered. Assuming a minimum approach temperature of 5 C for each loop, no pumping losses, and neglecting pumping power, the COPs would be about 80% of the reported values. Note that this is an idealized estimate and that the actual value would be much lower.

Although the COP of the classic reversed Stirling cycle model is the Carnot COP, the performance of actual Stirling refrigeration systems is much lower, as shown by the experimental results. The reasons are:

1. The classic ideal model does not accurately reflect the true thermodynamic processes in the actual cycle.

2. Inherent losses in the refrigeration system due to mechanical friction, fluid friction, imperfect heat transfer, motor losses, and leakage.

3. Additional losses due to heat transfer and work in the complete system which is in communication with the cooled space and the environment.

Closure

Reversed Stirling-type machinery is not considered to be a viable alternative to vapor-compression, particularly for air conditioning applications. Although the state of the art is such that hardware is commercially available for cryogenic applications, much development remains to be done to improve the useful life of the hardware and the cycle efficiency at source temperatures above −24 C. The system is best suited to applications requiring source temperatures between −193 C and −73 C [31], far below the temperatures needed for domestic and commercial refrigeration. Many of the technical improvements which must still be developed are applicable to vapor-compression technology. These technology improvements could be used to also improve the efficiency of vapor-compression refrigeration technology. For example, linear motors, dry lubrication, improved piston sealing, and minimizing in-cylinder heating are all directly applicable to vapor-compression compressor design. All of these concepts could be used to improve the efficiency of the vapor-compression cycle.

The numerical technical assessment ratings for Stirling-type refrigeration are given in Table 10.6.

Table 10.6: Technology assessment for reversed Stirling-type refrigeration.

Criteria	Ratings				
	Dom. AC.	Com. AC.	Mob. AC.	Dom. Ref.	Com. Ref.
State of art	3	3	3	3	3
Complexity	3	3	3	3	4
Size/Weight	4	4	4	4	4
Maintenance	3	3	3	3	3
Life	4	3	4	4	3
Energy Effic.	2	2	2	3	3

Reversed Brayton Refrigeration

Environmental Acceptability of Reversed Brayton Systems

The open-cycle reversed Brayton system uses air as the working fluid. Closed-cycle reversed Brayton systems can use air or other gases. Properties, such as specific heat and viscosity over the operating temperature range are considerations in selecting a working gas. Helium is an ideal choice due to its high specific heat and low viscosity. It is an inert gas with no ODP or direct GWP. Also, it will not react with the materials used to construct the system.

Noise generated by the compressor, expander, or high-velocity air is one consideration which may limit the type of application in which open-cycle Brayton systems can be used. Since escaping air is not a problem in closed-cycle reversed Brayton systems, the noise level may be acceptable, or it can be brought a to reasonable level using sound insulation or other acoustic engineering methods.

State of the Art

The basic technology used in reversed Brayton systems is well developed. Compressor and expander turbines are capable of reasonably long life, although not as long as that of compressors used in vapor-compression refrigeration systems.

Kauffeld et al. [22] performed a theoretical and experimental investigation of nine different reversed Brayton cycle systems. The cycle configurations included open and closed systems, multistage compression with intercooling, and regeneration. It was found that the highest COPs for air conditioning applications would be achieved by open cycle systems with multi-stage compression and intercooling. The COP of the reversed Brayton cycle is very sensitive to the isentropic efficiencies of the compressor and expander. Table 10.7 illustrates the effect of increasing the isentropic efficiency of the expander and compressor for a reversed Brayton cycle operating on air. The source temperature was 4 C and the sink temperature was 35 C.

For off-the-shelf hardware, the isentropic efficiency is between 63% and 88% for compressors and between 64% and 88% for expanders. Small capacity units have lower isentropic efficiencies. As the capacity increases, so does the isentropic efficiency. Some prototype expanders have achieved an isentropic efficiency of 87% [23].

The largest improvements of the reversed Brayton cycle COP would be realized if the isentropic efficiency of the expander and compressor could be raised. Turbine design is a mature technology. The state of the art is advanced as a result of application of turbomachinery in the gas turbine, jet engine, and electric power generation industries. Any further improvements in the efficiency of either the compressor or the expander will be small. Consequently, improvements in the COP of reversed

Table 10.7: COP of the theoretical reversed Brayton cycle for three different isentropic compressor and expander efficiencies. T_{source} = 4 C, T_{sink} = 35 C, Pressure ratio = 2.5.

Isentropic Efficiency (Compressor *and* Expander)	**Theoretical COP**	η_{Cycle}
0.60	0.12	0.014
0.65	0.19	0.021
0.70	0.27	0.031
0.75	0.39	0.044
0.80	0.55	0.062
0.85	0.79	0.088
0.90	1.16	0.130
0.95	1.82	0.204
1.00	3.30	0.369

Brayton-type refrigeration systems will be small, as well.

Complexity

The basic reversed Brayton system is composed of a compressor, expander, and one or two heat exchangers (depending upon whether the system cycle is open or closed). The COP of the basic system is low, as shown in Chapter 5.

Higher COPs can be achieved by incorporating regeneration, multistage compression, and intercooling between compression stages. Kauffeld et al. [22] determined that the COP for an air conditioning application could theoretically be raised from 0.59 (for a basic closed reversed Brayton cycle) to about 1.1 by using an open cycle with regeneration and two-stage compression with inter-stage cooling [22]. In all cases, an isentropic efficiency of 0.80 was assumed for the both the compressor and expander. A 10 C minimum approach temperature was assumed for external heat

exchangers. Although adding components to the system will increase the COP, it will also result in higher first cost maintenance costs.

For air conditioning, the volumetric flow rate of air would be approximately 2.3 $\frac{m^3}{minute}$ (81.7 cfm) per kW of refrigeration. Ducting to deliver cool air to the conditioned space would represent an additional first cost of the system. Ducts would need to be large enough to reduce the noise generated due the velocity of air entering the room in open-cycle refrigeration systems.

Since the highest COPs are achieved at relatively low pressure ratios (and over a narrow range of pressure ratios), the pressure drop in the duct system would be an important consideration requiring custom design of each application. The added engineering time would result in higher first costs.

A method of dehumidifying the air in open-cycle systems would be required applications where high relative humidity was encountered or at source temperatures below freezing ($T_{source} \leq 0$ C). The COP decreases as the relative humidity increases in the air.

Size/Weight

Given the low pressure ratios and high volumetric flow rates encountered in reversed Brayton refrigeration systems, the hardware volume would be large as compared to vapor-compression machinery of similar capacity. This would be particularly true of systems using multi-staging with or without intercooling and regeneration. If dehumidification were required to remove water from the air, additional space would be required to house a desiccant or dehumidification system.

Maintenance

The primary maintenance activities for open reversed Brayton cycle cooling equipment would include changing air filters and inspecting and cleaning heat exchanger and regenerator surfaces. These procedures would be necessary to insure that the maximum heat transfer rate could be maintained.

Open-cycle systems using a desiccant for moisture removal would require periodic cleaning and replacement of the desiccant media. If heat were used to remove the moisture from the media, the heat source may require additional attention.

In both open- and closed-cycle systems, field service of the compressors and expanders would involve replacement of the failed unit. Reconditioning of these replaced units would be done at a centralized location with specialized equipment and personnel. Field maintenance personnel would not require a high level of expertise.

Useful Life

The reversed Brayton cycle uses at least two rotating machines, the compressor and the expander. If multi-staged compression or expansion were used, each additional stage would be considered another rotating machine. Although the life of high-speed rotating turbomachinery used in compressors and expanders is reasonably long, it is doubtful that the useful life of reversed Brayton turbomachinery can exceed that of the compressor used in vapor-compression systems of equivalent thermal capacity. Even if the life of each rotating component used in a reversed Brayton system were as long, there would be increased probability of system failure due to the additional rotating machines in the system.

Energy/Efficiency

The COP is low for reversed Brayton cycles operating in the temperature ranges used in refrigeration and air conditioning applications. The primary reasons for the low COP are the thermodynamic behavior of the working gas (non-isothermal heat acceptance and rejection), non-isentropic compression and expansion, frictional losses due to viscous flow of the working gas and bearing lubrication in the compressor and expander, and irreversible heat transfer.

Closure

The numerical technical assessment ratings for Brayton cycle refrigeration are given in Table 10.8.

Table 10.8: Technology assessment for reversed Brayton refrigeration.

Criteria	Ratings				
	Dom. AC.	Com. AC.	Mob. AC.	Dom. Ref.	Com. Ref.
State of art	3	3	3	3	4
Complexity	3	3	3	3	4
Size/Weight	2	2	2	2	3
Maintenance	3	3	3	3	3
Life	3	3	3	3	3
Energy Effic.	1	1	1	2	2

Absorption Refrigeration

Introduction

Absorption refrigeration uses heat transfer from a high-temperature source rather than mechanical work as the energy input. The working material is the *refrigerant* while the material with which it is in solution is the *absorbent.* The absorbent can be either in the liquid or solid phase. When separated from the absorbent, the refrigerant is generally in the liquid or vapor phase, depending upon which thermodynamic process it is undergoing in the refrigeration cycle. In a simple single-stage absorption cycle, heat from an external source is supplied to the generator causing some of the refrigerant to be vaporized from the binary mixture. The refrigerant vapor is condensed to the liquid phase in a condenser by rejecting heat to the environment. The liquid refrigerant is expanded by a throttling process through an expansion valve or capillary tube. In the evaporator the refrigerant accepts heat from the conditioned space. During the evaporation process, the refrigerant is returned to the vapor phase. The refrigerant vapor is re-absorbed by the binary mixture creating a strong solution in the absorber. Additional heat is rejected to the environment during the absorption process. A complete description of simple single-stage absorption cycles can be found in the *ASHRAE Handbook of Fundamentals* [86] or in thermodynamic textbooks such as *Fundamentals of Engineering Thermodynamics* [24].

Environmental Acceptability of Absorption Refrigeration Systems

Traditional absorbent-refrigerant pairs are aqua-ammonia ($H_2O - NH_3$) and lithium bromide-water ($LiBr - H_2O$). Ammonia is mildly toxic but has no ODP

or direct GWP. Both lithium bromide and water are non-toxic and have no ODP or direct GWP.

Other absorbent-refrigerant combinations have been studied. Iyoki and Uemura [89] studied the theoretical efficiencies of absorption cycles using water as the refrigerant and multi-component absorbents which were mixtures of the inorganic salts lithium chloride, zinc chloride, calcium chloride, and calcium bromide as the absorbent. Iyoki and Uemura also considered adding ethylene glycol ($C_2H_6O_2$) to the lithium bromide-water pair. None of these materials have an ODP or a direct GWP. Procedures for the proper disposal of used material would have to be established to prevent ground water contamination.

Organic absorbent-refrigerant pairs have also been studied. Huber [90] tested a heat pump using HCFC-22 as the refrigerant and DEGDME (diethylene glycol dimethyl ether, $(CH_3OCH_2CH_2)_2O$). Although HCFC-22 is slated for phase-out due to its high direct GWP, the question of finding an environmentally acceptable replacement organic refrigerant for use in absorption refrigeration remains.

State of the Art

The aqua-ammonia refrigeration cycle predates vapor-compression systems. Basic technologies used in liquid absorption refrigeration (such as boiling and condensation heat transfer) are well developed.

The level of development of technology which is specific to liquid absorption refrigeration differs with absorbent-refrigerant pair. Aqua-ammonia refrigeration systems have not been commercially produced on a large scale for twenty years. Consequently, little development work has been done. Development has continued on

lithium bromide-water absorption systems for chiller applications. Hitachi [91] manufactures 100 to 1500 ton $LiBr - H_2O$ chillers which are available as direct-fired, steam heated, or waste heated units. The waste heat driven system makes use of exhaust gases from gas turbines, diesel engines, and other heat sources with an outlet temperature of 288 C or higher.

Sanyo [92] produces $LiBr - H_2O$ chillers ranging from 35 kW to 4050 kW capacity. These systems use a double-effect generation cycle in which the vapor from the diluted absorbent in the high temperature generator is used to heat the intermediate-strength absorbent in the low temperature generator and boil off additional refrigerant from the intermediate-strength mixture.

More complex absorption systems using multiple stages or multiple effects have been developed. Siddiqui and Riaz [87] have developed a two-stage absorption cycle which could be operated using biogas as the fuel. The first stage uses a water-lithium bromide refrigerant/absorbent pair. Heat from the second stage absorber is accepted by the first stage evaporator. The second stage uses the ammonia and water refrigerant/absorbent pair. If the temperature of the re-absorption process is too low, the refrigerant can form solid crystals. This condition is generally considered to be a problem. However, a system in which the solid refrigerant crystals are allowed to form in a slurry and the latent heat of fusion is utilized in the cycle, has been patented [88].

Liquid absorption systems can take many configurations. They can have multiple stages and multiple effects. Each stage could utilize a different refrigerant-absorbent pair. With respect to the refrigerant-absorbent pairs themselves, much work remains to be done to determine:

- Which pairs, both inorganic and organic, have the highest COP for a given condensing and evaporation temperature.
- The ODP and direct GWP of the pair.
- Toxicity of the pair.

Complexity

Basic liquid absorption systems single-stage absorption systems require a generator, condenser, expansion device, evaporator and absorber. Other necessary equipment includes liquid pumps, piping, heat input system, and a temperature control system. Multiple staging and/or multiple effect systems could be used in absorption systems to:

1. Improve the COP of a system.
2. Enable the system to operate with the desired temperature lift.

Multiple staging and/or multiple effect cycles would require additional components to be added to the system, resulting in higher first costs.

In some systems, the potential for corrosion of components could be high due to the reactive nature of the refrigerant or the absorbent used in the system. The materials used in the system must be corrosion resistant for the particular refrigerant-absorbent pair chosen. For example, copper and copper alloys such as brass or bronze are susceptible to corrosion when exposed to ammonia. Intergranular corrosion is also known to occur in cast iron after prolonged exposure to an ammonia atmosphere. Corrosion will lead to eventual failure of the component, rendering the system inoperable. Therefore, none of the components selected for use in an aqua-ammonia

system should include copper-based or cast iron materials. This would include small off-the-shelf items such as pipe fittings. Corrosion-related material problems could be unique to a particular refrigerant-absorbent pair. The need for corrosion resistant materials will add to the first costs of the system.

Lithium bromide-water absorption systems reject heat from the absorber and condenser through a cooling tower. The cooling tower would add to the first cost and maintenance costs of the system.

Operating pressure would be another consideration related to the first costs and maintenance costs of the system. Generators operating at high pressures could fall under the ASME direct-fired pressure vessel code. Certification of the generator to assure that the design and construction was in compliance with the pressure vessel code would add to the first cost of the system. Periodic hydrostatic testing as required by the code would add to the maintenance costs.

Size/Weight

In general, absorption systems are characterized by a relatively large size and high weight per ton of cooling. Single-stage systems operating with a large temperature lift have a correspondingly high operating pressure, requiring heavier construction of the high pressure side of the system. If multi-staging is used to accomplish the same temperature lift, the number of components is compounded by the number of stages. Multi-staging adds to both the size and weight of the refrigeration system.

Multiple effect systems have an additional generator and heat exchanger for each "effect" added to the system to raise the COP. Generally, for a given temperature lift, capacity, and refrigerant-absorbent pair; systems with higher a COP would also

have a correspondingly larger size and greater weight.

Liquid absorption systems using water as the refrigerant require a cooling tower. Although these systems would be used in large commercial air conditioning systems (and possibly shipboard), finding a location either on or adjacent to the structure could be an important consideration in some applications.

Maintenance

Maintenance needed for liquid absorption refrigeration systems would include periodic repair of pumps, repairing leaks, and servicing the heating system. The system would require periodic inspection for internal and external corrosion. High pressure systems would require periodic inspection, testing, and re certification of the generator. Cooling towers would require additional maintenance, including repair of circulating pumps and fans, and cleaning water filters.

Useful Life

The useful life of liquid absorption refrigeration systems would be long assuming that proper materials had been chosen and proper maintenance had been performed. The primary driving component of a liquid absorption system is the high temperature generator. These units do not fail as a result of wear stemming from the relative motion of moving parts or mechanical stresses, as is the case with reciprocating or rotary compressors. The principle causes of failures in the system would be:

1. Corrosion.

2. Erosion.

3. Cracks.

Corrosion can be minimized by the selection of materials and the refrigerant-absorbent pair during the design of the system, and by maintenance to stop corrosion after the system has been placed in service. Erosion can be minimized by keeping the working fluid free of particulates and by assuring that a high enough pressure is maintained at the liquid pump inlet to prevent cavitation. The occurrence of cracks could be minimized by proper design and construction.

In summary, an absorption refrigeration system represents a high first cost per kilowatt investment. If a major component of the system fails, replacement can be justified due to the high replacement cost of the entire system. The primary components are not subject to mechanical wear as in mechanically driven refrigeration systems. If proper maintenance is performed throughout the life of the system, the useful life should be indefinite.

Energy/Efficiency

The ideal absorption refrigeration system would be a Carnot refrigerator driven by a Carnot heat engine. If it is assumed that the condenser and absorber reject heat to the sink at T_H, the COP for the ideal absorption cycle can be written as,

$$COP_{Ideal} = \left(\frac{T_G - T_H}{T_G}\right)\left(\frac{T_L}{T_H - T_L}\right). \tag{10.1}$$

where,

T_L = The absolute source temperature.

T_H = The absolute sink temperature.

T_G = The absolute generator temperature.

The theoretical COP of absorption refrigeration systems has been studied. Iyoki and Uemura [89] reported calculating theoretical COPs ranging from 0.8 to 0.88 for single-stage absorption cycles operating at the following temperatures:

$TL = 8$ C.

$T_H = 30$ C.

$T_G = 57$ C.

For double-effect absorption cycles Iyoki and Uemura calculated COPs ranging from 1.3 to 1.7 for the following operating temperatures:

$TL = 8$ C.

$T_H = 30$ C.

$T_{G1} = 117$ C to 127 C.

$T_{G2} = 80$ C.

Several refrigerant-absorbent pairs were assumed in their study, all of which used water as the refrigerant. The absorbent mixtures were lithium-based with differing mixtures of other substances added.

Siddiqui and Riaz [87] modeled a two-stage system using the water-lithium bromide pair in the first stage and ammonia-lithium trioxide in the second stage. A theoretical COP of 0.55 for $T_L = -10$ C and 0.48 for $T_L = -25$ C was calculated. The other system temperatures were:

$TA = 25$ C.

$T_H = 35$ C.

$T_{G1} = 90$ C.

$T_{G2} = 100$ C.

DeVault and Marsala [92] determined a theoretical cooling COP of 1.41 for a

triple-effect aqua-ammonia cycle operating at:

$TL = 2.2$ C.

$TA1 = 75$ C.

$TA2 = 110$ C.

$T_H = 28$ C.

$T_{G1} = 116$ C.

$T_{G2} = 219$ C.

Actual COPs for liquid absorption refrigeration systems in service found during this study were:

- Sanyo [93] claims a COP of 0.96 for their double-effect absorption chillers. No operating temperatures were reported.
- Hitachi [91] claims a COP of 1.21 for their two-stage chiller heated by steam. No operating temperatures were reported.

Closure

Liquid absorption refrigeration cycles using water as the refrigerant can not be used for most refrigeration applications due to the freezing temperature of water.

Absorption systems which are capable of operating at temperatures below freezing are characterized by:

1. Higher generator temperatures.
2. Lower COP.

3. Higher size and weight per kilowatt of cooling capacity due to the heavier construction to needed enable the system to operate at high pressures or due to multi-staging.

4. Higher first cost.

The numerical technical assessment ratings for absorption refrigeration are given in Table 10.9.

Table 10.9: Technology assessment for absorption refrigeration.

Criteria	Ratings				
	Dom. AC.	Com. AC.	Mob. AC.	Dom. Ref.	Com. Ref.
State of art	4	5	2	4	4
Complexity	3	4	2	3	3
Size/Weight	3	3	2	3	3
Maintenance	3	4	2	3	3
Life	3	4	2	3	3
Energy Effic.	5	5	5	5	5

Solid Sorption Refrigeration

Introduction

Solid absorption and adsorption are two alternative refrigeration technologies which utilize similar hardware but the manner in which the refrigerant bonds to the sorbent differs. Both technologies use heat transfer from a high-temperature source rather than mechanical work as the energy input. Solid systems using heat from direct-fired, steam, engine exhaust gas, and solar sources have been proposed [94].

Solid absorption refrigeration utilizes refrigerant-absorbent pairs which are in the solid phase after absorption. The refrigerant fluid is chemically bonded with the absorbent to form a third compound. When the compound containing the absorbed refrigerant is heated, during generation, refrigerant vapor is released by breaking the chemical bonds. Therefore, the composition of the solid compound changes due to the removal of the refrigerant.

In solid adsorption, the refrigerant molecules physically, rather than chemically, bond to the surface of the adsorbent when it is cooled below a given temperature. When the adsorbent is heated, the physical bonds between the adsorbent and refrigerant are broken causing the refrigerant vapor to be released.

Both solid sorption cycles use a batch process in which multiple cannisters are alternately heated and cooled. The cannisters serve as both generators and sorbers in the system, depending upon their temperature.

Once the refrigerant has been liberated from the sorbent by heating the canister, the most common sorption cycle (for both adsorption and absorption) is as follows:

1. The refrigerant vapor is condensed to the liquid state in a condenser.

2. The pressure of the condensed liquid is then lowered by a throttling process and expanded at a lower pressure in an evaporator.

3. The refrigerant vapor is next re-combined with the solid sorbent compound in another canister which has been cooled.

4. When the desorbing canister is exhausted and the absorbing canister is filled, valves switch the system to another pair of cannisters in the piping network and the cycle is repeated. The cannisters which were just taken off line being a

cooling or heating process (depending upon whether they are now charged with refrigerant, or empty) to prepare them for another turn in the cycle [95, 96].

An alternative solid absorption cycle could be achieved with some refrigerant-absorbent pairs without condensing the liquid. After liberating the refrigerant by heating a canister, the gaseous refrigerant would be lowered by passing it through a heat exchanger at ambient temperature. The cooled refrigerant gas would then be reabsorbed in a discharged canister which had also been allowed to cool. During this absorption process, heat would be accepted by the canister from the surroundings due to the endothermic chemical reaction occurring as the refrigerant and absorbent recombine.

Douse and Meunier [97] developed a novel cascaded adsorption cycle using two water-zeolite stages and a methanol-activated carbon third stage to enable the system to operate at temperatures below 0 C.

Environmental Acceptability of Solid Sorption Refrigeration Systems

Solid absorption uses ammoniates or oxides of materials such as calcium. These compounds have an affinity for water. When heated, the ammoniates release ammonia vapor as the refrigerant. Calcium-based compounds release water as the refrigerant when heated. Neither solid refrigerant-absorbent pair contributes to ozone depletion or direct global warming.

The three adsorbents most commonly used in adsorption systems are: zeolites, silica gel, and activated carbon. These materials do not have an ODP or direct GWP.

Adsorption refrigerants which have been proposed or tested are:

- Water.

- Ammonia.
- Methanol.
- HFC-134a.
- Blends of HFC-134a, HFC-152a, and HCFC-124.

State of the Art

The technology used in solid sorption refrigeration is known but presently immature. Some experimentation has been done, but no working prototypes of refrigeration systems for the marketplace have been built.

The primary areas of current solid sorption development are canister design, canister heating/cooling system design, and developing systems which could provide continuous, rather than batch refrigeration.

For both absorption and adsorption, the rate at which cannisters release and accept refrigerant is a function of the uniformity of the heat transfer within the canister. Portions of the canister volume with low heat flux will have a correspondingly low rate of sorption and desorption. Therefore, methods of enhancing the heat transfer into and out of the cannisters must be improved.

The maximum refrigerant capacity of a canister (as well as the rate of refrigerant release and acceptance) is a function of the sorbent surface area to volume ratio. Methods of increasing the ratio through sorbent particle shape and porosity are of current interest [98].

Most of the proposed solid sorption refrigeration system concepts use a circulating fluid to heat and cool the cannisters. Recovering heat to be used to heat another

canister which will be active in a future batch cycle will be important in improving the COP of solid sorption cycles.

Further technology development for solid absorption refrigeration would include:

- Identifying additional refrigerant and absorbent pairs which are environmentally acceptable.
- Improving the efficiency of the desorption and absorption processes.
- Developing an improved method of connecting absorbing cannisters with desorbing cannisters.
- Improving the heat within the canister volume during heating and cooling.

Complexity

Solid sorption systems will require multiple canister pairs if the system is to operate continuously. Cannisters will require some method of providing uniform heat flux through the interior volume. Fins, mesh, or some other heat transfer enhancement method will be necessary. The sorbent material would need to be uniformly distributed over the heat transfer surface within the canister. Like the catalytic converter used in automobile exhaust systems, this technology may be expensive when first introduced, and become cheaper over time.

Three piping and valve systems would be needed: One would direct the flow of refrigerant to and from the cannisters. The second would direct the heating fluid to the generating canister. The third would direct the cooling fluid to the absorbing canister. Controls would be necessary to direct the flow of each of the fluids to different cannisters as they reached the fully desorbed or absorbed charge condition.

The cooling capacity of the system would vary with the charge state of the desorbing and absorbing cannisters. The initial desorption and absorption processes would occur at a faster rate initially and then slow as the fully charged is approached. Therefore, a sophisticated logic system would be required to deliver a near-constant cooling capacity.

Condensers, evaporators, and expansion valves currently used for vapor compression systems could be used with some refrigerants. This would be true if organic refrigerants which are also used in vapor compression systems were chosen.

In summary, the complex piping and control system for a continuous-cooling solid absorption refrigeration system could be expensive, particularly for small capacity units since the cost of valves and controls do not vary appreciably with size. Solid sorption systems which provide intermittent (batch) cooling would not require multiple cannisters, complicated valve networks or sophisticated controls. Consequently, they can be produced at a much lower cost.

Size/Weight

System size would depend upon the cooling capacity and refrigerant flow rate of the cannisters. It would also depend upon the sorbent material density in the canister. Since adsorption is a process in which the refrigerant molecules are attached to the surface of the adsorbing material, the surface area must be large to achieve a reasonable refrigerant capacity and flow rate per cannister. For efficient heat transfer within the cannister during heating and cooling, a minimum temperature difference between the heat transfer fluid and the canister surface would be necessary. Sorption cannisters are likely to be large.

Maintenance

Valve failure may be a potential maintenance problem in continuous-cooling solid sorption refrigerating systems. Given the complexity of the three piping and valve systems, troubleshooting the system to determine the location of a malfunctioning valve could be a time consuming process.

Refrigerant leakage from the pipe fittings, valves, and cannisters would necessitate periodic recharging of the system with refrigerant.

Jones [99] reported that long-term corrosion is a common failure mode for solid absorption cannisters.

Alternate heating and cooling of the heat exchanger in the canister could result in two thermal stress related failure modes:

- The bond between the sorbent material and to the heat exchanger surface may fail, reducing contact (and thereby increasing thermal resistance between the two materials) or causing the sorbent material of fracture and flake way.
- Heat exchanger cracking resulting in leaks which would be very difficult to repair.

Solid sorbent cannisters may fail due to mechanical vibration. Vibration could cause cracks which cause leaks. Vibration could also cause the sorbent material to detach from the heat exchanger surface in the canister.

Sorbent materials may have a finite life linked to the number of cycles they have undergone and may require periodic replacement.

Maintenance costs for solid sorbent refrigeration are likely to be high.

Useful Life

If the cannisters represent a large portion of the first cost of the solid sorption refrigeration system, then the life of the system is expected to be short. On the other hand, if the cannisters could be made inexpensively and periodic replacement of the cannisters was an accepted maintenance procedure, then the life of the refrigeration system would be based upon the life of other system components. Considering the system life to be that of the valves, control system, piping, pumps, evaporator, condenser, and the heating unit, the useful life would be somewhat shorter than that of a vapor-compression refrigeration system.

Energy/Efficiency

The ideal COP for solid absorption and solid adsorption systems would be that of a Carnot refrigerator driven by a Carnot heat engine. If it is assumed that the condenser and absorber reject heat to the sink at T_H, the COP for ideal solid sorption cycles would be given by Equation 10.1.

Some theoretical COPs have been reported for solid sorption refrigeration systems.

Douss and Meunier [97] reported a theoretical COP of 1.06 for their cascading cycle solid adsorption refrigeration system. The temperatures were:

$T_L = 2$ C.

$T_H = 30$ C.

$T_G = 250$ C.

Critoph [98] predicted a COP of 0.57 for a single-stage solid absorption cycle

operating with a methanol-activated carbon refrigerant adsorbent pair. The cycle was assumed to operate with no heat recovery. The operating temperatures were assumed to be:

$T_L = -10$ C.

$T_H = 25$ C.

$T_G = 100$ C.

Jones [99] reported a predicted COP of 1.0 for an adsorption system studied by Tchernev in 1988. The theoretical system used water as the refrigerant and operated at the following temperatures:

$T_L = 4.4$ C.

$T_H = 37.8$ C.

$T_G =$ Not Given.

Jones [101] predicted a cooling COP of at least 0.92 for a heat pump operating with a 12 canister regenerative solid adsorption system using the HFC-134a and zeolite refrigerant-adsorbent pair. A cooling COP of 1.47 was predicted for a 6 canister regenerative system using the water-zeolite pair. The operating temperatures were assumed to be:

$T_L = 4.4$ C.

$T_H = 37.8$ C.

$T_G = 204.4$ C.

Closure

The numerical technical assessment ratings for solid sorption refrigeration are presented in Table 10.10.

Table 10.10: Technology assessment for solid sorption refrigeration.

Criteria	Ratings				
	Dom. AC.	Com. AC.	Mob. AC.	Dom. Ref.	Com. Ref.
State of art	2	2	2	2	2
Complexity	3	3	3	3	3
Size/Weight	3	3	2	3	3
Maintenance	3	3	3	3	3
Life	3	3	3	3	3
Energy Effic.	3	3	3	4	4

Technology Assessment for Vapor-Compression Refrigeration

Introduction

A technical assessment of vapor-compression refrigeration systems is presented to provide a measure of comparison for the alternative refrigeration cycles which have been studied during this project.

The vapor-compression refrigeration cycle uses mechanical work to drive the cycle. The source of the work input for domestic and commercial refrigeration and air conditioning is usually an electric motor. Work to drive mobile air conditioners is supplied by the vehicle's engine.

The vapor-compression cycle is currently used for all of the refrigeration and air conditioning applications considered in this study.

A detailed description of the basic vapor-compression cycle can be found in references such as:

- *The Fundamentals of Engineering Thermodynamics* [24].
- *ASHRAE Handbook of Fundamentals* [86].
- *Modern Refrigerating Machines* [25].

Environmental Acceptability of Vapor-Compression Systems

The most common working fluids for vapor-compression refrigeration have been CFC-11, CFC-12, HCFC-22, CFC-114, CFC-502, and CFC-13 (It is acknowledged that other refrigerants have been used in some applications). CFC-12 has been used almost universally in domestic refrigerators and mobile air conditioning applications. HCFC-22 is the principal refrigerant for domestic air conditioners. CFC-12, HCFC-22, CFC-502 and CFC-13 are used in commercial vapor-compression refrigeration. CFC-11 and CFC-114 are used in vapor-compression chillers. Alternative refrigerants which can be used in vapor-compression systems and which have no (or very low) ODP and low (to moderately low) direct GWP have been synthesized. The thermodynamic properties of many of these new refrigerants are similar to those of the refrigerants which they replace. Consequently, little or no modification of existing vapor-compression system designs is necessary to utilize these refrigerants.

State of the Art

Vapor-compression technology for domestic, commercial, and mobile refrigeration and air conditioning applications is mature. Even so, some opportunities exist

to further improve performance:

- New compressor designs. Oil-free, variable displacement, and linear compressors could offer higher compressor efficiencies.
- Heat transfer enhancement in the evaporator and condenser.
- New refrigerants.
- Variable speed motors.
- Check valves to prevent refrigerant migration between cycles.
- In the case of domestic refrigeration, two evaporator systems.

The use of nonazeotropic refrigerant mixtures (NARMs) is being investigated. Pure refrigerants and azeotropic mixtures condense and evaporate at a constant temperature. Large temperature differences exist between the air- and refrigerant-sides of the heat exchanger, resulting in the introduction of irreversibilities. NARMs, on the other hand, condense and evaporate with a temperature glide. The temperature of the refrigerant exiting the heat exchanger is different from that of the entering fluid. If the refrigerant and the air (or other fluid) are passed through the heat exchangers in counterflow, the temperature difference between the two fluids is reduced, thereby reducing the irreversibilities during heat transfer.

Complexity

With the possible exception of cabinetry or air handling units, the compression system is the most complex and expensive component in a vapor compression refri-

geration or air conditioning system. The expansion device, evaporator, and condenser are relatively simple and inexpensive devices to manufacture.

There is a high degree of technology transfer between the manufacturing of vapor-compression refrigeration systems and the manufacturing of many other items used in modern society. Liquid-to-air heat exchangers, similar to those used as vapor-compression system evaporators and condensers, are used in the transportation and heating industries. Reciprocating machinery is used not only on refrigeration compressors, but in internal combustion engines, liquid pumps, etc. Reciprocating, screw, and centrifugal machines are used for compressing air and other gases in a wide variety of commercial and industrial applications.

The first cost-per-kilowatt of refrigeration for a vapor-compression system is low as compared to other technologies. The primary reason is industry's present level of expertise in building the components used in vapor-compression refrigeration. A second reason is that the systems are built of inexpensive materials like cast iron, steel, and aluminum.

Size/Weight

On a size and weight-per-unit of refrigeration basis, vapor-compression refrigeration and air conditioning systems are relatively small and compact. The reasons are:

- Vapor-compression systems require relatively few components. Unlike the magnetic refrigeration and reversed Stirling cycles, secondary heat transfer systems are not needed to transport heat between the refrigeration system, the cooled space, and the environment.

- The primary heat has already been converted to mechanical work by a power plant (either electrical generating or an internal combustion engine). Therefore, the generator used in sorption or ejector cycles is not required.
- Although the refrigerants themselves have a high molecular weight, the refrigerant charge-per-unit of refrigeration is small. A secondary material such as a solid or liquid sorbent is not required.
- The organic refrigerants used for vapor-compression are non-reactive.
- The condensing and evaporating pressures of the system are low enough to allow the use of lightweight materials (like aluminum) and thin walls in the heat exchangers and piping.
- The organic refrigerants are dielectric; electric motors can be entirely enclosed with the compressor.

Maintenance

Domestic refrigerator vapor-compression systems are sealed. The recommended preventative maintenance is to periodically clean the condenser coils air-side surface to remove dust.

Domestic window air conditioners also use sealed refrigeration systems. Periodic maintenance would be to clean the condenser and evaporator air-side surfaces.

Central air conditioning systems have a condensing unit (compressor and condenser) mounted outside the building. The piping system connecting the condensing unit to the evaporator (located in the air handling unit) often contains fittings which can leak. Leak repair and recharging the system with refrigerant may be necessary

over the life of these systems. As with window units, periodic cleaning of the evaporator and condenser surfaces is a preventative maintenance function.

Small commercial refrigeration and air conditioning systems are often configured like their domestic counterparts. Consequently, the causes of refrigerant release from these systems are the same as for the domestic systems.

Large commercial systems often use compressors which are shaft-driven. The shaft is sealed. Refrigerant leakage can occur from the seal/shaft interface, releasing refrigerant to the environment. Large commercial systems use piping with joints which are also a potential cause of refrigerant leaks.

Mobile air conditioners also use shaft driven compressors and pipe joints. The shaft seals and pipe joints are potential refrigerant leakage points. Mobile systems are also subjected to engine and road vibration. Hoses are used to connect the compressor to the condenser and evaporator, these hoses are often a leak source. Leaks can develop in evaporators and condensers due to vibration induced cracks. Leaks can also develop in condensers due to contact with foreign objects. Finally, the entire system is subject to leaks as a result of a collision.

In general, vapor-compression refrigeration systems are characterized by low maintenance.

Useful Life

Studies have been conducted regarding the median life of hermetically sealed compressors used in commercial and residential vapor-compression heat pump applications. Ross [102] analyzed service work orders for 2820 sealed compressors used in cabinet-type perimeter heat pumps used in office buildings. Based upon these

data he found that 94% of the compressors remained in service after 9.3 years. Ross projected a median service life (50% failure rate) of 47 years. Ross also examined the service work orders for 518 larger hermetically sealed compressors used in what he termed 'core' heat pumps. He projected a median compressor service life of between 11.8 and 18 years.

Bucher et al. [103] analyzed the service records for 2184 residential vapor-compression heat pumps. The median service life was 14.5 years. These data also indicated that only 4.6% of the heat pumps had multiple compressor replacements.

Energy/Efficiency

The COP for an ideal vapor-compression cycle would be the Carnot COP. Even though nearly isothermal heat acceptance and rejection can be achieved, some practical characteristics of the hardware prevent approaching the ideal cycle condition:

1. Compressing two-phase mixtures of vapor and liquid can damage mechanical compressors. Therefore, a certain amount of superheating is required to assure that only refrigerant vapor enters the compressor.

2. Isentropic compression cannot be achieved due to heat transfer, fluid friction, and mechanical friction in the compressor.

3. The expansion process is not isentropic.

4. Fluid friction causes additional pressure losses in the piping and heat exchangers.

Literature regarding the theoretical COP for vapor-compression systems was studied. Bare et al. [104] studied the theoretical performance of fifteen NARMs in

a two-evaporator domestic refrigerator/freezer using a modified version of the single-evaporator refrigerator cycle (SERCLE) model. The COPs for these mixtures were compared to a baseline case using a single evaporator and CFC-12. They determined that the most promising NARMs exhibited a 30% higher COP than the baseline case. Increasing the evaporator area by 20% resulted in an additional COP improvement of 2.4% to 2.7%. A compressor efficiency of 55% was assumed for all cases. The temperatures were:

$T_{L1} = 3.35$ C.

$T_{L2} = -20$ C.

$T_H = 32.2$ C.

Radermacher and Jung [105] modeled a residential air-conditioner using binary and ternary refrigerant substitutes for HCFC-22. Their model assumptions were:

1. The compressor efficiency was 70%.
2. Pressure drops were neglected.
3. No evaporator subcooling.
4. No condenser superheating.
5. $T_L = 11.1$ C.
6. $T_H = 35$ C.

For the ternary blend consisting of 20% R-32, 20% R-152a, and 60% R-124, a theoretical COP of 3.8 was calculated. This was a 13.7% improvement over the baseline case using R-22.

Braun et al. [106] modeled the performance of a 19.3 MW (5500 ton) variable-speed chiller at the Dallas/Fort Worth airport. The model involved the simultaneous solution of a system of mass, momentum, and energy balance equations written for the components of the system. Assuming a mechanical efficiency of 91% and a polytropic compressor efficiency of 82%, they determined a COP of 5.0 for a 28 C temperature lift at full chiller load using the refrigerant R-500.

Bare [107] also modeled the theoretical performance of chillers using alternatives to CFC-11 and CFC-114. The model assumed constant pressure evaporation and condensation, isentropic compression, and adiabatic expansion. The COPs were between 6.2 and 6.9 when no subcooling or superheating was assumed for an evaporating temperature of 4.4 C and a condensing temperature of 40 C.

Sand et al. [108] reported experimental performance results for twelve refrigerants. The vapor-compression system was a small laboratory unit with a water cooled condenser and water-heated evaporator. The cooling COP was 2.6 for HFC-152a and 2.45 for HFC-134a. In comparison, the cooling COP for CFC-12 was 2.4 and 2.1 for HCFC-22. The condensing temperature was 35 C. and the evaporator water outlet temperature was 15 C.

For commercial frozen food freezers the actual COPs range between 1.0 and 1.2, 1.9 for dairy display cases, and 2.5 for produce display cases [109].

Multerer and Burton [14] report a COP of 2.35 for automotive vapor compression systems using CFC-12.

Closure

The numerical technical assessment ratings for vapor-compression refrigeration are presented in Table 10.11.

Table 10.11: Technology assessment for vapor-compression refrigeration.

Criteria	Ratings				
	Dom. AC.	Com. AC.	Mob. AC.	Dom. Ref.	Com. Ref.
State of art	5	5	5	5	5
Complexity	4	4	4	4	4
Size/Weight	4	4	4	4	4
Maintenance	5	5	4	4	4
Life	5	5	5	5	5
Energy Effic.	5	5	5	5	5

Ejector Refrigeration

Introduction

Ejector cycles using organic refrigerants were analyzed during this study. They were found to be environmentally unacceptable when using the best currently known working fluids. The most promising ejector systems use R11 or R114 as the refrigerant. Both of these materials have an ODP of 1.0 [3]. The 100 year time integrated GWP is relative to carbon dioxide. The 100 year GWP of R11 is 3500, the 100 year GWP of R114 is 6900.

If an environmentally suitable alternative refrigerant could be found (one possible alternative refrigerant could be HFC-236ea), the ejector refrigeration system would offer a relatively simple system which could be operated using waste heat.

Ejector refrigeration cycles use heat transfer from a high temperature source rather than mechanical work to drive the cycle. The basic operating principle of ejector refrigeration can be understood by thinking of the cycle as a vapor-compression cycle in which the compressor has been replaced by a pump generator (boiler), and ejector. Figure 10.1 is a schematic diagram for a simple ejector refrigeration cycle.

High pressure refrigerant vapor is produced in the generator by the addition of heat from an external high temperature source. The heat source could be from:

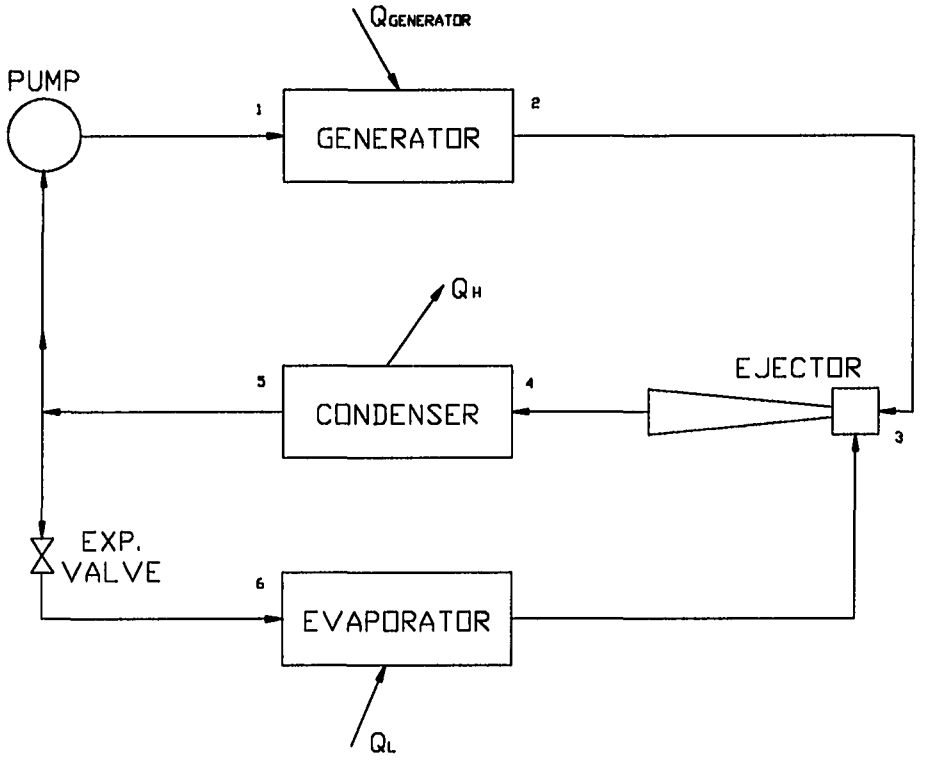

Figure 10.1: Schematic diagram of a simple ejector refrigeration system.

- A primary source such as a direct-fired burner.
- Waste heat.

- Solar.

The ejector consists of two sections, a nozzle and a mixing and diffuser section. The vapor from the generator is expanded through the converging-diverging nozzle. The resulting low pressure causes flow of the second stream for refrigerant from the evaporator. The two streams are mixed and the pressure of the mixed stream is increased in the mixing and diffuser section. The mixed stream of vapor is condensed in the condenser where heat is rejected to the environment. The condensed liquid is divided into two streams: One stream flows to the expansion device where it under goes a throttling process. The second stream flows to a pump which increases the pressure of the liquid to the generator pressure. The low pressure flow from the expansion valve is expanded in the evaporator, accepting heat from the cooling space during the process.

Sokolov and Hershgal considered two modified ejector refrigeration cycles for use with a low grade external heat source:

- A hybrid ejector/compression cycle in which a low pressure compressor was interposed between the evaporator and ejector to raise the pressure of the vapor at the ejector mixing chamber inlet [110].

- A multiple ejector cycle in which two or more ejectors are connected in parallel [111, 112].

The ideal COP for ejector refrigeration systems would be that of a Carnot refrigerator driven by a Carnot heat engine. If it is assumed that the condenser rejects heat to the environment at T_H, the COP for an ideal ejector refrigeration cycle would be given by Equation 10.1.

The theoretical COP for a simple ejector refrigeration system has been reported to be approximately 0.25. The theoretical COP for the hybrid system was 0.84 [110]. The system temperatures for both calculations were:

$T_L = -8$ C.

$T_H = 30$ C.

$T_G = 86$ C.

Sokolov and Hershgal [112] measured a COP of 0.42 for an experimental hybrid ejector refrigeration system using R114. The system temperatures were:

$T_L = 8.2$ C.

$T_H = 53.5$ C.

$T_G = 93$ C.

Closure

The numerical technical assessment data from the previous sections of this chapter were incorporated in a technical assessment computer program. The program is discussed in the next section.

Technical Assessment Ratings of the Refrigeration Technologies

To rate suitability of the refrigeration technologies for use in domestic, commercial, and mobile air conditioning and domestic and commercial refrigeration, an

algebraic expression was developed:

$$\Upsilon = \left(wf_A \times A\right) + \left(wf_B \times B\right) + \left(wf_C \times C\right) + \left(wf_D \times D\right) + \left(wf_E \times E\right) + \left(wf_F \times F\right). \tag{10.2}$$

where,

Υ = Overall technology rating.

wf_i = Technical assessment weighting factor for each criteria.

$A \rightarrow F$ = Criteria rating for each technology.

The technology assessment criteria rating data were presented in Tables 10.3 through 10.11.

The individual weighting factors, wf_i, are chosen so that the sum equals 1; i.e.,

$$\sum_{i=1}^{6} wf_i = 1. \tag{10.3}$$

A set weighting factors was developed for each application to weight the relative importance of each of the six criteria for each type of application. These weighting factors are given in Table 11.1 in Chapter 11.

A computer program using Equation 10.2 was developed to calculate the overall rating, Υ, and rank the refrigeration technologies from high to low based upon the value of Υ for each technology.

A description of the computer program usage and code are presented in Appendix D.

CHAPTER 11. RESULTS OF TECHNOLOGY ASSESSMENT AND SUMMARY OF CONCLUSIONS

Introduction

The alternative refrigeration technologies considered during this project were rated using the computer program in Appendix D. This program uses Equation 10.2 and the numerical technology assessment rating data presented in Tables 10.3 through 10.11 to rate the air conditioning and refrigeration technologies in the five application areas:

1. Domestic air conditioning.
2. Commercial air conditioning.
3. Mobile air conditioning.
4. Domestic refrigeration.
5. Commercial refrigeration.

The data in the computer program are a numerical summary of the patent search, numerical modeling, and technology assessment done during this project.

Technology Assessment Criteria Weighting Factors

A set of technology assessment criteria weighting factors were developed for each of the five applications areas. These weighting factors were selected to reflect the relative importance of the six criteria (State of the art, complexity, size/weight, maintenance, useful life, and energy efficiency) for each application.

Table 11.1 contains the weighting factors chosen for this study. Other weighting factor sets can be chosen to change the relative importance of the six technology assessment criteria.

Table 11.1: Technology assessment criteria weighting factors by application category.

Assessment Criteria	Dom. AC	Com. AC	Mobile AC	Dom. Ref.	Com. Ref.
State of Art	0.20	0.20	0.15	0.20	0.20
Complexity	0.15	0.10	0.20	0.20	0.10
Size/Weight	0.05	0.05	0.30	0.10	0.05
Maintenance	0.15	0.15	0.20	0.10	0.15
Useful Life	0.15	0.20	0.05	0.15	0.20
Energy/Effic.	0.30	0.30	0.10	0.25	0.30

Results

Table 11.2 contains the refrigeration technology ratings for domestic refrigeration. The technology ratings are distributed into three groups:

1. **High.** Vapor-compression was the most suitable refrigeration technology for domestic air conditioning applications. Absorption was also rated highly, but

was penalized because of additional complexity and increased maintenance.

2. **Medium.** The pulse/thermoacoustic, reversed Stirling, solid sorption, and reversed Brayton technologies had similar ratings. In terms of cycle efficiency, the most promising refrigeration technology in this group is solid sorption. Presently, it is an immature technology, which causes its rating to be lower.

3. **Low.** Thermoelectric and magnetic refrigeration have low cycle efficiencies. Presently, the amount of tellurium-based material for semi-conductors is limited. Therefore, the first cost of thermoelectric systems will be high. Magnetic refrigeration technology is immature. Regeneration is the principal area which must be developed.

Table 11.2: Ranking of domestic air conditioning technologies.

Ranking	Refrigeration Technology	Rating
1	Vapor-Compression	4.80
2	Absorption	3.80
3	Pulse/Thermoacoustic	2.95
4	Reversed Stirling	2.90
5	Solid Sorption	2.80
6	Reversed Brayton	2.35
7	Thermoelectric	2.05
8	Magnetic Refrigeration	1.95

Table 11.3 contains the refrigeration technology ratings for commercial refrigeration. The technologies are also distributed into three groups:

1. **High.** Vapor-compression was the most suitable refrigeration technology for commercial air conditioning applications. Absorption was also rated highly.

Since commercial refrigeration systems would generally have a larger cooling capacity and longer life expectancy than domestic systems, they were not penalized as heavily for additional complexity and increased maintenance.

2. **Medium.** The pulse/thermoacoustic, solid sorption, reversed Stirling, and reversed Brayton technologies had ratings above 2 and below 3. The gas-cycle technologies have low cycle efficiencies at the higher source temperatures (See Figures 7.7, 6.8, and 5.10). Solid sorption refrigeration has the highest cycle efficiency in the medium group.

3. **Low.** Thermoelectric and magnetic refrigeration have low cycle efficiencies. Presently, the amount of tellurium-based material for semi-conductors is limited. Therefore, the first cost of thermoelectric systems will be high. Magnetic refrigeration technology is immature. Regeneration is the principal area which must be developed.

Table 11.3: Ranking of commercial air conditioning technologies.

Ranking	Refrigeration Technology	Rating
1	Vapor-Compression	4.85
2	Absorption	4.45
3	Pulse/Thermoacoustic	3.10
4	Solid Sorption	2.80
5	Reversed Stirling	2.75
6	Reversed Brayton	2.35
7	Magnetic Refrigeration	2.05
8	Thermoelectric	1.95

The size/weight criteria was given a high weighting and the efficiency criteria was reduced for mobile air conditioning (Table 11.1). The useful life of mobile air conditioning systems is also shorter than for the other four application areas.

Vapor-compression was rated highest. Thermoelectric and magnetic refrigeration were rated lowest. The reasons for the low rating are low cycle efficiency an the need for a large electrical generation system aboard the vehicle.

Table 11.4: Ranking of mobile air conditioning technologies.

Ranking	Refrigeration Technology	Rating
1	Vapor-Compression	4.30
2	Reversed Stirling	3.25
3	Pulse/Thermoacoustic	2.65
4	Solid Sorption	2.55
5	Reversed Brayton	2.50
6	Absorption	2.30
7	Thermoelectric	2.15
8	Magnetic Refrigeration	1.25

For domestic refrigeration applications, four rating groups were observed:

1. Vapor-compression received the highest rating.

2. Absorption and reversed Stirling technologies were rated next-to-highest. Although absorption refrigeration is capable of high cycle efficiencies, it is not as attractive from the perspective of complexity, size and weight, and maintenance. Reversed Stirling technology was also in this group. It should be noted that the cycle efficiency of reversed Stirling-type refrigerators will be well below that of vapor-compression and absorption systems.

3. The solid sorption, reversed Brayton, and pulse/thermoacoustic refrigeration technologies were in the third group. Of these three, solid sorption is the most promising. It has a higher cycle efficiency than the other two and since it is a batch process, it could be adapted to domestic refrigeration applications.

4. Thermoelectric and magnetic refrigeration were in the lowest rating group for domestic refrigeration. The COP for thermoelectric refrigeration is very low at domestic refrigeration source temperatures (−18 C), as shown in Figure 8.1. The magnetic refrigeration rating was low due to immaturity of the technology, high complexity, and the low cycle efficiency of present systems.

Table 11.5: Ranking of domestic refrigeration technologies.

Ranking	Refrigeration Technology	Rating
1	Vapor-Compression	4.60
2	Absorption	3.70
3	Reversed Stirling	3.25
4	Solid Sorption	3.05
5	Reversed Brayton	2.65
6	Pulse/Thermoacoustic	2.60
7	Thermoelectric	2.20
8	Magnetic Refrigeration	1.95

Commercial refrigeration ratings were found to fall into four groups, as well:

1. Vapor-compression received the highest rating.

2. Absorption refrigeration was rated next-to-highest. Although absorption refrigeration is capable of high cycle efficiencies, it is not as attractive from the perspective of complexity, size and weight, and maintenance.

3. The gas-cycle refrigeration technologies, reversed Stirling, reversed Brayton, and pulse/thermoacoustic, were in the middle group.

4. Thermoelectric and magnetic refrigeration were in the lowest rating group for commercial refrigeration.

Table 11.6: Ranking of commercial refrigeration technologies.

Ranking	Refrigeration Technology	Rating
1	Vapor-Compression	4.70
2	Absorption	3.80
3	Reversed Stirling	3.15
4	Solid Sorption	3.10
5	Reversed Brayton	3.00
6	Pulse/Thermoacoustic	2.80
7	Magnetic Refrigeration	2.05
8	Thermoelectric	2.05

Conclusions

The present course for the development of environmentally suitable refrigeration technology for use in domestic, commercial, and mobile air conditioning and refrigeration applications appears to be the correct one. Vapor-compression refrigeration using non-CFC refrigerants is the most desirable technology for the five application areas considered in this study.

Absorption refrigeration is attractive for commercial refrigeration and air conditioning. If the complexity and maintenance levels can be reduced, they could also be attractive for domestic applications.

Solid sorption refrigeration technology is immature. This technology may have some advantages over absorption systems using liquid absorbents, particularly for domestic refrigeration and air conditioning applications. Cannister sorption and heat transfer efficiencies must be improved above present levels. Complete systems must be developed to demonstrate a reasonable useful life and acceptable maintenance levels. Solid sorption is the most promising new refrigeration technology in terms of technical feasibility, particularly for air conditioning and refrigeration applications were batch processes can be used.

The highest cycle efficiencies for the gas-cycle refrigeration technologies, reversed Stirling, reversed Brayton, and pulse/thermoelectric, occur at source temperatures below the lowest temperature considered in this study (−24 C). These technologies are best suited to low-temperature refrigeration.

The thermoelectric and magnetic refrigeration technologies are impractical for refrigeration and air conditioning applications, at this time.

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APPENDIX A. TECHNOLOGIES IDENTIFIED DURING PATENT AND LITERATURE SURVEYS

Category	Cycle Type	Item or Title	Source	Comments
Heat, Primary	Absorption	Refrigerating Apparatus	U.S. Patent 1369365 U.S. Patent 1369366	Uses vacuum to flash sub-cool liquid NH_3
Heat, Primary	Absorption	Refrigerator	U.S. Patent 1477127	Potassium carbonate and water used as refrigerant/absorbent mixture
Heat, Primary	Absorption	Apparatus for Refrigeration	U.S. Patent 1524297	Utilizes multiple generator tanks
Heat, Primary	Absorption	Absorption Refrigerating Apparatus	U.S. Patent 654395	Steam heated generator and steam driven pumps.
Heat, Primary	Absorption	Automatic Absorber Refrigerator	U.S. Patent 1273364	Intermittent regeneration at night to take advantage of favorable utility rates.
Heat, Primary	Absorption	Liquid Phase Separation in Absorption Refrigeration	U.S. Patent 4283918	Immiscibility property of refrigerant allows separation of refrigerant and absorbent in liquid phase.
Heat, Primary	Absorption	Hyperabsorption Space Condition Process and Apparatus	U.S. Patent 4413480 U.S. Patent 4487027	Generator employs liquid to solid crystallization of saturated salt solution to vaporize liquid refrigerant.
Heat, Primary	Absorption	Absorption Refrigeration Process	U.S. Patent 4475352 U.S. Patent 4475353	Process utilizes other binary mixtures as refrigerant/absorbent pairs.
Heat, Primary	Absorption	Liquid Phase Separation in Absorption Refrigeration	U.S. Patent 4283918	Refrigerant is methyl diethylamine. Absorbent is water.
Heat, Primary	Absorption	Absorption Type Heat Pump System	U.S. Patent 4448040	Utilizes two LiBr/water absorption cycles.
Heat, Primary	Absorption	Compressorless Air Conditioner	[Reference 1]	Evaporative cooling/ lithium bromide dehumidification of incoming air stream.

Category	Cycle Type	Item or Title	Source	Comments
Heat, Primary	Vapor-Compression	Refrigeration System	U.S. Patent 4378681	Uses an ejector as the compression device.
Heat, Primary	Vapor-Compression	Method of Cold Production and Devices for the Practical Application of Said Method	U.S. Patent 4070871	Employs a constant pressure expansion in a variable volume chamber.
Heat, Primary	Adsorption	Refrigeration Cycle Apparatus Having Refrigerant Separating System With Pressure Swing Adsorption	U.S. Patent 4972676	Refrigerant is a binary mixture of R-22 and R-114. The adsorbing tower is charged with activated alumina.
Heat, Primary	Adsorption	Regenerative Adsorbent Heat Pump	U.S. Patent 5046319	Uses multiple zeolite canisters as compressors.
Heat, Waste	Absorption	Process and Apparatus for Refrigeration	U.S. Patent 1265037	Requires no absorber or pump for the water side.
Heat, Waste	Vapor-Compression	Refrigeration Apparatus and Method	U.S. Patent 4345440	Mobile application using waste heat from engine exhaust gasses.
Heat, Waste	Vapor-Compression	Twin Reservoir Heat Transfer Circuit	U.S. Patent 4612782	Vapor is produced in a tank heated by a coil. The vapor passes through an ejector and then condensed. After evaporation the refrigerant is collected in an unheated tank.
Heat, Waste	Vapor-Compression	Device to Create Cooling Through Use of Waste Heat	U.S. Patent 4192148	Uses steam jet.
Heat, Other (Solar)	Adsorption	Modular Solar Powered Heat Pump	U.S. Patent 4199952	Uses a silica gel adsorber. Insolation generates heat to drive desorption process.
Heat, Other (Solar)	Absorption	Solar Powered Air Conditioning System Employing Hydroxide Water Solution	U.S. Patent 4151721	

Category	Cycle Type	Item or Title	Source	Comments
Heat, Other (Solar)	Absorption	Cooling Method and System Therefor	U.S. Patent 4488408	Refrigerant/ Absorbent pair is lithium/bromide and water.
Work, Phase Change	Vapor-Compression	Refrigerating or Ice Making Apparatus	U.S. Patent 1253895	Uses NH_3 as the refrigerant, multiple stage compression.
Work, Phase Change	Vapor-Compression	Process of Refrigeration	U.S. Patent 1264807 U.S. Patent 1379102	Series compressors with inter-cooling between stages.
Work, Phase Change	Vapor-Compression	Process of Refrigeration	U.S. Patent 1264845	Multi-stage compression with inter-cooling.
Work, Phase Change	Vapor-Compression	Process and Apparatus for Refrigeration	U.S. Patent 1337175	Use SO_2 as the refrigerant.
Work, Phase Change	Vapor-Compression	Refrigerating System	U.S. Patent 1455580	Series evaporators with liquid vapor separator between stages.
Work, Phase Change	Vapor-Compression	Artificial Refrigeration System	U.S. Patent 1520936	Refrigerant is SO_2 or CH_3Cl
Work, Phase Change	Vapor-Compression	Process and Apparatus for Multiple Stage Compression for Refrigeration	U.S. Patent 1471732	Inter-cooling between compression stages using water .
Work, Phase Change	Vapor-Compression	Refrigerating Process and Apparatus	U.S. Patent 1512133	Water driven compressor, water cooled condenser.
Work, Phase Change	Vapor-Compression	Refrigeration and Power System	U.S. Patent 1519353	Cascaded system using three separate systems, each using a different refrigerant; NH_3, SO_2, and CO_2.
Work, Phase Change	Vapor-Compression	Refrigerant	U.S. Patent 1547202	Methyl bromide

Category	Cycle Type	Item or Title	Source	Comments
Work, Phase Change	Vapor-Compression	Method of Improving Refrigerating Capacity and Coefficient of Performance in a Refrigerating System, and a Refrigerating System for Carrying Out Said Method	U.S. Patent 4014182	Refrigerant is flashed to vapor in an initially evacuated vessel.
Work, Phase Change	Vapor-Compression	Refrigeration Apparatus and Method	U.S. Patent 4019337	System utilizes two capillary tubes and a flow control which senses evaporator outlet temperature.
Work, Phase Change	Vapor-Compression	Direct Contact Heat Transfer System Using Magnetic Fluids	U.S. Patent 4078392	A ferrofluid is separated from a suitable refrigerant by magnetic means and circulated to the cooling load.
Work, Phase Change	Vapor-Compression	Dual Flash and Thermal Economized Refrigeration System	U.S. Patent 4141708	Uses a low temperature flash economizer and a high temperature flash economizer in conjunction with two compressors.
Work, Phase Change	Vapor-Compression	Hydraulic Refrigeration System	U.S. Patent 4157015 U.S. Patent 4251998 U.S. Patent 4424681	Vapor from evaporator is entrained in a vertically downward moving column of fluid (water). The vapor is compressed and condensed simultaneously.
Work, Phase Change	Vapor-Compression	Vapor Compression Refrigeration and Heat Pump Apparatus	U.S. Patent 4235079	Expansion valve is replaced with an expansion motor.
Work, Phase Change	Vapor-Compression	Refrigeration and Space Cooling Unit	U.S. Patent 4235080	Converts a portion of the latent heat into mechanical energy through a turbine operating between the vapor pressure and a vacuum.
Work, Phase Change	Vapor-Compression	Vapor Compression Refrigeration System and a Method of Operation Therefor	U.S. Patent 4258553	Series compressors with inter-cooling

Category	Cycle Type	Item or Title	Source	Comments
Work, Phase Change	Vapor-Compression	Refrigerant Sub-Cooling	U.S. Patent 4285205	Commercial application with multiple compressors in parallel. A heat exchanger increases the suction gas temperature and sub-cools the liquid prior to expansion.
Work, Phase Change	Vapor-Compression	Heat Exchange Method Using Natural Flow of Heat Exchange Medium	U.S. Patent 4295342	Uses natural convection to circulate refrigerant in a circuit to transfer heat from warm to cold. The system requires a difference in heat exchanger elevation and a by-pass valve and circuit around the compressor.
Work, Phase Change	Vapor-Compression	Gas Compression System	U.S. Patent 4311025	The refrigeration circuit utilizes a rotating disc compressor.
Work, Phase Change	Vapor-Compression	Method for Utilizing Gas-Solid Dispersions in Thermodynamic Cycles for Power Generation and Refrigeration	U.S. Patent 4321799	Utilizes a circulating dispersion of solid particles in a gaseous refrigerant.
Work, Phase Change	Vapor-Compression	Hybrid Heat Pump	U.S. Patent 4481783	The system makes use of both compression and absorption. Uses both a compressor and a generator.
Work, Phase Change	Vapor-Compression	Refrigerator Cooling and Freezing System	U.S. Patent 4513581	Single compressor, multiple series evaporators in freezer and food storage compartments.
Work, Phase Change	Vapor-Compression	Refrigeration System with Refrigerant Pre-Cooler	U.S. Patent 4577468	Single circuit vapor compression system with a condenser outlet sub-cooler.

Category	Cycle Type	Item or Title	Source	Comments
Work, Phase Change	Vapor-Compression	Refrigeration Process and Apparatus	U.S. Patent 4586344	Two immiscible or partly miscible refrigerants are mixed and evaporated in an evaporator. The absorption system absorbs a portion of the refrigerant, the compression system condenses the non-absorbed refrigerant vapor.
Work, Phase Change	Vapor-Compression	Apparatus for Maximizing Refrigeration Capacity	U.S. Patent 4599873	A pump is employed at the condenser outlet to prevent flashing, condenser temperature and pressure are allowed to fluctuate with ambient conditions.
Work, Phase Change	Vapor-Compression	Refrigeration System	U.S. Patent 4640100	A pump is employed to vary the condensing pressure as the ambient temperature fluctuates. System has multiple compressors in parallel for commercial applications
Work, Phase Change	Vapor-Compression	Refrigeration System with Hot Gas Pre-Cooler	U.S. Patent 4702086	A portion of the liquid refrigerant is evaporated to pre-cool the vapor between the compressor and condenser.
Work, Phase Change	Vapor-Compression	Chemically Assisted Mechanical Refrigeration System	U.S. Patent 4707996	A refrigerant/ solvent pair are separated into vapor and liquid phases respectively. Solvent is then used as a coolant to be circulated through a jacket around compressor. After heat exchange in the condenser and a pre-mixer the fluids are again combined.
Work, Phase Change	Vapor-Compression	Indirect Evaporative Cooling System	U.S. Patent 4827733	Water is evaporated to cool the incoming air and to condense a portion of the refrigerant vapor. A second heat exchanger (evaporator) cools the room air.

Category	Cycle Type	Item or Title	Source	Comments
Work, Phase Change	Vapor-Compression	Refrigerating System Incorporating A Heat Accumulator and Method of Operating the Same	U.S. Patent 4833893	Heat accumulator stores heat from vapor at compressor outlet for defrosting and lowering starting torque.
Work, Phase Change	Vapor-Compression	Refrigerating System Having A Compressor With An Internally and Externally Controlled Variable Displacement Mechanism	U.S. Patent 4882909	An axial piston, variable displacement compressor.
Work, Phase Change	Vapor-Compression	Binary Solution Compressive Heat Pump with Solution Circuit	U.S. Patent 4918945	Hybrid vapor compression/ absorption cycle.
Work, Phase Change	Vapor-Compression	Refrigerating Cycle Utilizing Cold Accumulation Material	U.S. Patent 4918936	A portion of the liquid refrigerant is evaporated to cool a thermal sink during small load conditions. The thermal sink then provides temporary additional capacity during high load conditions.
Work, Phase Change	Vapor-Compression	Binary Solution Compressive Heat Pump with Solution Circuit	U.S. Patent 4918945	Hybrid vapor compression/ absorption cycle.
Work, Phase Change	Vapor-Compression	Refrigerating Cycle Utilizing Cold Accumulation Material	U.S. Patent 4918936	A portion of the liquid refrigerant is evaporated to cool a thermal sink during small load conditions. The thermal sink then provides temporary additional capacity during high load conditions.
Work, Phase Change	Vapor-Compression	Binary Solution Compressive Heat Pump with Solution Circuit	U.S. Patent 4918945	Hybrid vapor compression/ absorption cycle.
Work, Phase Change	Vapor-Compression	Refrigerating Cycle Utilizing Cold Accumulation Material	U.S. Patent 4918936	A portion of the liquid refrigerant is evaporated to cool a thermal sink during small load conditions. The thermal sink then provides temporary additional capacity during high load conditions.

Category	Cycle Type	Item or Title	Source	Comments
Work, Phase Change	Vapor-Compression	Air Conditioning and Heat Pump System	U.S. Patent 49813023	Compressor housing and compressor outlet flow are cooled by a variable portion of the evaporator return flow.
Work, Phase Change	Vapor-Compression	Heat Pump Apparatus	U.S. Patent 4679403	Cycle utilizes a variable speed compressor and a refrigerant blend.
Work, Phase Change	Vapor-Compression	Process to Expand the Temperature Glide of a Non-Azeotropic Working Fluid Mixture in a Vapor Compression Cycle	U.S. Patent 4987751	Refrigerant not specified in this patent.
Work, No Phase Change	Gas	Refrigerating Machine	U.S. Patent 1508522	Air, Stirling cycle.
Work, No Phase Change	Gas	Refrigerating Apparatus Based Upon the Use of Air	U.S. Patent 1545587	Air, Stirling cycle.
Work, No Phase Change	Gas	Method and Apparatus for Inducing Heat Changes	U.S. Patent 1275507	Air, Stirling cycle.
Work, No Phase Change	Gas	Air Refrigerating Machine	U.S. Patent 1295724	Air, Brayton cycle.
Work, No Phase Change	Liquid	Heat Pump/ Refrigerator Using Liquid Working Fluid	U.S. Patent 4353218 Reference [2] Reference [3]	Multi-Engine Stirling cycle with regeneration.
Direct Electric	Peltier Effect		Reference [4]	Direct conversion of electrical to thermal energy.

Category	Cycle Type	Item or Title	Source	Comments
Direct Electric	Electrolytic	Electrolytic System of Refrigeration	U.S. Patent 1114006	Vapor is produced by applying an electric potential across an electrolyte in a cell. Also uses an evaporator and condenser.
Magnetic	Collapsing Field	Magnetic Refrigeration	U.S. Patent 4509334 U.S. Patent 4589953 Reference [5]	Helium gas is used as the heating medium.
Magnetic	Displaced Core		Reference [5]	Gadolinium core is displaced in and out of a non-collapsing magnetic field.
Other	Evaporative	Refrigerating Machine	U.S. Patent 1483990	Multi-stage system utilizing steam ejectors and a vacuum.
Other	Evaporative	Desiccant Air Conditioning Unit	Reference [6]	Uses desiccant dehumidification and air to air heat exchange.

References:

1. Norland, J. "$2.4 Million Financing for Compressorless A/C." *Air Conditioning, Heating & Refrigeration News*, November 11, 1992.
2. Swift, G. W. "Malone Refrigeration", *ASHRAE Journal*, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, New York. November 1990.
3. Swift, G. W. "A Stirling engine with a liquid working substance", *Journal of Applied Physics*, American Institute of Physics. Volume 65, Number 11.
4. Wood, B. D. *Applications of Thermodynamics*, 2nd Edition, Addison-Wesley, Reading, MA, p. 406-407.
5. Chen, F. C., Murphy, R. W., et al. "Loss Analysis of the Thermodynamic Cycle of Magnetic Heat Pumps." Oakridge National Laboratory, Oakridge, Tennessee, Report Number ORNL/TM-- 11608.
6. Miller-Picking Corporation. "Report on the Preliminary Design for a Desiccant Based A/C Unit", Ref.: T-14940, Miller-Picking Corp., Johnstown, PA, 1990.

APPENDIX B. ALTERNATIVE REFRIGERATION TECHNOLOGY MODELING PROGRAM

Introduction

The objective of this program is to determine the modeled coefficient of performance for different refrigeration thermodynamic cycles and the ideal (reversed Carnot) COP at a fixed sink temperature and over a range of source temperatures from -24 C to 28 C. All models assume steady state operation of a system in communication with two infinite thermal reservoirs, the source and sink. Both reservoirs are assumed to be at a fixed temperature which is unchanging over time or with the amount of heat removed from or added to them.

The program was developed to estimate the coefficient of performance of the following refrigeration cycles:

- Stirling
- Reversed Brayton
- Regenerative reversed Brayton
- Thermoelectric
- Pulse tube and thermoacoustic

- Magnetic Stirling
- Magnetic constant field/polytropic
- Magnetic combined constant field/isentropic

This program was written in FORTRAN. The source code can be compiled and used on any system having a FORTRAN compiler. The executable version we have furnished can be installed and run on IBM or IBM compatible personal computers.

The program is structured in an easy to use, interactive, menu driven format. The user is asked to supply information in a step by step process. Some of the input data are supplied as default values which reflect reasonable estimates, consistent with the present state of the art for each alternative technology. The user can substitute other data in place of the default values if they wish.

Validation of the Program

All thermodynamic models used in this program were validated by comparing the results of the numerical model with the results of hand colculations. The thermodynamic property subroutines were validated by comparing the results with tabulated values in Reynolds [26].

Program Structure

The program source code is contained in three files:

1. 1.FOR is the main program which contains the introductory screen formatted output statements, decision logic for the menus to select a particular refrigeration technology, and the default values for input data.

2. CYCLES.FOR contains the thermodynamic modeling subroutines used to calculate the theoretical COPs for the refrigeration cycles.

3. PROPS.FOR contains the thermodynamic property subroutines for air, helium, and gadolinium.

The main program (1.FOR) calls the appropriate cycle subroutine from CYCLES.FOR, which in turn calls a property subroutine from PROPS.FOR. The source code is well documented with comment statements indicating the purpose of each block of code.

System Requirements

This program was written in FORTRAN code which is compatible with MICROSOFT FORTRAN version 5.0. The executable version of the program has no special requirement as to micro-processor type; it can be run on computers using the 8086 through 80486 processors.

One feature of MICROSOFT FORTRAN which must be kept in mind when using this program is the choice of linking library options which are used to from the executable file during the compiling and linking process. MICROSOFT has developed separate libraries which are selected during the installation of their FORTRAN software. For computers equipped with the 8087, 80287, or 80387 math coprocessor the library LLIBFOR007 is used. Since the math coprocessor is incorporated on all 80486 chips, this library is utilized for these machines , as well. For computers using the 8086, 80286, and 80386 micro-processor without the 8087, 80287, or 80387 math coprocessor, the emulator library LLIBFORE is used. Therefore, if the program is

linked using the LLIBFOR007 library to form the executable file, it will not run on a computer that does not have a math coprocessor.

Program Installation

The program includes some screen clearing commands during execution. A line must be included in the computer's CONFIG.SYS file which reads exactly as follows:

DEVICE=C:\DOS\ANSI.SYS

If this line is not included, the code "2J]" will appear in the upper left corner of the monitor screen; however, the program can still be run and will provide correct results.

To install the program:

1. Choose or create a suitable directory on the hard disk.
2. Insert the diskette in the A drive and choose the directory entitled 1FOR.
3. Type the command:

COPY 1.EXE C:\(directory name)\1.EXE

Running the Program

To start the program, type "1" and press return. Each screen is self explanatory and prompts the user for the required input action (such as pressing return to refresh a screen), numerical input value, or choice (yes or no). The user is also prompted to furnish an output file name for the file to which the output data will be written.

At the end of a program sequence the user can choose to either start a new sequence or to exit the program by answering "Y" or "N" to the question appearing on the screen.

The data from each run will be found in the data file named during the run sequence. Each new case must have a unique file name. If the same file name is given, the data from the previous run will be overwritten. It is suggested that the file name be appended with a letter or number to indicate the order of the run. For example, the file names TE1.DAT, TE2.DAT, and TE3.DAT could be used for the data files for the first, second, and third runs used to consider different cases for a thermoelectric cooling system. A sample data file is included as Appendix C.

```
C     ************************************************************************
C
C                    REFRIGERATION PERFORMANCE COMPARISON ROUTINE
C
C
C
C                                 PREPARED BY:
C
C
C                          MECHANICAL ENGINEERING DEPARTMENT
C
C                               IOWA STATE UNIVERSITY
C
C                                    AMES, IOWA
C
C     ************************************************************************
C
C     THIS ROUTINE COMPARES THE PERFORMANCE OF VARIOUS ALTERNATIVE
C     REFRIGERATION CYCLES USING THERMODYNAMIC MODELS.  THE PROGRAM
C     IS IN THREE PARTS:
C
C           1)  A MAIN PROGRAM (1.FOR) IN WHICH THE TEMPERATURES AND
C               SPECIFIC PARAMETERS FOR THE MODELS ARE ENTERED, AND THE
C               DESIRED MODEL IS SELECTED.
C
C           2)  A SECOND FILE (CYCLES.FOR) CONTAINS A COLLECTION OF
C               THERMODYNAMIC MODELS IN SUBROUTINE FORM.
C
C           3)  THE THIRD FILE (PROPS.FOR) CONTAINS A COLLECTION OF
C               THERMODYNAMIC PROPERTY SUBROUTINES WHICH ARE CALLED
C               BY THE APPROPRIATE MODEL IN CYCLES.FOR.
C
C     THE PROGRAM IS STRUCTURED IN AN INTERACTIVE MANNER IN WHICH THE USER
C     PROMPTED TO ANSWER A SERIES OF QUESTIONS REGARDING THE CHOICE OF A
C     SINK TEMPERATURE, HX APPROACH TEMPERATURES, OUTPUT DATA FILE NAME,
C     REFRIGERATION CYCLE MODEL CHOICE, AND MODEL-SPECIFIC PARAMETERS.
C
C     WHERE POSSIBLE, REASONABLE DEFAULT VALUES OF PARAMETERS HAVE BEEN
C     INCLUDED AS A STARTING POINT.  THE USER CAN CHANGE THESE VALUES AS
C     THEY WISH.
C
C     WE HAVE TRIED TO MAKE THE PROGRAM AS "CRASHPROOF" AS POSSIBLE;
C     HOWEVER, IN SOME INSTANCES THE PROGRAM EXECUTION WILL STOP IF
C     CERTAIN MODEL-SPECIFIC CONSTRAINTS ARE EXCEEDED.
C
C     SYSTEM REQUIREMENTS:
C
C     THE PROGRAM IS WRITTEN IN FORTRAN AND CAN BE COMPILED ON AN
```

```
C     IBM OR IBM COMPATIBLE PERSONAL COMPUTER.  WE USED MICROSOFT
C     FORTRAN VERSION 5.0 TO CREATE THE EXECUTABLE FILE.
C
C     A LINE SHOULD BE INSERTED INTO THE CONFIG.SYS FILE WHICH
C     LISTS THE ANSI.SYS FILE AS A DEVICE.  THE LINE SHOULD READ:
C
C                        DEVICE= ANSI.SYS
C
C     THE COMPUTER SHOULD THEN BE RE-BOOTED.  IF THIS LINE IS NOT
C     PRESENT IN THE CONFIG.SYS FILE, THE PROGRAM WILL STILL FUNCTION.
C     HOWEVER, A CHARACTER STRING "2J]" WILL APPEAR IN THE UPPER LEFT-
C     HAND CORNER OF THE SCREEN, AND THE SCREEN CLEARING FEATURE BET-
C     WEEN PARAMETER CHOICES MAY NOT FUNCTION CORRECTLY.
C
C
C     ********* ALL TEMPERATURE INPUTS MUST BE IN DEGREES CELSIUS! ***********
C
C     ************************************************************************
C
C     VARIABLE DECLARATION:
C
      IMPLICIT REAL*8(A-H,O-Z)
      INTEGER NCYCLE,II,JJ,KTT2
      CHARACTER*50 OUTPUTFILE,WARN
      CHARACTER*1 CHOICE1,CHOICE2,CHOICE3,CHOICE4,CHOICE5,CHOICE6,
     &            CHOICE7,CHOICE8
C
C     ************************************************************************
C
C     CLEAR THE SCREEN:
C
      JJ = 27
      WRITE(6,500) JJ
500   FORMAT(1X,A1,'[2J')

C
C     ************************************************************************
C
C     INTRODUCTORY SCREEN:
C
      WRITE(6,*) ' '
      WRITE(6,*) ' '
      WRITE(6,501) 'REFRIGERATION TECHNOLOGY COMPARISON ROUTINE'
501   FORMAT (16X,A43,//)
C
      WRITE(6,502) 'DEPARTMENT OF MECHANICAL ENGINEERING'
502   FORMAT(19X,A36,/)
C
```

```
      WRITE(6,21) 'IOWA STATE UNIVERSITY'
21    FORMAT(27X,A21)
C
      WRITE(6,504) 'AMES, IOWA 50011'
504   FORMAT(29X,A16,///////////)
C
C     ***********************************************************************
C
C     CLEAR THE SCREEN:
C
      II = CHAR(13)
      WRITE(6,505)'PRESS RETURN'
505   FORMAT(32X,A12)
C
      READ(6,506) II
506   FORMAT(A1)
C
      WRITE(6,500) JJ
C
C     ***********************************************************************
C
      WRITE(6,45) 'THIS PROGRAM CAN BE USED TO COMPARE DIFFERENT'
45    FORMAT(15X,A45)
C
      WRITE(6,48) 'REFRIGERATION TECHNOLOGIES AT A FIXED SINK TEMP.'
48    FORMAT(15X,A48)
      WRITE(6,509) 'OVER A RANGE OF SOURCE TEMPERATURES'
509   FORMAT(15X,A35,//////////)

      WRITE(6,505)'PRESS RETURN'
C
C     ***********************************************************************
C
C     OPTION TO CHANGE THE SINK TEMPERATURE:
C
      READ(6,506) II
      WRITE(6,500) JJ
C
C
C     USER OPPORTUNITY TO CHANGE TO A DIFFERENT SINK TEMPERATURE:
C
4444  WRITE(6,37) 'THE DEFAULT SINK TEMPERATURE FOR THIS'
37    FORMAT(15X,A37)
      WRITE(6,539) 'APPLICATION IS 35.0 DEGREES C.'
539   FORMAT(15X,A30,/////)
      WRITE(6,540) 'DO YOU WISH TO ACCEPT THIS TEMPERATURE?  Y OR N'
540   FORMAT(15X,A47,//////)
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
```

```
31    FORMAT(15X,A31)
C
4001  READ(6,519) CHOICE1
519   FORMAT(A1)
C
      WRITE(6,500) JJ
      IF ((CHOICE1 .EQ. 'Y').OR.(CHOICE1 .EQ. 'y')) THEN
         T1C = 35.0
         CONTINUE
      ELSEIF ((CHOICE1 .EQ. 'N').OR.(CHOICE1 .EQ. 'n')) THEN
         WRITE(6,42) 'ENTER THE NEW SINK TEMPERATURE IN DEG. C.'
42       FORMAT(15X,A42)
         READ(6,*) T1CN
         WRITE(6,500) JJ
         WRITE(6,38) 'YOU HAVE CHOSEN A NEW SINK TEMPERATURE'
38       FORMAT(15X,A38)
         WRITE(6,543) 'OF: ',T1CN,' DEGREES C.'
543      FORMAT(15X,A4,F6.3,A11,//////)
         T1C = T1CN
      ELSE
            WRITE(6,752)'TYPE Y OR N AND PRESS RETURN'
752         FORMAT(15X,A29)
            GO TO 4001
      ENDIF
C
C     ************************************************************************
C
      WRITE(6,43)'THE RANGE OF SOURCE TEMPERATURES OVER WHICH'
      WRITE(6,43)'THE COP WILL BE CALCULATED IS -24 C TO 28 C.'
43    FORMAT(15X,A44)
C
C     ************************************************************************
C
C     CLEAR THE SCREEN:
C
      WRITE(6,*) ' '
      WRITE(6,*) ' '
      WRITE(6,*) ' '
      WRITE(6,*) ' '
      WRITE(6,*) ' '
      WRITE(6,*) ' '
      WRITE(6,505)'PRESS RETURN'
C
      READ(6,506) II
      WRITE(6,500) JJ
C
C     ************************************************************************
C
```

```
C     MINIMUM APPROACH TEMPERATURE SELECTION:
C
      DELTL = 5.0
      DELTH = 5.0
C
      WRITE(6,48)'THIS PROGRAM USES A MINIMUM APPROACH TEMPERATURE'
      WRITE(6,43)'TO ACCOUNT FOR HEAT EXCHANGER EFFECTIVENESS.'

      WRITE(6,47)'THE DEFAULT VALUE IS 5 DEGREES CELSIUS FOR BOTH'
47    FORMAT(15X,A47)
      WRITE(6,436)'THE SOURCE AND SINK HEAT EXCHANGERS.'
436   FORMAT(15X,A36,////)
      WRITE(6,649)'DO YOU WISH TO ACCEPT THESE TEMPERATURES? Y OR N'
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
C
1001  READ(6,519) CHOICE2
C
      WRITE(6,500) JJ
      IF ((CHOICE2 .EQ. 'Y').OR.(CHOICE2 .EQ. 'y')) THEN
           CONTINUE
      ELSEIF ((CHOICE2 .EQ. 'N').OR.(CHOICE2 .EQ. 'n')) THEN
C
      WRITE(6,50)'ENTER THE SOURCE MINIMUM APPROACH TEMP. IN DEG. C.'
50    FORMAT(15X,A50)
         READ(6,*) DELTL
      WRITE(6,48) 'YOU HAVE CHOSEN A NEW SOURCE MIN. APPROACH TEMP.'
C
         WRITE(6,543) 'OF: ',DELTL,' DEGREES C.'
C
      WRITE(6,51) 'ENTER THE SINK MINIMUM APPROACH TEMPERATURE DEG. C.'
51    FORMAT(15X,A51)
C
         READ(6,*) DELTH
         WRITE(6,39) 'YOU HAVE CHOSEN A NEW SINK MIN. APPROACH TEMP.'
39    FORMAT(15X,A46)
C
         WRITE(6,543) 'OF: ',DELTH,' DEGREES C.'
C
            WRITE(6,*) ' '
            WRITE(6,*) ' '
            WRITE(6,*) ' '
            WRITE(6,505)'PRESS RETURN'
            READ(6,506) II
            WRITE(6,500) JJ
      ELSE
            WRITE(6,752)'TYPE Y OR N AND PRESS RETURN'
            GO TO 1001
      ENDIF
```

```
C
C     ***********************************************************************
C
C     INPUT THE NUMBER OF THE REFRIGERATION TECHNOLOGY TO BE CONSIDERED:
C
      WRITE(6,545)'ENTER THE NUMBER OF THE CYCLE FROM THE MENU'
545   FORMAT(15X,A43,///)
      WRITE(6,546) 'STIRLING                                  1'
      WRITE(6,546) 'REVERSED BRAYTON                          2'
      WRITE(6,546) 'REVERSED BRAYTON WITH REGENERATION        3'
      WRITE(6,546) 'THERMOELECTRIC                            4'
      WRITE(6,546) 'PULSE TUBE                                5'
      WRITE(6,546) 'MAGNETIC HEAT PUMP                        6'
546   FORMAT(15X,A43,/)
C
      READ(6,*) NCYCLE
      WRITE(6,500) JJ
C
C
      WRITE(6,48)'YOU WILL BE GIVEN THE OPPORTUNITY TO CHANGE SOME'
      WRITE(6,49)'OF THE DEFAULT VALUES OF PARAMETERS FOR THE CYCLE'
49    FORMAT(15X,A49)
      WRITE(6,115)'YOU HAVE CHOSEN'
115   FORMAT(15X,A15,////)
1700  WRITE(6,45) 'FIRST, ENTER THE NAME OF THE DATA OUTPUT FILE'
      READ(6,548) OUTPUTFILE
      WRITE(6,500) JJ
548   FORMAT(A50)
C
      OPEN(11,FILE=OUTPUTFILE,STATUS='UNKNOWN')
      WRITE(11,517)'SINK TEMPERATURE =',T1C,'CELSIUS'
517   FORMAT(15X,A18,F3.0,2X,A7,/)
      WRITE(11,518)'HIGH TEMP HX DELTA T =',DELTH,'CELSIUS'
518   FORMAT(15X,A22,F3.0,2X,A7,/)
      WRITE(11,522)'LOW TEMP HX DELTA T =',DELTL,'CELSIUS'
522   FORMAT(15X,A21,F3.0,2X,A7,////)
C
C     ***********************************************************************
C
C     CHOOSE REFRIGERATION CYCLES AND SPECIFIC PARAMETERS:
C
      IF(NCYCLE .EQ. 1) THEN
C
      WRITE(6,46)'YOU HAVE CHOSEN THE STIRLING CYCLE.  THE IDEAL'
      WRITE(6,46)'THEORETICAL STIRLING CYCLE PROVIDES THE CARNOT'
46    FORMAT(15X,A46)
      WRITE(6,47)'COP.  HOWEVER, THE COP CALCULATED BY THIS MODEL'
      WRITE(6,51)'WILL BE LOWER DUE TO THE IRREVERSIBILTIY INTRODUCED'
```

```
      WRITE(6,24)'IN THE HEAT EXCHANGERS.'
24    FORMAT(15X,A24)
      WRITE(6,*) ' '
      WRITE(6,*) ' '
      WRITE(6,*) ' '
      WRITE(6,45)'AS A COMPARISON-THE BEST COPs CURRENTLY BEING'
      WRITE(6,48)'OBTAINED EXPERIMENTALLY IN FREE PISTON STIRLING'
      WRITE(6,50)'REFRIGERATORS IS ABOUT 30% OF CARNOT WHEN OPERATED'
      WRITE(6,125)'IN THIS TEMPERATURE RANGE'
125   FORMAT(15X,A25,///)
C
      WRITE(6,505)'PRESS RETURN'
      READ(6,506) II
      WRITE(6,500) JJ


      WRITE(11,515)'STIRLING CYCLE RESULTS'
515   FORMAT(15X,A22,//)
      WRITE(11,899)'SOURCE TEMP.','CARNOT COP','COP','COP/COPC',
     &      'COMMENTS'
899   FORMAT(7X,A12,2X,A10,6X,A3,4X,A8,10X,A8,/)
C
      GO TO 3333
C     ***********************************************************************
C
C     REVERSED BRAYTON CYCLE:
C
      ELSE IF(NCYCLE .EQ. 2) THEN
C
      WRITE(6,43)'YOU HAVE CHOSEN THE REVERSED BRAYTON CYCLE.'
C
C     ********************************************************************
C
C     ASSIGN VALUES TO THE PRESSURE RATIO, COMPRESSOR EFFICIENCY, AND
C     EXPANDER EFFICIENCY:
C
      WRITE(6,47)'THIS PROGRAM USES A PRESSURE RATIO WHICH CAN BE'
      WRITE(6,47)'CHANGED BY THE USER.  THE DEFAULT VALUE IS 3.0.'
      WRITE(6,51)'IT IS RECOMMENDED THAT THE PRESS. RATIO NOT EXCEED'
      WRITE(6,49)'4.0 IN THIS PROGRAM DO TO LOW TEMP. THERMODYNAMIC'
      WRITE(6,321)'PROPERTY LIMITATIONS.'
321   FORMAT(15X,A21,///)
      WRITE(6,649)'DO YOU WISH TO ACCEPT THE PRESSURE RATIO, Y OR N?'
649   FORMAT(15X,A49,//////)
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
C
C
C     ********************************************************************
```

```
C
1002  READ(6,519) CHOICE3
      WRITE(6,500) JJ
      IF ((CHOICE3 .EQ. 'Y').OR.(CHOICE3 .EQ. 'y')) THEN
C
            PRATIO = 3.0
            CONTINUE
C
      ELSEIF ((CHOICE3 .EQ. 'N').OR.(CHOICE3 .EQ. 'n')) THEN
C
            WRITE(6,28) 'ENTER THE NEW PRESSURE RATIO'
28          FORMAT (15X,A28)
            READ(6,*) PRATION
            WRITE(6,500) JJ
            WRITE(6,36) 'YOU HAVE CHOSEN A NEW PRESSURE RATIO'
36          FORMAT(15X,A36)
            WRITE(6,543) 'OF: ',PRATION,' '
            PRATIO = PRATION
C
      ELSE
C
            WRITE(6,752)'TYPE Y OR N AND PRESS RETURN'
            GO TO 1002
C
      ENDIF
C
C     ***********************************************************************
C
      WRITE(6,46)'ISENTROPIC EFFICIENCIES FOR THE COMPRESSOR AND'
      WRITE(6,46)'EXPANDER ARE USED TO ACCOUNT FOR THE IRREVERS-'
      WRITE(6,49)'IBILITIES PRESENT IN THESE COMPONENTS.  PRESENTLY'
      WRITE(6,51)'THE DEFAULT VALUES ARE 0.85 FOR BOTH THE COMPRESSOR'
      WRITE(6,313)'AND EXPANDER.'
313   FORMAT(15X,A13,///)
C
      WRITE(6,343)'DO YOU WISH TO ACCEPT THESE VALUES, Y OR N?'
343   FORMAT(15X,A43,///)
C
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
C
C     CLEAR THE SCREEN:
C
1003  READ(6,519) CHOICE4
      WRITE(6,500) JJ
      IF ((CHOICE4 .EQ. 'Y').OR.(CHOICE4 .EQ. 'y')) THEN
C
            ETAE = 0.85
            ETAC = 0.85
```

```
C
      ELSEIF ((CHOICE4 .EQ. 'N').OR.(CHOICE4 .EQ. 'n')) THEN
C
            WRITE(6,42) 'ENTER THE COMPRESSOR ISENTROPIC EFFICIENCY'
            READ(6,*) ETACN
            WRITE(6,43) 'YOU HAVE CHOSEN A NEW COMPRESSOR EFFICIENCY'
            WRITE(6,543) 'OF: ',ETACN,' '
C
            WRITE(6,43) 'ENTER THE EXPANDER ISENTROPIC EFFICIENCY'
            READ(6,*) ETAEN
            WRITE(6,41)'YOU HAVE CHOSEN A NEW EXPANDER EFFICIENCY'
41          FORMAT(15X,A41)
            WRITE(6,543) 'OF: ',ETAEN,' '
            ETAC = ETACN
            ETAE = ETAEN
            CONTINUE
C
      ELSE
            WRITE(6,752)'TYPE Y OR N AND PRESS RETURN'
            GO TO 1003
      ENDIF
C
C     ************************************************************************
C
C     WRITE THE REVERSED BRAYTON CYCLE RESULTS:
C
      WRITE(11,520)'REVERSED BRAYTON CYCLE RESULTS'
520   FORMAT(15X,A32,/)
      WRITE(11,590)'PRESSURE RATIO =',PRATIO
      WRITE(11,590)'COMP. EFF.=',ETAC
      WRITE(11,590)'EXPANDER EFF. =',ETAE
590   FORMAT(12X,A16,F10.3,//)
      WRITE(11,899)'SOURCE TEMP.','CARNOT COP','COP','COP/COPC',
     &      'COMMENTS'

C
      WRITE(6,820)'EXECUTING REVERSED BRAYTON CYCLE MODEL'
820   FORMAT(15X,A38,/)
      WRITE(6,590)'PRESSURE RATIO =',PRATIO
      WRITE(6,590)'COMP. EFF.=',ETAC
      WRITE(6,590)'EXPANDER EFF. =',ETAE
890   FORMAT(15X,A16,F10.3)
C
      GO TO 3333
C     ************************************************************************
C
C     REVERSED BRAYTON CYCLE WITH REGENERATION:
C
```

```
      ELSE IF(NCYCLE .EQ. 3) THEN
C
      WRITE(6,48)'YOU HAVE CHOSEN THE REVERSED BRAYTON CYCLE WITH'
      WRITE(6,113)'REGENERATION.'
113   FORMAT(15X,A13,/)
C
C     ***********************************************************************
C
C     ASSIGN VALUES TO THE PRESSURE RATIO, COMPRESSOR EFFICIENCY, AND
C     EXPANDER EFFICIENCY:
C
      WRITE(6,47)'THIS PROGRAM USES A PRESSURE RATIO WHICH CAN BE'
      WRITE(6,47)'CHANGED BY THE USER.  THE DEFAULT VALUE IS 1.5.'
      WRITE(6,51)'IT IS RECOMMENDED THAT THE PRESS. RATIO NOT EXCEED'
      WRITE(6,49)'4.0 IN THIS PROGRAM DO TO LOW TEMP. THERMODYNAMIC'
      WRITE(6,321)'PROPERTY LIMITATIONS.'
      WRITE(6,649)'DO YOU WISH TO ACCEPT THE PRESSURE RATIO, Y OR N?'
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
C
C     ************************************************************************
C
1004  READ(6,519) CHOICE3
      WRITE(6,500) JJ
      IF ((CHOICE3 .EQ. 'Y').OR.(CHOICE3 .EQ. 'y')) THEN
C
            PRATIO = 1.5
            CONTINUE
C
      ELSEIF ((CHOICE3 .EQ. 'N').OR.(CHOICE3 .EQ. 'n')) THEN
            WRITE(6,28) 'ENTER THE NEW PRESSURE RATIO'
            READ(6,*) PRATION
            WRITE(6,500) JJ
            WRITE(6,36) 'YOU HAVE CHOSEN A NEW PRESSURE RATIO'
            WRITE(6,543) 'OF: ',PRATION,' '
            PRATIO = PRATION
C
      ELSE
C
            WRITE(6,752)'TYPE Y OR N AND PRESS RETURN'
            GO TO 1004
C
      ENDIF
C
C     ***********************************************************************
C
      WRITE(6,46)'ISENTROPIC EFFICIENCIES FOR THE COMPRESSOR AND'
      WRITE(6,46)'EXPANDER ARE USED TO ACCOUNT FOR THE IRREVERS-'
      WRITE(6,49)'IBILITIES PRESENT IN THESE COMPONENTS.  PRESENTLY'
```

```
      WRITE(6,51)'THE DEFAULT VALUES ARE 0.85 FOR BOTH THE COMPRESSOR'
      WRITE(6,313)'AND EXPANDER.'
      WRITE(6,343)'DO YOU WISH TO ACCEPT THESE VALUES, Y OR N?'
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
C
C
1005  READ(6,519) CHOICE4
      WRITE(6,500) JJ
      IF ((CHOICE4 .EQ. 'Y').OR.(CHOICE4 .EQ. 'y')) THEN
C
            ETAE = 0.85
            ETAC = 0.85
C
      ELSEIF ((CHOICE4 .EQ. 'N').OR.(CHOICE4 .EQ. 'n')) THEN
C
            WRITE(6,42) 'ENTER THE COMPRESSOR ISENTROPIC EFFICIENCY'
            READ(6,*) ETACN
            WRITE(6,43) 'YOU HAVE CHOSEN A NEW COMPRESSOR EFFICIENCY'
            WRITE(6,543) 'OF: ',ETACN,' '
C
            WRITE(6,43) 'ENTER THE EXPANDER ISENTROPIC EFFICIENCY'
            READ(6,*) ETAEN
            WRITE(6,41)'YOU HAVE CHOSEN A NEW EXPANDER EFFICIENCY'
            WRITE(6,543) 'OF: ',ETAEN,' '
            ETAC = ETACN
            ETAE = ETAEN
            CONTINUE
C
      ELSE
            WRITE(6,752)'TYPE Y OR N AND PRESS RETURN'
            GO TO 1005
      ENDIF
C
C     ************************************************************************
C
      WRITE(6,45)'THIS PROGRAM USES A REGENERATOR EFFECTIVENESS'
      WRITE(6,52)'TO ACCOUNT FOR IRREVERSIBILITIES IN THE REGENERATOR.'
52    FORMAT(15X,A52)
      WRITE(6,138)'THIS VALUE CAN BE CHANGED BY THE USER.'
138   FORMAT(15X,A38,/)
      WRITE(6,226)'THE DEFAULT VALUE IS 0.88.'
226   FORMAT(15X,A26,//)
      WRITE(6,641)'DO YOU WISH TO ACCEPT THIS VALUE, Y OR N?'
641   FORMAT(15X,A41,//////)
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
C
C     CLEAR THE SCREEN:
C
```

```
C     ***********************************************************************
C
1006  READ(6,519) CHOICE5
      WRITE(6,500) JJ
c
      IF ((CHOICE5 .EQ. 'Y').OR.(CHOICE5 .EQ. 'y')) THEN
C
            ETAR = 0.88
            CONTINUE
C
      ELSEIF ((CHOICE5 .EQ. 'N').OR.(CHOICE5 .EQ. 'n')) THEN
            WRITE(6,35)'ENTER THE REGENERATOR EFFECTIVENESS'
35       FORMAT (15X,A35)
            READ(6,*) ETARN
       WRITE(6,48) 'YOU HAVE CHOSEN A NEW REGENERATOR EFFECTIVENESS'
            WRITE(6,543) 'OF: ',ETARN,' '
            ETAR = ETARN
C
      ELSE
            WRITE(6,752)'TYPE Y OR N AND PRESS RETURN'
            GO TO 1006
      ENDIF
C
C     ***********************************************************************
C
C     WRITE THE REVERSED BRAYTON CYCLE WITH REGENERATION RESULTS:
C
      WRITE(11,620)'REVERSED BRAYTON CYCLE WITH REGENERATION RESULTS'
620   FORMAT(16X,A49,/)
      WRITE(11,690)'PRESSURE RATIO = ',PRATIO
      WRITE(11,690)'COMP. EFF.= ',ETAC
      WRITE(11,690)'EXPANDER EFF. = ',ETAE
      WRITE(11,690)'REGEN. EFF. = ',ETAR
690   FORMAT(12X,A17,F6.3,/)
      WRITE(11,899)'SOURCE TEMP.','CARNOT COP','COP','COP/COPC',
     &      'COMMENTS'
C
      WRITE(6,821)'EXECUTING REVERSED BRAYTON CYCLE WITH REGEN. MODEL'
821   FORMAT(15X,A50,/)
      WRITE(6,590)'PRESSURE RATIO =',PRATIO
      WRITE(6,590)'COMP. EFF.=',ETAC
      WRITE(6,590)'EXPANDER EFF. =',ETAE
      WRITE(6,690)'REGEN. EFF. = ',ETAR
C
      GO TO 3333
C
C     ***********************************************************************
C
```

```
      ELSE IF(NCYCLE .EQ. 4) THEN
C
      WRITE(6,46)'YOU HAVE CHOSEN THE THERMOELECTRIC CYCLE.  THE'
      WRITE(6,46)'MODEL CONTAINS A FIGURE OF MERIT PARAMETER (Z)'
      WRITE(6,48)'WHICH CAN BE VARIED BY THE USER.  THIS PARAMETER'
      WRITE(6,49)'IS A FUNCTION OF THE SEMI-CONDUCTOR MATERIAL PAIR'
      WRITE(6,46)'USED IN THE SYSTEM. THE DEFAULT VALUE IS 0.003'
      WRITE(6,45)'WHICH IS CURRENTLY THE HIGHEST VALUE ACHIEVED'
      WRITE(6,115)'EXPERIMENTALLY.'
      WRITE(6,341)'HIGHER VALUES OF Z WILL INCREASE THE COP.'
341   FORMAT(15X,A41,///)
C
C     ***********************************************************************
C
C     FIGURE OF MERIT, Z:
C
      WRITE(6,646)'DO YOU WISH TO ACCEPT THE VALUE OF Z, Y OR N?'
646   FORMAT(15X,A46,//////)
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
C
C     CLEAR THE SCREEN:
C
C     ***********************************************************************

C
1007  READ(6,519) CHOICE6
      WRITE(6,500) JJ
      IF ((CHOICE6 .EQ. 'Y').OR.(CHOICE6 .EQ. 'y')) THEN
C
         Z = 0.003
         CONTINUE
C
      ELSEIF ((CHOICE6 .EQ. 'N').OR.(CHOICE6 .EQ. 'n')) THEN
C
            WRITE(6,224) 'ENTER THE NEW VALUE OF Z'
224         FORMAT (15X,A24,//)
            READ(6,*) ZN
            WRITE(6,32)'YOU HAVE CHOSEN A NEW Z VALUE OF'
32          FORMAT(15X,A32)
            WRITE(6,543)'OF: ',ZN,' '
            Z = ZN
C
      ELSE
            WRITE(6,752)'TYPE Y OR N AND PRESS RETURN'
            GO TO 1007
      ENDIF
C
C     ***********************************************************************
```

```
C
      WRITE(11,619)'THERMOELECTRIC CYCLE RESULTS'
619   FORMAT(15X,A28,//)
C
      WRITE(11,596)'FIGURE OF MERIT (Z) =',Z
596   FORMAT(12X,A22,F5.4,/)
      WRITE(11,899)'SOURCE TEMP.','CARNOT COP','COP','COP/COPC',
     &      'COMMENTS'
C
      WRITE(6,822)'EXECUTING THERMOELECTRIC CYCLE MODEL'
822   FORMAT(15X,A36,/)
      WRITE(6,690)'          Z = ',Z
      GO TO 3333
C
C     ***********************************************************************
C
C     PULSE TUBE CYCLE:
C
      ELSE IF(NCYCLE .EQ. 5) THEN
C
      WRITE(6,39)'YOU HAVE SELECTED THE PULSE TUBE CYCLE.'
      WRITE(6,44)'THE MODEL IS AN IDEALIZED STEADY STATE MODEL'
44    FORMAT(15X,A44)
      WRITE(6,46)'OPERATING WITH HELIUM AS THE WORKING MATERIAL.'
C
      WRITE(6,47)'THIS PROGRAM USES A PRESSURE RATIO WHICH CAN BE'
      WRITE(6,47)'CHANGED BY THE USER.  THE DEFAULT VALUE IS 2.5.'
      WRITE(6,*) ' '
      WRITE(6,48)'CHANGING THE PRESSURE RATIO SIMULATES INCREASING'
      WRITE(6,47)'THE PRESSURE RATIO IN A PULSE TUBE REFRIGERATOR'
      WRITE(6,47)'OR INCREASING THE AMPLITUDE OF THE DIAPHRAGM IN'
      WRITE(6,230)'A THERMOACOUSTIC REFRIGERATOR.'
230   FORMAT(15X,A30,//)
      WRITE(6,649)'DO YOU WISH TO ACCEPT THE PRESSURE RATIO, Y OR N?'
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
C
C     CLEAR THE SCREEN:
C
C     ***********************************************************************
C
1008  READ(6,519) CHOICE3
      WRITE(6,500) JJ
      IF ((CHOICE3 .EQ. 'Y').OR.(CHOICE3 .EQ. 'y')) THEN
C
            PRATIO = 2.5
            CONTINUE
C
      ELSEIF ((CHOICE3 .EQ. 'N').OR.(CHOICE3 .EQ. 'n')) THEN
```

```
C
            WRITE(6,28) 'ENTER THE NEW PRESSURE RATIO'
            READ(6,*) PRATION
            WRITE(6,500) JJ
            WRITE(6,36) 'YOU HAVE CHOSEN A NEW PRESSURE RATIO'
            WRITE(6,543) 'OF: ',PRATION,' '
            PRATIO = PRATION
C
      ELSE
C
            WRITE(6,752)'TYPE Y OR N AND PRESS RETURN'
            GO TO 1008
C
      ENDIF
C
C     ***********************************************************************
C
      WRITE(6,48)'THE ISENTROPIC EFFICIENCY OF THE COMPRESSION AND'
      WRITE(6,50)'EXPANSION PROCESSES CAN BE SPECIFIED IN THE MODEL.'
      WRITE(6,42)'THIS EFFICIENCY IS USED TO ACCOUNT FOR THE'
      WRITE(6,51)'IRREVERSIBILITIES RESULTING DURING THESE PROCESSES.'
      WRITE(6,226)'THE DEFAULT VALUE IS 0.80.'
      WRITE(6,343)'DO YOU WISH TO ACCEPT THESE VALUES, Y OR N?'
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
C
C
1011  READ(6,519) CHOICE4
      WRITE(6,500) JJ
      IF ((CHOICE4 .EQ. 'Y').OR.(CHOICE4 .EQ. 'y')) THEN
C
            ETAE = 0.80
            ETAC = 0.80
C
      ELSEIF ((CHOICE4 .EQ. 'N').OR.(CHOICE4 .EQ. 'n')) THEN
C
            WRITE(6,31) 'ENTER THE ISENTROPIC EFFICIENCY'
            READ(6,*) ETACN
            WRITE(6,29) 'YOU HAVE CHOSEN AN EFFICIENCY'
29          FORMAT(15X,A29)
            WRITE(6,543) 'OF: ',ETACN,' '
C
            ETAC = ETACN
            ETAE = ETACN
            CONTINUE
C
      ELSE
            WRITE(6,752)'TYPE Y OR N AND PRESS RETURN'
            GO TO 1011
```

```
      ENDIF
C
C     ************************************************************************
C
      WRITE(6,45)'THIS PROGRAM USES A REGENERATOR EFFECTIVENESS'
      WRITE(6,52)'TO ACCOUNT FOR IRREVERSIBILITIES IN THE REGENERATION'
      WRITE(6,51)'PROCESS OCCURRING AT THE TUBE WALL.  THIS VALUE CAN'
      WRITE(6,251)'BE CHANGED BY THE USER.  THE DEFAULT VALUE IS 0.80.'
251   FORMAT(15X,A51,//)
      WRITE(6,641)'DO YOU WISH TO ACCEPT THIS VALUE, Y OR N?'
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
C
C     CLEAR THE SCREEN:
C
C     ************************************************************************
C
1012  READ(6,519) CHOICE5
      WRITE(6,500) JJ
c
      IF ((CHOICE5 .EQ. 'Y').OR.(CHOICE5 .EQ. 'y')) THEN
C
            ETAR = 0.80
            CONTINUE
C
      ELSEIF ((CHOICE5 .EQ. 'N').OR.(CHOICE5 .EQ. 'n')) THEN
            WRITE(6,36)'ENTER THE REGENERATION EFFECTIVENESS'
            READ(6,*) ETARN
       WRITE(6,48) 'YOU HAVE CHOSEN A NEW REGENERATION EFFECTIVENESS'
            WRITE(6,543) 'OF: ',ETARN,' '
            ETAR = ETARN
C
      ELSE
            WRITE(6,752)'TYPE Y OR N AND PRESS RETURN'
            GO TO 1012
      ENDIF
C

C     ************************************************************************
C
      WRITE(11,677)'PULSE TUBE CYCLE RESULTS'
677   FORMAT(15X,A24,/)
      WRITE(11,690)'PRESSURE RATIO = ',PRATIO
      WRITE(11,690)'COMP./EXP. EFF.= ',ETAC
      WRITE(11,690)'REGEN. EFF. = ',ETAR
      WRITE(11,899)'SOURCE TEMP.','CARNOT COP','COP','COP/COPC',
     &      'COMMENTS'
      WRITE(6,823)'EXECUTING PULSE TUBE CYCLE MODEL'
823   FORMAT(15X,A32,/)
```

```
      WRITE(6,590)'PRESSURE RATIO =',PRATIO
      GO TO 3333
C
C     ***********************************************************************
C
C     MAGNETIC HEAT PUMP:
C
      ELSE IF(NCYCLE .EQ. 6) THEN
C
      WRITE(6,147)'YOU HAVE SELECTED THE MAGNETIC REFRIGERATION CYCLE.'
15    FORMAT(15X,A51,/)
      WRITE(6,44)'THE MODEL IS AN IDEALIZED STEADY STATE MODEL'
      WRITE(6,46)'OPERATING BETWEEN TWO MAGNETIC FIELD STRENGTHS'
      WRITE(6,140)'USING GADOLINIUM AS THE WORKING MATERIAL.'
140   FORMAT(15X,A40,/)
      WRITE(6,45)'THE DEFAULT FIELD STRENGTHS ARE CURRENTLY SET'
      WRITE(6,45)'AT THE MAXIMUM LIMITS - 0 AND 7 TESLAS. SEVEN'
      WRITE(6,43)'TESLAS IS PRESENTLY THE PRACTICAL LIMIT FOR'
      WRITE(6,42)'FIELD STRENGTH WITH TODAYS TECHNOLOGY.  NO'
      WRITE(6,147)'PROPERTY DATA IS AVAILABLE FOR FIELDS ABOVE 7T.'
147   FORMAT(15X,A47,/)
      WRITE(6,45)'THE USER CAN CHANGE THE FIELD STRENGTH LIMITS'
      WRITE(6,316)'BETWEEN 0 AND 7.'
316   FORMAT(15X,A16,///)
      WRITE(6,643)'DO YOU WISH TO ACCEPT THESE LIMITS, Y OR N?'
643   FORMAT(15X,A43,//////)
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
C
C     CLEAR THE SCREEN:
C
C     ***********************************************************************

C
1009  READ(6,519) CHOICE7
      WRITE(6,500) JJ
C
      IF ((CHOICE7 .EQ. 'Y').OR.(CHOICE7 .EQ. 'y')) THEN
C
         HL = 0.0
         HH = 7.0
         CONTINUE
C
      ELSEIF ((CHOICE7 .EQ. 'N').OR.(CHOICE7 .EQ. 'n')) THEN
C
      WRITE(6,44) 'ENTER THE FIELD STRENGTH VALUES, HL AND HH.'
      READ(6,*)HLN,HHN
      WRITE(6,41)'YOU HAVE CHOSEN NEW FIELD STRENGTH LIMITS'
      WRITE(6,943)'OF: ',HLN,' AND ',HHN,' TESLAS'
```

```
943   FORMAT(15X,A4,F6.2,A5,F6.2,A7)
      HL = HLN
      HH = HHN
      CONTINUE
C
      ELSE
C
            WRITE(6,752)'TYPE Y OR N AND PRESS RETURN'
            GO TO 1009
C
      ENDIF
C
C     ***********************************************************************
C
      WRITE(11,617)'MAGNETIC REFRIGERATION CYCLE RESULTS'
617   FORMAT(15X,A36,/)
      WRITE(11,454)'HIGH FIELD =',HH,'TESLAS'
454   FORMAT(12X,A12,1X,F4.2,1X,A6,/)
      WRITE(11,454)'LOW FIELD  =',HL,'TESLAS'
      WRITE(11,899)'SOURCE TEMP.','CARNOT COP','COP','COP/COPC',
     &      'COMMENTS'
C
      WRITE(6,824)'EXECUTING MAGNETIC REFRIGERATION CYCLE MODEL'
824   FORMAT(15X,A44,/)
      WRITE(6,590)'HIGH FIELD =',HH
      WRITE(6,590)'LOW FIELD =',HL
C
      GO TO 3333
C
      ENDIF
C
C     ***********************************************************************
C
C     DO LOOP TO INCREMENT SOURCE TEMPERATURES:
C
3333  TOC = -24.0                                !     CELSIUS
      DO 9999 I = 1,27

C
C     ***********************************************************************
C
C     CONVERT TO ABSOLUTE TEMPERATURES:
C
      TO = TOC + 273.15
      T1 = T1C + 273.15
C
C     ***********************************************************************
C
```

```
C     CALCULATE THE CARNOT COP:
C
      CC = T0/(T1 - T0)
      TLL = T0 - DELTL
      THH = T1 + DELTH
C
      IF(NCYCLE .EQ. 1) THEN
            WRITE(6,980)'ITERATION NUMBER',I,'OF 27'
980         FORMAT(38X,A16,1X,I2,1X,A5)
            GO TO 1000
      ELSE IF(NCYCLE .EQ. 2) THEN
            WRITE(6,980)'ITERATION NUMBER',I,'OF 27'
            GO TO 1100
      ELSE IF(NCYCLE .EQ. 3) THEN
            WRITE(6,980)'ITERATION NUMBER',I,'OF 27'
            GO TO 1200
      ELSE IF(NCYCLE .EQ. 4) THEN
            WRITE(6,980)'ITERATION NUMBER',I,'OF 27'
            GO TO 1300
      ELSE IF(NCYCLE .EQ. 5) THEN
            WRITE(6,980)'ITERATION NUMBER',I,'OF 27'
            GO TO 1400
      ELSE IF(NCYCLE .EQ. 6) THEN
            WRITE(6,980)'ITERATION NUMBER',I,'OF 27'
            GO TO 1410
      END IF
C
1000  CALL STIRLING(T0,DELTL,T1,DELTH,COP)
      GO TO 1500
C
1100  KTT2 = 1
      CALL SBRAY(TLL,THH,ETAC,ETAE,PRATIO,COP,KTT2)
C
      IF(KTT2 .EQ. 0) THEN
            COP=0
            GO TO 1500
      ELSE
            GO TO 1500
      ENDIF
C
1200  KTT2 = 1
      CALL SBRAYR(TLL,THH,ETAC,ETAE,ETAR,PRATIO,COP,KTT2)
C
      IF(KTT2 .EQ. 0) THEN
            COP=0
            GO TO 1500
      ELSE
            GO TO 1500
```

```
      ENDIF
C
1300  CALL STE (TLL,THH,Z,COP)
      GO TO 1500
1400  KTT2 = 1
      CALL SPTSUB (TLL,THH,ETAC,ETAE,ETAR,PRATIO,COP,KTT2)
C
      IF(KTT2 .EQ. 0) THEN
            COP=0
            GO TO 1500
      ELSE
            GO TO 1500
      ENDIF
C
1410  CALL SMHP (TLL,THH,HL,HH,COP)
      GO TO 1500
C
C     ***********************************************************************
C
1500  IF (COP .LT. 0.0) THEN
            WARN = 'TEMPERATURES OUT OF RANGE'
      ELSE IF ((COP .GT. CC).AND.(NCYCLE .EQ. 2)) THEN
            WARN = 'PRESSURE RATIO IS TOO LOW'
      ELSE IF ((KTT2 .EQ. 0).AND.(NCYCLE .EQ. 2)) THEN
            WARN = 'COMP. OUTLET TEMP LESS THAN TH'
      ELSE IF ((COP .GT. CC).AND.(NCYCLE .EQ. 3)) THEN
            WARN = 'PRESSURE RATIO IS TOO LOW'
      ELSE IF ((KTT2 .EQ. 0).AND.(NCYCLE .EQ. 3)) THEN
            WARN = 'COMP. OUTLET TEMP LESS THAN TH'
      ELSE IF ((COP .GT. CC).AND.(NCYCLE .EQ. 5)) THEN
            WARN = 'PRESSURE RATIO IS TOO LOW'
      ELSE IF ((KTT2 .EQ. 0).AND.(NCYCLE .EQ. 5)) THEN
            WARN = 'COMP. OUTLET TEMP LESS THAN TH'
      ELSE
            WARN = ' '
      ENDIF
C
      RATIO = COP/CC
      WRITE(11,552) TOC,CC,COP,RATIO,WARN
552   FORMAT(5X,F10.3,2X,F10.3,2X,F10.3,2X,F10.3,3X,A30)
C
      TOC = TOC + 2.0
9999  CONTINUE
      WRITE(6,215)'MODEL COMPLETED'
215   FORMAT(15X,A15,//)
      WRITE(6,750)'OUTPUT IS IN',OUTPUTFILE
750   FORMAT(15X,A12,2X,A50,///)
C
```

```
C     ************************************************************************
C
C     CHOICE TO CONTINUE PROGRAM, OR NOT:
C
      WRITE(6,632)'DO YOU WISH TO CONTINUE WITH ANOTHER CASE, Y OR N?'
632   FORMAT(15X,A50,///)
      WRITE(6,31)'MAKE SELECTION AND PRESS RETURN'
C
C     CLEAR THE SCREEN:
C
C     ************************************************************************

C
1010  READ(6,519) CHOICE8
      WRITE(6,500) JJ
C
      IF ((CHOICE8 .EQ. 'Y').OR.(CHOICE8 .EQ. 'y')) THEN
C
            GO TO 4444
C
      ELSE
C
            CONTINUE
C
      ENDIF
C
      END
```

```
      SUBROUTINE STIRLING (TL,DELTL,TH,DELTH,COP)
C
      IMPLICIT REAL*8(A-H,O-Z)
C
C     THIS SUBROUTINE ESTIMATES THE COEFFICIENT OF PERFORMANCE OF AN IDEAL
C     STIRLING REFRIGERATION CYCLE GIVEN THE ABSOLUTE TEMPERATURES OF THE
C     SOURCE AND SINK AND THE MINIMUM APPROACH TEMPERATURE.
C
C     ***********************************************************************
C
      TERM1 = (TH + DELTH)/(TL - DELTL)
      TERM2 = TERM1 - 1.0
      COP = 1/TERM2
C
      RETURN
      END


      SUBROUTINE SBRAY (TL,TH,ETAC,ETAE,PRATIO,COP,JTT2)
C
      IMPLICIT REAL*8(A-H,O-Z)
      INTEGER NOP,JTT2
C
C     ***********************************************************************
C
C                              DON GAUGER
C
C                              JUNE  1992
C
C
C     THIS SUBROUTINE IS USED TO CALCULATE THE COEFFICIENT OF
C     PERFORMANCE OF THE REVERSED BRAYTON CYCLE WITHOUT
C     REGENERATION.
C
C     AIR IS THE WORKING FLUID AND ASSUMED TO BE AN IDEAL GAS.
C
C     THE COMPRESSOR EFFICIENCY AND EXPANDER EFFICIENCY ARE SPECIFIED
C     BY THE USER IN THE MAIN PROGRAM.
C
C     THE LOW PRESSURE IS FIXED 1 ATMOSPHERE.
C
C     ALL PRESSURES ARE IN KILOPASCALS.
C
C     ALL TEMPERATURES ARE IN DEGREES KELVIN.
C
C     ********************************************************************
C
```

```
C     SET THE LOW PRESSURE:
C
      P1 = 101.325                              !KPa
C
C     ***********************************************************************
C     ACCOUNT FOR THE MINIMUM RECOVERY TEMPERATURES IN THE
C     HIGH AND LOW TEMPERATURE HEAT EXCHANGERS:
C

      T1 = TL
      T3 = TH
C
C
C     ********************************************************************
C
C     STATE ONE - COMPRESSOR INLET/LOW TEMP. HEAT EXCHANGER OUTLET:
C
      NOP = 1                   ! T1 AND P1 KNOWN
C
      CALL APROP (T1,P1,V1,U1,H1,S1,NOP)
C
C     ********************************************************************
C
C     STATE TWO - COMPRESSOR OUTLET/HIGH TEMP. HEAT EXCH. INLET (ISENTROPIC):
C
      P2 = P1 * PRATIO
C
      S2S = S1
C
      NOP = 4                   ! P2 AND S2S (= S1) KNOWN
C
      CALL APROP (T2S,P2,V2S,U2S,H2S,S2S,NOP)
C
C     ********************************************************************
C
C     STATE TWO (ACTUAL):
C
      H2 = (((H2S - H1) / ETAC) + H1)
C
      NOP = 2                   ! P2 AND H2 KNOWN
C
      CALL APROP (T2,P2,V2,U2,H2,S2,NOP)
C
      IF(T2 .LT. T3) THEN
            JTT2 = 0
            COP=0.0
            GO TO 8989
      ELSE
```

```
            JTT2 = 1
            CONTINUE
      ENDIF
C
C     ***********************************************************************
C
C     STATE THREE - HIGH TEMP. HEAT EXCH. OUTLET/ EXPANDER INLET:
C
C     T3 WAS ESTABLISHED AS TH ON LINE 60.
C
      P3 = P2
C
      NOP = 1                  ! T3 (FROM TH) AND P3=P2 KNOWN
C
      CALL APROP (T3,P3,V3,U3,H3,S3,NOP)
C
C     ***********************************************************************
C
C     STATE FOUR - EXPANDER OUTLET/ LOW TEMP. HEAT EXCH. INLET (ISENTROPIC):
C
      S4S = S3
C
      P4 = P1
C
      NOP = 4                  ! P1 AND S4S (= S3) KNOWN
C
      CALL APROP (T4S,P4,V4S,U4S,H4S,S4S,NOP)
C
C     ***********************************************************************
C
C     STATE FOUR (ACTUAL):
C
      H4 = (((H4S - H3) * ETAE) + H3)
C
      NOP = 2                  ! P4 AND H4 KNOWN
C
      CALL APROP (T4,P4,V4,U4,H4,S4,NOP)
C
C     ***********************************************************************
C
C     COEFFICIENT OF PERFORMANCE:
C
      COP = ((H1 - H4) / ((H2 - H1) - (H3 - H4)))
C
C     ***********************************************************************
C
8989  RETURN
      END
```

```
      SUBROUTINE SBRAYR (TL,TH,ETAC,ETAE,ETAR,PRATIO,COP,JTT2)
C
      IMPLICIT REAL*8(A - H,O - Z)
      INTEGER JTT2
C
C
C                              DON GAUGER
C
C                              JUNE  1992
C
C     THIS SUBROUTINE IS USED TO CALCULATE THE COEFFICIENT OF
C     PERFORMANCE OF THE CLOSED REVERSED BRAYTON CYCLE WITH RE-
C     GENERATION.
C
C     AIR IS THE WORKING FLUID AND ASSUMED TO BE AN IDEAL GAS.
C
C     THE COMPRESSOR EFFICIENCY, EXPANDER EFFICIENCY, AND REGENERATOR
C     EFFECTIVENESS ARE SPECIFIED BY THE USER.
C
C     THE LOW PRESSURE IS FIXED AT 1 ATMOSPHERE.
C
C     THE PRESSURE RATIO MUST BE SPECIFIED.
C
C     ALL PRESSURES ARE IN KILOPASCALS.
C
C     ALL TEMPERATURES ARE IN DEGREES KELVIN.
C
C     ***********************************************************************
C
C     SET THE LOW PRESSURE:
C
      PL = 101.325                                !KPa
C
      CGAS = 1
C     ***********************************************************************
C
      TB = TL
      TA = TH
C
C     ***********************************************************************
C
C     STATE A - HIGH TEMP. HEAT EXCHANGER OUTLET/ REGEN. INLET:
C
C     TA AND PA = PL * PRATIO ARE KNOWN
```

```
C
      PA = PL * PRATIO
C
      NOP = 1
C
      CALL APROP (TA,PA,VA,UA,HA,SA,NOP)
C
C     ************************************************************************
C
C     STATE B - LOW TEMP. HEAT EXCHANGER OUTLET/ REGEN. INLET:
C
C     TB AND PB = PL ARE KNOWN
C
      PB = PL
C
      NOP = 1
C
      CALL APROP (TB,PB,VB,UB,HB,SB,NOP)
C
C     ************************************************************************
C
C     STATE THREE - REGENERATOR OUTLET/ EXPANDER INLET:
C
C     THE REGENERATOR EFFECTIVENESS IS DEFINED AS:
C
C                     (H3 - HA)/(HA - HB) = ETAR
C
C
C     THE KNOWN PRESSURE IS P3 = PA.
C
      P3 = PA
C
      H3 = HA - ETAR * (HA - HB)
C
      NOP = 2
C
      CALL APROP (T3,P3,V3,U3,H3,S3,NOP)
C
C     ************************************************************************
C
C     STATE FOUR - EXPANDER OUTLET/ LOW TEMP. HEAT EXCH. INLET (ISENTROPIC):
C
      S4S = S3
C
      P4 = PB
C
      NOP = 4
C
```

```
      CALL APROP (T4S,P4,V4S,U4S,H4S,S4S,NOP)
C
C     ***********************************************************************
C
C     STATE FOUR (ACTUAL):
C
      P4 = PB
C
      H4 = ((H4S - H3) * ETAE) + H3
C
      NOP = 2
C
      CALL APROP (T4,P4,V4,U4,H4,S4,NOP)
C
C     ***********************************************************************
C
C     STATE ONE - REGENERATOR EXIT/ COMPRESSOR INLET:
C
      P1 = PB
C
      H1 = (HA - H3) + HB
C
      NOP = 2
C
      CALL APROP (T1,P1,V1,U1,H1,S1,NOP)
C
C     ***********************************************************************
C
C     STATE TWO - COMPRESSOR EXIT/ HIGH TEMP. HEAT EXCH. INLET (ISENTROPIC):
C
      P2 = PA
C
      S2S = S1
C
      NOP = 4
C
      CALL APROP (T2S,P2,V2S,U2S,H2S,S2S,NOP)
C
C     ***********************************************************************
C
C     STATE TWO (ACTUAL):
C
      P2 = PA
C
      H2 = ((H2S - H1) / ETAC) + H1
C
      NOP = 2
C
```

```
      CALL APROP (T2,P2,V2,U2,H2,S2,NOP)
C
      IF(T2 .LT. TA) THEN
            JTT2 = 0
            COP=0.0
            GO TO 8990
      ELSE
            JTT2 = 1
            CONTINUE
      ENDIF
C
C     ***********************************************************************
C
C     COEFFICIENT OF PERFORMANCE:
C
      COP = ((HB - H4) / ((H2 - H1) - (H3 - H4)))
C
8990  RETURN
      END



C
      SUBROUTINE SPTSUB (TLL,THH,ETAC,ETAE,ETAR,PRATIO,COP,JTT2)
C
      IMPLICIT REAL*8 (A - H,O - Z)
      INTEGER JTT2
C
C     THIS SUBROUTINE IS USED TO CALCULATE THE COEFFICIENT OF
C     PERFORMANCE OF THE IDEAL PULSE TUBE REFRIGERATION CYCLE.
C
C     HELIUM IS THE WORKING FLUID.
C
C     THE COMPRESSOR EFFICIENCY, AND REGENERATOR
C     EFFECTIVENESS ARE SPECIFIED.
C
C     THE LOW PRESSURE IS 1 ATMOSPHERE.
C
C     THE PRESSURE RATIO MUST BE SPECIFIED.
C
C     ALL PRESSURES ARE IN KILOPASCALS.
C
C     ALL TEMPERATURES ARE IN DEGREES KELVIN.
C
C     ***********************************************************************
C
C     SET THE LOW PRESSURE:
C
```

```
      PL = 101.325                              !KPa
C
C     ********************************************************************
C
      TB = TLL
      TA = THH
C
C     ******************************************************************
C
C     STATE A - HIGH TEMP. HEAT EXCHANGER OUTLET/ REGEN. INLET:
C
C     TA AND PA = PL * PRATIO ARE KNOWN
C
      PA = PL * PRATIO
      NOP = 1
      CALL HPROP (TA,PA,VA,UA,HA,SA,NOP)
C
C     ******************************************************************
C
C     STATE B - LOW TEMP. HEAT EXCHANGER / REGEN. INLET:
C
C     TB AND PB = PL ARE KNOWN
C
      PB = PL
C
      NOP = 1
C
      CALL HPROP (TB,PB,VB,UB,HB,SB,NOP)
C
C     ******************************************************************
C
C     STATE THREE - REGENERATOR OUTLET/ EXPANDER INLET:
C
C     THE REGENERATOR EFFECTIVENESS IS DEFINED AS:
C
C                     (H3 - HA)/(HA - HB) = ETAR
C
C
C     THE KNOWN PRESSURE IS P3 = PA.
C
      P3 = PA
      H3 = HA - ETAR * (HA - HB)
      NOP = 2
      CALL HPROP (T3,P3,V3,U3,H3,S3,NOP)
C
C     ******************************************************************
C
C     STATE FOUR - EXPANDER OUTLET/ LOW TEMP. HEAT EXCH. INLET (ISENTROPIC):
```

```
C
      S4S = S3
C
      P4 = PB
C
      NOP = 3
C
      CALL HPROP (T4S,P4,V4S,U4S,H4S,S4S,NOP)
C
C     ******************************************************************
C
C     STATE FOUR (ACTUAL):
C
      H4 = ((H4S - H3) * ETAE) + H3
C
      NOP = 2
C
C      CALL HPROP (T4,P4,V4,U4,H4,S4,NOP)
C
C     ******************************************************************
C
C     STATE ONE - REGENERATOR EXIT/ COMPRESSOR INLET:
C
      P1 = PB
C
      H1 = (H3 - HA) + HB
C
      NOP = 2
C
      CALL HPROP (T1,P1,V1,U1,H1,S1,NOP)
C
C     ******************************************************************
C
C     STATE TWO - COMPRESSOR EXIT/ HIGH TEMP. HEAT EXCH. INLET (ISENTROPIC):
C
      P2 = PA
C
      S2S = S1
C
      NOP = 3
C
      CALL HPROP (T2S,P2,V2S,U2S,H2S,S2S,NOP)
C
C     ******************************************************************
C
C     STATE TWO (ACTUAL):
C
      P2 = PA
```

```
C
      H2 = ((H2S - H1) / ETAC) + H1
C
      NOP = 2
C
      CALL HPROP(T2,P2,V2,U2,H2,S2,NOP)
C
      IF(T2 .LE. TA) THEN
            JTT2 = 0
            COP=0.0
            GO TO 8991
      ELSE
            JTT2 = 1
            CONTINUE
      ENDIF
C
C     ********************************************************************
C
C     COEFFICIENT OF PERFORMANCE:
C
      COP = (HB - H4) / (H2 - H1)
C
C     ********************************************************************
C
8991  RETURN
      END


      SUBROUTINE STE(TL,TH,Z,COPTE)
      IMPLICIT REAL*8(A-H,O-Z)
C
C     *********************************************************************
C
C     THERMOELECTRIC REFRIGERATION COEFFICIENT OF PERFORMANCE SUBROUTINE
C
C
C                              DON GAUGER
C
C                         IOWA STATE UNIVERSITY
C
C
C                              28 JULY 1992
C
C     *********************************************************************
C
C     THIS SUBROUTINE IS USED TO CALCULATE THE MAXIMUM COEFFICIENT OF
C     PERFORMANCE FOR A THERMOELECTRIC REFRIGERATION SYSTEM.
C
```

```
C     ***********************************************************************
C
C     CALCULATE THE AVERAGE TEMPERATURE, TBAR:
C
      TBAR = (TH+TL)/2.0
C
C     ***********************************************************************
C
C     CALCULATE THE CARNOT COP:
C
      COPC = TL/(TH - TL)
C
C     ***********************************************************************
C
C     CALCULATE THE MAXIMUM IDEAL COP FOR THE THERMOELECTRIC REFRIGERATOR:
C
      XX = (1.0 + (Z*TBAR))**0.5
C
      TERMA = (XX - (TH/TL))
      TERMB = (XX + 1.0)
      TERMC = TERMA/TERMB
C
      COPTE = COPC*TERMC
C
      RETURN
      END


      SUBROUTINE SMHP(TL,TH,HL,HH,COP)
      IMPLICIT REAL*8(A-H,O-Z)
C
C     ***********************************************************************
C
C
C                              DON GAUGER
C
C                         IOWA STATE UNIVERSITY
C
C                              DECEMBER 1992
C
C     ***********************************************************************
C
C     THIS SUBROUTINE IS INTENDED TO CALCULATE THE COP OF AN IDEALIZED
C     MAGNETIC HEAT PUMP CYCLE OPERATING AT STEADY STATE IN A CONSTANT
C     FIELD CYCLE (TWO ISOFIELD AND TWO ISOTHERMAL PROCESSES). THE
C     MAGNETIC SOLID IS GADOLINIUM.  THE CONSTANT FIELD STRENGTHS ARE
C     BETWEEN 0AND 7 TESLAS.  THE SOURCE AND SINK TEMPERATURES ARE
```

```
C     BETWEEN 280 AND 320 K.
C
C     ***********************************************************************
C
C     READ THE FIELD STRENGTHS:
C
      HL = 0.
      HH = 7.
C
C     ***********************************************************************
C
C
C     CALCULATE AREA THREE:
C
      CALL GD(HL,TH,S2)
      CALL GD(HH,TH,S3)
      A3 = (S2-S3)*TH                          !     THIRD AREA ON TS DIAGRAM
C
C     ***********************************************************************
C
C     CALCULATE DELTA T:
C
      N = 50
      DELT = (TH-TL)/N
C
C     ***********************************************************************
C
C     CALCULATE THE FOURTH AREA:
C
      A = 0.0
      AT = 0.0
      SL = 0.0
      SR = 0.0
      TC = TL
      HC = HL
      CALL GD(HC,TC,SL)
C
      DO 1000 I = 1,N
            TC = TC + DELT
            CALL GD(HC,TC,SR)
            TMP = TC - DELT/2.0
            A = (SR - SL)*TMP
            AT = AT + A
            SL = SR
1000  CONTINUE
C
      A4 = AT
C
```

```
C     ***********************************************************************
C
C     CALCULATE THE ENTROPY AT STATE ONE:
C
C     THE FIRST AREA MUST EQUAL THE FOURTH AREA
C
      A1 = 0.0
      A = 0.0
      SL = 0.0
      SR = S3
      TC = TH
      HC = HH
      DELTS = 1.0E-01
C
999   DIFF = (A4 - A1)
C
      IF (DIFF .GT. 1.0E-04) THEN
            TC = TC - DELTS
            CALL GD(HC,TC,SL)
            TMP = TC + DELTS/2.0
            A = (SR - SL)*TMP
            A1 = A1 + A
            SR = SL
            GO TO 999
      ELSE
            CONTINUE
      ENDIF
C
      S4 = SL
C
C     ***********************************************************************
C
C     CALCULATE THE HEAT ACCEPTED FROM THE SINK:
C
C
      CALL GD(HL,TL,S1)
      QC = (S1-S4)*TL                          !     J/KG.
      A2 = QC                              !     AREA TWO ON TS DIAGRAM
C
C     ***********************************************************************

C     CALCULATE THE WORK:
C
C
      WORK = A1 + A3 - (A2 + A4)
C
C     ***********************************************************************
C
```

```
C     CALCULATE THE COEFFICIENT OF PERFORMANCE:
C
      COP = QC/WORK
      RETURN
      END
```

```
      SUBROUTINE APROP(TS,PS,VS,US,HS,SS,NOP)
      IMPLICIT REAL*8(A-H,O-Z)
      INTEGER NOP,KTR
C
C     ******************************************************************
C
C     THIS SUBROUTINE CALCULATES THE THERMODYNAMIC PROPERTIES
C     OF GASES USING THE IDEAL GAS EQUATION OF STATE. THE CONSTANT
C     PRESSURE SPECIFIC HEAT FUNCTION WAS DERIVED FROM DATA FROM:
C     REYNOLDS, W.,C., THERMODYNAMIC PROPERTIES IN SI, DEPARTMENT
C     OF MECHANICAL ENGINEERING STANFORD UNIVERSITY, STANFORD, CA.
C
C
C                           DON GAUGER
C
C                            JUNE 1992
C
C     THE REFERENCE STATE IS ONE ATMOSPHERE AND ZERO DEGREES KELVIN
C
C     THE UNITS ARE AS FOLLOWS:
C
C     TEMPERATURE       DEGREES KELVIN
C     PRESSURE          KPa
C     SPECIFIC VOLUME   M^3/KILOGRAM
C     INTERNAL ENERGY   KILOJOULES/KILOGRAM
C     ENTHALPY          KILOJOULES/KILOGRAM
C     ENTROPY           KILOJOULES/KILOGRAM*K.
C
C     ******************************************************************
C
C     SELECT OPTIONS:
C
      IF (NOP .EQ. 1) THEN                  ! P AND T KNOWN
        GO TO 30
      ELSE IF (NOP .EQ. 2) THEN             ! H AND P KNOWN
        GO TO 20
      ELSE IF (NOP .EQ. 3) THEN             ! U AND P KNOWN
        GO TO 10
      ELSE IF (NOP .EQ. 4) THEN             ! S AND P KNOWN
        CONTINUE
      ENDIF
C
C     ******************************************************************
C
C     ROUTINE TO ITERATE AND FIND T, V, U, AND H KNOWING S AND P:
C
      PT = PS
      KTR = 0
```

```
      TTL = 150.0
C
      DELT = 20.0
C
      CALL AIR (TTL,PT,VT,UT,HT,ST)
C
11    DELS =  SS - ST
      KTR = KTR + 1
C
      IF (ABS(DELS) .LT. 1.5E-04) THEN
            TS = TTL
            GO TO 30
      ELSE IF (DELS .GT. 0.0) THEN
            TTL = TTL + DELT
            CALL AIR (TTL,PT,VT,UT,HT,ST)
            GO TO 11
      ELSE IF (DELS .LT. 0.0) THEN
            DELT = DELT/2.0
            TTL = TTL - DELT
            CALL AIR (TTL,PT,VT,UT,HT,ST)
            GO TO 11
            CONTINUE
      END IF
C
C     ******************************************************************
C
C     ROUTINE TO ITERATE AND FIND T, V, U, AND S KNOWING H AND P:
C
20    PT = PS
      KTR = 0
      TTL = 150.0
C
      DELT = 20.0
C
      CALL AIR (TTL,PT,VT,UT,HT,ST)
C
15    DELH =  HS - HT
      KTR = KTR + 1
C
      IF (ABS(DELH) .LT. 1.5E-03) THEN
            TS = TTL
            GO TO 30
      ELSE IF (DELH .GT. 0.0) THEN
            TTL = TTL + DELT
            CALL AIR (TTL,PT,VT,UT,HT,ST)
            GO TO 15
      ELSE IF (DELH .LT. 0.0) THEN
            DELT = DELT/2.0
```

```
            TTL = TTL - DELT
            CALL AIR (TTL,PT,VT,UT,HT,ST)
            GO TO 15
            CONTINUE
      END IF
C
C     ******************************************************************
C
C     ROUTINE TO ITERATE AND FIND T, V, H, AND S KNOWING U AND P:
C
10    PT = PS
      KTR = 0
      TTL = 150.0
C
      DELT = 20.0
C
      CALL AIR (TTL,PT,VT,UT,HT,ST)
C
25    DELU =  US - UT
      KTR = KTR + 1
C
      IF (ABS(DELU) .LT. 1.5E-03) THEN
            TS = TTL
            GO TO 30
      ELSE IF (DELU .GT. 0.0) THEN
            TTL = TTL + DELT
            CALL AIR (TTL,PT,VT,UT,HT,ST)
            GO TO 25
      ELSE IF (DELU .LT. 0.0) THEN
            DELT = DELT/2.0
            TTL = TTL - DELT
            CALL AIR (TTL,PT,VT,UT,HT,ST)
            GO TO 25
            CONTINUE
      END IF
C
C     ******************************************************************
C
C     ROUTINE TO FIND V, U, H, AND S KNOWING T AND P:
C
30    CALL AIR (TS,PS,VS,US,HS,SS)
C
      RETURN
      END


      SUBROUTINE AIR (XT,XP,XV,XU,XH,XS)
C
```

```
      IMPLICIT REAL*8(A-H,O-Z)
C
C     THIS SUBROUTINE CALCULATES THE THERMODYNAMIC PROPERTIES
C     OF AIR USING THE IDEAL GAS EQUATION OF STATE.
C
C     THE REFERENCE STATE IS ONE ATMOSPHERE AND ZERO DEGREES KELVIN.
C
C     THE UNITS ARE AS FOLLOWS:
C
C     TEMPERATURE       DEGREES KELVIN
C     PRESSURE          KILOPASCALS
C     SPECIFIC VOLUME   M^3/KILOGRAM
C     INTERNAL ENERGY   KILOJOULES/KILOGRAM
C     ENTHALPY          KILOJOULES/KILOGRAM
C     ENTROPY           KILOJOULES/KILOGRAM*K.
C
C
      PO = 101.325                              ! KPa
C
C     ******************************************************************
C
C     UNIVERSAL GAS CONSTANT:
C
      RU = 8.314                                ! KJ/KMOL*K.
C
C     ******************************************************************
C
C     CONSTANTS FOR AIR:
C
      WTM = 28.97                               ! KG/KMOL
      ALPHA = 0.101630E01
      BETA = -0.137524E-03
      GAMMA = 0.277805E-06
      DELTA = -0.203442E-09
      EPSILON = -0.221745E-12
C
C     ******************************************************************
C
      R = RU/WTM                                ! KJ/KG*K.
C
C     CALCULATE THE SPECIFIC VOLUME:
C
      XV = (R*XT)/XP
C
C     ******************************************************************
C
C     EVALUATE THE ENTHALPY:
C
```

```
      IF (XT .LT. 150.) THEN
C
      WRITE(6,*) 'T4 MUST BE GREATER THAN 150 K., TRY A LOWER PRATIO',XT
            STOP
      ELSE
         CONTINUE
      ENDIF
C
      XH = ((ALPHA*XT) + (BETA/2.)*(XT**2) + (GAMMA/3.)
     &     *(XT**3) +(DELTA/4.)*(XT**4) +(EPSILON/5.)
     &     *(XT**5))
C
C     ********************************************************************
C
C     EVALUATE THE ENTROPY:
C
      XS =  (ALPHA*LOG(XT) + BETA*XT + GAMMA/2.*(XT**2) +
     &      (DELTA/3.)*(XT**3) + (EPSILON/4)*(XT**4)
     &     - R*LOG(XP/PO))
C
C     ********************************************************************
C
C     EVALUATE THE INTERNAL ENERGY:
C
      XU = XH - R*XT
C
      RETURN
      END


      SUBROUTINE HPROP (TS,PS,VS,US,HS,SS,NOP)
C
      IMPLICIT REAL*8 (A-H,O-Z)
C
C     ********************************************************************
C
C     THIS SUBROUTINE CALCULATES THE THERMODYNAMIC PROPERTIES
C     OF GASES USING THE IDEAL GAS EQUATION OF STATE. THE CONSTANT
C     PRESSURE SPECIFIC HEAT CONSTANT IS FROM:
C     REYNOLDS, W.,C., THERMODYNAMIC PROPERTIES IN SI, DEPARTMENT
C     OF MECHANICAL ENGINEERING STANFORD UNIVERSITY, STANFORD, CA.
C
C
C                              DON GAUGER
C
C                               JUNE 1992
C
C     THE REFERENCE STATE IS ONE ATMOSPHERE AND ZERO DEGREES KELVIN
```

```
C
C     THE UNITS ARE AS FOLLOWS:
C
C     TEMPERATURE        DEGREES KELVIN
C     PRESSURE           KPa
C     SPECIFIC VOLUME    M^3/KILOGRAM
C     INTERNAL ENERGY    KILOJOULES/KILOGRAM
C     ENTHALPY           KILOJOULES/KILOGRAM
C     ENTROPY            KILOJOULES/KILOGRAM*K.
C
C     ***********************************************************************
C
C     SELECT OPTIONS:
C
      IF (NOP .EQ. 1) THEN                    ! P AND T KNOWN
        GO TO 30
      ELSE IF (NOP .EQ. 2) THEN               ! H AND P KNOWN
        GO TO 20
      ELSE IF (NOP .EQ. 3) THEN               ! S AND P KNOWN
        CONTINUE
      ENDIF
C
C     ***********************************************************************
C
C     ROUTINE TO ITERATE AND FIND T, V, U, AND H KNOWING S AND P:
C
      PT = PS
      KTR = 0
      TTL = 1.0
C
      DELT = 20.0
C
      CALL HELIUM(TTL,PT,VT,UT,HT,ST)
C
10    DELS =  SS - ST
      KTR = KTR + 1
C
      IF (ABS(DELS) .LT. 1.5E-04) THEN
            TS = TTL
            GO TO 30
      ELSE IF (DELS .GT. 0.0) THEN
            TTL = TTL + DELT
            CALL HELIUM(TTL,PT,VT,UT,HT,ST)
            GO TO 10
      ELSE IF (DELS .LT. 0.0) THEN
            DELT = DELT/2.0
            TTL = TTL - DELT
            CALL HELIUM(TTL,PT,VT,UT,HT,ST)
```

```
            GO TO 10
            CONTINUE
      END IF
C
C     ******************************************************************
C
C     ROUTINE TO ITERATE AND FIND T, V, U, AND S KNOWING H AND P:
C
20    PT = PS
      KTR = 0
      TTL = 1.0
C
      DELT = 20.0
C
      CALL HELIUM(TTL,PT,VT,UT,HT,ST)
C
15    DELH =  HS - HT
C
      KTR = KTR + 1
C
      IF (ABS(DELH) .LT. 1.5E-04) THEN
            TS = TTL
            GO TO 30
      ELSE IF (DELH .GT. 0.0) THEN
            TTL = TTL + DELT
            CALL HELIUM(TTL,PT,VT,UT,HT,ST)
            GO TO 15
      ELSE IF (DELH .LE. 0.0) THEN
            DELT = DELT/2.0
            TTL = TTL - DELT
            CALL HELIUM(TTL,PT,VT,UT,HT,ST)
            GO TO 15
            CONTINUE
      END IF
C
C     ******************************************************************
C
C     ROUTINE TO FIND V, U, H, AND S KNOWING T AND P:
C
30    CALL HELIUM(TS,PS,VS,US,HS,SS)
C
      END


      SUBROUTINE HELIUM (THE,PHE,VHE,UHE,HHE,SHE)
C
      IMPLICIT REAL*8 (A-H,O-Z)
```

```
C
C
C     THIS SUBROUTINE CALCULATES THE THERMODYNAMIC PROPERTIES
C     OF HELIUM USING THE IDEAL GAS EQUATION OF STATE.
C
C     THE REFERENCE STATE IS ONE ATMOSPHERE AND ZERO DEGREES KELVIN.
C
C     THE UNITS ARE AS FOLLOWS:
C
C     TEMPERATURE        DEGREES KELVIN
C     PRESSURE           KILOPASCALS
C     SPECIFIC VOLUME    M^3/KILOGRAM
C     INTERNAL ENERGY    KILOJOULES/KILOGRAM
C     ENTHALPY           KILOJOULES/KILOGRAM
C     ENTROPY            KILOJOULES/KILOGRAM*K.
C
C     ********************************************************************
C
C     UNIVERSAL GAS CONSTANT:
C
      RU = 8.314                                ! KJ/KMOL*K.
      PO = 101.325                              ! KPa
C
C     ********************************************************************
C
C     CONSTANTS FOR HELIUM:
C
      WTM = 4.0026                              ! KG/KMOL
      ALPHA = 5.2328746                         ! KJ/(KG-MOL)(K.)
C
C     ********************************************************************
C
      R = RU/WTM                                ! KJ/KG*K.
C
C     CALCULATE THE SPECIFIC VOLUME:
C
      VHE = (R*THE)/PHE
C
C     ********************************************************************
C
C     EVALUATE THE ENTHALPY:
C
C
      HHE = (ALPHA*THE)
C
C     ********************************************************************
C
C     EVALUATE THE ENTROPY:
```

```
C
      RS1 = (ALPHA*LOG(THE))
      RS2 = PHE/PO
C
      IF (RS2 .LE. 1E-06) THEN
        RS2 = 1.0
        ELSE
      CONTINUE
      ENDIF
C
      RS3 = (R*LOG(RS2))
C
      SHE =  RS1 - RS3
C
C     ********************************************************************
C
C     EVALUATE THE INTERNAL ENERGY:
C
      UHE = HHE - R*THE
C
      RETURN
      END


      SUBROUTINE GD(YH,TT,SS)
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION TERM(9)
C
C     ********************************************************************
C
C     GADOLINIUM ENTROPY ROUTINE FOR USE WITH MAGNETIC HEAT PUMP MODEL
C
C
C                              DON GAUGER
C
C                         IOWA STATE UNIVERSITY
C
C
C                              16 DECEMBER 1992
C
C     ********************************************************************
C
C     THIS SUBROUTINE IS USED TO CALCULATE THE ENTROPY OF GADOLINIUM AS A
C     FUNCTION OF ABSOLUTE TEMPERATURE AND MAGNETIC FIELD STRENGTH.  THE
C     DATA USED FOR THE CURVE FIT WERE TAKEN FROM: CHEN,F.C.,ET AL.,"LOSS
C     ANALYSIS OF THE THERMODYNAMIC CYCLE OF MAGNETIC HEAT PUMPS", U.S. DEPT.
C     OF ENERGY REPORT ORNL/TM--11608, FEBRUARY 1991, FIGURE 2.2, PAGE 39.
```

```
C
C     ss = J/KG-K.
C     TT = K.
C     YH = TESLAS
C
C     ************************************************************************
C
C     SCALE VARIABLES USING SCALING FACTORS FROM CURVE FIT:

      YO = -0.77777779E+00
      RY = .77777778E+01
      TO = .25333333E+03
      RT = .66666667E+02
C
      YYH = ((YH-YO)/RY)
      TTT = ((TT-TO)/RT)
C
C     ************************************************************************
C
C     ASSEMBLE TERMS:
C
      TERM(1) = 3.862186E-04
      TERM(2) = (8.144588E-05)*TTT
      TERM(3) = (-2.630953E-05)*(TTT**2)
      TERM(4) = (-1.408800E-05)*YYH
      TERM(5) = (-6.804424E-06)*(YYH*TTT)
      TERM(6) = (2.192644E-05)*(TTT**2)*YYH
      TERM(7) = (4.165092E-06)*(YYH**2)
      TERM(8) = (-2.592406E-06)*(TTT*(YYH**2))
      TERM(9) = (-9.646256E-06)*(YYH**2)*(TTT**2)
C
C     ************************************************************************
C
C     CALCULATE SS:
C
      SS= 0.0
C
      DO 40 I=1,9
           SS = SS + TERM(I)
   40 CONTINUE
C
      RETURN
      END
```

APPENDIX C. SAMPLE DATA FROM ALTERNATIVE REFRIGERATION TECHNOLOGY CYCLE PROGRAM

SINK TEMPERATURE =35. CELSIUS

HIGH TEMP HX DELTA T = 5. CELSIUS

LOW TEMP HX DELTA T = 5. CELSIUS

REVERSED BRAYTON CYCLE RESULTS

PRESSURE RATIO = 2.500

COMP. EFF.= .850

EXPANDER EFF. = .850

SOURCE TEMP.	CARNOT COP	COP	COP/COPC	COMMENTS
-24.000	4.223	-.284	-.067	TEMPERATURES OUT OF RANGE
-22.000	4.406	-.199	-.045	TEMPERATURES OUT OF RANGE
-20.000	4.603	-.118	-.026	TEMPERATURES OUT OF RANGE
-18.000	4.814	-.042	-.009	TEMPERATURES OUT OF RANGE
-16.000	5.042	.031	.006	
-14.000	5.289	.100	.019	
-12.000	5.556	.165	.030	
-10.000	5.848	.228	.039	
-8.000	6.166	.287	.047	
-6.000	6.516	.345	.053	
-4.000	6.901	.398	.058	
-2.000	7.328	.451	.062	
.000	7.804	.500	.064	
2.000	8.338	.549	.066	
4.000	8.940	.594	.066	
6.000	9.626	.638	.066	
8.000	10.413	.680	.065	
10.000	11.326	.721	.064	
12.000	12.398	.760	.061	
14.000	13.674	.798	.058	
16.000	15.218	.833	.055	
18.000	17.126	.868	.051	
20.000	19.543	.902	.046	
22.000	22.704	.936	.041	
24.000	27.014	.966	.036	
26.000	33.239	.996	.030	

28.000	43.021	1.026	.024

reference

APPENDIX D. ALTERNATIVE REFRIGERATION CYCLE TECHNICAL ASSESSMENT PROGRAM

Introduction

The objective of this program is to compare refrigeration and air conditioning technologies on the basis of the technical assessment criteria established for this project. The program uses a three dimensional array which contains all of the ratings for the alternative refrigeration technologies discussed in Chapter 10.

The program was developed to estimate the coefficient of performance of the following refrigeration cycles:

1. Reversed Stirling.
2. Reversed Brayton.
3. Thermoelectric.
4. Pulse tube and thermoacoustic.
5. Magnetic.
6. Liquid absorption.
7. Solid sorption.

8. Vapor compression.

This program was written in FORTRAN. The source code can be compiled and used on any system having a FORTRAN compiler. The executable version we have furnished can be installed and run on IBM or IBM compatible personal computers.

The program is structured in an easy to use, interactive, menu driven format. The user is asked to supply information in a step by step process. The user first asked to select the refrigeration application they wish to consider. On subsequent screen, the user selects which of the technical assessment criteria they wish to use to assess the technologies. The user is then asked to weight the criteria in terms of relative importance. A default weighting of equal importance can be selected. Based upon the criteria which have been selected, the weighting, and the ratings established during the technical assessment, the program ranks the technologies from high to low in order of the calculated rating.

Validation of the Program

The program was validated by comparing the results with hand calculations.

Program Structure

The program source code is contained in a single file, TEKA.FOR.

System Requirements

This program was written in FORTRAN code which is compatible with MICROSOFT FORTRAN version 5.0. The executable version of the program has no

special requirement as to micro-processor type; it can be run on computers using the 8086 through 80486 processors.

One feature of MICROSOFT FORTRAN which must be kept in mind when using this program is the choice of linking library options which are used to from the executable file during the compiling and linking process. MICROSOFT has developed separate libraries which are selected during the installation of their FORTRAN software. For computers equipped with the 8087, 80287, or 80387 math co-processor the library LLIBFOR007 is used. Since the math co-processor is incorporated on all 80486 chips, this library is utilized for these machines , as well. For computers using the 8086, 80286, and 80386 micro-processor without the 8087, 80287, or 80387 math co-processor, the emulator library LLIBFORE is used. Therefore, if the program is linked using the LLIBFOR007 library to form the executable file, it will not run on a computer that does not have a math co-processor.

Program Installation

The program includes some screen clearing commands during execution. A line must be included in the computer's CONFIG.SYS file which reads exactly as follows:

DEVICE=C:\DOS\ANSI.SYS

If this line is not included, the code "2J]" will appear in the upper left corner of the monitor screen; however, the program can still be run and will provide correct results.

To install the program:

1. Choose or create a suitable directory on the hard disk.

2. Insert the diskette in the A drive and choose the directory entitled 1FOR.

3. Type the command:

COPY 1.EXE C:\(directory name)\TEKA.EXE.

Running the Program

To start the program, type "TEKA" and press return. Each screen is self explanatory and prompts the user for the required input action (such as pressing return to refresh a screen), numerical input value, or choice (yes or no). The user is also prompted to furnish an output file name for the file to which the output data will be written.

At the end of a program sequence the user can choose to either start a new sequence or to exit the program by answering "Y" or "N" to the question appearing on the screen.

The data from each run will be found in the data file named during the run sequence. Each new case must have a unique file name. If the same file name is given, the data from the previous run will be overwritten. It is suggested that the file name be appended with a letter or number to indicate the order of the run. For example, the file names TEKA1.DAT, TEKA2.DAT, and TEKA3.DAT could be used for the data files for the first, second, and third runs used to consider different cases.

Changing the Technical Assessment Ratings

Program currently used the ratings for each technical assessment criteria and application which were given in the tables at the beginning of each technical assessment section in Chapter 10 . If the user wishes to change the theses ratings, it must be done by altering the FORTRAN code. The technical assessment ratings are located in a three dimensional array (A(I,J,K)). Each element has been individually assigned a value (rather than using DATA statements). The method of entering the data as individual array elements was chosen to simplify the process of changing the ratings. Comment statements at the beginning of the code clearly identify how the array element values are arranged and assigned.

The code can then be re-compiled to create a new executable version with the new ratings.

```
C     ***********************************************************************
C
C           ALTERNATIVE REFRIGERATION CYCLE TECHNICAL ASSESSMENT ROUTINE
C
C
C                                 DON  C.  GAUGER
C
C                         MECHANICAL ENGINEERING DEPARTMENT
C
C                               IOWA STATE UNIVERSITY
C
C                                      1993
C
C     ***********************************************************************
C
C     THIS ROUTINE COMPARES SIX KEY TECHNICAL ASSESSMENT CRITERIA FOR
C     ALTERNATIVE REFRIGERATION CYCLES USING A WEIGHTING SYSTEM.
C
C     THE FORMAT IS MENU DRIVEN.
C
C     THE USER IS ASKED TO MAKE CHOICES REGARDING THE TYPE OF REFRIGERATION
C     APPLICATION TO BE CONSIDERED, WHICH TECHNICAL ASSESSMENT CRITERIA ARE
C     TO BE CONSIDERED, AND THE WEIGHTING EACH CRITERIA IS TO BE GIVEN.
C
C     THE PROGRAM WILL RANK THE ALTERNATIVE TECHNOLOGIES FROM BEST TO WORST
C     BASED UPON THIS INFORMATION.
C
C     ***********************************************************************
C
C     VARIABLE DECLARATION:
C
      REAL WC(6),TOTAL,DIFF,A(6,5,10),RATE(10),RMAX,DWF
      INTEGER I,J,K,L,N
      CHARACTER*50 OUTPUTFILE
      CHARACTER*1 CHOICE1,CHOICE2,CHOICE4,CHOICE5,CHOICE6,CHOICE7,
     &            CHOICE8,CHOICE9,CHOICE10,CHOICE11
      CHARACTER*2 CHOICE3
      CHARACTER*10 C2,C3,C5
      CHARACTER*11 C6,C7,C8
      CHARACTER*16 C1,C4,C9
C
C     ***********************************************************************
C
C     INTEGER INDICATING THE NUMBER OF TECHNOLOGIES:
C
      N = 8                   !  IF TECHNOLOGIES ARE ADDED, INCREASE THIS
C                                 NUMBER ACCORDINGLY.
C
```

```
C     ************************************************************************
C
C     INPUT THE DATA FOR THE TECHNICAL ASSESSMENT ARRAYS:
C
C     ARRAY FORMAT:
C
C     THE TECH. ASSESSMENT ARRAYS ARE THREE DIMENSIONAL, 6 BY 5 BY "N" IN SIZE.
C     THE ROWS ARE THE CRITERIA, THE COLUMNS ARE THE APPLICATIONS, THE
C     RANKS ARE THE DIFFERENT TECHNOLOGIES.
C
C     DEFINITION OF ARRAY ELEMENTS:
C
C     ROWS (ARRAY "I" TERM):   TECH
C
C     ROW 1 = STATE OF THE ART.
C     ROW 2 = COMPLEXITY.
C     ROW 3 = SIZE/WEIGHT.
C     ROW 4 = MAINTENANCE.
C     ROW 5 = USEFUL LIFE.
C     ROW 6 = CYCLE EFFICIENCY.
C
C
C     COLUMNS (ARRAY "J" TERM):
C
C     COLUMN 1 = DOMESTIC AIR CONDITIONING.
C     COLUMN 2 = COMMERCIAL AIR CONDITIONING.
C     COLUMN 3 = MOBILE AIR CONDITIONING.
C     COLUMN 4 = DOMESTIC REFRIGERATION.
C     COLUMN 5 = COMMERCIAL REFRIGERATION.
C
C
C     RANK (ARRAY "K" TERM):
C
C     RANK 1 = MAGNETIC REFRIGERATION.
C     RANK 2 = THERMOELECTRIC REFRIGERATION.
C     RANK 3 = PULSE/THERMOACOUSTIC REFRIGERATION.
C     RANK 4 = REVERSED STIRLING REFRIGERATION.
C     RANK 5 = REVERSED BRAYTON REFRIGERATION.
C     RANK 6 = ABSORPTION REFRIGERATION.
C     RANK 7 = SOLID SORPTION REFRIGERATION.
C     RANK 8 = VAPOR COMPRESSION REFRIGERATION.
C     ************************************************************************
C     ************************************************************************
C
C     TECHNICAL ASSESSMENT DATA ENTRIES FOR THE DIFFERENT REFRIGERATION
C     TECHNOLOGIES IS ENTERED HERE:
C
C     ************************************************************************
```

```
C
C     TECH ASSESSMENT ARRAY FOR MAGNETIC REFRIGERATION:
C
      A(1,1,1) = 1
      A(1,2,1) = 1
      A(1,3,1) = 1                          ! STATE OF ART
      A(1,4,1) = 1
      A(1,5,1) = 1
C
      A(2,1,1) = 2
      A(2,2,1) = 2
      A(2,3,1) = 1                          ! COMPLEXITY
      A(2,4,1) = 2
      A(2,5,1) = 2
C
      A(3,1,1) = 2
      A(3,2,1) = 2
      A(3,3,1) = 1                          ! SIZE/WEIGHT
      A(3,4,1) = 2
      A(3,5,1) = 2
C
      A(4,1,1) = 3
      A(4,2,1) = 3
      A(4,3,1) = 2                          ! MAINTENANCE
      A(4,4,1) = 3
      A(4,5,1) = 3
C
      A(5,1,1) = 4
      A(5,2,1) = 4
      A(5,3,1) = 2                          ! USEFUL LIFE
      A(5,4,1) = 4
      A(5,5,1) = 4
C
      A(6,1,1) = 1
      A(6,2,1) = 1
      A(6,3,1) = 1                          ! CYCLE EFFICIENCY
      A(6,4,1) = 1
      A(6,5,1) = 1
C
C     ***********************************************************************
C
C     TECH ASSESSMENT ARRAY FOR THERMOELECTRIC REFRIGERATION:
C
      A(1,1,2) = 3
      A(1,2,2) = 3
      A(1,3,2) = 3                          ! STATE OF ART
      A(1,4,2) = 2
      A(1,5,2) = 2
```

```
C
      A(2,1,2) = 2
      A(2,2,2) = 2
      A(2,3,2) = 2                          ! COMPLEXITY
      A(2,4,2) = 1
      A(2,5,2) = 1
C
      A(3,1,2) = 5
      A(3,2,2) = 5
      A(3,3,2) = 3                          ! SIZE/WEIGHT
      A(3,4,2) = 5
      A(3,5,2) = 5
C
      A(4,1,2) = 5
      A(4,2,2) = 5
      A(4,3,2) = 3                          ! MAINTENANCE
      A(4,4,2) = 5
      A(4,5,2) = 5
C
      A(5,1,2) = 5
      A(5,2,2) = 5
      A(5,3,2) = 4                          ! USEFUL LIFE
      A(5,4,2) = 5
      A(5,5,2) = 5
C
      A(6,1,2) = 1
      A(6,2,2) = 1
      A(6,3,2) = 1                          ! CYCLE EFFICIENCY
      A(6,4,2) = 1
      A(6,5,2) = 1
C
C     ************************************************************************
C
C     TECH ASSESSMENT ARRAY FOR PULSE/THERMOACOUSTIC REFRIGERATION:
C
      A(1,1,3) = 2
      A(1,2,3) = 2
      A(1,3,3) = 2                          ! STATE OF ART
      A(1,4,3) = 2
      A(1,5,3) = 2
C
      A(2,1,3) = 3
      A(2,2,3) = 2
      A(2,3,3) = 3                          ! COMPLEXITY
      A(2,4,3) = 3
      A(2,5,3) = 3
C
      A(3,1,3) = 3
```

```
      A(3,2,3) = 3
      A(3,3,3) = 2                          ! SIZE/WEIGHT
      A(3,4,3) = 3
      A(3,5,3) = 3
C
      A(4,1,3) = 2
      A(4,2,3) = 2
      A(4,3,3) = 2                          ! MAINTENANCE
      A(4,4,3) = 2
      A(4,5,3) = 2
C
      A(5,1,3) = 3
      A(5,2,3) = 3
      A(5,3,3) = 3                          ! USEFUL LIFE
      A(5,4,3) = 3
      A(5,5,3) = 3
C
      A(6,1,3) = 1
      A(6,2,3) = 1
      A(6,3,3) = 1                          ! CYCLE EFFICIENCY
      A(6,4,3) = 1
      A(6,5,3) = 1
C
C     ************************************************************************
C
C     TECH ASSESSMENT ARRAY FOR REVERSED STIRLING REFRIGERATION:
C
      A(1,1,4) = 3
      A(1,2,4) = 3
      A(1,3,4) = 3                          ! STATE OF ART
      A(1,4,4) = 3
      A(1,5,4) = 3
C
      A(2,1,4) = 3
      A(2,2,4) = 3
      A(2,3,4) = 3                          ! COMPLEXITY
      A(2,4,4) = 3
      A(2,5,4) = 4
C
      A(3,1,4) = 4
      A(3,2,4) = 4
      A(3,3,4) = 4                          ! SIZE/WEIGHT
      A(3,4,4) = 4
      A(3,5,4) = 4
C
      A(4,1,4) = 3
      A(4,2,4) = 3
      A(4,3,4) = 3                          ! MAINTENANCE
```

```
      A(4,4,4) = 3
      A(4,5,4) = 3
C
      A(5,1,4) = 4
      A(5,2,4) = 3
      A(5,3,4) = 4                          ! USEFUL LIFE
      A(5,4,4) = 4
      A(5,5,4) = 3
C
      A(6,1,4) = 2
      A(6,2,4) = 2
      A(6,3,4) = 2                          ! CYCLE EFFICIENCY
      A(6,4,4) = 3
      A(6,5,4) = 3
C
C     ************************************************************************
C
C     TECH ASSESSMENT ARRAY FOR REVERSED BRAYTON REFRIGERATION:
C
      A(1,1,5) = 3
      A(1,2,5) = 3
      A(1,3,5) = 3                          ! STATE OF ART
      A(1,4,5) = 3
      A(1,5,5) = 4
C
      A(2,1,5) = 3
      A(2,2,5) = 3
      A(2,3,5) = 3                          ! COMPLEXITY
      A(2,4,5) = 3
      A(2,5,5) = 4
C
      A(3,1,5) = 2
      A(3,2,5) = 2
      A(3,3,5) = 2                          ! SIZE/WEIGHT
      A(3,4,5) = 2
      A(3,5,5) = 3
C
      A(4,1,5) = 3
      A(4,2,5) = 3
      A(4,3,5) = 3                          ! MAINTENANCE
      A(4,4,5) = 3
      A(4,5,5) = 3
C
      A(5,1,5) = 3
      A(5,2,5) = 3
      A(5,3,5) = 3                          ! USEFUL LIFE
      A(5,4,5) = 3
      A(5,5,5) = 3
```

```
C
      A(6,1,5) = 1
      A(6,2,5) = 1
      A(6,3,5) = 1                          ! CYCLE EFFICIENCY
      A(6,4,5) = 2
      A(6,5,5) = 2
C
C     ***********************************************************************
C
C     TECH ASSESSMENT ARRAY FOR ABSORPTION REFRIGERATION:
C
      A(1,1,6) = 4
      A(1,2,6) = 5
      A(1,3,6) = 2                          ! STATE OF ART
      A(1,4,6) = 4
      A(1,5,6) = 4
C
      A(2,1,6) = 3
      A(2,2,6) = 4
      A(2,3,6) = 2                          ! COMPLEXITY
      A(2,4,6) = 3
      A(2,5,6) = 3
C
      A(3,1,6) = 3
      A(3,2,6) = 3
      A(3,3,6) = 2                          ! SIZE/WEIGHT
      A(3,4,6) = 3
      A(3,5,6) = 3
C
      A(4,1,6) = 3
      A(4,2,6) = 4
      A(4,3,6) = 2                          ! MAINTENANCE
      A(4,4,6) = 3
      A(4,5,6) = 3
C
      A(5,1,6) = 3
      A(5,2,6) = 4
      A(5,3,6) = 2                          ! USEFUL LIFE
      A(5,4,6) = 3
      A(5,5,6) = 3
C
      A(6,1,6) = 5
      A(6,2,6) = 5
      A(6,3,6) = 5                          ! CYCLE EFFICIENCY
      A(6,4,6) = 5
      A(6,5,6) = 5
C
C     ***********************************************************************
```

```
C
C     TECH ASSESSMENT ARRAY FOR SOLID SORPTION REFRIGERATION:
C
      A(1,1,7) = 2
      A(1,2,7) = 2
      A(1,3,7) = 2                          ! STATE OF ART
      A(1,4,7) = 2
      A(1,5,7) = 2
C
      A(2,1,7) = 3
      A(2,2,7) = 3
      A(2,3,7) = 3                          ! COMPLEXITY
      A(2,4,7) = 3
      A(2,5,7) = 3
C
      A(3,1,7) = 3
      A(3,2,7) = 3
      A(3,3,7) = 2                          ! SIZE/WEIGHT
      A(3,4,7) = 3
      A(3,5,7) = 3
C
      A(4,1,7) = 3
      A(4,2,7) = 3
      A(4,3,7) = 3                          ! MAINTENANCE
      A(4,4,7) = 3
      A(4,5,7) = 3
C
      A(5,1,7) = 3
      A(5,2,7) = 3
      A(5,3,7) = 3                          ! USEFUL LIFE
      A(5,4,7) = 3
      A(5,5,7) = 3
C
      A(6,1,7) = 3
      A(6,2,7) = 3
      A(6,3,7) = 3                          ! CYCLE EFFICIENCY
      A(6,4,7) = 4
      A(6,5,7) = 4
C
C     ***********************************************************************
C
C     TECH ASSESSMENT ARRAY FOR VAPOR COMPRESSION REFRIGERATION:
C
      A(1,1,8) = 5
      A(1,2,8) = 5
      A(1,3,8) = 5                          ! STATE OF ART
      A(1,4,8) = 5
      A(1,5,8) = 5
```

```
C
      A(2,1,8) = 4
      A(2,2,8) = 4
      A(2,3,8) = 4                          ! COMPLEXITY
      A(2,4,8) = 4
      A(2,5,8) = 4
C
      A(3,1,8) = 4
      A(3,2,8) = 4
      A(3,3,8) = 4                          ! SIZE/WEIGHT
      A(3,4,8) = 4
      A(3,5,8) = 4
C
      A(4,1,8) = 5
      A(4,2,8) = 5
      A(4,3,8) = 4                          ! MAINTENANCE
      A(4,4,8) = 4
      A(4,5,8) = 4
C
      A(5,1,8) = 5
      A(5,2,8) = 5
      A(5,3,8) = 5                          ! USEFUL LIFE
      A(5,4,8) = 5
      A(5,5,8) = 5
C
      A(6,1,8) = 5
      A(6,2,8) = 5
      A(6,3,8) = 5                          ! CYCLE EFFICIENCY
      A(6,4,8) = 5
      A(6,5,8) = 5
C


C     ************************************************************************
C
C     CLEAR THE SCREEN:
C
      JJ = 27
      WRITE(6,500) JJ
500   FORMAT(1X,A1,'[2J')

C
C     ************************************************************************
C
C     INTRODUCTORY SCREEN:
C
      WRITE(6,*) ' '
```

```
      WRITE(6,*) ' '
      WRITE(6,501)'REFRIGERATION TECHNOLOGY ASSESSMENT'
501   FORMAT (22X,A35)
      WRITE(6,1501)'COMPARISON ROUTINE'
1501  FORMAT(30X,A18,//)
C
      WRITE(6,502) 'DEPARTMENT OF MECHANICAL ENGINEERING'
502   FORMAT(19X,A36,/)
C
      WRITE(6,503) 'IOWA STATE UNIVERSITY'
503   FORMAT(27X,A21)
C
      WRITE(6,504) 'AMES, IOWA 50011'
504   FORMAT(29X,A16,///////////)
C
C     ************************************************************************
C
C     CLEAR THE SCREEN:
C
      II = CHAR(13)
      WRITE(6,505)'PRESS RETURN'
505   FORMAT(32X,A12)
C
      READ(6,506) II
506   FORMAT(A1)
C
      WRITE(6,500) JJ
C
C     ************************************************************************
C
4567  WRITE(6,507) 'THIS PROGRAM CAN BE USED TO COMPARE DIFFERENT'
507   FORMAT(15X,A45)
C
      WRITE(6,508) ' REFRIGERATION TECHNOLOGIES IN SEVERAL APPLICATIONS'
508   FORMAT(12X,A51,///)
C
      WRITE(6,509) 'FIRST, THE APPLICATION MUST BE CHOSEN'
509   FORMAT(19X,A37,//)
C
      WRITE(6,510) 'THE CHOICES ARE:'
510   FORMAT(29X,A18,/)
C
      WRITE(6,511) 'DOMESTIC REFRIGERATION'
511   FORMAT(25X,A22)
C
      WRITE(6,512) 'COMMERCIAL REFRIGERATION'
512   FORMAT(25X,A24,/)
C
```

```
      WRITE(6,513) 'DOMESTIC AIR-CONDITIONING'
513   FORMAT(25X,A25)
C
      WRITE(6,514) 'COMMERCIAL AIR-CONDITIONING'
514   FORMAT(25X,A27)
C
      WRITE(6,515) 'MOBILE AIR-CONDITIONING'
515   FORMAT(25X,A23,////)
C
C     ************************************************************************
C
C     CLEAR THE SCREEN:
C
      WRITE(6,505) 'PRESS RETURN'
C
      READ(6,506) II
C
      WRITE(6,500) JJ
C
C     ************************************************************************
C
C     CHOOSE REFRIGERATION OR AIR-CONDITIONING:
C
911   WRITE(6,516) 'TO CONSIDER A REFRIGERATION APPLICATION, TYPE "R"'
516   FORMAT(15X,A49)
      WRITE(6,517)'TO CONSIDER AN AIR-CONDITIONING APPLICATION,TYPE "A"'
517   FORMAT(15X,A51,//////////)
C
      WRITE(6,518)'MAKE SELECTION AND PRESS RETURN'
518   FORMAT(22X,A31)
C
      READ(6,519) CHOICE1
519   FORMAT(A1)
C
      WRITE(6,500) JJ
C
      IF(CHOICE1 .EQ. 'R') THEN
            GO TO 600
      ELSE IF (CHOICE1 .EQ. 'r') THEN
            GO TO 600
      ELSE IF (CHOICE1 .EQ. 'A') THEN
            GO TO 603
      ELSE IF (CHOICE1 .EQ. 'a') THEN
      GO TO 603
      ELSE
      GO TO 911
      ENDIF
C
```

```
C     ************************************************************************
C
C     CHOOSE DOMESTIC OR COMMERCIAL REFRIGERATION:
C
600   C1='REFRIGERATION   '
      C3='           '
      WRITE(6,520) 'YOU HAVE SELECTED REFRIGERATION APPLICATIONS'
520   FORMAT(13X,A44,///)
C
2222  WRITE(6,521) 'TO SELECT DOMESTIC APPLICATIONS, TYPE "D"'
521   FORMAT(15X,A41)
C
      WRITE(6,522) 'TO SELECT COMMERCIAL APPLICATIONS, TYPE "C"'
522   FORMAT(15X,A43,////////)
C
      WRITE(6,518)'MAKE SELECTION AND PRESS RETURN'
C
      READ(6,519) CHOICE2
C
      WRITE(6,500) JJ
C
      IF (CHOICE2 .EQ. 'D') THEN
        C2='DOMESTIC  '
        K = 4
        GO TO 601
      ELSE IF (CHOICE2 .EQ. 'd')THEN
        C2='DOMESTIC  '
        K = 4
        GO TO 601
      ELSE IF(CHOICE2 .EQ. 'C') THEN
        GO TO 222
      ELSE IF(CHOICE2 .EQ. 'c') THEN
        GO TO 222
      ELSE
        GO TO 2222
      ENDIF
C
222   C2='COMMERCIAL'
      C3='           '
      K = 5
      GO TO 601
C
C     ************************************************************************
C
C     CHOOSE DOMESTIC, COMMERCIAL, OR MOBILE AIR-CONDITIONING:
C
603   C1='AIR CONDITIONING'
      WRITE(6,527) 'YOU HAVE SELECTED AIR-CONDITIONING APPLICATIONS'
```

```
527   FORMAT(13X,A47,///)
C
707   WRITE(6,528) 'TO SELECT DOMESTIC APPLICATIONS, TYPE "DA"'
528   FORMAT(15X,A43)
      WRITE(6,*) ' '
      WRITE(6,529) 'TO SELECT COMMERCIAL APPLICATIONS, TYPE "CA"'
529   FORMAT(15X,A44,/)
      WRITE(6,530) 'TO SELECT MOBILE APPLICATIONS, TYPE "MA"'
530   FORMAT(15X,A39,//////////)
C
      WRITE(6,518)'MAKE SELECTION AND PRESS RETURN'
C
      READ(6,531) CHOICE3
531   FORMAT (A2)
C
      WRITE(6,500) JJ
C
      IF (CHOICE3 .EQ. 'DA') THEN
            C3='DOMESTIC  '
            K = 1
      ELSE IF(CHOICE3 .EQ. 'da') THEN
            C3='DOMESTIC  '
            K = 1
            GO TO 601
      ELSE IF(CHOICE3 .EQ. 'dA') THEN
            C3='DOMESTIC  '
            K = 1
            GO TO 601
      ELSE IF(CHOICE3 .EQ. 'Da') THEN
            C3='DOMESTIC  '
            K = 1
            GO TO 601
      ELSE IF(CHOICE3 .EQ. 'CA') THEN
            C3='COMMERCIAL'
            K = 2
            GO TO 601
      ELSE IF(CHOICE3 .EQ. 'ca') THEN
            C3='COMMERCIAL'
            K = 2
            GO TO 601
      ELSE IF(CHOICE3 .EQ. 'Ca') THEN
            C3='COMMERCIAL'
            K = 2
            GO TO 601
      ELSE IF(CHOICE3 .EQ. 'cA') THEN
            C3='COMMERCIAL'
            K = 2
            GO TO 601
```

```
      ELSE IF(CHOICE3 .EQ. 'MA') THEN
            C3='MOBILE    '
            K = 3
            GO TO 601
      ELSE IF(CHOICE3 .EQ. 'ma') THEN
            C3='MOBILE    '
            K = 3
            GO TO 601
      ELSE IF(CHOICE3 .EQ. 'mA') THEN
            C3='MOBILE    '
            K = 3
            GO TO 601
      ELSE IF(CHOICE3 .EQ. 'Ma') THEN
            C3='MOBILE    '
            K = 3
            GO TO 601
      ELSE
            GO TO 707
      ENDIF
C
C     ************************************************************************
C
C     TECHNICAL ASSESSMENT CRITERIA:
C
601   WRITE(6,20) 'THE TECHNICAL ASSESSMENT CRITERIA ARE:'
20    FORMAT(15X,A39,//)
      WRITE(6,21)'STATE OF THE ART.'
21    FORMAT(20X,A17)
      WRITE(6,22)'COMPLEXITY.'
22    FORMAT(20X,A11)
      WRITE(6,23)'SIZE/WEIGHT.'
23    FORMAT(20X,A12)
      WRITE(6,23)'MAINTENANCE.'
      WRITE(6,23)'USEFUL LIFE.'
      WRITE(6,21)'CYCLE EFFICIENCY.'
C
      WRITE(6,*) '  '
      WRITE(6,*) '  '
      WRITE(6,*) '  '
C
      WRITE(6,40)'THE OZONE DEPLETION POTENTIAL (ODP) AND DIRECT GLOBAL'
40    FORMAT(12X,A53)
      WRITE(6,40)'WARMING POTENTIAL (GWP) OF THE WORKING MATERIALS IS  '
      WRITE(6,40)'ZERO FOR ALL REFRIGERATION TECHNOLOGIES CONSIDERED IN'
      WRITE(6,41)'THIS TECHNICAL ASSESSMENT.'
41    FORMAT(12X,A26)
      WRITE(6,*) '  '
      WRITE(6,*) '  '
```

```
      WRITE(6,*) '  '
      WRITE(6,505) 'PRESS RETURN'
      READ(6,506) II
      WRITE(6,500) JJ
C
8787  WRITE(6,24)'SELECT THE CRITERIA YOU WISH TO CONSIDER BY ANSWERING'
24    FORMAT(15X,A53,//)
      WRITE(6,26)'YES (Y) OR NO (N) TO EACH OF THE FOLLOWING QUESTIONS'
26    FORMAT(15X,A53,///)
C
      J= 0
C
101   WRITE(6,25)'STATE OF THE ART ?          Y OR N'
25    FORMAT(20X,A33)
      READ(6,519) CHOICE4
C      WRITE(6,500) JJ
C
      IF(CHOICE4 .EQ. 'Y') THEN
      J= J + 1
      C4='STATE OF THE ART'
      GO TO 102
      ELSE IF (CHOICE4 .EQ. 'y') THEN
      J= J + 1
      C4='STATE OF THE ART'
      GO TO 102
      ELSE IF (CHOICE4 .EQ. 'N') THEN
      C4='                '
      GO TO 102
      ELSE IF (CHOICE4 .EQ. 'n') THEN
      C4='                '
      GO TO 102
      ELSE
      GO TO 101
      ENDIF
C
102   WRITE(6,25)'COMPLEXITY ?                Y OR N'
      READ(6,519) CHOICE5
C      WRITE(6,500) JJ
C
      IF(CHOICE5 .EQ. 'Y') THEN
      J = J + 1
      C5='COMPLEXITY'
      GO TO 103
      ELSE IF (CHOICE5 .EQ. 'y') THEN
      J = J + 1
      C5='COMPLEXITY'
      GO TO 103
      ELSE IF (CHOICE5 .EQ. 'N') THEN
```

```
      C5='          '
      GO TO 103
      ELSE IF (CHOICE5 .EQ. 'n') THEN
      C5='          '
      GO TO 103
      ELSE
      GO TO 102
      ENDIF
C
103   WRITE(6,25)'SIZE/WEIGHT ?                Y OR N'
      READ(6,519) CHOICE6
C      WRITE(6,500) JJ
C
      IF(CHOICE6 .EQ. 'Y') THEN
      J = J + 1
      C6='SIZE/WEIGHT'
      GO TO 104
      ELSE IF (CHOICE6 .EQ. 'y') THEN
      J = J + 1
      C6='SIZE/WEIGHT'
      GO TO 104
      ELSE IF (CHOICE6 .EQ. 'N') THEN
      C6='          '
      GO TO 104
      ELSE IF (CHOICE6 .EQ. 'n') THEN
      C6='          '
      GO TO 104
      ELSE
      GO TO 103
      ENDIF
C
104   WRITE(6,25)'MAINTENANCE ?                Y OR N'
      READ(6,519) CHOICE7
C      WRITE(6,500) JJ
C
      IF(CHOICE7 .EQ. 'Y') THEN
      J = J + 1
      C7='MAINTENANCE'
      GO TO 105
      ELSE IF (CHOICE7 .EQ. 'y') THEN
      J = J + 1
      C7='MAINTENANCE'
      GO TO 105
      ELSE IF (CHOICE7 .EQ. 'N') THEN
      C7='          '
      GO TO 105
      ELSE IF (CHOICE7 .EQ. 'n') THEN
      C7='          '
```

```
      GO TO 105
      ELSE
      GO TO 104
      ENDIF
C
105   WRITE(6,25)'USEFUL LIFE ?                Y OR N'
      READ(6,519) CHOICE8
C      WRITE(6,500) JJ
C
      IF(CHOICE8 .EQ. 'Y') THEN
      J = J + 1
      C8='USEFUL LIFE'
      GO TO 106
      ELSE IF (CHOICE8 .EQ. 'y') THEN
      J = J + 1
      C8='USEFUL LIFE'
      GO TO 106
      ELSE IF (CHOICE8 .EQ. 'N') THEN
      C8='            '
      GO TO 106
      ELSE IF (CHOICE8 .EQ. 'n') THEN
      C8='            '
      GO TO 106
      ELSE
      GO TO 105
      ENDIF
C
106   WRITE(6,25)'CYCLE EFFICIENCY ?           Y OR N'
      READ(6,519) CHOICE9
C      WRITE(6,500) JJ
C
      IF(CHOICE9 .EQ. 'Y') THEN
      J = J + 1
      C9='CYCLE EFFICIENCY'
      GO TO 107
      ELSE IF (CHOICE9 .EQ. 'y') THEN
      J = J + 1
      C9='CYCLE EFFICIENCY'
      GO TO 107
      ELSE IF (CHOICE9 .EQ. 'N') THEN
      C9='                '
      GO TO 107
      ELSE IF (CHOICE9 .EQ. 'n') THEN
      C9='                '
      GO TO 107
      ELSE
      GO TO 106
      ENDIF
```

```
C
107   IF (J .EQ. 0) THEN
            WRITE(6,109)'YOU MUST SELECT AT LEAST 1 CRITERIA!'
109         FORMAT(15X,A36,///)
            GO TO 8787
      ELSE
            CONTINUE
      ENDIF
C
      WRITE(6,99)'NUMBER OF CRITERIA SELECTED =',J
99    FORMAT(///,20X,A29,I1,////)
      WRITE(6,50)'YOU HAVE CHOSEN THE FOLLOWING APPLICATION:'
50    FORMAT(12X,A42,/)
C
      IF(CHOICE1 .EQ. 'R') THEN
            GO TO 1600
      ELSE IF (CHOICE1 .EQ. 'r') THEN
            GO TO 1600
      ELSE IF (CHOICE1 .EQ. 'A') THEN
            GO TO 1603
      ELSE IF (CHOICE1 .EQ. 'a') THEN
      GO TO 1603
      ENDIF
C
1600  WRITE(6,60) C2,' ',C1
60    FORMAT(20X,A10,A1,A16,//)
      GO TO 1602
1603  WRITE(6,61) C3,' ',C1
61    FORMAT(20X,A10,A1,A16,//)
1602  CONTINUE
C
C     ************************************************************************
C
      WRITE(6,505)'PRESS RETURN'
      READ(6,506) II
      WRITE(6,500) JJ
C
C
      WRITE(6,80)'YOU HAVE CHOSEN TO CONSIDER   ',J
80    FORMAT(20X,A30,I1)
      WRITE(6,81)'TECHNICAL ASSESSMENT CRITERIA:'
81    FORMAT(20X,A30,///)
      WRITE(6,82)C4
82    FORMAT(25X,A16)
      WRITE(6,83)C5
83    FORMAT(25X,A10)
      WRITE(6,84)C6
84    FORMAT(25X,A11)
```

```
      WRITE(6,84)C7
      WRITE(6,84)C8
      WRITE(6,82)C9
C
      DO 90, I = 1,6
            WC(I) = 0.0
90    CONTINUE
C
      WRITE(6,500) JJ
      WRITE(6,812)'THE NEXT STEP IS TO WEIGHT THE TA CRITERIA.'
812   FORMAT(15X,A43,//)
      WRITE(6,814)'YOU HAVE TWO OPTIONS:'
814   FORMAT(15X,A21,/)
      WRITE(6,815)'ACCEPT THE DEFAULT VALUE OF EQUAL WEIGHTING.'
815   FORMAT(20X,A44)
      WRITE(6,816)'WEIGHT THE TA CRITERIA YOURSELF.'
816   FORMAT(20X,A32)
      WRITE(6,5638)'DO YOU WISH TO ACCEPT THE DEFAULT, Y OR N ?'
5638  FORMAT(///,15X,A43,//)
      WRITE(6,518)'MAKE SELECTION AND PRESS RETURN'
C
      READ(6,519) CHOICE11
      WRITE(6,500) JJ
C
      IF ((CHOICE11 .EQ. 'Y').OR.(CHOICE11 .EQ. 'y')) THEN
C
            DWF = (1.0/(J))
C
            IF ((CHOICE4 .EQ. 'Y').OR.(CHOICE4 .EQ. 'y')) THEN
                  WC(1) = DWF
            END IF
C
            IF ((CHOICE5 .EQ. 'Y').OR.(CHOICE5 .EQ. 'y')) THEN
                  WC(2) = DWF
            END IF
C
            IF ((CHOICE6 .EQ. 'Y').OR.(CHOICE6 .EQ. 'y')) THEN
                  WC(3) = DWF
            END IF
C
            IF ((CHOICE7 .EQ. 'Y').OR.(CHOICE7 .EQ. 'y')) THEN
                  WC(4) = DWF
            END IF
C
            IF ((CHOICE8 .EQ. 'Y').OR.(CHOICE8 .EQ. 'y')) THEN
                  WC(5) = DWF
            END IF
C
```

```
            IF ((CHOICE9 .EQ. 'Y').OR.(CHOICE9 .EQ. 'y')) THEN
                  WC(6) = DWF
            END IF
            GO TO 3030
C
      ENDIF
C
      WRITE(6,85)'YOU MAY WEIGHT THESE CRITERIA AS YOU WISH. HOWEVER'
85    FORMAT(15X,A50)
3333  WRITE(6,86)'THE SUM OF THE WEIGHTING FACTORS FOR THE ',J
86    FORMAT(15X,A41,I1)
      WRITE(6,87)'CRITERIA MUST EQUAL 1.'
87    FORMAT(15X,A22,///)
C
      IF(CHOICE4 .EQ. 'Y') THEN
            WRITE(6,88)C4,' ?'
88          FORMAT(20X,A16,A2)
            READ(6,*) WC(1)
      ELSE IF (CHOICE4 .EQ. 'y') THEN
            WRITE(6,88)C4,' ?'
            READ(6,*) WC(1)
      ENDIF
C
      IF(CHOICE5 .EQ. 'Y') THEN
           WRITE(6,89)C5,' ?'
89          FORMAT(20X,A10,A2)
            READ(6,*) WC(2)
      ELSE IF (CHOICE5 .EQ. 'y') THEN
            WRITE(6,89)C5,' ?'
            READ(6,*) WC(2)
      ENDIF
C
      IF(CHOICE6 .EQ. 'Y') THEN
            WRITE(6,9000)C6,' ?'
9000        FORMAT(20X,A11,A2)
            READ(6,*) WC(3)
      ELSE IF (CHOICE6 .EQ. 'y') THEN
            WRITE(6,9000)C6,' ?'
            READ(6,*) WC(3)
      ENDIF
C
      IF(CHOICE7 .EQ. 'Y') THEN
            WRITE(6,9000)C7,' ?'
            READ(6,*) WC(4)
      ELSE IF (CHOICE7 .EQ. 'y') THEN
            WRITE(6,9000)C7,' ?'
            READ(6,*) WC(4)
      ENDIF
```

```
C
      IF(CHOICE8 .EQ. 'Y') THEN
            WRITE(6,9000)C8,' ?'
            READ(6,*) WC(5)
      ELSE IF (CHOICE8 .EQ. 'y') THEN
            WRITE(6,9000)C8,' ?'
            READ(6,*) WC(5)
      ENDIF
C
      IF(CHOICE9 .EQ. 'Y') THEN
            WRITE(6,88)C9,' ?'
            READ(6,*) WC(6)
      ELSE IF (CHOICE9 .EQ. 'y') THEN
            WRITE(6,9000)C9,' ?'
            READ(6,*) WC(6)
      ENDIF
C
C     ***********************************************************************
C
C     CHECK:
C
      TOTAL=0.0
      DO 2000 I=1,6
            TOTAL=TOTAL + WC(I)
2000  CONTINUE
C
      DIFF=TOTAL - 1.0
C
      IF(DIFF .GT. 0.05) THEN
        WRITE(6,91)'SUM OF WEIGHTING FACTORS EXCEEDS 1..TRY AGAIN!'
91      FORMAT(12X,A46,//)
        GO TO 3333
      ELSE IF (DIFF .LT. -0.05) THEN
        WRITE(6,92)'SUM OF WEIGHTING FACTORS LESS THAN 1..TRY AGAIN!'
92      FORMAT(12X,A48,//)
        GO TO 3333
      ELSE
        CONTINUE
      ENDIF
C
      WRITE(6,500) JJ
C
3030        WRITE(6,45) 'ENTER THE NAME OF THE DATA OUTPUT FILE'
45          FORMAT(15X,A38,//)
            READ(6,548) OUTPUTFILE
            WRITE(6,500) JJ
548         FORMAT(A50)
            OPEN(11,FILE=OUTPUTFILE,STATUS='UNKNOWN')
```

```
C
C     ************************************************************************
C
C     TECH ASSESSMENT EQUATION:
C
      DO 6666 L=1,N
            RATE(L) = 0.0
                  DO 7000 I = 1,6
                        RATE(L)= RATE(L) + (WC(I)) * (A(I,K,L))
7000              CONTINUE
C
6666  CONTINUE
C
C     ************************************************************************
C
C     RANK THE TECHNOLOGIES FROM HIGH TO LOW AND DISPLAY OUTPUT:
C
      WRITE(6,737)'RANKING OF ALTERNATIVE REFRIGERATION TECHNOLOGIES'
      WRITE(11,737)'RANKING OF ALTERNATIVE REFRIGERATION TECHNOLOGIES'
737   FORMAT(15X,A49,/)
C
      IF(CHOICE1 .EQ. 'R') THEN
            GO TO 2600
      ELSE IF (CHOICE1 .EQ. 'r') THEN
            GO TO 2600
      ELSE IF (CHOICE1 .EQ. 'A') THEN
            GO TO 2603
      ELSE IF (CHOICE1 .EQ. 'a') THEN
      GO TO 2603
      ENDIF
C
2600  WRITE(6,8601) 'FOR ',C2,' ',C1,' ',' APPLICATIONS'
      WRITE(11,8601) 'FOR ',C2,' ',C1,' ',' APPLICATIONS'
8601   FORMAT(20X,A4,A10,A1,A16,A1,A13,////)
      GO TO 2602
2603  WRITE(6,611) 'FOR ',C3,' ',C1,' APPLICATIONS'
      WRITE(11,611) 'FOR ',C3,' ',C1,' APPLICATIONS'
611   FORMAT(20X,A4,A10,A1,A16,A13,////)
2602  CONTINUE
C
      WRITE(6,80)'YOU HAVE CHOSEN TO CONSIDER   ',J
      WRITE(11,80)'YOU HAVE CHOSEN TO CONSIDER   ',J
C
      WRITE(6,81)'TECHNICAL ASSESSMENT CRITERIA:'
      WRITE(11,81)'TECHNICAL ASSESSMENT CRITERIA:'
C
      WRITE(6,822)C4,WC(1)
      WRITE(11,822)C4,WC(1)
```

```
822   FORMAT(25X,A16,4X,F4.2)
      WRITE(6,833)C5,WC(2)
      WRITE(11,833)C5,WC(2)
833   FORMAT(25X,A10,10X,F4.2)
      WRITE(6,844)C6,WC(3)
      WRITE(11,844)C6,WC(3)
844   FORMAT(25X,A11,9X,F4.2)
      WRITE(6,844)C7,WC(4)
      WRITE(11,844)C7,WC(4)
      WRITE(6,844)C8,WC(5)
      WRITE(11,844)C8,WC(5)
      WRITE(6,822)C9,WC(6)
      WRITE(11,822)C9,WC(6)
      WRITE(6,*)' '
      WRITE(6,*)' '
      WRITE(6,*)' '
      WRITE(11,*)' '
      WRITE(11,*)' '
      WRITE(11,*)' '
C
      WRITE(6,505) 'PRESS RETURN'
      READ(6,506) II
      WRITE(6,500) JJ
C
      WRITE(6,738)'RANKING','REFRIGERATION TECHNOLOGY','RATING'
      WRITE(11,738)'RANKING','REFRIGERATION TECHNOLOGY','RATING'
738   FORMAT(20X,A7,2X,A24,6X,A6,//)
C
      DO 4444 L = 1,N
            RMAX = -1E2
C
            DO 5555 K = 1,N
                  RMAX = MAX(RMAX,RATE(K))
5555        CONTINUE
C
            IF (RMAX .EQ. RATE(1)) THEN
                  WRITE(6,739)L,'MAGNETIC REFRIGERATION',RATE(1)
                  WRITE(11,739)L,'MAGNETIC REFRIGERATION',RATE(1)
739               FORMAT(23X,I1,5X,A22,10X,F4.2)
                  RATE(1) = 0.0
            ELSE IF (RMAX .EQ. RATE(2)) THEN
                  WRITE(6,740)L,'PULSE/THERMOACOUSTIC',RATE(2)
                  WRITE(11,740)L,'PULSE/THERMOACOUSTIC',RATE(2)
740               FORMAT(23X,I1,5X,A20,12X,F4.2)
                  RATE(2) = 0.0
            ELSE IF (RMAX .EQ. RATE(3)) THEN
                  WRITE(6,741)L,'THERMOELECTRIC',RATE(3)
                  WRITE(11,741)L,'THERMOELECTRIC',RATE(3)
```

```
741             FORMAT(23X,I1,5X,A14,18X,F4.2)
                RATE(3) = 0.0
          ELSE IF (RMAX .EQ. RATE(4)) THEN
                WRITE(6,742)L,'REVERSED STIRLING',RATE(4)
                WRITE(11,742)L,'REVERSED STIRLING',RATE(4)
742             FORMAT(23X,I1,5X,A17,15X,F4.2)
                RATE(4) = 0.0
          ELSE IF (RMAX .EQ. RATE(5)) THEN
                WRITE(6,743)L,'REVERSED BRAYTON',RATE(5)
                WRITE(11,743)L,'REVERSED BRAYTON',RATE(5)
743             FORMAT(23X,I1,5X,A16,16X,F4.2)
                RATE(5) = 0.0
          ELSE IF (RMAX .EQ. RATE(6)) THEN
                WRITE(6,744)L,'ABSORPTION',RATE(6)
                WRITE(11,744)L,'ABSORPTION',RATE(6)
744             FORMAT(23X,I1,5X,A10,22X,F4.2)
                RATE(6) = 0.0
          ELSE IF (RMAX .EQ. RATE(7)) THEN
                WRITE(6,745)L,'SOLID SORPTION',RATE(7)
                WRITE(11,745)L,'SOLID SORPTION',RATE(7)
745             FORMAT(23X,I1,5X,A14,18X,F4.2)
                RATE(7) = 0.0
          ELSE IF (RMAX .EQ. RATE(8)) THEN
                WRITE(6,746)L,'VAPOR COMPRESSION',RATE(8)
                WRITE(11,746)L,'VAPOR COMPRESSION',RATE(8)
746             FORMAT(23X,I1,5X,A10,22X,F4.2)
                RATE(8) = 0.0
          END IF
4444  CONTINUE
C
      CLOSE (11)
C
C     ***********************************************************************
C
C     CHOICE TO CONTINUE PROGRAM, OR NOT:
C
      WRITE(6,5632)'DO YOU WISH TO CONTINUE WITH ANOTHER CASE, Y OR N?'
5632  FORMAT(///,15X,A50,//)
      WRITE(6,518)'MAKE SELECTION AND PRESS RETURN'
C
C     ***********************************************************************
C
1010  READ(6,519) CHOICE10
      WRITE(6,500) JJ
C
      IF ((CHOICE10 .EQ. 'Y').OR.(CHOICE10 .EQ. 'y')) THEN
C
          GO TO 4567
```

```
C
      ELSEIF ((CHOICE10 .EQ. 'N').OR.(CHOICE10 .EQ. 'n')) THEN
C
            CONTINUE
C
      ENDIF
C
      END
```

APPENDIX E. SAMPLE DATA FROM THE TECHNOLOGY ASSESSMENT PROGRAM

RANKING OF ALTERNATIVE REFRIGERATION TECHNOLOGIES

FOR DOMESTIC AIR CONDITIONING APPLICATIONS

YOU HAVE CHOSEN TO CONSIDER 6
TECHNICAL ASSESSMENT CRITERIA:

STATE OF THE ART	.20
COMPLEXITY	.15
SIZE/WEIGHT	.05
MAINTENANCE	.15
USEFUL LIFE	.15
CYCLE EFFICIENCY	.30

RANKING	REFRIGERATION TECHNOLOGY	RATING
1	VAPOR COMP	4.80
2	ABSORPTION	3.80
3	PULSE/THERMOACOUSTIC	2.95
4	REVERSED STIRLING	2.90
5	SOLID SORPTION	2.80
6	REVERSED BRAYTON	2.35
7	THERMOELECTRIC	2.05
8	MAGNETIC REFRIGERATION	1.95

RANKING OF ALTERNATIVE REFRIGERATION TECHNOLOGIES

FOR COMMERCIAL AIR CONDITIONING APPLICATIONS

YOU HAVE CHOSEN TO CONSIDER 6
TECHNICAL ASSESSMENT CRITERIA:

STATE OF THE ART	.20
COMPLEXITY	.10
SIZE/WEIGHT	.05
MAINTENANCE	.15
USEFUL LIFE	.20
CYCLE EFFICIENCY	.30

RANKING	REFRIGERATION TECHNOLOGY	RATING
1	VAPOR COMP	4.85
2	ABSORPTION	4.45
3	PULSE/THERMOACOUSTIC	3.10
4	SOLID SORPTION	2.80
5	REVERSED STIRLING	2.75
6	REVERSED BRAYTON	2.35
7	MAGNETIC REFRIGERATION	2.05
8	THERMOELECTRIC	1.95

RANKING OF ALTERNATIVE REFRIGERATION TECHNOLOGIES

FOR MOBILE AIR CONDITIONING APPLICATIONS

YOU HAVE CHOSEN TO CONSIDER 6
TECHNICAL ASSESSMENT CRITERIA:

STATE OF THE ART	.15
COMPLEXITY	.20
SIZE/WEIGHT	.30
MAINTENANCE	.20
USEFUL LIFE	.05
CYCLE EFFICIENCY	.10

RANKING	REFRIGERATION TECHNOLOGY	RATING
1	VAPOR COMP	4.30
2	REVERSED STIRLING	3.25
3	PULSE/THERMOACOUSTIC	2.65
4	SOLID SORPTION	2.55
5	REVERSED BRAYTON	2.50
6	ABSORPTION	2.30
7	THERMOELECTRIC	2.15
8	MAGNETIC REFRIGERATION	1.25

RANKING OF ALTERNATIVE REFRIGERATION TECHNOLOGIES

FOR DOMESTIC REFRIGERATION APPLICATIONS

YOU HAVE CHOSEN TO CONSIDER 6
TECHNICAL ASSESSMENT CRITERIA:

STATE OF THE ART	.20
COMPLEXITY	.20
SIZE/WEIGHT	.10
MAINTENANCE	.10
USEFUL LIFE	.15
CYCLE EFFICIENCY	.25

RANKING	REFRIGERATION TECHNOLOGY	RATING
1	VAPOR COMP	4.60
2	ABSORPTION	3.70
3	REVERSED STIRLING	3.25
4	SOLID SORPTION	3.05
5	REVERSED BRAYTON	2.65
6	PULSE/THERMOACOUSTIC	2.60
7	THERMOELECTRIC	2.20
8	MAGNETIC REFRIGERATION	1.95

RANKING OF ALTERNATIVE REFRIGERATION TECHNOLOGIES

FOR COMMERCIAL REFRIGERATION APPLICATIONS

YOU HAVE CHOSEN TO CONSIDER 6
TECHNICAL ASSESSMENT CRITERIA:

STATE OF THE ART	.20
COMPLEXITY	.10
SIZE/WEIGHT	.05
MAINTENANCE	.15
USEFUL LIFE	.20
CYCLE EFFICIENCY	.30

RANKING	REFRIGERATION TECHNOLOGY	RATING
1	VAPOR COMP	4.70
2	ABSORPTION	3.80
3	REVERSED STIRLING	3.15
4	SOLID SORPTION	3.10
5	REVERSED BRAYTON	3.00
6	PULSE/THERMOACOUSTIC	2.80
7	MAGNETIC REFRIGERATION	2.05
8	THERMOELECTRIC	2.05