Comparative study of advanced heat pumps

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COMPARATIVE STUDY OF ADVANCED HEAT PUMPS

by

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A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering

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ABSTRACT

A numerical simulation study is reported on the thermodynamic performance of several non-CFC refrigeration devices. The study includes complex compound absorption, Brayton, Stirling, and thermoelectric devices. Comparisons are made to the more commonly applied vapor compression systems, including those using R-134a. The study examines the effect of thermal resistances between the device and the heat rejection or heat absorption space. A cool side temperature difference between 0 and 20° C is investigated, and this temperature difference accounts for both thermal resistance and cooling load. An outside temperature ranging between 35° C and 46° C is considered in the calculations, with a cooled space temperature of 22° C assumed throughout. Evaluations of the coefficients of performance for each of the units show the vapor compression machines demonstrate superior performance over the complete range of operating conditions examined. However, additional requirements, such as maintenance and environmental factors, indicate other desirable options.
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CHAPTER 1

GENERAL OVERVIEW

Introduction

Society today demands a comfortable living climate. Suitable temperatures certainly constitute a major part of that requirement. Society further demands that the method used must consider economic and environmental concerns which ultimately may become intertwined. A diverse set of machines and chemical processes emulate thermodynamic cycles to produce the necessary cooling or heating effect. By comparing these systems, the most satisfactory solution to all the concerns can be determined.

Heat pumps use an input of heat or work to drive a device which can produce either a cooling effect (heat transferred out of a space) or a heating effect (heat transferred into a space). Currently, the vapor compression cycle is the most common using primarily dichlorodifluoromethane (R-12) as the working fluid. Chlorofluorocarbons (CFCs) represent the bulk of fluids
employed by vapor compression cycles. Recent studies contend the
discharge of CFCs into the atmosphere has led to ozone layer depletion.
Whether scientifically proven or not, the international political community has
spoken by banning the production of all CFCs by the end of 1995. This
action has prompted the search for substitute fluids. The leading candidate
for replacement is 1,1,1,2-tetrafluoroethane (R-134a), seemingly harmless to
the protective ozone layer. However, the search continues and the long term
detrimental ramifications cannot be foreseen.

Previously, the vapor compression cycle seemed to have all the
advantages: high efficiency, low toxicity, low cost, and a simple mechanical
embodiment. Now, the environmental disadvantages outweigh the benefits,
making way for other heat pump technologies which until now have mostly
been disregarded. As environmental concerns grow, alternative technologies
which use inert gasses or no fluid at all become attractive solutions.

The devices modeled include: vapor compression, complex compound
absorption, Brayton, Stirling, and thermoelectric. The vapor compression
obviously bears importance in any comparative discussion. The others were
chosen based on their ability to become useful, short term, non-CFC heat
pump devices. The traditional absorption heat pump’s high level of
characterization resulted in omitting the device from consideration. Other advanced technologies such as thermoacoustic and magnetic heat pumps do not seem viable solutions at this time; though they certainly warrant more investigation.

In this study, computer modeling of the heat pump technologies provides the output used for the comparisons. The coefficient of performance, defined as the ratio of the useful effect to the energy purchased, provides the best measure for evaluating the systems. The important parameters common to all systems are varied to cover a range of operating criteria. In the following, a description of each cycle introduces these parameters and the methodology used for finding the coefficient of performance. Finally, an analysis of the results ranks the systems, giving insight to the strengths and weaknesses of each.

Literature Survey

The literature tends to focus on a particular device or even a specific component but does not cover the broader aspect of comparison. The lack of information produces a hindrance in verifying the results; however, the results become that much more interesting.
A recent study by Herbas et al. (1993) presents a steady state simulation of a vapor compression heat pump. Mathematical models of each component result in a nonlinear set of equations numerically solved. Specific equipment characteristics (compressor volumes, speed, etc.) design the exact system. The model used two refrigerants, R-12 and R-134a. The simulation attained good agreement with experimental data for R-12. This model evolved from an earlier study by Parise (1986) adding condenser losses and an improved numerical method (Herbas et al. 1993). The model by Parise (1986) produced an error of 10% compared to experimental data.

Bisio (1993) presented a paper considering an ideal cycle with finite capacity heat reservoirs. The paper asserts referring to an ideal Carnot cycle has poor meaning with regard to a vapor compression heat pump. The Lorenz cycle provides a more realistic ideal reference cycle. Non-azeotropic refrigerant mixtures allow sliding condensation and evaporation temperatures approximating the Lorenz cycle.

The paper by Graebel, Rockenfeller, and Kirol (1991) discusses complex compound absorption. The text offers the technology as a viable alternative to CFC refrigeration. The discussion lists system advantages and disadvantages while presenting typical values for the Coefficient of
Performance. The paper presents the general theory behind complex compound absorption.

Sisto (1978) modeled the reversed Brayton cycle. Using realistic values for compressor and turbine efficiencies along with the heat exchanger effectiveness, the paper presents the heating coefficient of performance. Sisto concludes the potential exists for quite reasonable COP values with the regenerative heat pump clearly superior to the basic cycle.

Two papers present Stirling cycle refrigerators for home appliance use. Kim, Cho, and Chung (1993) constructed a 250 watt refrigerator and analyzed the performance effects of various parameters such as pressure, speed, and temperature. Otaka, Saito, and Saito (1993) constructed a 100 watt cooler and examined similar effects on COP concluding the refrigerator will be a promising candidate for CFC system replacement.

Vitale and Vincent (1992) studied the Stirling refrigeration cycle for low temperature ratio refrigeration applications, i.e. residential and commercial heating and air conditioning. The paper addresses some of the low temperature ratio implications on design and performance. HFAST1, a proprietary Stirling cycle analysis code, evaluated the performance for a variety of input parameters. The results were 15% below measured test data.
The simulation verified cycle trends, for example decreased cycle performance with increased cooling flux and with decreased cycle temperature ratio.

Rix (1989) created a Stirling cycle model investigating the performance in an industrial waste heat recovery application. Further, a practical prototype provided verification of the results. The study produced encouraging performance results for this application (taking waste heat at 60° C and returning it at 160° C).

Stockholm and Stockholm (1992) model a thermoelectric device with heat exchangers on both sides. The model provides a way to find the module performance with a heat sink of known thermal resistance. Manufacturer data gives performance based on the module face temperatures, usually unknown quantities. This paper demonstrates the effect on performance of adding heat exchangers.

Another paper discussing thermoelectric modeling investigates cooling parked aircraft (Gwilliam et al. 1992). The study considers evaporative cooling of the waste heat side air to improve performance. The new approach substantially improved performance.
CHAPTER 2

DEVICE DESCRIPTION

Each of the heat pumps differ, sometimes dramatically, in their thermodynamic cycle. This section presents how the cycles were considered for modeling. A temperature versus entropy (T-s) diagram provides a common way of conceptualizing many cycles and a schematic diagram illustrates the physical configuration of each system. Presentation of these diagrams where appropriate along with the pertinent equations aids in understanding the method of development.

Vapor Compression

The Rankine cycle approximates the vapor compression system which has four components: 1. compressor, 2. condenser, 3. throttling valve, and 4. evaporator. Figure 1 schematically shows the components of the vapor compression system with the cycle state points marked for reference to Figure 2.
Figure 1. Vapor Compression Schematic Diagram

Figure 2. Vapor Compression Cycle Temperature-Entropy Diagram
Component Description

1. Compressor

The compressor increases the pressure of the vapor which also increases the temperature. This device inputs work to the system. An ideal compressor (100% efficient) performs a constant entropy process from state 1 to 2s as shown in Figure 2. As with all machines, real compressors have an efficiency below 100% resulting in an increase in entropy. Compressor efficiency is defined as a ratio of enthalpy differences,

$$\eta = \frac{h_{2s} - h_{1}}{h_{2} - h_{1}}$$  \hspace{1cm} (1)

2. Condenser

The condenser condenses the fluid isobarically (at constant pressure) from a superheated vapor at state 2 to a saturated liquid at state 3 by rejecting heat through a heat exchanger to the hot temperature reservoir. Realistically a pressure drop across the condenser is needed to cause the fluid to flow, though this is neglected by the present model. The heat rejected equals the product of the mass flow rate multiplied by the change in enthalpy,

$$Q_{n} = \dot{m}(h_{1} - h_{2})$$  \hspace{1cm} (2)
3. Throttling Valve

The throttling process reduces the pressure, and thus temperature, through a constant enthalpy process from state 3 to 4.

$$h_3 = h_4$$  \hspace{1cm} (3)

4. Evaporator

The evaporator changes the fluid isobarically from a wet mixture to a saturated vapor through the addition of heat according to

$$Q_c = \dot{m}(h_1 - h_4)$$  \hspace{1cm} (4)

The outlet of the evaporator, state 1, coincides with the inlet of the compressor thus completing the cycle (this assumes that the fluid does not undergo superheating before it enters the compressor).

Coefficient of Performance

The coefficient of performance (COP) gives a quantitative value for the performance of a heat pump. COP is defined as the ratio of the useful effect to the energy purchased. For cooling, the COP is given by

$$COP_c = \frac{Q_c}{W} = \frac{h_1 - h_4}{h_2 - h_1}$$  \hspace{1cm} (5)

The heat and work terms can be expressed as the mass flow rate multiplied by the change in enthalpy. In the ratio the mass flow rate cancels, leaving the
Thermal Resistance

Two temperatures of practical importance to a heat pump are the ambient temperature and the room temperature. These temperatures differ from those within the cycle due to thermal resistance between the fluid inside and the temperature reservoir outside the respective heat exchangers. This resistance depends on the material conduction properties, convection heat transfer coefficients, heat exchanger design and configuration, etc.

\[ R = \frac{1}{k_{\text{cond}}} + \frac{1}{h_{\text{conv}} A_{\text{in}}} + \frac{1}{h_{\text{conv}} A_{\text{out}}} + \frac{1}{R_{\text{mixing}}} \]

**Figure 3.** Thermal Resistance

In order to account for this resistance while not making it specific to one design, an overall resistance can be used which lumps all the individual parameters into a single term, \( R \). Figure 3 shows some of the resistances included in \( R \), for example the convection and conduction resistances which include most elements of heat exchanger design and a mixing resistance which accounts for the mixing of the air coming from over the heat exchanger.
with the room or ambient air. In this study the resistance is varied from zero, an ideal situation, to a relatively large value to see the effect on cycle performance. The resistance concept applies to both the condenser and evaporator. To narrow the study’s scope, the condenser and evaporator resistances were considered equal. The heat flow equals the conductance, the inverse of the resistance, multiplied by the temperature difference.

\[ Q_c = \frac{1}{R} (T_{room} - T_{evap}) \]  

\[ Q_{H} = \frac{1}{R} (T_{cond} - T_{amb}) \]

These two equations use the ambient and room temperatures along with the cooling load, \( Q_c \), to determine the appropriate cycle temperatures.

### Complex Compound Absorption

A simple complex compound absorption system consists of five parts: 1. desorbing vessel, 2. condenser, 3. expansion valve, 4. evaporator, and 5. absorbing vessel. This is essentially a vapor compression system with the compressor replaced by the sorption process. Commonly, the device uses a metal salt as the sorbant and ammonia as the refrigerant. The schematic diagram of Figure 4, shows the physical relationship between the parts. The next section concludes describing the noncyclic operation of the vessels.
**Figure 4.** Complex Compound Absorption Schematic Diagram

**Figure 5.** Complex Compound Absorption Temperature-Entropy Diagram
Component Description

1. Desorbing Vessel (Vessel #1)

Referring to the temperature-entropy diagram in Figure 5, state 1 can be determined using an equation developed by Nernst relating the temperature and pressure of the complex compound and refrigerant (Graebel et al. 1991).

\[
\log P = a - \frac{b}{T}
\]  

The constants \( a \) and \( b \) are determined experimentally based on the type of complex compound. If the thermal resistance is neglected, the temperature in the saturation region is known (ambient temperature), and hence so is the pressure. Considering the condenser as an isobaric process, the pressure at state 1 becomes known. Solving this equation for temperature provides the second property needed for determination of the thermodynamic state.

2. Condenser - 4. Evaporator

The theory and equations for this section of the cycle identically match the vapor compression cycle discussed previously.

5. Absorbing Vessel (Vessel #2)

The same equation used for state 1 applies to state 4P. Once again pressure is known from saturation and solving for temperature determines the second property.
State 1 and state 4P represent two separate vessels, completely absorbed and completely desorbed respectively. The process begins by desorbing the ammonia from the complex compound at state 1 through the addition of heat. The heat needed is given by the product of the mass flow rate, the constant $a$, and the universal gas constant.

$$Q_{\text{DESR}} = -2.303\dot{n}aR$$

Similarly the complex compound in vessel #2 absorbs the ammonia at state 4P with the rejection of heat. The constant 2.303 is a logarithmic conversion.

$$Q_{\text{ABSOR}} = 2.303\dot{n}aR$$

Once all the ammonia has been desorbed from vessel #1 and absorbed by vessel #2 the vessels switch places through valving of the connecting lines. Vessel #1 becomes vessel #2 by lowering #1’s temperature; conversely, vessel #2 becomes vessel #1 by raising #2’s temperature and the process begins again. Ideally, the heat associated with raising the vessel temperature can be completely recovered.

Coefficient of Performance

The cooling COP equals the useful energy divided by the heat added for desorption assuming an ideal process for raising the vessel temperature.
The process of raising the vessel temperature requires some heat addition. The complex compound, the refrigerant, and the vessel all contribute to the total amount of heat necessary which depends on density, volume, specific heat, and the temperature difference divided by time.

\[ Q = \rho V c_p \frac{\Delta T}{\Delta t} \]  \hspace{2cm} (12)

The unrecoverable fraction of the total represents the external heat added. This term would appear as energy input to the system, adding to the denominator in equation (11), decreasing COP.

In this investigation, complex compound absorption operates on an input of heat while the other devices use an input of work. All the devices can be heat driven with an engine generating the work input; however, not all the devices can be work driven because of complex compound absorption. The large number of engines available, each with different efficiencies, makes the inclusion of this parameter an unnecessary complication to the problem. As an approximation engines convert heat into work with an efficiency of 33%. For reference, decreasing the COP of the other devices by one-third would estimate the result of operating each with heat rather than work.
Thermal Resistance

The thermal resistance directly corresponds to the vapor compression cycle discussion.

Brayton

Three components comprise the open Brayton cycle considered in this study: 1. compressor, 2. heat exchanger, and 3. turbine. As illustrated by the schematic in Figure 6, the air exits the cycle and goes directly into the room. A closed Brayton cycle would have an additional heat exchanger inside the space making a closed loop. Generally, the closed cycle is not employed due to the poor performance of the air to air heat exchangers.

![Figure 6. Brayton Schematic Diagram](image-url)
NOTE: $4_{RM}$ is final state inside space. Using inside air for cycle, state 1 would be the same point as state $4_{RM}$

Figure 7. Brayton Temperature-Entropy Diagram

Component Description

1. Compressor

The compressor brings in either inside room air or outside ambient air increasing the air pressure and thus temperature. The process from state 1 to 2s and state 3 to 4s in Figure 7 is a polytropic process. This means that the pressure multiplied by the specific volume to the $n^{th}$ power equals a constant.

$$Pv^n = \text{const.}$$ (13)
The polytropic exponent, \( n \), has different values depending on the process. In the adiabatic case, no heat crossing the system boundary, \( n \) is equal to the ratio of specific heat at constant pressure to specific heat at constant volume.

\[
n = \gamma = \frac{c_p}{c_v}
\]  

(14)

Air, considered an ideal gas, obeys the ideal gas law,

\[
P_v = RT
\]  

(15)

Using this equation and the polytropic equation between states 1 and 2, the following result is obtained:

\[
\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{(\gamma - 1)}{\gamma}}
\]  

(16)

and similarly for state 3 to 4,

\[
\frac{T_{4s}}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{(\gamma - 1)}{\gamma}}
\]  

(17)

As with the vapor compression cycle, the work input device (the compressor) does not operate 100% efficiently. For an ideal gas with constant specific heat, the enthalpy difference equals the specific heat multiplied by the temperature difference.

\[
\Delta h = c_p \Delta T
\]  

(18)

Substituting into the original equation for efficiency, the specific heat cancels
leaving the more accommodating relation,

\[ \eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{T_{2s} - T_1}{T_2 - T_1} \]  

in terms of temperature differences.

2. Heat Exchanger

The heat exchanger decreases the air temperature isobarically from state 2 to 3 by rejecting heat to the ambient temperature heat sink. Again, realistically a pressure drop would be required to move the fluid, but it is neglected. Using the enthalpy - temperature relation the heat rejected equals:

\[ Q_n = \dot{m}(h_3 - h_2) = \dot{m}c_p(T_3 - T_2) \]

3. Turbine

The turbine expands the gas from state 3 to 4 producing work while lowering the temperature and pressure. The work produced goes directly into the compressor reducing the amount of external work required. As a work device, the turbine also operates at an efficiency,

\[ \eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{T_3 - T_4}{T_3 - T_{4s}} \]  

As mentioned, the air exiting the turbine directly enters the conditioned space. The desired effect results from the two masses of air at different temperatures mixing to yield the room temperature at state 4_{RM}. The same
effect occurs in the other cycles with the air passing over the heat exchanger mixing with the room air. The absence of a heat exchanger and the refrigerant (air) directly influencing the space make this process unique. If the cycle uses inside room air, state $4_{RM}$ coincides with state 1.

### Coefficient of Performance

The Brayton cycle coefficient of performance for cooling is:

$$ COP_c = \frac{h_{RM} - h_4}{(h_2 - h_1) - (h_3 - h_4)} = \frac{T_{RM} - T_4}{(T_2 - T_1) - (T_3 - T_4)} $$

### Thermal Resistance

Determination of the thermal resistance for the Brayton cycle takes special consideration. Even without a heat exchanger,

$$ Q_c = \frac{1}{R} (T_{RM} - T_4) = \dot{m}C_p(T_{RM} - T_4) $$

(23)

to remain consistent with the other devices. Obviously the temperature difference cancels when solving for $R$ leaving,

$$ R = \frac{1}{\dot{m} C_p} $$

(24)

The resistance results from the amount of air injected and its specific heat. By increasing one or both the resistance decreases, a logical conclusion with only air mixing occurring.
Characterizing the heat rejection also has complications not encountered before. The heat exchange process does not occur isothermally (at constant temperature) as in the previous cycles where the refrigerant was in the saturation region. To accurately examine the heat exchanger effectiveness, the log mean temperature difference relation must be used rather than just the strict temperature difference.

\[ Q_H = \frac{1}{R_{in}} \Delta T_{in} = m c_p (T_3 - T_2) \]  

\[ \Delta T_{in} = \frac{(T_{AMB} - T_3) - (T_{AMB} - T_2)}{\ln \left( \frac{T_{AMB} - T_3}{T_{AMB} - T_2} \right)} \]  

As done previously, the resistances on both sides are set equal to narrow the scope of the problem and remain consistent. From the three preceding equations, setting \( R_{in} \) equal to \( R \) results in

\[ \ln \left[ \frac{T_{AMB} - T_3}{T_{AMB} - T_2} \right] = -1 \]  

**Stirling**

The Stirling cycle heat pump consists of the following components shown in Figure 8: 1. expansion space, 2. compression space, 3. cold heat exchanger, 4. hot heat exchanger, and 5. regenerator.
Figure 8. Stirling Schematic Diagram

Component Description

1. Expansion Space

The displacer motion controls the expansion space volume. Expansion of the gas absorbs heat through the cold heat exchanger.
2. Compression Space

The relative motion between the piston and the displacer controls the compression space volume. Compression of the gas rejects heat through the hot heat exchanger.

3. Cold Heat Exchanger

The cold heat exchanger absorbs heat from the low temperature reservoir.

4. Hot Heat Exchanger

The hot heat exchanger rejects heat to the high temperature reservoir.

5. Regenerator

The regenerator stores heat from the gas during one part of the cycle and returns the heat during another part. The regenerator represents the most important part of the Stirling cycle by making the high theoretical efficiencies possible.

Operational Description

The operation of the continuous Stirling cycle machine is somewhat complicated. In the other cycles, relatively simple models developed from the temperature-enthalpy diagram describe the actual embodiment. Unlike those cycles, the actual Stirling system does not follow the ideal temperature-
enthalpy diagram, Figure 9, closely. For example the ideal cycle considers the heat rejection and absorption steps as isothermal. In a real machine, gas expansion and compression more closely approximates adiabatic conditions requiring the addition of heat exchangers which decrease performance by adding dead space (space which does not undergo expansion or compression) (Urieli 1984). Reader and Hooper (1983) along with Organ (1992) discuss this further, illustrating an idealized P-v (pressure versus specific volume) diagram for a Stirling engine. This more realistic, though still highly idealized, cycle replaces each isothermal process with an adiabatic process and an isochoric process. The model for the Stirling uses this basic principle, but does not trace the state diagram. The model develops the conservation equations describing the system and uses the specific mechanical drive volume variation (e.g. sinusoidal) as an input parameter to drive the solution. This procedure renders more accurate results since such a large deviation exists in this system between reality and the ideal conditions described.

Numerous equations describe the system. The computer model for this cycle came from Stirling Cycle Engine Analysis (Urieli 1984). While the program was modified the equations given in that text remain unchanged and it should be consulted for further reference.
Referring to Figures 8 and 9, the following outlines the ideal cycle steps for a general description:

State 1 to 2

The expansion space volume decreases, fluid pressure and temperature increase. Heat transfers to the fluid from the regenerator while the fluid moves to the compression space.

State 2 to 3

The piston compresses the fluid while the displacer reduces the expansion space volume to a minimum. The hot heat exchanger rejects heat to the high temperature reservoir.
State 3 to 4

The piston and displacer come together reducing the compression space volume to a minimum, fluid pressure and temperature decrease. The regenerator absorbs heat while the fluid moves to the expansion space.

State 4 to 1

The expansion space volume increases expanding the fluid. The cold heat exchanger absorbs heat from the low temperature reservoir.

The transient nature of the fluid flow and the complicated gas dynamics make the cycle difficult to model numerically and virtually impossible to solve analytically. In fact, the Schmidt analysis done by Gustav Schmidt in 1871 provides the only analytical solution. This solution relies on the idealized assumption of isothermal conditions and allows only sinusoidal volume variations. The isothermal condition does not describe the actual operation, as discussed previously, rendering the solution only marginally useful.

Coefficient of Performance

The COP again equals the ratio of the useful effect to the rate of work input.
Thermal Resistance

As with the previous discussions, an overall thermal resistance was added to each heat exchanger to more realistically portray the operation between the two known temperature reservoirs.

Thermoelectric

In every other heat pump discussed, a fluid transports the heat from one temperature reservoir to the other. In thermoelectric devices, electronic phenomena provide the mode of heat transfer. The device consists of the following: 1. P-type semiconductor, 2. N-type semiconductor, 3. heat source, 4. heat sink, and 5. direct current (DC) power source.

Component Description

1. P-type Semiconductor and 2. N-type Semiconductor

The two semiconductors provide the dissimilar properties necessary to exhibit thermoelectric behavior.

3. Heat Source

The heat source absorbs heat from the low temperature reservoir.

4. Heat Sink

The heat sink rejects heat to the high temperature reservoir.
5. Power Source

The DC voltage source supplies the energy necessary to pump the heat.

![Diagram of Thermoelectric Schematic Diagram](image)

**Figure 10.** Thermoelectric Schematic Diagram

**Operational Description**

Referring to Figure 10, the voltage source’s positive terminal connects to the N-type semiconductor, the N-type and P-type semiconductors are joined electrically, and finally the other side of the P-type semiconductor connects to the voltage source’s negative terminal completing the circuit. The side of each semiconductor connected to the power source provides the location of the heat sink for heat rejection. The P-N junction provides the location of the heat source for heat addition.
The Seebeck effect relates the voltage drop across the P-N junction to a temperature difference.

\[ \Delta V = \alpha_{P-N} \Delta T \]  

(28)

The constant of proportionality, \( \alpha_{P-N} \), is a material property.

The Peltier effect pumps heat from one junction to the other. The rate of heat transfer is proportional to the current,

\[ Q_{P-N} = \pi_{P-N} I \]  

(29)

where

\[ \pi_{P-N} = \alpha_{P-N} T \]  

(30)

Three terms comprise the heat absorbed and rejected by the module: the heat generated by the electrical circuit, the heat transported via the Peltier effect, and the heat conducted across the device. The total heat from the circuit equals the electrical resistance and the length to area ratio multiplied by the current squared,

\[ Q_{gen} = \left( R_p \frac{L_p}{A_p} + R_N \frac{L_N}{A_N} \right) I^2 = R_{P-N} I^2 \]  

(31)

The electrical resistances for the two semiconductors add directly because the electric circuit connects in series. From Fourier's law, the heat conducted
equals the material’s thermal conductivity and the area to length ratio
multiplied by the temperature difference,

\[ Q_{\text{cond}} = \left( k_p \frac{A_p}{L_p} + k_N \frac{A_N}{L_N} \right) (T_H - T_C) \]  \hspace{1cm} (32)

The thermal conductances for the two semiconductors add directly because the thermal circuit connects in parallel. Summing these terms, the heat absorbed,

\[ Q_C = -\frac{1}{2} Q_{\text{gen}} + \alpha_{p-N} IT_C - Q_{\text{cond}} \]  \hspace{1cm} (33)

Similarly the heat rejected,

\[ Q_H = \frac{1}{2} Q_{\text{gen}} + \alpha_{p-N} IT_H - Q_{\text{cond}} \]  \hspace{1cm} (34)

with the heat generation term dissipated equally through each.

**Coefficient of Performance**

The cooling COP equals the useful energy divided by the rate of electrical work input, or power.

\[ COP_c = \frac{Q_C}{\text{Power}} = \frac{Q_C}{I[\alpha_{p-N}(T_H - T_C) + IR_{p-N}]} \]  \hspace{1cm} (35)

The denominator accounts for the power required for the Seebeck effect and the power dissipated due to the resistivity of the circuit, respectively.
Thermal Resistance

The same overall thermal resistance term, $R$, describes this system.

The absence of an internal fluid circulating through a heat exchanger does not change the similarity with regard to the other resistances.
CHAPTER 3

COMPUTER MODELS

Vapor Compression

Software from the book *Thermodynamic Properties* models the Rankine refrigeration cycle (Software Systems Corp. 1988). The Quick BASIC program was modified to include thermal resistance allowing the user to specify the ambient and room temperatures rather than the condenser and evaporator temperatures, respectively.

The model input includes: the two temperatures mentioned, the thermal resistance of the condenser and evaporator, the degrees of superheat for the fluid entering the compressor, the compressor efficiency, the load, and the refrigerant. The program presents a choice of eleven refrigerants: R-11, R-12, R-13, R-14, R-22, R-23, R-113, R-114, R-500, R-502, and NH₃ (ammonia). The most notable absence from this list is R-134a, the present favorite for future use in this cycle. A more current software package would no doubt include this refrigerant. Hand calculations and literature (Petersson
and Thorsell 1989, Devotta and Gopichand 1991) insured the quantities for the comparison fell within upper and lower bounds found using the previously mentioned refrigerants. The omission of this refrigerant does not present a significant setback because this study compares broad concepts, not the specifics of any one cycle.

The outside ambient temperature and the effect of thermal resistance represent the most important parameters. The effect of ambient temperature can be evaluated easily. The thermal resistance, though, is more complicated because the values are not so straightforward and their effect depends on the cooling load. Recall the governing equation presented in Chapter 2

\[ Q_c = \frac{1}{R} (T_{room} - T_{evap}) \]

or alternatively

\[ \Delta T = Q_c R \]

By selecting a range of \( \Delta T \) the effect of two parameters, the cooling load and thermal resistance, are lumped into a single term producing thermodynamically equivalent but more generic results. The value of the cooling load and thermal resistance independently does not matter so long as the product remains constant. Note that if the pressure drop were included in the analysis this simplification would break down because the higher mass
flow rates associated with larger cooling loads would require more input energy to achieve.

**Complex Compound Absorption**

The complex compound absorption model resulted from further modification of the above program. The inputs remain the same with two exceptions: 1. the compressor efficiency can be omitted and 2. the constants for the complex compound equation must be added.

**Brayton**

The model for the Brayton cycle was written in FORTRAN 77. The inputs include: turbine efficiency, compressor efficiency, ambient temperature, room temperature, thermal resistances, cooling load, and whether inside or outside air enters the cycle.

**Stirling**

One program evaluating the Stirling cycle comes from *Stirling Cycle Engine Analysis* (Urieli, 1984). The program allows three types of analysis isothermal, adiabatic, and quasi-steady. The quasi-steady model most accurately predicts real cycle operation, although the results remain somewhat idealized compared to experimental measurements.
The FORTRAN 77 code given in the book overwrote memory locations inexplicably, stopping program execution. A modified version significantly streamlined the program allowing it to run properly. The program provides a detailed simulation of a variety of Stirling engines, a sinusoidal, rhombic, or yoke drive with a choice of heat exchangers. The working fluids available include air, hydrogen, and helium. The dimensions of all components (e.g. diameters, lengths, volumes, etc.) must be specified as well as the frequency, pressure, and temperatures. Further modifications to the program accounted for the thermal resistances and the load.

_Stirling and Vuilleumier Heat Pumps_ provides two programs for simulating an ideal, isothermal Stirling heat pump (Wurm, 1990). One program uses a piston-piston configuration, the other a piston-displacer configuration. Pressure, temperatures, volumes, and the amount of working fluid must be specified. The usefulness of these programs seems limited due to the type of simulation. An isothermal analysis grossly approximates the real cycle, for example as stated earlier the machine does not need heat exchangers. An adiabatic simulation more accurately describes a real engine, although still highly idealized. Neither approach provides much useful
information for more realistic comparisons. As a result, only the modified quasi-steady Urieli program supplied useful information for this study.

**Thermoelectric**

William P. Graebel, Ph.D., P.E. wrote the FORTRAN 77 program modeling the thermoelectric device. The program requires: the manufacturer, thermocouples per module, design current or geometric factor (manufacturer specifications), ambient temperature, room temperature, thermal resistance, and load. Manufacturer data allows the program to calculate the necessary material properties.
CHAPTER 4

DISCUSSION

Individual Results

The programs were all run using the same input parameters to insure an equal basis for comparison. A summary of these values follows:

Room Temperature: $T_{ROOM} = 22^\circ C \approx 72^\circ F$

Outside Ambient Temperature: $35 \leq T_{AMB} \leq 46^\circ C$

or $95 \leq T_{AMB} \leq 115^\circ F$

Temperature Difference ($\Delta T = Q_C R$): $0 \leq \Delta T \leq 20^\circ C$

or $0 \leq \Delta T \leq 68^\circ F$

from which for convenience: $Q_C = 20 \text{ kW} = 68,260 \text{ Btu/hr}$

and $0 \leq R \leq 1 \text{ \degree C/kW}$ or $0 \leq R \leq 0.001 \text{ \degree F/Btu/hr}$

Compressor and Turbine Efficiencies: $\eta = 0.80 = 80\%$

In this section, detailed results for each device illustrate individual performance. The next section draws comparisons between all the devices.
Vapor Compression

Figures 11 and 12 illustrate the performance of the vapor compression cycle for a variety of fluids. Only five fluids were chosen due to the temperatures involved and the close grouping of the curves. The coefficient of performance decreases rapidly with increasing ambient temperature at low temperature differences. As $\Delta T$ increases, the curves level off while the COP drops dramatically.

**Figure 11.** Vapor Compression Cycle COP vs. $T_{AMB}$ at $\Delta T = Q_c R = 0^\circ C$
The order of the fluids with respect to COP remains consistent throughout with R-11 exhibiting the highest COP and R-22 the lowest. The most environmentally notorious fluid, R-12, lies in the middle of the group. From this point the discussion will concentrate only on R-11 due to its high performance eliminating the obvious redundancy associated with examining all the operating fluids.
Figure 13 shows a more detailed illustration of COP versus $T_{AMB}$ for R-11 as $\Delta T$ increases. Note the large change in COP for $\Delta T = 0^\circ$ C over the ambient temperature range and how that difference progressively becomes less as $\Delta T$ increases.

![Figure 13. Vapor Compression COP vs. $T_{AMB}$ for R-11](image)

The COP versus $\Delta T$ plot for R-11, Figure 14, provides a clearer picture of increased resistance and/or load on the performance. The COP drops
significantly at low $\Delta T$ while changing less at larger $\Delta T$ where the curves for various ambient temperatures approach one another. This suggests increasing the temperature difference or the ambient temperature has a diminished effect. Extending the range for $\Delta T$ would not add appreciably to the understanding or usefulness of the results. The COP diminishing by a factor of 6 demonstrates the extreme sensitivity of this system to the temperature difference and thus to the thermal resistance along with the cooling load.

Figure 14. Vapor Compression COP vs. $\Delta T$ for R-11
Complex Compound Absorption

The complex compound absorption simulation used strontium bromide as the complex compound and ammonia as the refrigerant. The constants associated with this compound can be entered into equation (8), yielding

\[
\log P = \left(4 - \frac{1200}{T}\right) \log 15 \quad (8a)
\]

Figure 15. Complex Compound Absorption COP vs. \(T_{AMB}\)

In Figure 15, COP versus ambient temperature, the COP changes only slightly, 5%, with increasing \(T_{AMB}\). Graebel et al. (1991) give the COP as on
the order of 0.4. The results compare well with that figure considering the idealized assumption of complete heat recovery with respect to raising the vessel temperature.

Figure 16 shows the curves of COP versus $\Delta T$ almost paralleling one another in contrast to Figure 14 for the vapor compression cycle. The COP decreases a modest 14%. Neither the temperature difference, $\Delta T$, nor the ambient temperature seems to dominate the trend.

![Figure 16. Complex Compound Absorption COP vs. $\Delta T$](image-url)
Brayton

The Brayton results considered only indoor air for cycle operation which as expected gave higher performance than outdoor air. The plot of COP versus ambient temperature, Figure 17, deviates from the established tendency. COP still decreases with increasing ambient temperature; but, the influence of ΔT does not follow previous expectations.

![Brayton COP vs. T_{AMB}](image)

**Figure 17.** Brayton COP vs. $T_{AMB}$
First, COP = 0 for $\Delta T = 0$ unlike the other examples where the maximum COP occurred at $\Delta T = 0$. Close examination of the equations and the temperature-entropy diagram support the validity of this result. Next, the COP progressively gets larger with an increase in the temperature difference until a peak occurs near $14^\circ C$ as evident in Figure 18. The influence of the temperature difference on COP diminishes as $\Delta T$ increases. Preceding to the peak the temperature difference obviously dominates the performance.

![Brayton COP vs. $\Delta T$](image)

**Figure 18.** Brayton COP vs. $\Delta T$
The results show the same trend as those presented by Sisto (1978) exhibiting a maximum COP. However, Sisto plots COP versus the isentropic temperature ratio for a heating application. The isentropic temperature ratio relates directly to the thermal resistance equating the independent parameters, i.e. increasing the thermal resistance increases the temperature ratio. A spot check to compare specific numbers required the Brayton heating COP. Selecting input comparable with Sisto’s parameters, the program calculated a heating COP of 1.21 compared to Sisto’s 1.5. The same compressor and turbine efficiencies were used (80%); however, Sisto’s heat exchanger effectiveness equaled 0.8 while the program’s effectiveness equaled 0.63. Given this discrepancy, the results seem highly compatible.

Stirling

As previously mentioned, the Stirling program required specific design specifications, e.g. volumes, diameters, materials, etc. The influence of these parameters would distinguish between different Stirling cycle embodiments; however, in this comparison their contribution does not add any insight and actually only complicates the situation. Since the importance of these parameters is minimal, most any realistic values should give adequate results for a more generic Stirling simulation. Rather than designing a complete
machine, a monumental task, one of the examples given in Urieli's text will be used changing only the parameters important to this study.

The text gives all the necessary information for three Stirling engines previously developed, the Ford-Philips 4-215 sinusoidal drive, the General Motors GPU-3 rhombic drive, and the Ross yoke drive. Of these the Ford-Philips 4-215 engine was chosen for this analysis because the engine size was compatible with the assigned value of $Q_c$. Some concern may arise from the fact that these are engines and not heat pumps. While there are differences in such things as construction materials and parasitic heat losses the method of solution remains the same. Again these specific design differences will not jeopardize the results required for this comparison.

From the program's standpoint, the roles of the expansion space and compression space reverse insuring that the system absorbs and rejects heat through the correct heat exchangers for a heat pump. The frequency was set to 1500 rpm, an arbitrary, but realistic figure. An important point to note: the performance depends on the frequency and can vary dramatically for different fluids. The performance for air, or nitrogen, plummets as frequency increases (refer to Reader and Hooper 1983). This factor was not investigated to avoid further complication at this point. The fluids should be compatible for
comparison at the specified operating frequency according to the literature and several trials.

The cycle pressure and frequency are analogous to the mass flow rate in the vapor compression cycle. Equation (4) yields the vapor compression mass flow rate directly. The Stirling does not accommodate this type of simplicity. With the frequency fixed, the average operating pressure, a required input, was found by iteration based on \( Q_C \). Figure 19 plots the COP as

![Figure 19](image)

**Figure 19.** Stirling COP vs. \( T_{AMB} \), Pressure Drop Excluded
a function of ambient temperature excluding the pressure drop for the three available fluids. Hydrogen yields the highest COP followed by air and then helium.

Figure 20 more clearly identifies the performance of the Stirling by showing only a single gas. Hydrogen was the chosen working fluid based on the level of performance. The COP decreases almost linearly with respect to ambient temperature dropping moderately.

**Figure 20.** Stirling COP vs. $T_{AMB}$, H$_2$ with Pressure Drop Excluded
The plot of COP versus $\Delta T$ shows the same trend as in the vapor compression case though the initial slope is less severe. The curves in Figure 21 seem to approach one another as $\Delta T$ increases. The COP reduces nearly one-half. A higher sensitivity to $\Delta T$ than to the ambient temperature appears apparent.

**Figure 21.** Stirling COP vs. $\Delta T$, H$_2$ with Pressure Drop Excluded
The pressure drop up to this point has been neglected. The validity of this assumption becomes questionable because the Stirling cycle's regenerator resembles a porous media, consisting of a close-packed wire mesh. Compared to flow in a tube, a substantial pressure drop should result from this type of flow detrimentally affecting the performance. The program gives the option of including the dissipation terms. Figure 22 plots COP versus ambient temperature for hydrogen with the pressure drop included. Notice the COP has dropped by a factor of 2 compared to the previous case. The lowest curve in Figure 20 ($\Delta T = 20$) has higher values than the highest curve in Figure 22 ($\Delta T = 0$). The pressure drop impacted the results more significantly than the largest value of $\Delta T$. This affirms the presumption that including the pressure drop significantly decreases the performance.

The results with pressure drop included provide the most realistic case for a comparison with experimental measurements. Otaka et al. (1993) give the COP for a 100 W, 700 rpm refrigerator as 1.05. The device used helium as the working fluid with a cold heat exchanger wall temperature of 253° K and hot heat exchanger wall temperature of 303° K. Although the operating parameters differ moderately, the experimental COP lies within the curves shown in Figure 22 suggesting reasonable output.
A Melcor (manufacturer) thermoelectric device was chosen from a company catalog. The deciding factor was the module’s high heat pumping capacity. The module contains 31 thermocouples and has a geometric factor of 1.255. Other modules were tried; but, the COPs were equal or slightly less suggesting the specific module picked does not substantially alter the results.
Figure 23. Thermoelec. COP vs. $T_{AMB}$, Melcor 31 thermocouples, $G = 1.255$

Figure 23 shows an interesting characteristic; the module cannot accommodate large temperature differences. The curve for $\Delta T = 7^\circ C$ stops at an ambient temperature of $41^\circ C$. The temperature difference across the faces of the module has become too large and device can no longer pump heat between the two temperature reservoirs. Specifically, the Peltier effect cannot overcome the more dominant joule heating and heat conduction.
The COP versus $\Delta T$ plot, Figure 24, demonstrates the unusual trend associated with increasing the temperature difference. The COP rapidly decreases with $\Delta T$. Instead of leveling off, the curves become steeper just before the device no longer performs. $\Delta T$ dominates the performance; although, both parameters have magnified importance in this case with the direct tie to joule heating and heat conduction. The increased face temperatures correspond to serious performance deficiencies.

**Figure 24.** Thermoelec. COP vs. $\Delta T$, Melcor 31 thermocouples, $G = 1.255$
Stockholm and Stockholm (1992) provide a 3-D plot of COP versus temperature difference across the module and electrical current. The maximum COP (over the series of electrical currents) for the module face temperature difference range of interest (≥ 13° C) decreased from 3 to 0. The limiting value 13° C came from subtracting the room temperature (22° C) from the lowest ambient temperature (35° C). The module face temperature difference must increase as ΔT increases from 0. More exact COP values were not discernible from the graph presented. In general, these values correspond well to the COPs determined in this analysis.

**Comparisons**

Figure 25 compares all the devices on a plot of COP versus ambient temperature for ΔT = 0° C. The vapor compression cycle obviously far surpasses the other concepts in this idealized case. The Stirling cycle without a pressure drop takes second place followed by the thermoelectric device, the Stirling cycle with a pressure drop, the complex compound absorption, and finally the Brayton cycle. This idealized case somewhat unfairly portrays the Brayton cycle. Due to the nature of the equations, the COP = 0; however,
even the optimum performance of the Brayton cycle would rank last thus not affecting its relative position.

![Graph](image.png)

**Figure 25.** COP vs. $T_{AMB}$, $\Delta T = 0^\circ C$

The next graph, Figure 26, compares the devices with $\Delta T = 7^\circ C$. The overall COP for the vapor compression decreased by a factor of 2, but the cycle still leads the pack. The most important aspect of this graph is the performance of the thermoelectric device falling below both the Stirling cycle
without a pressure drop and the complex compound absorption. The thermoelectric module cannot even perform above an ambient temperature of 41° C. The graphical depiction translates the nonperformance to a COP = 0.0. The Brayton cycle COP increased from the previous case but still remains low when compared to the other devices.

Figure 26. COP vs. $T_{AMB}$, $\Delta T = 7^\circ C$
In Figure 27 $\Delta T = 14^\circ$ C where according to the individual results the maximum Brayton cycle COP occurs. Unfortunately, the Brayton cycle still significantly lags behind the others. The thermoelectric device COP has gone to 0 for all ambient temperatures. The vapor compression COP decreased by almost a factor of 2 again, but remains double the value for the Stirling cycle without a pressure drop.

![Figure 27: COP vs. $T_{AMB}, \Delta T = 14^\circ$ C](image-url)
Finally Figure 28 presents the comparison for the highest temperature difference, $\Delta T = 20^\circ C$. The vapor compression COP still decreases, though not as dramatically. The device ranking remains the same.

**Figure 28.** COP vs. $T_{AMB}$, $\Delta T = 20^\circ C$
CHAPTER 5

CONCLUSION

This study examined various thermodynamic devices for the purpose of refrigeration. Chapter 2 presented the theory behind each concept along with the appropriate equations. Computer models generated the results used to characterize each device individually. The devices were then compared to one another to give insight into the interrelationship of the present technologies.

The vapor compression cycle certainly maintains the best coefficient of performance throughout. With higher cold side temperature differences and thus resistances and/or cooling loads, the margin between devices narrows though no real challenge to the vapor compression’s superiority arises. The environmental concerns regarding the refrigerants could be the one equalizing factor negating the performance advantages. Presently, the new refrigerant R-134a appears environmentally sound; however, the fluid has not been investigated fully and in some cases only over time can any determination be
made. Perhaps some undetectable problem exists or a problem no one has thought to investigate. The closest alternative, the Stirling cycle, poses no environmental problems using common, natural gasses. The thermoelectric device uses no fluid at all. Performance criteria alone cannot determine the best solution; though it remains one of the most important considerations.

From a practical engineering standpoint, each device has particular advantages and disadvantages aside from the performance aspect. The lack of a contained fluid such as in the Brayton cycle and the thermoelectric device provides a substantial benefit for long term storage. Conversely, the Stirling cycle requires a highly pressurized fluid which would perform poorly over long term storage with certain leakage through degrading seals. The problems would compound for a hydrogen operated heat pump where the gas could diffuse through the metal causing metallurgical problems. The complex compound absorption and thermoelectric device contain no moving parts a significant advantage for continued maintenance. The thermoelectric device, advantageous in the previous examples, degrades with continued on and off cycling a generally common occurrence in air conditioning applications. This short summary of advantages and disadvantages presents other considerations
which would go into the decision making process regarding the application of these devices.

Table 1 outlines the major advantages and disadvantages of each device. Every application carries unique pros and cons for example residential, mobil (automobile), long term storage between uses, and industrial (storage warehouses) would each have different important criteria.

**Table 1. Device Advantages and Disadvantages**

<table>
<thead>
<tr>
<th>Device</th>
<th>Advantages</th>
<th>Disadvantages</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vapor Compression</td>
<td>Very good performance</td>
<td>Environmentally unsound refrigerants</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Refrigerant may leak out in storage</td>
</tr>
<tr>
<td>Complex Compound</td>
<td>Few moving parts</td>
<td>Non-cyclic operation</td>
</tr>
<tr>
<td>Absorption</td>
<td>Environmentally acceptable refrigerant</td>
<td>Toxic refrigerant</td>
</tr>
<tr>
<td></td>
<td>Heat driven</td>
<td>Refrigerant may leak out in storage</td>
</tr>
<tr>
<td>Brayton</td>
<td>Air used as the refrigerant</td>
<td>Poor performance</td>
</tr>
<tr>
<td></td>
<td>Can run off of available compressed air</td>
<td>Many mechanical parts</td>
</tr>
<tr>
<td>Stirling</td>
<td>Good performance possible</td>
<td>Mechanically complex</td>
</tr>
<tr>
<td></td>
<td>Environmentally favorable refrigerants</td>
<td>High pressures</td>
</tr>
<tr>
<td></td>
<td>Heat driven</td>
<td>Close tolerances</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Seals degrade in storage</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Helium and hydrogen can leak out over time</td>
</tr>
<tr>
<td>Thermoelectric</td>
<td>No refrigerant</td>
<td>Precise manufacturing</td>
</tr>
<tr>
<td></td>
<td>No moving parts</td>
<td>Only a small temperature difference across module</td>
</tr>
<tr>
<td></td>
<td>Compact</td>
<td>Large DC voltage</td>
</tr>
<tr>
<td></td>
<td>Quiet</td>
<td>On and off cycling</td>
</tr>
<tr>
<td></td>
<td>No degradation with time</td>
<td>reduces reliability</td>
</tr>
</tbody>
</table>
The lack of major research hinders the current alternatives. The vapor compression cycle received an incredible amount of attention over an entire era because of the commercial development, for a good reason based on the results of this comparison. Considering the new information, the other devices warrant more attention which may increase their competitiveness through novel developments. The Stirling for example is much more complicated than the vapor compression just from a basic understanding. With improved understanding and a concentrated effort perhaps improved performance could be engineered.

In this study, the number of parameters was limited to focus on a brief, but useful result. Changing temperatures and allowing the thermal resistances on the cold side to be different from the hot side would generate new insight. This study provided important trends and comparisons from which more advanced investigations could build and benefit.
APPENDIX

FLOWCHARTS

This section contains brief flowcharts for each of the modified computer programs used in this study. The flowcharts show the relationships between the major inputs, outputs, and calculations. Where appropriate a demarcation line designates the modified portion from the original program.
INPUT Refrigerant, Ambient Temperatures, Deg. of Superheat, and Compressor Eff.

CALCULATE Enthalpies, Entropies, Volumes, Cooling and Heating COPs

INPUT Cooling Load, Thermal Resistance, and Outdoor Ambient Temperature Range

CALCULATE New Condenser and Evaporator Temps from Thermal Resistance and Load

CALCULATE Enthalpies, Entropies, Volumes, Cooling and Heating COPs

CALCULATE New Condenser Temperature

If $|T_{\text{cond}} - T_{\text{cond, old}}| < \text{error}$ then Yes, OUTPUT Cooling COP

If $T_{\text{amb}} > T_{\text{amb, max}}$ then Yes, End

Figure 29. Vapor Compression Flowchart
INPUT Ambient Temperatures (Existing) and Complex Compound Constants a and b (New)

CALCULATE Enthalpies, Entropies, Volumes, Cooling and Heating COPs

INPUT Cooling Load, Thermal Resistance, and Outdoor Ambient Temperature Range

CALCULATE New Condenser and Evaporator Temps from Thermal Resistance and Load

CALCULATE Enthalpies, Entropies, Volumes, Cooling and Heating COPs

CALCULATE New Condenser Temperature

\[ |T_{\text{COND}} - T_{\text{COND, OLD}}| \leq \text{error} \]

Yes

OUTPUT Cooling COP

\[ T_{\text{AMB}} > T_{\text{AMB, MAX}} \]

Yes

End

Figure 30. Complex Compound Absorption Flowchart
INPUT Ambient Temperature Range, Room Temperature, Cooling Load, Thermal Resistance, and choose Inside or Outside Air

CALCULATE Temperature of each State Point and Cooling COP

OUTPUT Cooling COP

Figure 31. Brayton Flowchart
Start

INPUT Type of Drive and Physical Specifications (volumes, lengths, etc.)

INPUT Type of Heater/Cooler with Physical Specifications

CALCULATE Void Volumes, Wetted Areas, and Hydraulic Diameters

INPUT Type of Regenerator with Physical Specifications inc. Porosity and Wire Dia.

INPUT Gas, Mean Pressure, Frequency, Room and Ambient Temperature, Thermal Resistance, and Cooling Load

CALCULATE Schmidt Analysis Results

INPUT Convergence Accelerating Factor, No. of Increments, Printouts, and Cycles, and Pressure Drop Included or Excluded

Continued in Figure 33

**Figure 32.** Stirling Flowchart (part a)
Continued from Figure 32

- CALCULATE Volume Variation and Solve the Algebraic/Differential Equation Set using 4th Order Runge-Kutta for the Differential Equations
- CALCULATE New Regenerator Matrix Temperatures

- Number of Cycles > Set Value
  - No
  - Yes
    - CALCULATE New Heat Exchanger Temperatures from the Thermal Resistance and the Calculated Heats (Absorbed and Rejected)
      - Both HX
        - No
          - |T_{HX} - T_{HX, OLD}| < error
          - CALCULATE New Mean Pressure
        - Yes
          - |Q_{COLD} - Q_{LOAD}| < error
            - No
              - Yes
                - OUTPUT Cooling COP
            - Yes
              - End

Figure 33. Stirling Flowchart (part b)
INPUT Manufacturer, Number of Thermocouples per Module, the Geometric Factor or Design Current (depending on manufacturer), Ambient Temperatures, Thermal Resistance, and Cooling Load

CALCULATE Properties of the Module and Face Temperatures

CALCULATE Cooling and Heating Load per Module, Number of Modules, Cooling COP, and Hot Side Thermal Resistance for a Range of Electric Currents
Find Maximum Hot Side Thermal Resistance

CALCULATE New Hot Face Temp.
Increase if \( R_{\text{HOT, MAX}} < R_{\text{COLD}} \)
else Decrease

\[
\left| R_{\text{HOT, MAX}} - R_{\text{COLD}} \right| < \text{error}
\]

OUTPUT Cooling COP

\[
T_{\text{AMB}} > T_{\text{AMB, MAX}}
\]

End

Figure 34. Thermoelectric Flowchart
BIBLIOGRAPHY


