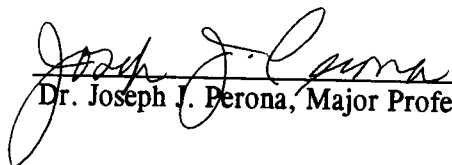
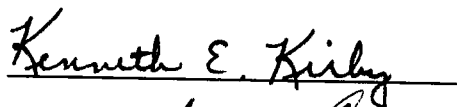
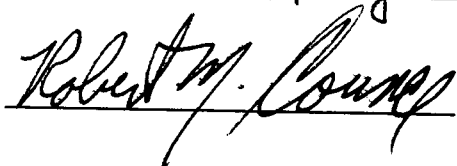


To the Graduate Council:

I am submitting herewith a thesis written by Patricia Welesko Adcock entitled "Component Modeling and Preliminary Optimization of LiBr/H₂O Absorption Cycles". I have examined the final copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Chemical Engineering.


Dr. Joseph J. Perona, Major Professor

We have read this thesis and
recommend its acceptance:

Accepted for the Council:


Associate Vice Chancellor and
Dean of The Graduate School

**COMPONENT MODELING
AND PRELIMINARY OPTIMIZATION
OF LiBr/H₂O ABSORPTION CYCLES**

A Thesis
Presented for the
Master of Science
Degree
The University of Tennessee, Knoxville

Patti Welesko Adcock
May, 1995

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Abstract

In an effort to validate Lithium Bromide/Water (LiBr/H₂O) absorption chiller design specifications using theoretical cycle calculations, a computer code was written to calculate overall heat transfer coefficient times area (UA) values in heat exchangers with horizontal tube bundles. Specifically, component models were developed for absorbers, generators, evaporators, and condensers. These component models were linked to the DOE/ORNL-sponsored ABSIM computer model to integrate the design specifications with the cycle analysis. ABSIM is a user-oriented computer model for simulation of absorption systems.

The UA values from the component models were compared to experimentally determined values based on operating data from a double-effect LiBr/H₂O absorption chiller. These comparisons were used as a tool to validate the component models. Since the triple-effect absorption chillers are simply modular extensions of the double-effect absorption machine, it can be inferred that the component models are valid for these cycles.

The calculated UA values matched the experimentally determined UA values within 20% for the evaporator, 10% for the condenser, 15% for the absorber, and within 45% for the generator. The experimental data used for comparisons was primarily taken over a narrow range of operating conditions. It would be desirable to obtain follow-on experimental data over a larger range of load conditions to perform additional comparisons between experimental and calculated values. It is also desirable to obtain data from chillers of alternate designs to perform additional comparisons, and to develop an alternate correlation, either empirical or theoretical, for the generator.

UA values can be used to calculate cycle performance. The utility of valid component UA values was demonstrated using a parametric analysis by varying component UA values and calculating cycle performance using ABSIM. Based on the cycle performance calculations, optimum sized components for the base operating case triple-effect cycle are recommended. The optimum sized components performed at an efficiency comparable to the base case at a lower relative cost across all heat loads. In addition, the optimized cycle offers the ability to run the chiller at much higher heat loads than the base case.

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Acronyms

ABSIM	ABSorber SIMulation Model
ARI	American Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
CFC	Chlorofluorocarbons
COP	Coefficient of Performance = Refrigeration Effect/Net Work Input
DCC	Double-Condenser Coupled
DOE	Department of Energy
EES	Engineering Equation Solver
FY	Fiscal Year
GAX	Generator-Absorber Heat Exchange
HCFC	Hydrochlorofluorocarbons
HFC	Hydrofluorocarbons
Hg	Mercury
H ₂ O	Water
LiBr	Lithium Bromide
LMTD	Log Mean Temperature Difference
NH ₃	Ammonia
NO _x	Nitric Oxides
ORNL	Oak Ridge National Laboratory
PTX	Pressure, Temperature, Concentration
R&D	Research and Development
U	Overall Heat Transfer Coefficient
UA	Overall Heat Transfer Coefficient Times Area

1. Overview

1.1 Background

This thesis project involves the study of the triple-effect absorption chiller, which is the next step beyond the current state-of-the-art double-effect absorption chillers. Absorption chillers are heat-operated refrigeration machines that operate on one of the earliest known principles of refrigeration. The cycle uses a refrigerant (known as the primary fluid), and an absorbent (known as the secondary fluid). The refrigerant (primary fluid) is chemically and physically absorbed by the absorbent (secondary fluid) for the purpose of transferring heat. The evaporation of the primary fluid removes heat, thus providing the refrigeration effect.

The absorption cycle can be compared to the more familiar mechanical compression cycle in that both cycles evaporate and condense a refrigerant liquid at two or more pressures within the unit [1]. The absorption cycle uses a heat-operated generator, a heat-rejecting absorber, and a liquid solution pump. The mechanical compression cycle uses a compressor to increase the refrigerant pressure. The absorption cycle substitutes a physiochemical process for a purely mechanical compressor. Both cycles require energy input for operation--heat and a small amount of mechanical energy for the absorption cycle compared to strictly mechanical energy for the compression cycle.

The application of the absorption cycle for air conditioning has been used for over 50 years. Ammonia-water absorption equipment has been used in large capacity industrial applications that require low temperatures for process cooling. The air-cooled ammonia-water-hydrogen absorption

cycle, which uses ammonia as the refrigerant and water as the absorbent, is used in a domestic refrigerator application to keep food and liquid cold. Lithium bromide/water absorption equipment is used to produce chilled water and also is used to produce hot water for indoor comfort heating, process heating, and domestic purposes. This cycle can also be used as a temperature booster or heat transformer by delivering heat at a temperature higher than that of the driving heat source.

Many single-effect LiBr/H₂O absorption chillers using low pressure steam or hot water as the heat source, have been installed in commercial buildings to produce chilled water for air conditioning. Single-effect systems are also used to cool fluids in industrial processes, often using waste heat to power the system. The thermal efficiency of single-effect absorption systems is low. As natural gas prices increased relative to electricity over the past 20 years, the market for such chillers has decreased significantly. Although the technology is sound, the low efficiency has reduced the cost effectiveness of single-effect systems. Most new single-effect machines are now installed in applications where waste heat is readily available.

The desire for higher efficiencies in absorption chillers led to the development of double-effect LiBr/H₂O systems. These systems have been designed and manufactured to be powered by gas fired combustors or high pressure steam. In the United States, double-effect absorption chillers are used for air conditioning and process cooling in regions where the cost of electricity is high relative to natural gas. Double-effect absorption chillers are also used in applications where high pressure steam is readily available (e.g. district heating), or in areas where seasonal load peaks cause utilities to subsidize gas cooling.

From an environmental perspective, absorption chillers provide several benefits. They use LiBr and H₂O as the working fluids, rather than chlorofluorocarbons (CFCs), hydrochlorofluorocarbons (HCFCs), or hydrofluorocarbons (HFCs), refrigerants that contribute to ozone depletion and/or global warming. For natural gas fired systems, the CO₂ (greenhouse gas) emission from natural gas combustion is relatively low compared to the CO₂ evolved in the production of electricity produced from burning coal. An additional benefit, when using natural gas as the heat source, is the much lower NO_x emissions as compared to coal produced electricity. For these reasons, the market for double-effect absorption chillers is growing, but has not taken over the chiller market. Electrical vapor-compression chillers, the dominant form of competition, offer lower initial production cost.

Three-stage or "triple-effect" absorption chillers are proposed as the next step in the evolution of absorption technology. Triple-effect systems offer the possibility of thermal efficiencies equal to those of electrical chillers, while offering the ability to reduce peak electric loading and capitalize on the environmental benefits. The higher efficiency levels would open wider markets for absorption chillers and help to reduce the investment required in new electrical generation facilities by summer-peaking utilities.

1.2 Absorption: The Beginning

Nairne [2] reported in 1777 on his experiments with sulfuric acid and water. He placed two bowls, one filled with water and one filled with sulfuric acid, side by side under the receiver of an air pump. As air was sucked into the receiver, part of the water evaporated and was absorbed by the sulfuric acid. The temperature of the evaporating water thus decreased until ice was formed.

Simultaneously the sorption heat received by the sulfuric acid could have been used for heating purposes. This was not done at that time, because the primary interest was ice production. In 1810, an apparatus using the same principles was built by John Leslie, who used the machine for ice production in his laboratory. Forty years later, Edmond Carré constructed a sulfuric acid-water engine which was used in coffee-houses in Paris for ice fabrication. In 1859, a breakthrough occurred with the patenting, by Ferdinand Carré, brother of Edmond, of an absorption machine using ammonia-water as the working pair. This patent, and 14 additional patents by F. Carré, described a periodically operating low energy machine for ice fabrication in households and a continuously operating machine for industrial applications. From 1880 on, the vapor compression machines of Carl Linde began to displace the absorption machines. The absorption machines experienced a renaissance in the 1920's. At that time, the Europeans became aware that energy was an expensive commodity and that a great amount of energy could be saved by utilizing waste energy. The first paper on the use of absorption processes for heat pumps was published by E. Altenkirch in 1920, who laid the principles of reversible heating. A tremendous amount of absorption technology currently used today was prescribed in papers by Altenkirch.

The earliest reported absorption refrigeration research in the United States was carried out by McKelvy and Isaacs of the U.S. Bureau of Standards in 1920 [3]. The subject of their study was *"The Identification and Prevention of Non-Condensable Gases in Ammonia-Water Absorption Refrigeration Machines"*. During the 1920's and 1930's, most of the R&D activity in the U.S. on heat-operated absorption refrigeration and air conditioning was carried out by equipment manufacturers. The manufacturers were motivated to search for alternatives to compression style cycles for residential applications due to the limitations of compression air conditioning units: poor load factor, high owning and operating costs, and somewhat noisy performance. On the other

hand, the success of the gas-fired refrigerator demonstrated an attractive potential for a system that operated virtually noise-free, using low cost energy for the era, and with a potential for trouble-free operation. The gas fired refrigerator was used for domestic purposes in the 1920s-1940s. Vapor compression units pushed them out of the market in the 1950s. Today, absorption refrigerators are still used in niche markets such as hotel rooms and recreational vehicles.

By the end of World War II, Servel had completed the development of the first marketable steam-fired, water-cooled, residential size LiBr/H₂O absorption chiller. In addition, Servel began R&D on steam-fired units for the commercial market. The Williams Company designed a 10-ton unit aimed at the small-commercial air conditioning market. Also in this time frame, Carrier completed field testing of a marketable large capacity LiBr/H₂O absorption chiller unit. Various chiller machines using Ammonia/Water or LiBr/H₂O absorption pairs for both commercial and residential use were built and patented during the 1950's, 1960's, and 1970's.

With the onset of the energy crisis in 1974, many new R&D projects emerged. Developments that occurred in Europe are described in Hodgett [4]. Developments that occurred in the United States are reviewed by Hanna and Wilkinson [5]. Several of these programs were sponsored by the United States Department of Energy. This thesis project is an outgrowth of the program initiated by the Department of Energy during this time period.

1.3 Single-Effect Cycle

Figure 1 represents a typical schematic diagram of a single-effect chiller that is commercially available in capacities ranging from 50 to 1500 tons. It is common industrial practice to represent

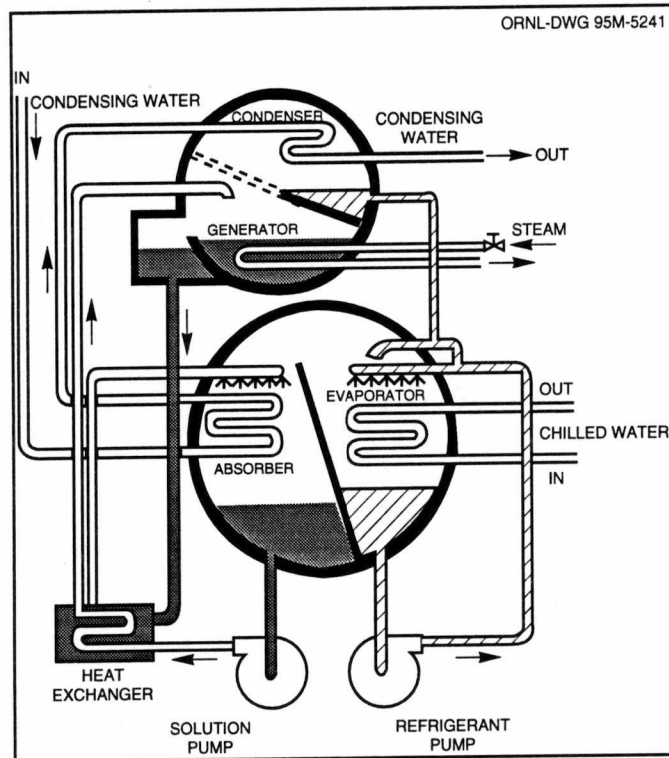


Figure 1. Two-Shell, Single-Effect Lithium Bromide/Water Cycle Absorption Chiller
(Reprinted by permission from ASHRAE Handbook 1988 Equipment published by American Society of Heating, Refrigerating, and Air-Conditioning Engineers.)

the four major components (evaporator, absorber, condenser, and generator) on a pressure versus temperature diagram, as shown in Figure 2. Figure 3 represents a single-effect LiBr/H₂O absorption cycle drawn with the computer model named ABSIM. ABSIM provides the capability to represent all flow paths and component interfaces for each component in the cycle. Additionally, ABSIM provides cycle analysis capabilities as used in this thesis and described in Chapters 3 and 7.

To describe the operation of a single-effect absorption cycle, the schematic diagram in Figure 3 is used. The single-effect cycle uses a LiBr/H₂O solution as an absorbent and water as the

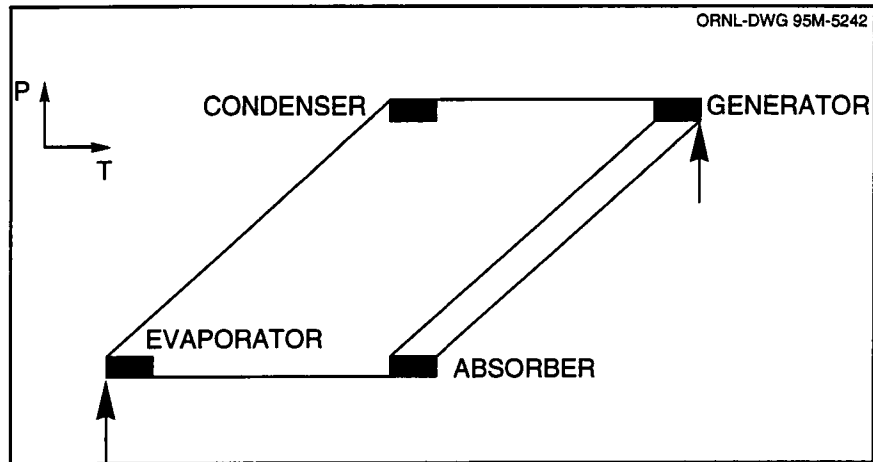


Figure 2. Single-Effect Lithium Bromide/Water Absorption Cycle Represented on a Pressure versus Temperature Diagram

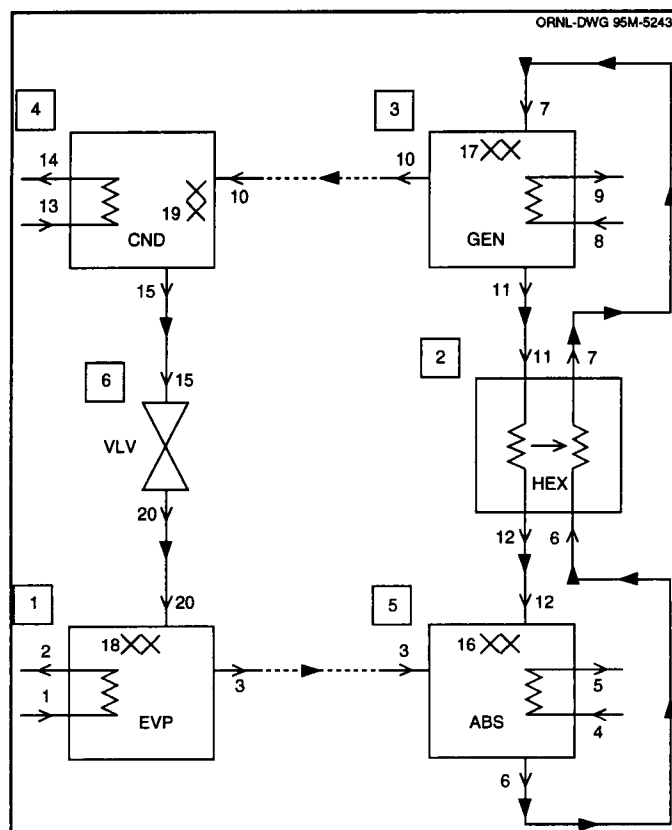


Figure 3. Single-Effect, Lithium Bromide/Water Absorption Cycle (Drawn Using ABSIM)

refrigerant. The cycle operates as follows (see Figure 3): Hot, concentrated absorbent solution (point 11), in equilibrium with the condenser pressure, leaves the generator. This solution is cooled in the solution heat exchanger by the incoming solution and flows to the absorber (point 16). Cold, concentrated absorbent solution absorbs low-pressure refrigerant (point 3) in the absorber, which is in equilibrium with the evaporator pressure. The absorbent solution (point 6) is then pumped to the generator via the solution heat exchanger, where heat is recovered from the solution leaving the generator. Hot, low concentration lithium bromide absorbent solution (point 7) enters the generator, where heat is added to distill the H_2O refrigerant which leaves as a vapor stream (point 10). This high pressure refrigerant vapor condenses in the condenser (point 19). Hot, liquid refrigerant (point 15) is expanded into the evaporator (point 18), where it is evaporated at low pressure and temperature. The heat absorbed by the refrigerant provides the desired cooling effect of the chilled water (points 1 and 2). Cold, low-pressure refrigerant vapor (point 3) is absorbed by the solution in the absorber. Stream 8-9 is either hot water or steam used as the energy supply for the generator. Streams 4-5 and 13-14 are cooling tower water used to remove heat. [6]

A detailed pressure, temperature, concentration diagram is shown in Figure 4. This diagram is helpful to show how the operating conditions of the single-stage chiller can be set. The temperature of the evaporator is set by the chilled water (tubeside) temperature. Nominal output chilled water temperature to the air conditioned space is 45 °F. With an approximate 3 °F approach temperature, the refrigerant in the evaporator (shell side) is 42 °F. The pure refrigerant water line is at the left axis of the curve. Point 1 represents the pure refrigerant temperature of 42 °F. This point fixes the evaporator pressure at approximately 7.5 mm Hg. The condenser temperature is set by the cooling tower water temperature. The nominal temperature for cooling

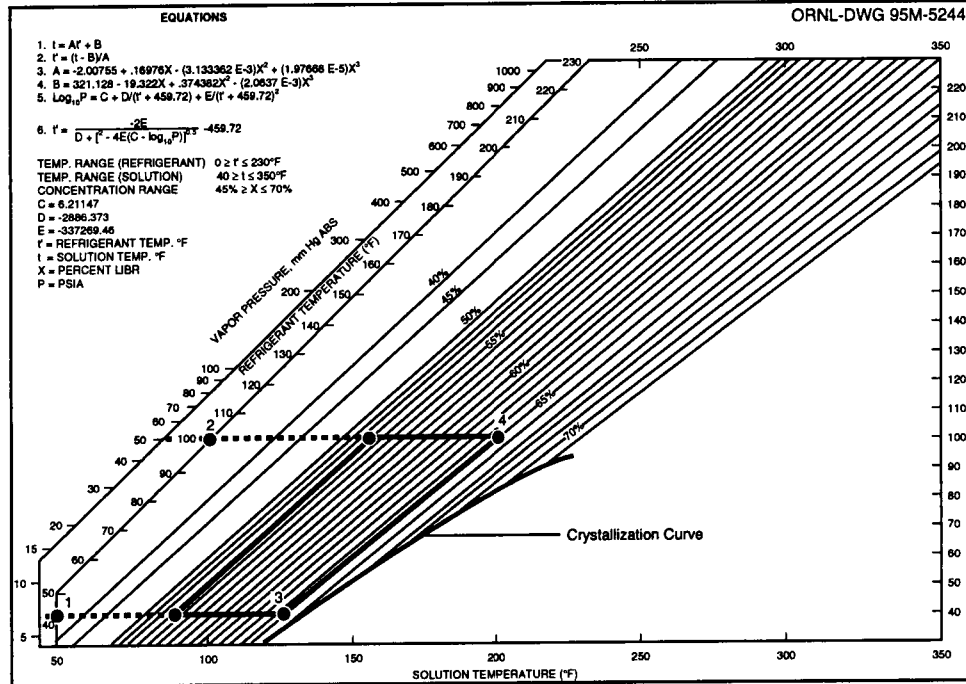


Figure 4. Single-Effect, Lithium Bromide/Water Absorption Cycle Represented on a Pressure, Temperature, Concentration (PTX) Diagram

tower water (tubeside) leaving the condenser is 95°F . With an approximate 5°F approach temperature, the refrigerant in the condenser (shell side) is 100°F . Point 2 represents the pure refrigerant temperature of 100°F . This point fixes the condenser pressure at approximately 50 mm Hg. The absorber operates at the same pressure as the evaporator, and the generator operates at the same pressure as the condenser. The absorber operating conditions (point 3) are constrained by the crystallization line of the LiBr/H₂O solution. The absorber operating conditions influence where the remaining component, the generator (point 4), can be positioned on the PTX diagram. The corresponding solution temperature for the generator can be read from the bottom axis. With a corresponding approach temperature, the heat input required to drive the generator can now be deduced.

Efficiencies in absorption chillers are described by the term coefficient of performance (COP), which is defined as the refrigeration effect/net work input. Single-stage absorption machines have COPs of approximately 0.6-0.8 out of a possible 1.0 (i.e. for every unit of heat input to the generator, 0.6-0.8 units of cooling output are produced in the evaporator). The inefficiencies can be accounted for as second law thermodynamic losses in the chiller. Since the COPs are less than one, the single-effect chillers are normally used in applications that recover waste heat such as waste steam from power plants or boilers.

1.4 Double-Effect Cycle

The world's first working double-effect LiBr/H₂O absorption chiller was built in the late 1950's by J. S. Swearingen and E. P. Whitlow [7].

Figure 5 represents the double-effect absorption cycle on a pressure versus temperature diagram. Figures 6 and 7 represent typical schematic diagrams of a double-effect absorption chiller that is commercially available with capacities ranging from 100 to 1500 tons. There are several variations on this cycle, but the one shown is the most commonly used in manufactured products. The double-effect chiller differs from the single-effect in that there are two condensers and two generators instead of only one of each. The higher temperature generator receives the externally-supplied heat (eg. steam) that boils the refrigerant from the weak absorbent. This refrigerant vapor from the high temperature generator is condensed and the heat produced is used to provide the heat to the low temperature generator. Due to the additional recovery of heat, two units of refrigerant vapor recovery are available for only one unit of heat input. The second unit of refrigerant vapor recovery comes from the additional recovery of heat at the lower temperature.

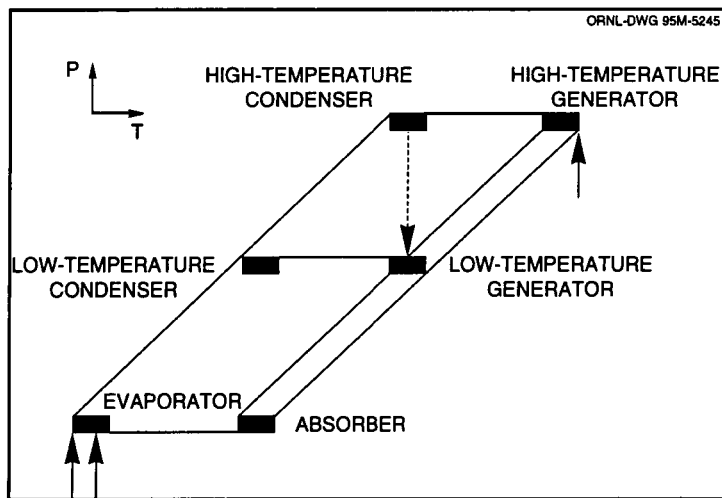


Figure 5. Double-Effect, Lithium Bromide/Water Absorption Cycle Represented on a Pressure versus Temperature Diagram

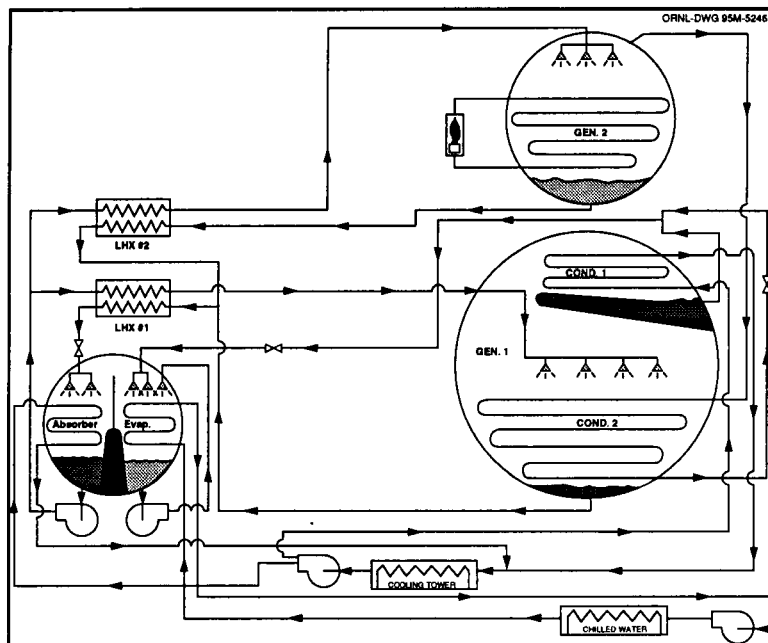


**Figure 6. Double-Effect Lithium Bromide/Water Chiller Design
(Drawn Using Freelance Graphics)**
(Reprinted by permission from R. C. DeVault, Oak Ridge National Laboratory)

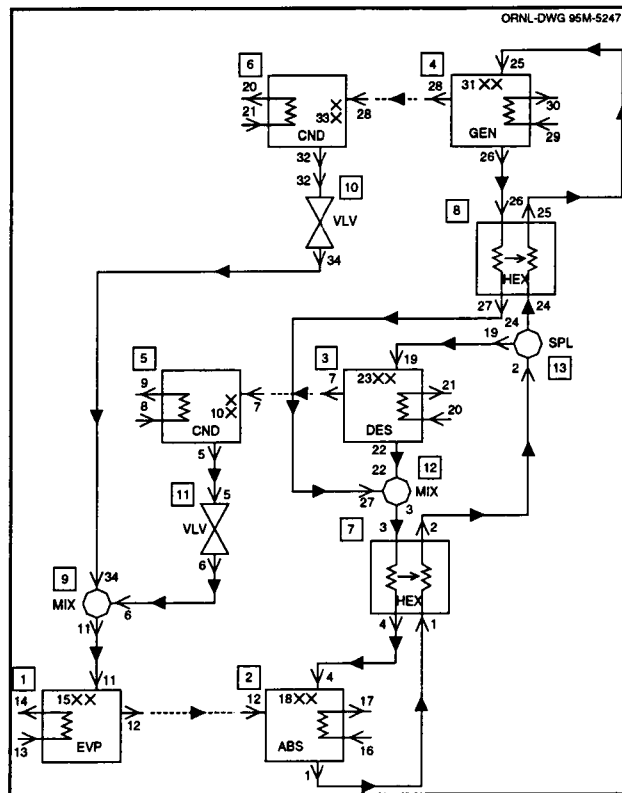


**Figure 7. Double-Effect Lithium Bromide/Water Absorption Cycle
(Drawn Using ABSIM)**

Double-effect absorption chillers have COPs of approximately 1.0-1.2 out of a possible 2, again because of second law thermodynamic losses.

Although the double-effect machines are more efficient than single-effect machines, they have a higher initial manufacturing cost related to special materials consideration because of increased corrosion rates (higher operating temperatures than single-effect machines), larger heat exchanger surface areas, much higher pressures, more complicated control systems, and the related increased manufacturing costs.

1.5 Triple-Effect Cycle

Currently, there are no triple-effect absorption chillers manufactured in the world. Previous work has shown that there are theoretically a large number of cycles that fall into the category of "triple-efficiency" [8, 9, 10, 11, 12, 13]. A basic three-condenser-three-generator triple-effect cycle was patented by Oouchi et. al. in 1985 (a pressure versus temperature representation is shown in Figure 8). An alternate triple-effect cycle, the double condenser coupled (DCC) cycle, was patented by DeVault and Biermann in 1993 (a pressure versus temperature representation is shown in Figure 9). An absorption chiller that uses this cycle could look like the schematic in Figure 10. LiBr solution is used as the absorbent and H_2O as the refrigerant. The cycle can be described as follows: a strong (refrigerant rich) LiBr/ H_2O solution is pumped from an absorber to low, medium, and high temperature generators (in series). In each generator some of the H_2O refrigerant is boiled off and leaves as hot vapor. The refrigerant vapor from the high and medium temperature generators is condensed and this heat is used to provide the heat input to the next lower temperature generator. The refrigerant from all three condensers flows to an evaporator where it absorbs more heat. The refrigerant evaporates and the cold vapor is returned to the absorber. In the absorber, the LiBr solution (refrigerant weak) that is returned from the three generators reabsorbs the refrigerant. This now refrigerant strong solution is then pumped to the three generators where it repeats the cycle.

In the triple-effect chiller, three units of refrigerant vapor recovery are available for only one unit of heat input. However, because of second law thermodynamic losses, cycle performance calculations predict coefficients of performance of approximately 1.5 out of a possible 3.

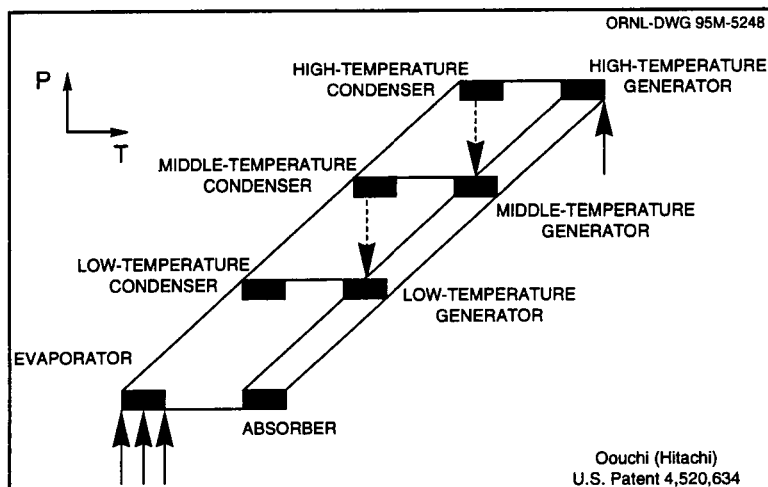


Figure 8. Triple-Effect Lithium Bromide/Water Absorption Cycle Represented on a Pressure versus Temperature Diagram (Oouchi, et al patent)

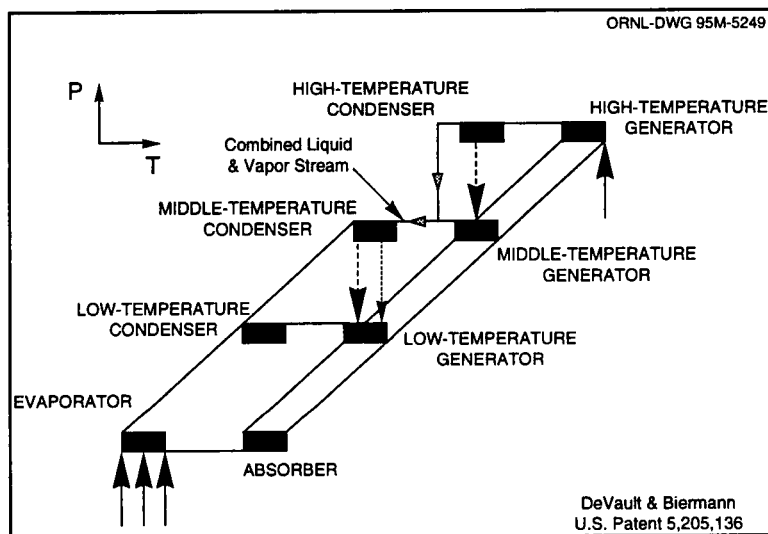


Figure 9. Alternate Triple-Effect Lithium Bromide/Water Absorption Cycle Represented on a Pressure versus Temperature Diagram (DeVault and Biermann patent)

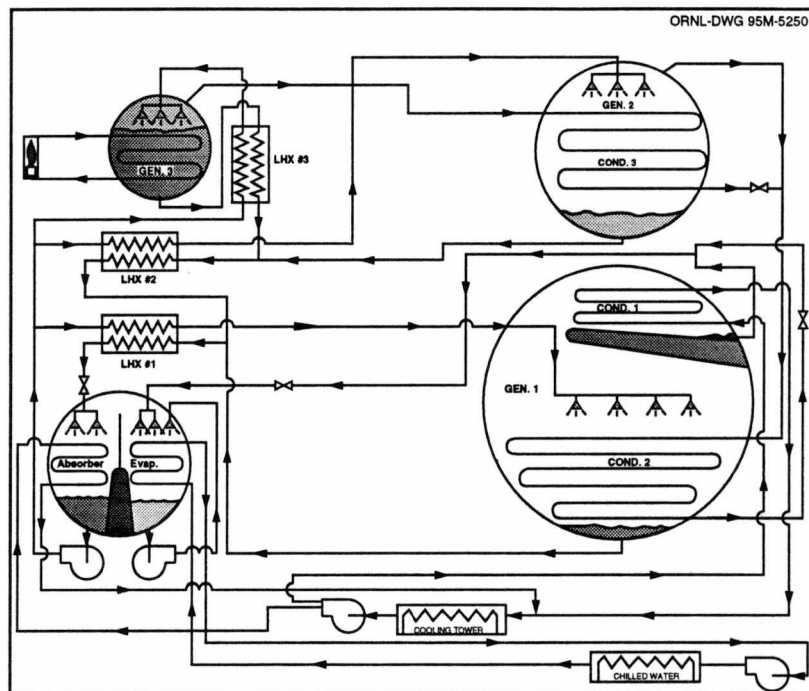


Figure 10. Triple-Effect Lithium Bromide/Water Absorption Chiller Design
(Drawn Using Freelance Graphics)
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Predicted initial manufacturing cost of the triple-effect machine is somewhat higher than that of the double-effect machines. These higher costs are related to increased corrosion rates and more expensive materials of construction (higher operating temperatures than double-effect machines), larger heat exchanger surface areas (in some cases), much higher pressures, more complicated control systems, and the related increased manufacturing costs.

1.6 Current Technology

Current American manufacturers of absorption chillers are Carrier Corporation, McQuay International, the Trane Company, and York International. Figure 11 represents an actual

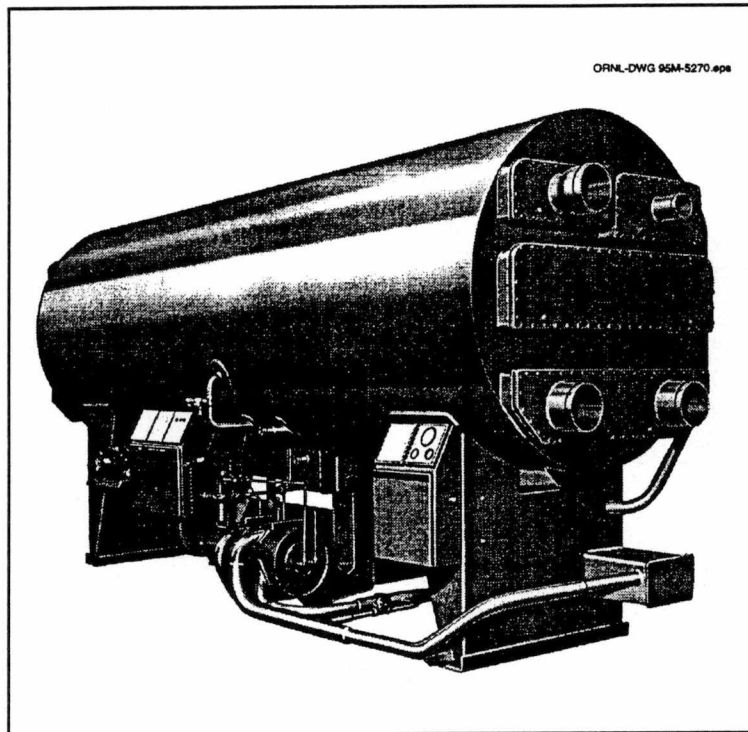


Figure 11. Single-Effect, Lithium Bromide/Water Absorption Chiller Hardware
(Reprinted by permission from The Trane Company)

operating single-effect LiBr/H₂O absorption chiller currently being built in the United States.

Figure 12 represents a double-effect LiBr/H₂O absorption chiller currently being built in the United States.

Carrier [14] was the first to manufacture large single-effect chillers for the commercial market beginning in the mid-1940s. Currently Carrier manufactures single-and double-effect absorption machines in Monterrey, Mexico. The single-effect machines use steam or hot water as the heat source and offer a COP of 0.7. These chillers come in sizes from 100 to 680-tons nominal capacity, and since they use low pressure steam or hot water as the heat source, they are ideal for

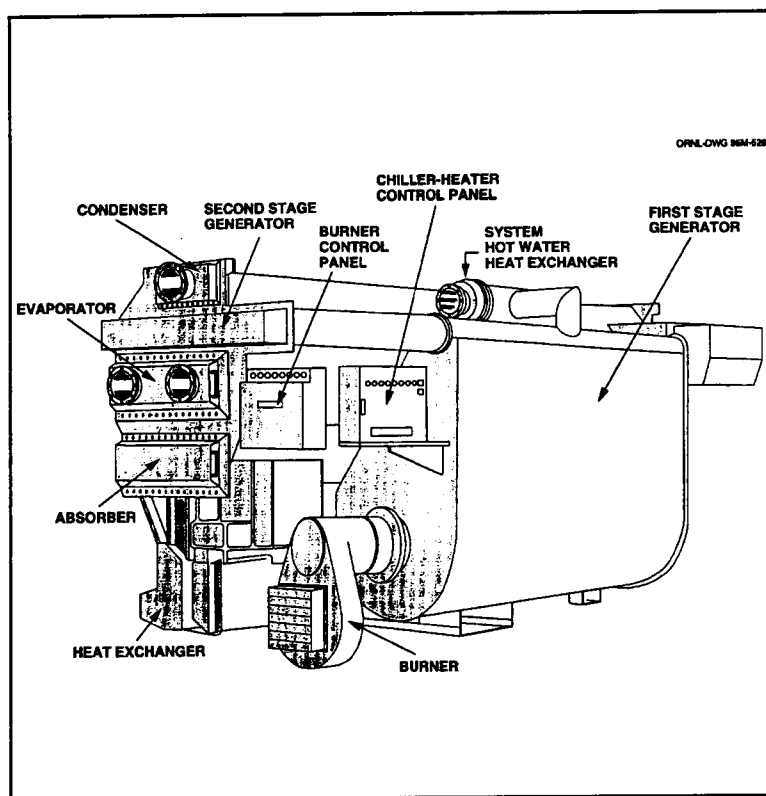


Figure 12. Double-Effect, Lithium Bromide/Water Absorption Chiller Hardware
(Reprinted by permission from York International.)

cogeneration or waste heat applications. The double-effect machine designs are licensed from Ebara. Spanning a 100 to 1700 ton nominal capacity range, the double-effect steam-driven machine offers a COP of 1.2. The direct fired double-effect chiller, with capacities from 135 to 1000 tons, offers a COP of 0.97 for gas-fired chillers and 1.0 when oil is used as the heat source.

York International [15] builds single-effect absorption chillers primarily in their Durango, Mexico plant and builds double-effect absorption chillers primarily in their Houston, Texas facility, with a few units built in York, Pennsylvania. The single-effect models use steam or hot water as the heat source. They range in capacity from 100 to 1400 tons and have COPs of 0.69 for steam and 0.67 for hot water. The double-effect models are produced using a design licensed from Hitachi, and

are available in direct-fired models from 180 to 1000 tons capacity, and capacities from 450 to 1500 tons in steam supplied models. This large range of models and capacities is intended to meet a wide range of cooling and heating requirements. The direct-fired units have a COP of approximately 1.0 and the steam fired units have a COP of approximately 1.19.

The Trane Company [16] builds their absorption chillers in LaCrosse, Wisconsin. Single-effect units are steam or hot water driven. They range in capacity from 100 to 1600 tons and offer COPs of 0.66 to 0.67. The double-effect model uses steam or hot water as the heat source. They range in capacity from 380 to 1000 tons and offer a COP of 1.2. Currently, the direct-fired units are manufactured in Japan. This unit has been licensed from Kawasaki and will be built in LaCrosse, Wisconsin in late 1995. The direct-fired units have COPs ranging from 0.97 to 0.98 and range in capacity from 100 to 1100 tons.

Starting January, 1995, McQuay International [17] began building double-effect chillers in Stanton, Virginia, using a licensed design from Sanyo. The steam heated units range in capacity from 100 to 1500 tons and have a COP of 1.2. The gas direct-fired models range in capacity from 100 to 1500 tons and have a COP of 0.99.

Manufacturers outside of the United States include: Sanyo, Kawasaki, Daiken, Hitachi-Zosen, Hitachi, Mitsubishi, Yazaki, and Ebara.

2. Introduction

2.1 Justification of Work

2.1.1 The United States Department of Energy Project History

The United States Department of Energy began a research program on absorption heat pumps in FY-1977. The Oak Ridge National Laboratory manages this program for the Department of Energy. One of the products of the program includes the development of a prototype residential ammonia/water absorption heat pump by the subcontractor Phillips Engineering. In 1994, this heat pump was licensed for manufacture by Carrier Corporation.

In 1981, ORNL-Energy Division personnel decided that additional research and development for advanced absorption cycles was needed. Figure 13 represents the relative energy use for the single-effect, double-effect, and triple-effect chillers versus the actual or predicted coefficient of performance. It can be clearly seen that the performance benefit is large if one goes from the single- to double- to triple-effect chillers. Currently, only single- and double- effect chillers are on the market. ORNL conducted scoping studies aimed toward improving the cooling performance of large commercial chillers by looking at triple-effect cycles. These studies led to DOE-owned patents for two distinctly different triple-effect absorption chiller cycles [10, 11] which are potentially 30% to 50% more efficient than the best existing absorption chillers. In the late 1960's, U.S. manufacturers had 100% of the world market for absorption chillers using the standard single-effect cycle. Currently, Japanese manufacturers dominate worldwide sales of absorption chillers using single-effect and double-effect cycles. This trend is shown in Figure 14.

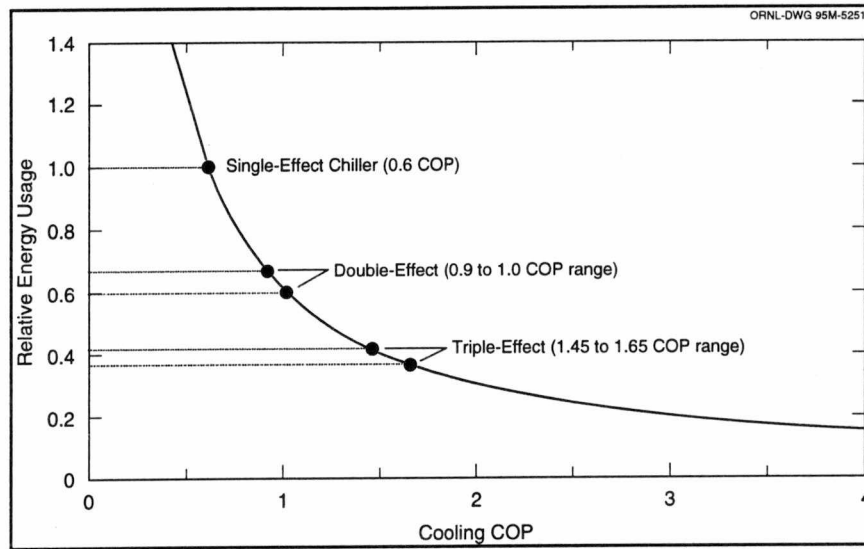


Figure 13. Relative Energy Usage versus Cooling COP for Large Commercial Chillers

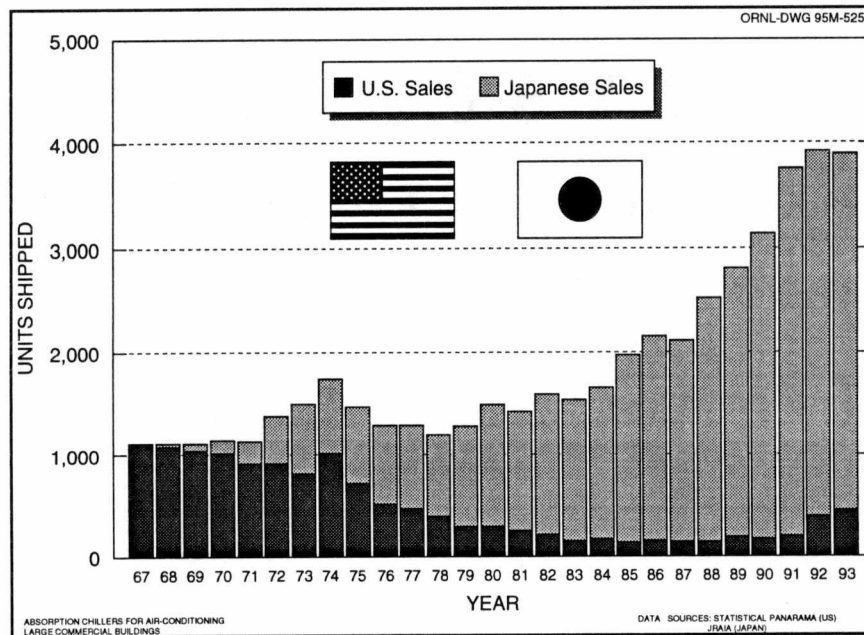


Figure 14. U.S. versus Japanese World-Wide Sales of Absorption Chillers

As part of an ongoing effort to promote U.S. production of absorption chillers, DOE sponsored research and development on triple-effect cycles. After completion of Phase II of an Advanced Absorption Cycles Heat Pump subcontract from ORNL, Trane licensed one of the two patented triple-effect cycles, the dual-loop cycle, in 1988 and is currently testing a laboratory prototype. Separately, in 1990, a request for proposal was issued to develop and proof-test basic technology needed for high temperature operation of LiBr/H₂O for absorption chiller cycles (i.e. triple-effect cycles). The contractor selected to do the research was Battelle Memorial Institute (Columbus, Ohio) with York International (York, Pennsylvania) as a subcontractor. Battelle performed materials and additive evaluations. York International provided insight and guidance from a manufacturing perspective.

As the work progressed towards the development stage, ORNL subcontracted with York International in 1992 to evaluate several cycle possibilities for LiBr/H₂O triple-effect absorption chillers. Figure 15 [18] represents the task outline of the York Triple-Effect Program. The objective of the triple-effect program is to build fully functional hardware to demonstrate a practical triple-effect chiller for commercial air conditioning applications. The design goal for the prototype will be to achieve 400 tons of cooling capacity. The final goal is a U.S. manufactured triple-effect absorption chiller product.

2.1.2 The Oak Ridge National Laboratory Triple-Effect Program

The Oak Ridge National Laboratory manages the triple-effect absorption chiller program for the United States Department of Energy. The program involves research, development, testing, and

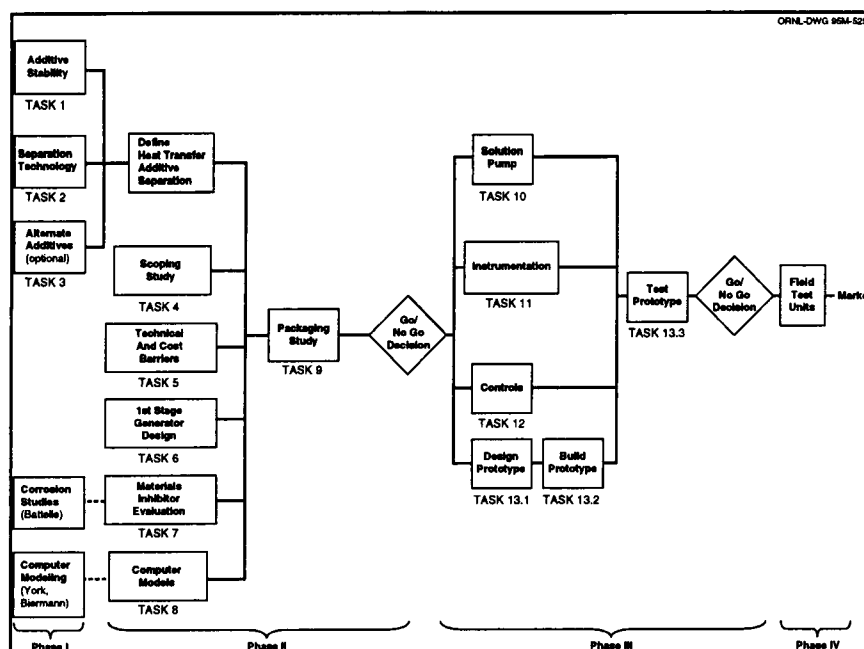


Figure 15. Task Outline of the DCC Triple-Effect Absorption Program

evaluation of multiple triple-effect designs. As part of the program, ORNL subcontracted with York International to review several cycle possibilities for LiBr/H₂O triple-effect absorption chillers. The expected outcome of the triple-effect program is to build a prototype triple-effect absorption chiller to be field tested and ultimately manufactured. Figure 15, Task 8 represents a computer modeling effort in which analytical tools will be used to predict the performance of alternative cycles and to guide design and testing of prototype hardware. This thesis falls under the effort required to complete Task 8.

It is anticipated that the most significant problems to be solved in developing a triple-effect absorption chiller will be associated with the higher expected operating temperatures compared to the double-effect. Problems related to these higher temperatures could include corrosion, (both in the upper stage generator and in the hot side of the upper stage solution heat exchanger), and

thermal decomposition of the heat/mass transfer additive. Several cycles will be analyzed with the generator temperature being a variable; the selected design will strike a balance between initial production cost and improved operating efficiency.

In an effort to optimize system component design and to shorten equipment production cycle time, ORNL uses computer modeling techniques to predict cycle efficiency. Currently ORNL uses ABSIM (a computer model), to perform cycle analysis. In addition, industry utilizes component design models based on empirical results. The component models provide the capability to evaluate and modify component designs when they affect overall system efficiency or cost. Currently, there is no link between the modeling of components and the overall cycle analysis performed by ABSIM. In this project, the ABSIM program was integrated with component models to bridge the gap between theoretical cycle analysis and practical applications. The model was validated using data taken from tests performed on an existing double-effect absorption chiller. The model will be useful in the equipment design for the triple-effect absorption chiller, the future four-effect absorption chiller, and follow on programs.

2.2 Scope of Work

2.2.1 Development of a Computer Model

The ABSIM computer model was used as a basis for the overall cycle performance calculations. To augment the ABSIM program with a powerful modeling tool, computer code for specific component design was developed and integrated with the ABSIM model. Heat transfer

coefficients were based on known correlations, either from industrial empirical data, text books, or open literature.

Several cycle configurations were studied: the patented DCC triple-effect cycle [11], several proprietary triple-effect cycle(s) proposed by York, and alternate triple-effect cycles. The base operating case used for performance calculations for this thesis is the patented DCC cycle. A parametric analysis was performed to evaluate performance versus initial manufacturing costs.

Initial manufacturing cost is based on the following assumptions:

- 1) components (pumps, valves, etc.) and shells that are identical in both systems are neglected, and
- 2) for the absorber, generator, condenser, and evaporator components: cost is based on the amount of material (proportional to area) in the components.

A parametric analysis was performed by varying the UA values of each major component independently and calculating the resulting coefficient of performance and capacity.

2.2.2 Comparison With Field Data

Double-effect absorption chillers are currently being built and sold by several vendors including York International. Although the instrumentation is not available at the customer's installation to extensively monitor system performance, York does have test stands in their manufacturing plants in Houston, Texas and York, Pennsylvania. Performance requirements on each chiller are tested and validated prior to shipment to the customer. Additionally, test stands are utilized to perform

developmental tests. These test stands are instrumented with thermocouples, flowmeters, pressure gauges, and sampling ports, which provide adequate data to validate the component models. Data was taken from a double-effect chiller that was operated on a test stand at the York, Pennsylvania, manufacturing plant for validation of the component models.

2.3 Value Added

This thesis project integrates component design models with a system performance model, providing an important link between academia and industry. Although this work specifically supports development of a triple-effect absorption chiller, it may be possible to extend it to other absorption chiller projects currently underway or to those planned for the future.

In addition, this thesis report can be used as a primer for engineers entering into this field of study and practice. It discusses absorption history, current technology, absorption basics, current work related to modeling similar systems, and discusses results of component modeling.

3. Current Approach

3.1 Modular Cycle Models

3.1.1 ORNL ABSIM

The ABSIM code [19, 20] was developed for simulation of absorption systems, and has been employed successfully to simulate a variety of chillers, heat pumps and heat transformers with different working fluids and in different cycle configurations. Because each absorption system consists of a number of standard components (e.g., absorber, condenser, etc.), the code has been structured around unit subroutines, each of which contains the governing equations for the particular component. The user specifies which components are to be modeled and all connections to other components and systems (chilled water, steam, etc.). This is all performed with an "easy-to-use" graphical interface to allow ease of development and modification of system models. The component subroutines are activated by a main program, which interprets the user input, calls the units, and links them in an order corresponding to the equipment model specified by the user. Each unit subroutine, when activated, addresses a property data base for the thermodynamic properties of the working fluids. The equations generated by the unit subroutines are solved simultaneously.

One of the major challenges solved by the ABSIM computer model is the simultaneous solution of numerous thermodynamic relationships. Recent improvements in the code [21] have significantly enhanced the program's convergence capability with large and complex cycles.

ABSIM is presently used extensively at the Oak Ridge National Laboratory [22, 23] and by several ORNL-associated developers of absorption equipment to simulate absorption cycles [24, 25, 26, 27, 28, 29].

3.1.2 Other Equation Solvers

EES

EES (pronounced 'ease') is an acronym for Engineering Equation Solver. It was copyrighted by S. A. Klein and F. L. Alvarado, professors at the University of Wisconsin at Madison [30]. The basic function provided by EES is the solution of numerous sets of algebraic equations. EES can also be used to solve initial value differential equations. The major difference between EES and other existing equation-solving programs is that EES provides many built in mathematical and physical property functions useful for engineering calculations. For example, the steam tables are implemented such that any thermodynamic property can be obtained from a built-in call function in terms of any two other properties. A similar capability is provided for most CFC refrigerants, ammonia, methane, carbon dioxide and other fluids, including LiBr/H₂O and NH₃/H₂O. With such capabilities, it essentially provides a LiBr/H₂O PTX diagram on computer. The EES language supports user-written functions and procedures, similar to those in Pascal and FORTRAN. The functions and procedures can be saved in library files which are automatically read in when EES is started. Compiled functions and procedures, written in a high-level language such as Pascal, C or FORTRAN, can be dynamically-linked into EES using the library capability incorporated into the Windows operating system.

Recently, EES has been used to develop a steady-state computer model simulation of a direct-fired double-effect LiBr/H₂O absorption chiller in parallel flow. Unknown model parameters such as heat transfer coefficients were determined by matching the model's calculated state points and COP against nominal full-load operating data and COP obtained from a manufacturer's catalog. The model was used to investigate the effect of varying the water circuit flow rates with the chiller load, and to optimize part-load performance of the double-effect chiller with respect to the distribution and flow of the weak solution. [31]

The component models developed for this thesis project can be used as a stand alone subroutine for use with the EES program.

Vissim

Vissim [32] is a visual environment for systems design and simulation developed with the goal of increasing productivity. Using a technique called visual programming, the user designs and tests systems quickly and accurately. Vissim comes with over 75 predefined blocks for modeling nonlinear systems, including summing functions, gains, six different numerical integration methods, transcendental functions, data import and export, and file-based signal mapping with linear interpolation and extrapolation. It also supports imbedded plots, graphics, buttons, displays, and meters for input and output.

Vissim is being used by United Technologies Carrier [33] to do performance modeling of the generator-absorber heat exchange (GAX) absorption chiller for another program funded by the Department of Energy through the Oak Ridge National Laboratory.

ASPEN

Other more traditional equation solvers are in use by industry. ASPEN [34] was evaluated during a previous subcontract with DOE and it was found to be less efficient because it works on a sequential basis and requires time consuming iterations to close the balance.

3.2 System Specific Models for LiBr/H₂O Systems

3.2.1 Biermann and He

Wendell Biermann has performed extensive cycle analysis using computer models, written in QuickBasic, that he developed for DOE and industry over the last 20 years [35]. These models are steady state models; each one written for a specific absorption cycle. The double- and triple- effect models build upon the single-effect models. Actual design information and operating conditions are provided as input and system performance is given as output.

Xiaoyu He has utilized the Biermann models to perform cycle analysis at York International. The models were modified to make them more user friendly [36].

3.2.2 Miller

Bill Miller has completed tests of falling film absorption in a vertical-tube configuration using advanced surfaces [37]. Currently he is performing tests in a horizontal-tube configuration. Correlations are being developed to model simultaneous heat and mass transfer internal to the

absorber. The results of Miller's work could be used as future input into the computer code written for this masters thesis.

3.2.3 Others

References [38] through [48] include recent studies in the LiBr/H₂O and NH₃/H₂O area.

4. Calculation of Heat Transfer Coefficients

A generalized diagram of the temperature gradient from a hot fluid to a cold fluid through a tube wall is shown in Figure 16 [49]. For turbulent flow, there is a laminar zone of fluid adjacent to the surface of the tube wall called the film. The resistance of heat flow through the film can vary, depending on its thickness. The turbulent fluid core and the transition zone between the core and the film also introduce resistance to heat flow. The resistance is dependent upon the degree of turbulence and the fluid properties.

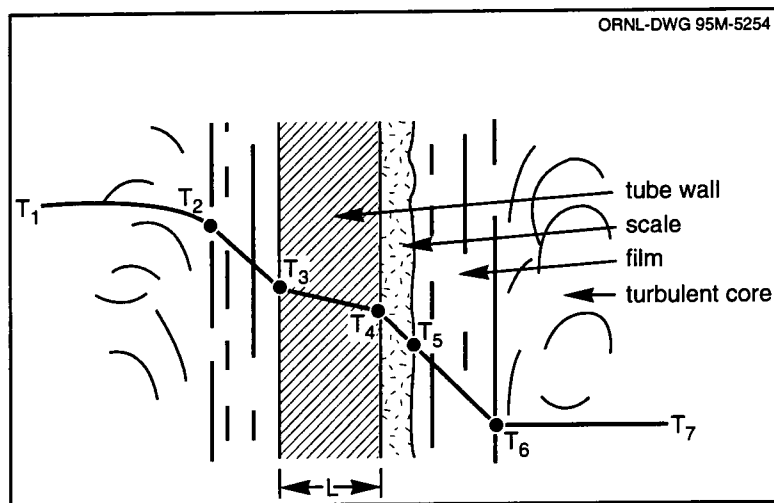


Figure 16. Temperature Gradients in Forced Convection

(Taken from the Chemical Engineering Reference Manual, by Randall N. Robinson, with permission from the publisher Professional Publications, Inc., Belmont, CA, (415)593-9119, copyright 1987.)

The overall resistance to the flow of heat is a result of separate resistances occurring in series: the film resistance inside the tube, the resistance of the tube itself, the fouling resistance, and the film resistance outside the tube.

The overall resistance can be calculated as the sum of the individual resistances:

$$\frac{1}{U} = \frac{1}{h_i} + A_i \left(\frac{\Delta}{kA} \right)_{\text{material}} + A_i \left(\frac{\Delta}{kA} \right)_{\text{fouling}} + \frac{1}{h_o} \frac{A_i}{A_o}$$

In this equation, U is the overall coefficient of heat transfer, as defined in the equation $Q = UA \cdot \text{LMTD}$.

The inside and outside resistances are calculated by theoretical correlations typically described in textbooks and utilized in the next few sections.

The thermal resistance of the tube material (copper) is reported in Perry's Handbook [50] as 2700 BTU-in/ft²-hr-°F. The thickness of the tube varies with the component design. A typical copper tube thickness is 0.0375 inches.

The standard fouling factor for refrigeration equipment is reported in the ASHRAE Handbook [51] as .0005 ft²-hr-°F/BTU. Fouling factors can also be calculated by using ARI standard 480 [52].

4.1 Evaporator

The Dittus and Boelter Equation is used to calculate the turbulent tubeside (inside) coefficient.

This equation applies to fully developed turbulent flow ($N_{Re} > 10,000$) of fluids for $0.7 < N_{Pr} < 700$.

In pipes for which $L/D > 60$, the entrance effects are negligible.

All fluid properties are evaluated at the arithmetic mean bulk temperature:

$$\frac{h_i D}{k} = 0.023 \left(\frac{D v \rho}{\mu} \right)^{0.8} \left(\frac{C_p \mu}{k} \right)^{0.3 \text{ or } 0.4}$$

where h_i = average inside heat transfer coefficient in BTU/hr-ft²-°F
 D = tube inside diameter in ft
 k = thermal conductivity in BTU/hr-ft-°F
 v = velocity in ft/hr
 ρ = density in lb/ft³
 μ = absolute viscosity in lb/hr-ft
 C_p = specific heat in BTU/lb-°F

which can be rewritten as:

$$\frac{h_i D}{k} = 0.023 N_{Re}^{0.8} N_{Pr}^{0.3 \text{ or } 0.4}$$

where N_{Re} = dimensionless Reynold's Number
 N_{Pr} = dimensionless Prandtl Number

This equation correlates data for heating of the fluid when the exponent on the Prandtl number is 0.4 and for cooling of the fluid when the exponent is 0.3. In the evaporator, the chilled water is being cooled, so the exponent of 0.3 is used.

For the outside (shell side) coefficient, relationships have been defined by Chun and Seban [53], as follows. For laminar flow, the heat transfer coefficient for evaporation from the surface of water films is approximated as:

$$h_o = 0.606 \left(\frac{k^3 \rho^2 g}{\mu^2} \right)^{0.333} \left(\frac{\Gamma}{\mu} \right)^{0.22}$$

where h_o = average outside heat transfer coefficient in BTU/hr-ft²-°F
 k = thermal conductivity in BTU/hr-ft-°F
 ρ = density in lb/ft³
 g = gravitational acceleration in ft/hr²
 v = velocity in ft/hr
 Γ = loading per linear foot in lb/ft-hr
 μ = absolute viscosity in lb/hr-ft

which can be rewritten as:

$$h_o = 0.606 \left(\frac{k^3 \rho^2 g}{\mu^2} \right)^{0.333} \left(\frac{\Gamma}{\mu} \right)^{-0.22} \left(4 \right)^{-0.22} \left(\frac{1}{4} \right)^{0.22}$$

$$= 0.822 \left(\frac{k^3 \rho^2 g}{\mu^2} \right)^{0.333} N_{Re}^{-0.22} \text{ where } N_{re} \left(= \frac{4\Gamma}{\mu} \right)$$

For turbulent flow, Chun and Seban approximate the outside heat transfer coefficient with the following relation:

$$h_o = 0.0038 \left(\frac{k^3 \rho^2 g}{\mu^2} \right)^{0.333} N_{Re}^{0.4} N_{Pr}^{0.65}$$

These relationships were developed from experiments of water films flowing along the outside surface of vertical tubes. Kern [54] found that when vertical tube correlations were extended to horizontal tubes, the following relationship applies:

$$\Gamma = \frac{m}{L N_t^{0.667}}$$

where m = mass flow rate in lb/hr
 L = length of tube in ft
 N_t = number of tubes

The overall heat transfer coefficient is calculated from the following equations:

$$\frac{1}{UA_i} = \frac{1}{h_i A_i} + \left(\frac{\Delta}{kA} \right)_{\text{material}} + \left(\frac{\Delta}{kA} \right)_{\text{fouling}} + \frac{1}{h_o A_o}$$

or

$$\frac{1}{U} = \frac{1}{h_i} + A_i \left(\frac{\Delta}{kA} \right)_{\text{material}} + A_i \left(\frac{\Delta}{kA} \right)_{\text{fouling}} + \frac{1}{h_o} \frac{A_i}{A_o}$$

4.2 Condenser

As in the evaporator, the Dittus and Boelter equation is used to calculate the turbulent tubeside (inside) coefficient:

$$\frac{h_i D}{k} = 0.023 N_{Re}^{0.8} N_{Pr}^{0.3 \text{ or } 0.4}$$

In the condenser, the cooling tower water is being heated, so the exponent of 0.4 is used.

For the outside (shell side) coefficient, McCabe and Smith [55] and McAdams [56] approximate the heat transfer coefficient for film-type condensation on horizontal tubes by the Nusselt theoretical equation. All fluid properties are evaluated at the arithmetic mean film temperature:

$$h_o = 1.51 \left(\frac{4\Gamma}{\mu} \right)^{-0.333} \left(\frac{k^3 \rho^2 g}{\mu^2} \right)^{.333}$$

where Γ is approximated by Kern as:

$$\Gamma = \frac{m}{L N_t^{0.667}}$$

The overall heat transfer coefficient is calculated from the following equations:

$$\frac{1}{UA_i} = \frac{1}{h_i A_i} + \left(\frac{\Delta}{kA} \right)_{\text{material}} + \left(\frac{\Delta}{kA} \right)_{\text{fouling}} + \frac{1}{h_o A_o}$$

or

$$\frac{1}{U} = \frac{1}{h_i} + A_i \left(\frac{\Delta}{kA} \right)_{\text{material}} + A_i \left(\frac{\Delta}{kA} \right)_{\text{fouling}} + \frac{1}{h_o} \frac{A_i}{A_o}$$

4.3 Absorber

As in the evaporator and condenser, the Dittus and Boelter equation is used to calculate the turbulent tubeside (inside) coefficient:

$$\frac{h_i D}{k} = 0.023 N_{Re}^{0.8} N_{Pr}^{0.3 \text{ or } 0.4}$$

In the absorber, the cooling tower water is being heated, so the exponent of 0.4 is used.

For the outside (shell side) coefficient, Biermann [57] approximates the heat transfer coefficient by the following empirical relation:

$$h_o = 273.3 + 0.00777 * Q/A$$

where h_o = outside heat transfer coefficient in BTU/hr-ft²-°F
 Q = absorber load in BTU/hr
 A = area in ft²

This correlation was developed from averages of data taken over several years and in multiple machines. The absorber coefficient, h_o , was derived from the overall absorption coefficient of a lithium bromide machine containing 2-ethyl-1-hexanol, using log mean temperature differences between the absorber solution and coolant.

The overall heat transfer coefficient is calculated from the following equations:

$$\frac{1}{UA_i} = \frac{1}{h_i A_i} + \left(\frac{\Delta}{kA} \right)_{\text{material}} + \left(\frac{\Delta}{kA} \right)_{\text{fouling}} + \frac{1}{h_o A_o}$$

or

$$\frac{1}{U} = \frac{1}{h_i} + A_i \left(\frac{\Delta}{kA} \right)_{\text{material}} + A_i \left(\frac{\Delta}{kA} \right)_{\text{fouling}} + \frac{1}{h_o} \frac{A_i}{A_o}$$

4.4 Generator

For steam condensing inside tubes, the heat transfer coefficient is approximated by McCabe and Smith [55] and McAdams [56] according to the following relation:

$$h_i = 1.51 \left(\frac{4\Gamma}{\mu} \right)^{-0.333} \left(\frac{k^3 \rho^2 g}{\mu^2} \right)^{.333} \quad \text{where}$$

$$\text{where} \quad \Gamma = \frac{m}{L N_t}$$

For the outside (shell side) coefficient, Biermann [57] approximates the heat transfer coefficient by the following empirical relation:

$$h_o = 176.1 + 0.04945 * Q/A$$

$$\begin{aligned} \text{where} \quad h_o &= \text{outside heat transfer coefficient in BTU/hr-ft}^2\text{-}^\circ\text{F} \\ Q &= \text{generator load in BTU/hr} \\ A &= \text{area in ft}^2 \end{aligned}$$

This correlation was developed from averages of data taken over several years and in multiple machines. The generator coefficient, h_o , was derived from the overall heat transfer coefficient of a

lithium bromide machine, using log mean temperature differences between the generator solution and hot water.

The overall heat transfer coefficient is calculated from the following equations:

$$\frac{1}{UA_i} = \frac{1}{h_i A_i} + \left(\frac{\Delta}{kA} \right)_{\text{material}} + \left(\frac{\Delta}{kA} \right)_{\text{fouling}} + \frac{1}{h_o A_o}$$

or

$$\frac{1}{U} = \frac{1}{h_i} + A_i \left(\frac{\Delta}{kA} \right)_{\text{material}} + A_i \left(\frac{\Delta}{kA} \right)_{\text{fouling}} + \frac{1}{h_o} \frac{A_i}{A_o}$$

5. Development of the Component Models

Because ABSIM was written in FORTRAN, the component codes for this thesis were also written in FORTRAN.

A listing of variables is contained in Appendix 1.

A listing of unit conversions is contained in Appendix 2.

The FORTRAN code for each component is contained in Appendix 3.

Linking of Code

The FORTRAN component codes were compiled and linked to the ABSIM code via the FORTRAN linking program. An additional variable was added to the component input file to allow the user to select the FORTRAN code discussed above if the UA calculations were desired. The ABSIM modifications were completed under subcontract to the Department of Energy.

A sample of the input file is contained in Appendix 4.

A sample of the output file is contained in Appendix 5.

6. Data Validation

The theoretical relationships developed in Chapter 4, Calculation of Heat Transfer Coefficients, were used to calculate overall U values for each component using data collected for a series of runs on a 400 ton double-effect absorption chiller on a test block at York International. The theoretically calculated component overall resistance, which is the inverse of U, is the summation of the individual heat transfer resistances across the tube. The resistances across the tube are: 1) the inside tubeside resistance, 2) the resistance of the tube material, 3) the resistance due to fouling, and 4) the outside film resistance. The inside and outside resistances are calculated using the correlations described in Chapter 4. The fouling resistance is taken as half the standard resistance reported in the ASHRAE handbook, because the experimental data was taken on a relatively new chiller. The resistance of the copper tube material is calculated from the thermal conductivity of the material divided by the thickness, where the thermal conductivity is taken from Perry's Handbook.

The theoretically calculated overall heat transfer resistances, $1/U$ values, described in the previous paragraph were compared to values determined experimentally from the operational runs. The experimental U values were calculated as follows:

$$Q = UA \cdot \text{LMTD}$$

The Q values for the condenser, evaporator, absorber, and generator were calculated from the ΔT values of the cooling tower water, chilled water, cooling tower water, and steam, respectively. The

LMTD values were calculated from measured temperatures. The component areas (A) were calculated from design information.

The figures on pages 42 through 45 represent the comparisons (for each component) of overall heat transfer resistance values calculated from the summation of individual resistances derived in the theoretical heat transfer correlations to overall resistance values calculated from the overall heat balance for each experimental run. The ordinate represents $1/U$ values and the abscissa represents the heat load (Q) of the component. These scales have been normalized to the manufacturer's rated condition to protect the proprietary nature of the data. The circles represent the values calculated as the summation of the individual heat transfer resistances (thesis calculated values). The squares represent the values calculated from the overall heat balance (experimental values).

Figure 17 represents data on the evaporator. At near-full load conditions ($Q \sim 1$), the correlations match well with the experimental data. At part load conditions, the calculated values are somewhat lower. The inside resistance ($1/h_i$) and the resistance of the tube material (Δ/k) are relatively constant across the full range of loads, as would be expected. The resistance due to fouling is also constant over the full range of loads. The outside resistance ($1/h_o$) should vary across the operating range due to the decreased flow of the refrigerant through the evaporator. The calculated outside resistance data confirmed this trend (i.e. as the heat load increases, the outside resistance also increases). Since the outside resistance is a smaller factor in the overall resistance due to the area ratio, the change in outside resistance did not produce as significant a change in the overall resistance ($1/U$) as would otherwise be expected.

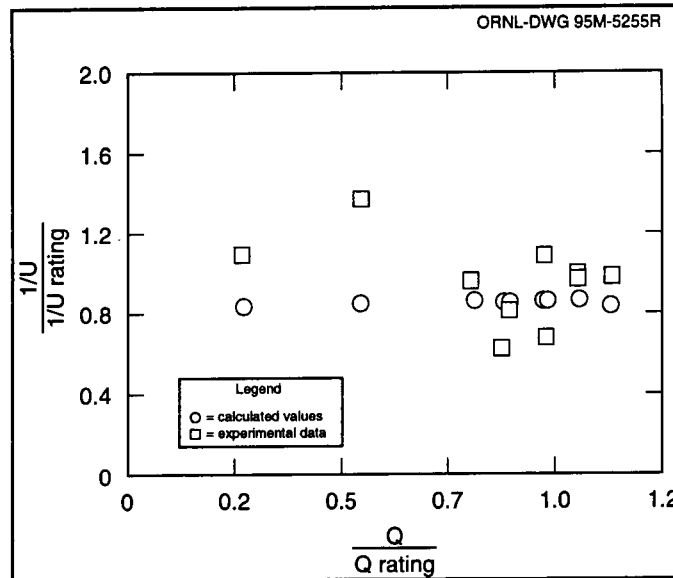
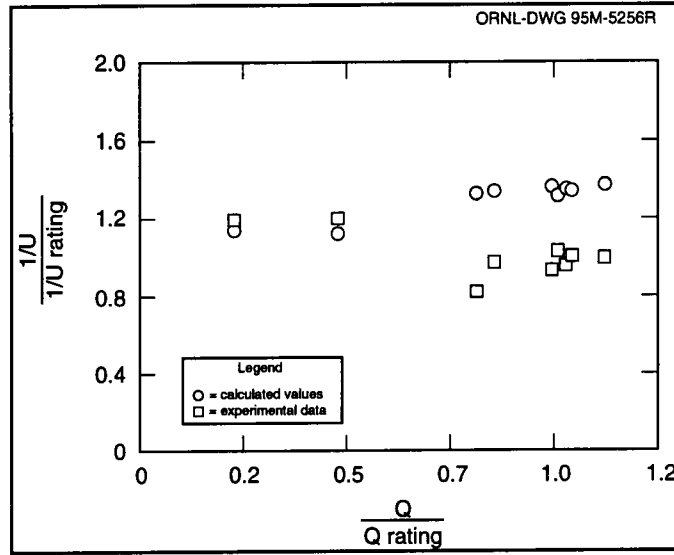


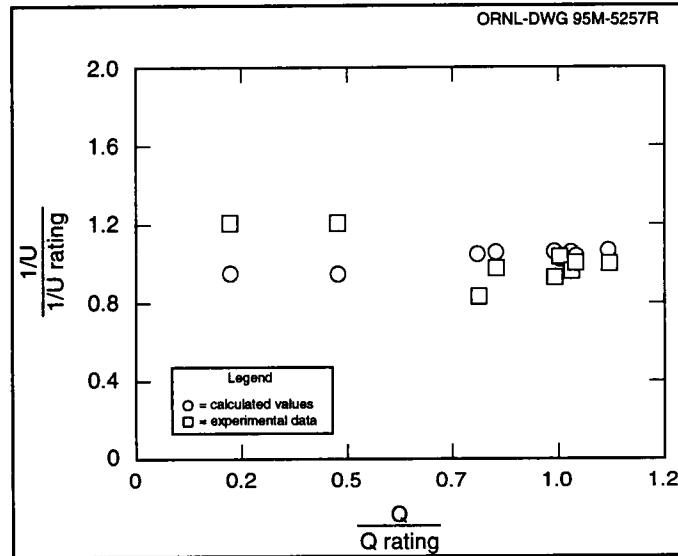
Figure 17. Overall Normalized Heat Transfer Resistance versus Normalized Load for the Evaporator

Figures 18a. and 18b. represent data on the condenser. Figure 18a. is the overall resistance using the Kern relationship to calculate h_o with Γ being inversely proportional to $N_t^{0.667}$, due to incomplete wetting of the tubes. Figure 18b. is the overall resistance using Γ based on an inverse proportionality to N_t . The calculated values on Figure 18a. more closely match the experimental data at low heat loads. The calculated values on Figure 18b. more closely match the experimental data at high heat loads. It is proposed that this can be accounted for due to incomplete tube wetting at low loads as a result of lower refrigeration flow.

Figure 19 represents data on the absorber. Over the full range of operating conditions, the calculated values match relatively well with the experimental data. The calculated values do not vary across the operating range as much as the experimental values. This could be attributed to the empirical correlations that were used to determine the calculated values of the outside resistance.



a. Kern Relationship for Γ



b. McCabe and Smith Relationship for Γ

Figure 18. Overall Normalized Heat Transfer Resistance versus Normalized Load for the Condenser

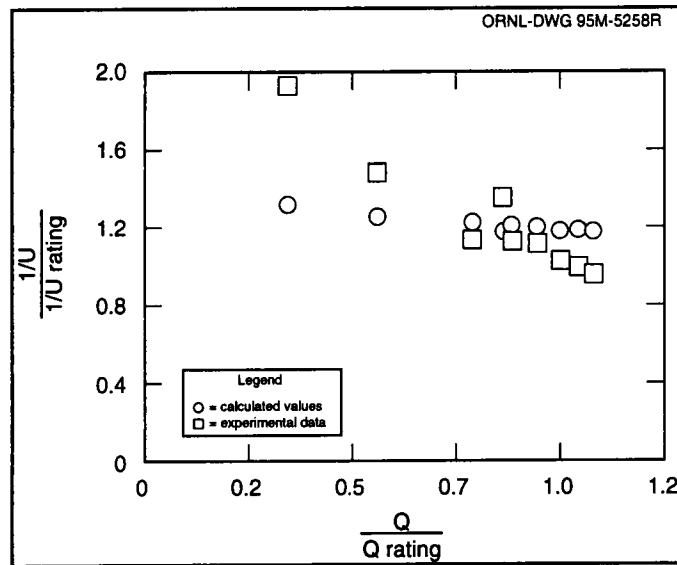


Figure 19. Overall Normalized Heat Transfer Resistance versus Normalized Load for the Absorber

Figure 20 represents data on the generator. Over the full range of operating conditions, the calculated values are lower than the experimental data. Several correlations were tested as alternates (i.e. Rohsenow and Forster and Zuber as published in Perry's [50] and the ASHRAE [1] Handbooks and Chun and Seban as described in Section 4.1) to the Biermann correlation for the outside resistance, however these correlations gave even lower calculated values for the outside resistance contribution. It is desired to perform future work related to the development of the generator model.

A Wilson plot analysis can also be used to calculate individual heat transfer coefficients, however the operating conditions under which the experimental data was taken did not lend themselves to this technique. Additional work may include the use of this type of analysis.

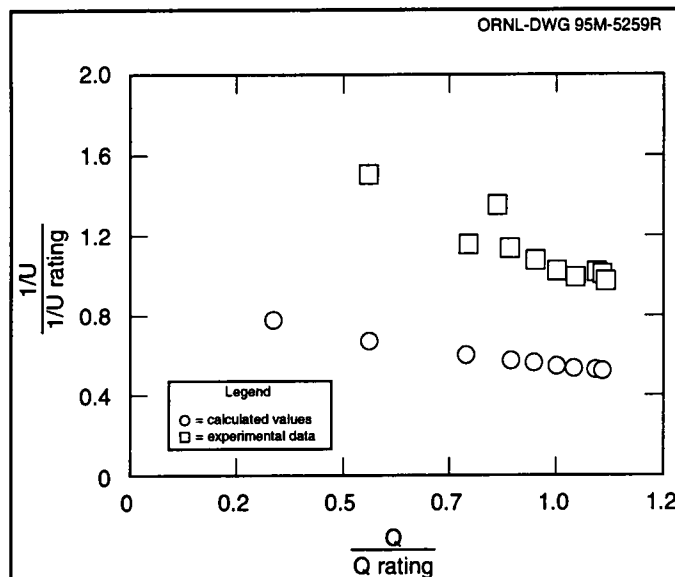


Figure 20. Overall Normalized Heat Transfer Resistance versus Normalized Load for the Generator

7. Cycle Simulations

7.1 Base Case

The DCC triple-effect cycle was chosen as the base case for the cycle simulations. A schematic of the DCC cycle chosen is shown in Figure 21. The pressure versus temperature diagram was previously shown in Figure 9. Note that the high temperature condenser and middle temperature condenser have been combined with the next lower temperature generator, which is the way the actual operating equipment is built. (The highest temperature refrigerant from the high temperature generator is the heat source for the middle temperature generator and actually

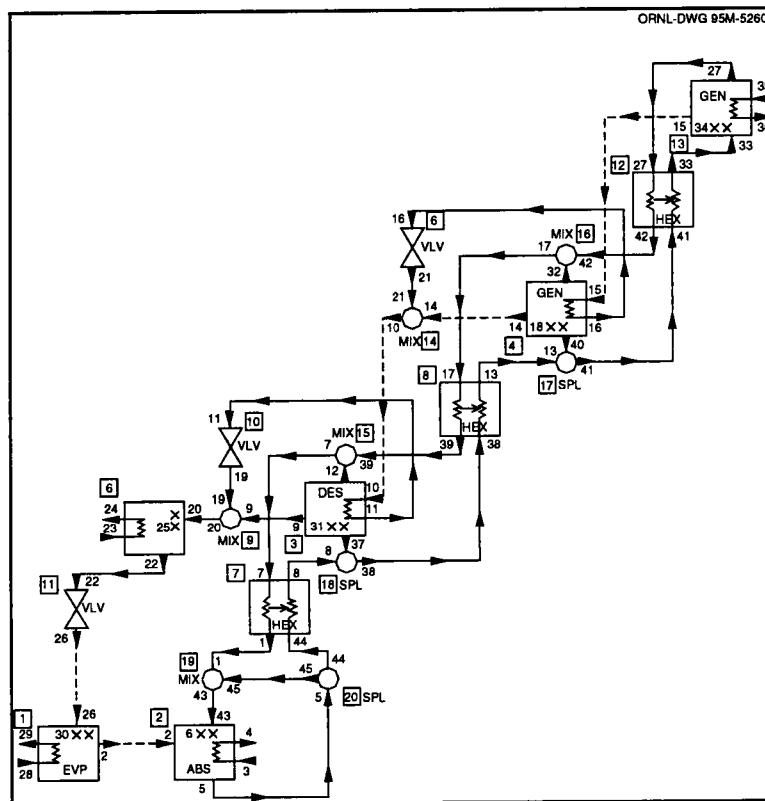


Figure 21. Triple-Effect, Lithium Bromide/Water Absorption Cycle
(Drawn Using ABSIM)

condenses inside the tubes of the middle temperature generator.) Table 1 lists the initial operating values, which were chosen based on recommended conditions from industrial counterparts. A sample input/output file is contained in Appendix 5.

7.2 Optimization Runs

An approach used for optimization of components was outlined by McGahey and Christensen [24]. Each component UA was sized independently so that the effect on system COP could be determined. If the system were optimized solely for performance, each heat exchanger would be designed so its effectiveness would approach 100%. These extremely large components would then make the system so expensive that the differential initial manufacturing cost between this machine and a competing machine could never be recovered. Two additional criteria are added in McGahey and Christensen's optimization: the first is to find the point where making a particular component larger does not yield an equivalent improvement in performance. The second is to find a set of components that will operate at the base case conditions.

Figure 22 [24] depicts McGahey and Christensen's approach used for parameter optimization. The curve should intersect the y-axis at the point where the component UA value is equal to zero and it should approach a horizontal asymptote as the UA value approaches infinity. The point where the curve breaks, and starts to approach the horizontal asymptote is called the knee of the curve. Moving past the knee represents the point of diminished return.

Triple-Effect DCC Cycle Baseline Operating Conditions

Absorber	Temperature: Inlet = 85 °F Outlet = 92 °F Flowrate = 13,884 lb/min
Condenser	Temperature: Inlet = 85 °F Outlet = 92 °F Flowrate = 3,098 lb/min

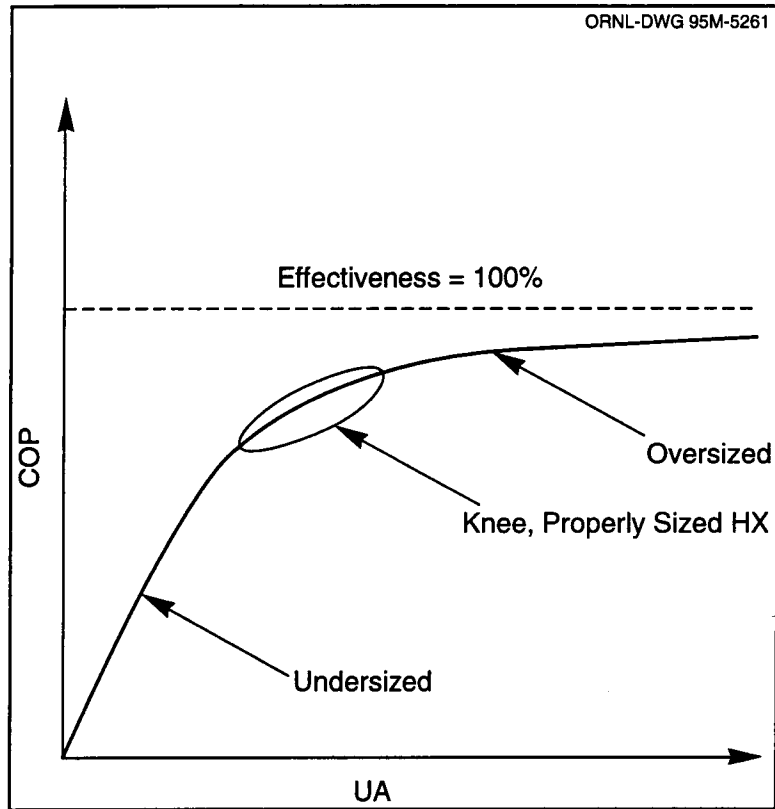


Figure 22. Approach Used by McGahey and Christensen for Parameter Optimization.

The approach used for optimization in this thesis was to evaluate how the initial manufacturing cost of the DCC cycle changed while varying the component sizes (UA values) of the evaporator, absorber, condenser, and each of the three generators independent of each other. Initial manufacturing cost is calculated from an equation presented by Herold, et.al. [58]:

$$Z = \sum c_i UA_i$$

where Z = cost function in \$

c_i = relative cost of each component in \$/BTU/hr-F

UA_i = component heat transfer coefficient times area in BTU/hr-F

This equation can be rewritten for the current chiller cycle under study, as follows:

$$Z = (cUA)_{\text{evaporator}} + (cUA)_{\text{absorber}} + (cUA)_{\text{condenser}} + (cUA)_{\text{generator1}} + (cUA)_{\text{heat exchanger1}} + \\ (cUA)_{\text{generator2}} + (cUA)_{\text{heat exchanger2}} + (cUA)_{\text{generator3}} + (cUA)_{\text{heat exchanger3}}$$

Since actual costing information for each component studied here was not available, relative cost values were determined as follows. The cost of each component is proportional to the area of the component using the following relationships.

1. The cost of each component is proportional to the number of tubes used in the component and the relative cost of the material of each component. Labor costs are proportional to the number of tubes in each component.
2. The shell material cost for each component is neglected in these calculations for two reasons: 1) It is common practice in industry to utilize the same shell for several components (i.e. the low temperature generator, condenser, evaporator and absorber are all in the same shell in some double-effect chillers), and 2) Manufacturers often use the same shell for a range of operating loads, and vary the number of tubes to change the area and corresponding UA for that component.

A material cost multiplier of 1.0 is used for each of the components that are made of copper tubes.

A material cost multiplier of 6.86 [59] is used for components that are constructed of 70-30

Copper-Nickel. These components include the highest temperature generator and highest temperature heat exchanger.

Utilizing the above assumptions produces the following relationship:

$$Z \propto UA_{\text{evaporator}} + UA_{\text{absorber}} + UA_{\text{condenser}} + UA_{\text{generator1}} + UA_{\text{heat exchanger1}} \\ + UA_{\text{generator2}} + UA_{\text{heat exchanger2}} + 6.86 * UA_{\text{generator3}} + 6.86 * UA_{\text{heat exchanger3}}$$

To perform the parametric analysis, numerous runs were performed using the ABSIM cycle model. The base case conditions were held constant, while varying the UA value of one component. The UA value was varied to represent a change in size of the component. The evaporator load was calculated for each value of UA and then plotted on a curve of Z/Q versus Q to obtain a minimum inflection point. The evaporator load was chosen for the heat load value (Q) because that is the output of the system for which the customer pays (eg., chilled water for the absorption chiller). Further, standard industrial practice is to sell chiller units in \$/ton.

Figures 23 to 28 depict the relative cost (using Z calculated with the above equation divided by Q) versus evaporator heat load (Q) for each of the major components. From each figure, the individual component UA value that corresponded to the lowest relative cost was selected as the optimum UA value for that component. Finally, each component's optimum UA value was used in cycle calculations using ABSIM. Table 2 depicts one operating condition for these cycle calculations.

Figure 29 represents relative cost per ton for the base case and the optimized case versus heat load. This curve demonstrates that the optimized machine is cheaper per unit ton across all operating ranges. In addition, larger capacities can be obtained by using the optimized cycle.

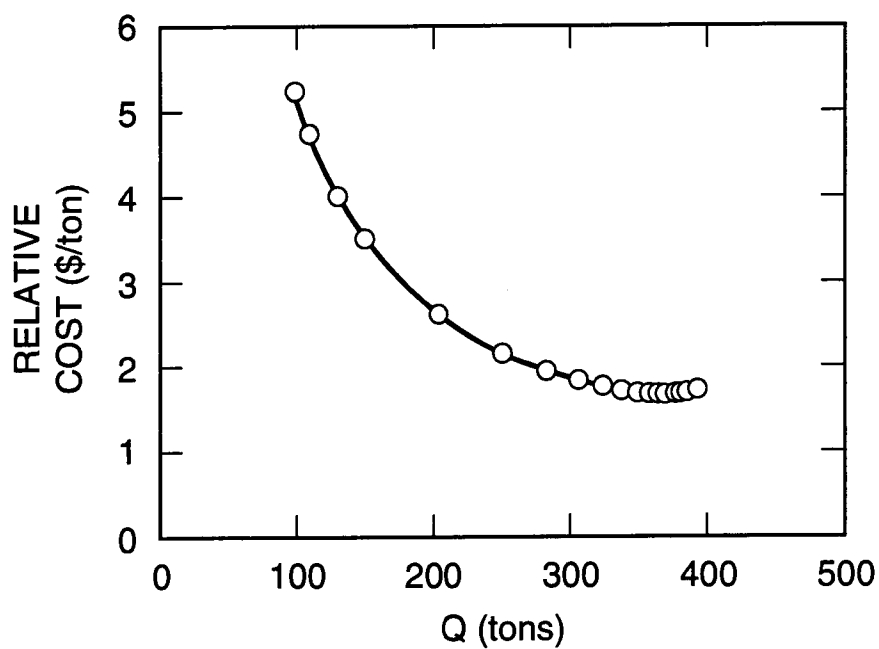


Figure 23. Relative Cost versus Evaporator Heat Load over a range of Evaporator UA values

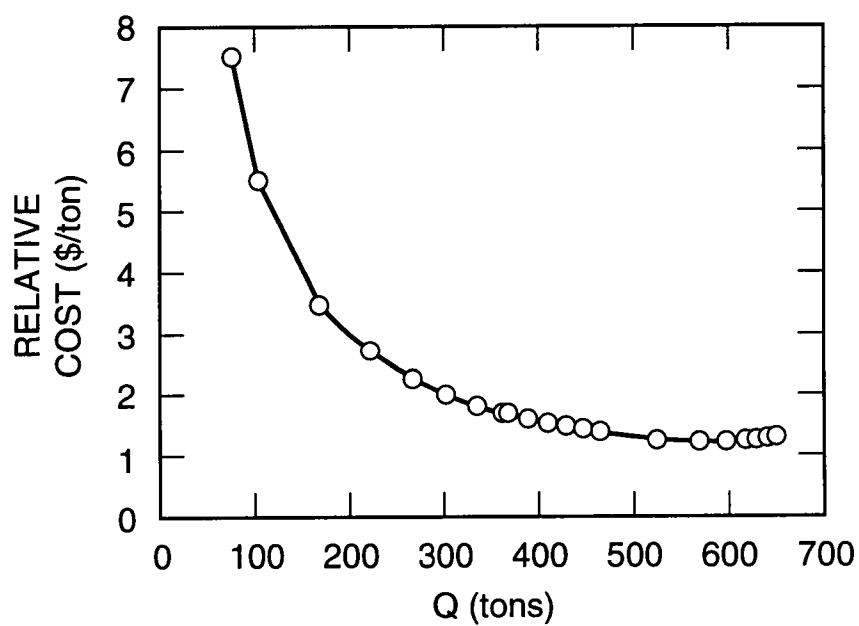


Figure 24. Relative Cost versus Evaporator Heat Load over a range of Absorber UA values

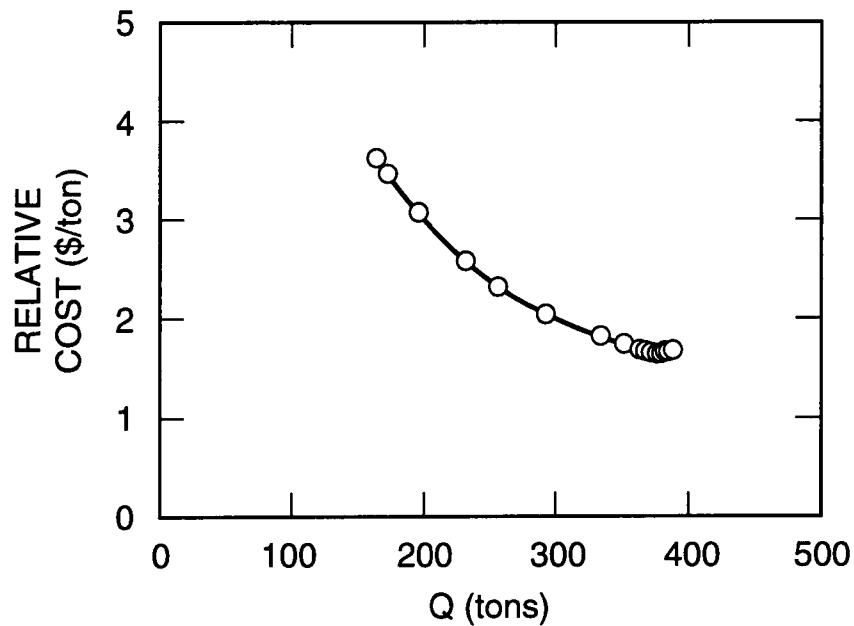


Figure 25. Relative Cost versus Evaporator Heat Load over a range of Condenser UA values

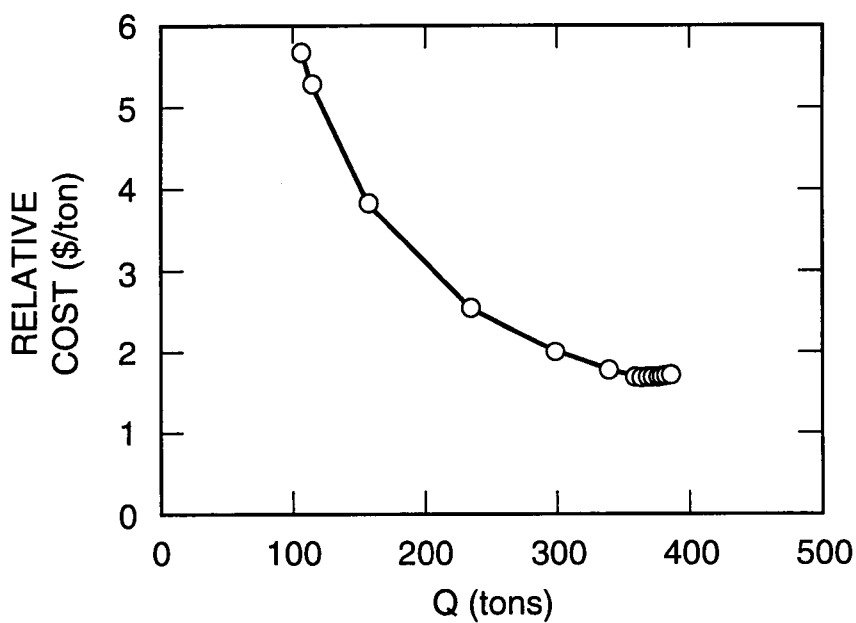


Figure 26. Relative Cost versus Evaporator Heat Load over a range of Lowest Temperature Generator UA values

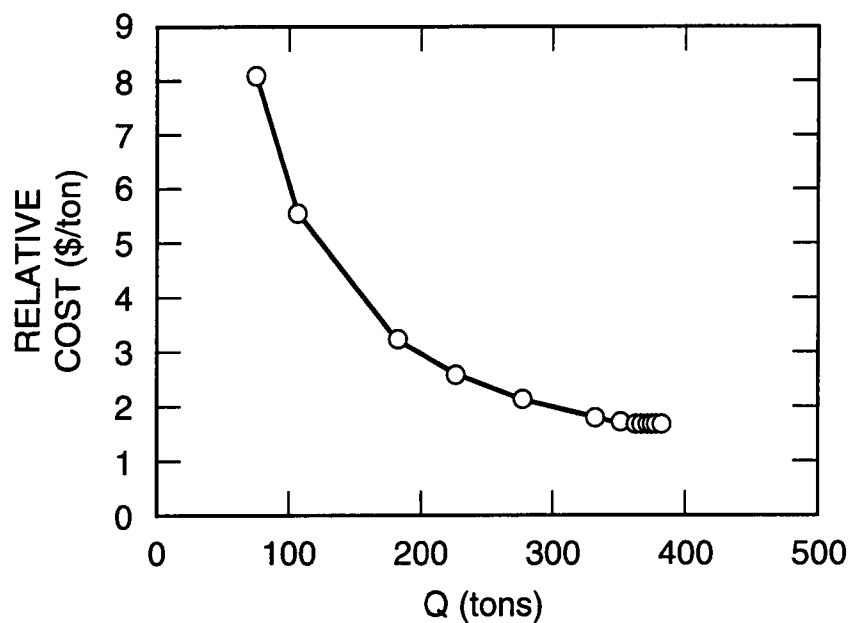


Figure 27. Relative Cost versus Evaporator Heat Load over a range of Middle Temperature Generator UA values

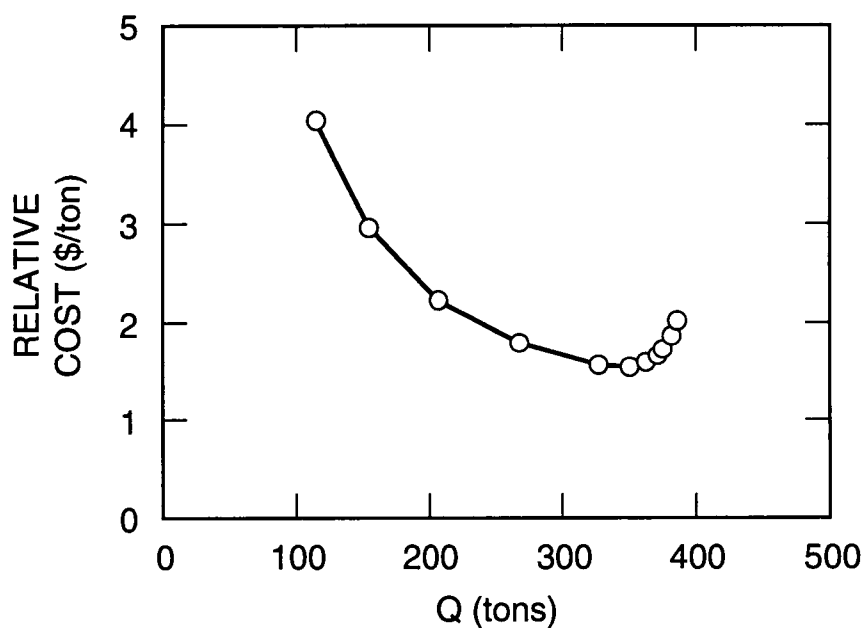


Figure 28. Relative Cost versus Evaporator Heat Load over a range of Highest Temperature Generator UA values

Triple-Effect DCC Cycle Operating Conditions After Cost Optimization

Flowrate = 4,683 lb/min

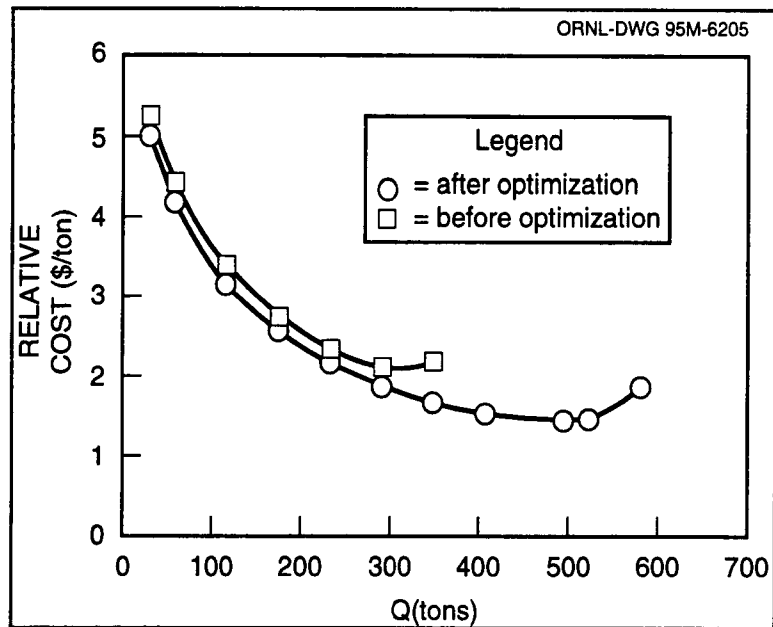


Figure 29. Relative Cost Versus Heat Load for Base Case and Optimized Case Triple-Effect Absorption Cycles

A common measure used in this industry to compare cycles is COP. Figure 30 represents COP versus heat load for the optimized case and the base case. This plot demonstrates that comparable COPs were obtained relative to the base case. The advantage of the optimized cycle is that it has a cheaper initial manufacturing cost (as discussed above) and offers the ability to run the chiller at much higher heat loads than the base case.

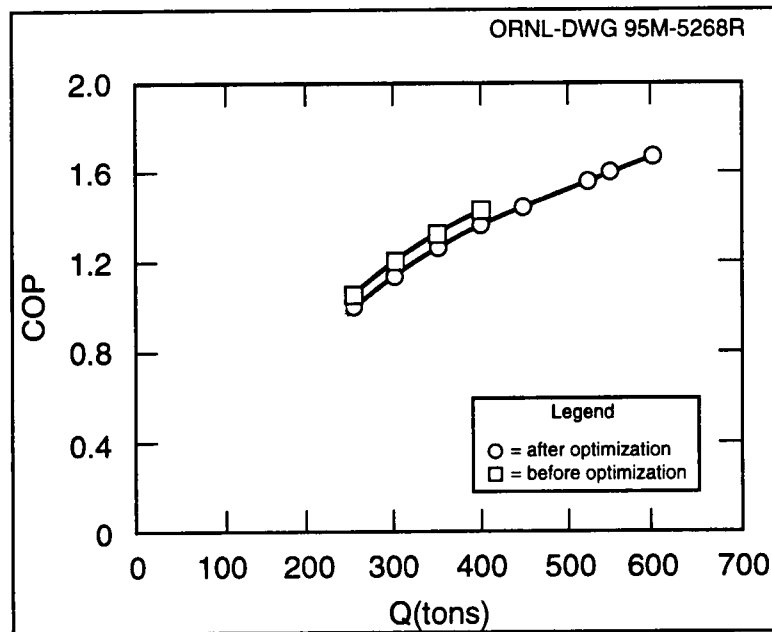


Figure 30. COP versus Heat Load for Base Case and Optimized Case Triple-Effect Absorption Cycles

8. Conclusions & Recommendations

Theoretical and empirical correlations were used to develop component models that calculate individual and overall heat transfer coefficients for the evaporator, condenser, absorber, and generator components of an absorption chiller system. The UA values from the component models were compared to experimentally determined values based on operating data from a double-effect LiBr/H₂O absorption chiller. The calculated UA values matched the experimentally determined UA values within 20% for the evaporator, 10% for the condenser, 15% for the absorber, and within 45% for the generator. The experimental data used for comparisons was primarily taken over a narrow range of operating conditions. It would be desirable to obtain follow-on experimental data over a larger range of load conditions to perform additional comparisons between experimental and calculated values. It is also desirable to obtain data from chillers of alternate designs to perform additional comparisons, and perhaps develop an alternate correlation, either empirical or theoretical, for the generator.

These component models were linked to the DOE/ORNL-sponsored ABSIM computer model to integrate the design specifications with the cycle analysis. ABSIM is a user-oriented computer model for simulation of absorption systems.

UA values can be used to calculate cycle performance. The utility of valid component UA values was demonstrated using a parametric analysis by varying component UA values and calculating cycle performance using ABSIM. A preliminary cost optimization was performed assuming the cost of the components was proportional to the type and amount of material used for the heat exchange surface. Based on these calculations, optimum sized components for the base operating

case triple-effect cycle are recommended. The optimum sized components performed at an efficiency comparable to the base case at a lower relative cost than the base case across all heat loads. In addition, the optimized cycle offers the ability to run the chiller at much higher heat loads than the base case.

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Appendices

Appendix 1. Variable Definitions

Note: Most of the variables are common to each of the component codes. In the component codes, the condenser variables are marked with a C_, the absorber variables are marked with an A_, and the generator variables are marked with a G_.

area_inner [ft²] = inner surface area of tube bundle
chilw_flowE [lb/min] = mass flowrate of chilled water stream (tubeside)
CPLIQ [BTU/lb-°F] = specific heat capacity
DENL [lb/ft³] = density
fac_foul [BTU/ft²-hr-°F] = fouling factor
HIEV [BTU/ft²-hr-°F] = tubeside, turbulent, heat transfer coefficient in evaporator
HOEV [BTU/ft²-hr-°F] = outside film coefficient in evaporator
mass_flowrE [lb/hr] = mass flowrate of shellside fluid
MUL [lb/ft-hr] = absolute viscosity
NPR = dimensionless Prandtl number
NRE = dimensionless Reynolds number
num_passes = number of passes
num_tubes = number of tubes
pi = constant pi (3.141592653)
ratio_surf = outside to inside area ratio
REV [ft²-hr-°F/BTU] = inverse of U
T [C] = temperature, varies by stream
temp_chilw [C] = chilled water average temperature
temp_chilwE [F]
temp_evap [C] = temperature of the evaporator/component
temp_evapE [F]
THCL [BTU/ft-hr-°F] = thermal conductivity of heat transfer fluid
therm_cond [BTU/ft²-hr-°F/in] = thermal conductivity of tube material
tube_diamE [in] = tube inner diameter
tube_lengthE [ft] = tube length
tube_thickE [in] = tube wall thickness
U [BTU/ft²-hr-°F] = Heat transfer coefficient
value = marker used for IF/THEN statement
vel [ft/sec] = velocity of tubeside fluid
vel_ref [ft/sec] = velocity of shellside fluid

Appendix 2. Unit Conversions

The actual lines from the evaporator computer component model were pulled out and checked for unit accuracies. Any equations that were different in the condenser, absorber, and generator are added at the end of this appendix.

Evaporator

The Dittus and Boelter Equation is used to calculate the turbulent tubeside (inside) coefficient. This equation applies to fully developed turbulent flow ($N_{Re} > 10,000$) of fluids for $0.7 < N_{Pr} < 700$. In pipes for which $L/D > 60$, the entrance effects are negligible. All fluid properties are evaluated at the arithmetic mean bulk temperature:

$$\frac{h_i D}{k} = 0.023 \left(\frac{D v \rho}{\mu} \right)^{0.8} \left(\frac{C_p \mu}{k} \right)^{0.3 \text{ or } 0.4}$$

where h_i = average inside heat transfer coefficient in BTU/hr-ft²-°F
 D = tube inside diameter in ft
 k = thermal conductivity in BTU/hr-ft-°F
 v = velocity in ft/hr
 ρ = density in lb/ft³
 μ = absolute viscosity in lb/hr-ft
 C_p = specific heat in BTU/lb-°F

$$\frac{\frac{\text{BTU}}{\text{hr ft}^2 \text{ F}} \cdot \text{ft}}{\frac{\text{BTU}}{\text{hr ft F}}} = 0.023 \left(\frac{\text{ft} \cdot \frac{\text{ft}}{\text{hr}} \cdot \frac{\text{lb}}{\text{ft}^3}}{\frac{\text{lb}}{\text{hr ft}}} \right)^{0.8} \left(\frac{\frac{\text{BTU}}{\text{lb F}} \cdot \frac{\text{lb}}{\text{hr ft}}}{\frac{\text{BTU}}{\text{hr ft F}}} \right)^{0.3}$$

which can be rewritten as:

$$\frac{h_i D}{k} = 0.023 N_{Re}^{0.8} N_{Pr}^{0.3 \text{ or } 0.4}$$

where N_{Re} = dimensionless Reynold's Number
 N_{Pr} = dimensionless Prandtl Number

This equation correlates data for heating of the fluid when the exponent on the Prandtl number is 0.4 and for cooling of the fluid when the exponent is 0.3. In the evaporator, the chilled water is being cooled, so the exponent of 0.3 is used.

For the outside (shell side) coefficient:

For laminar flow, Chun and Seban [53] approximate the heat transfer coefficient for evaporation from the surface of water films as:

$$h_o = 0.606 \left(\frac{k^3 \rho^2 g}{\mu^2} \right)^{0.333} \left(\frac{\Gamma}{\mu} \right)^{-0.22}$$

where h_o = average outside heat transfer coefficient in BTU/hr-ft²-°F

k = thermal conductivity in BTU/hr-ft-°F

ρ = density in lb/ft³

g = gravitational acceleration in ft/hr²

v = velocity in ft/hr

Γ = loading per linear foot in lb/ft-hr

μ = absolute viscosity in lb/hr-ft

$$\frac{\text{BTU}}{\text{hr ft}^2 \text{F}} = 0.606 \left(\frac{\frac{\text{BTU}^3}{\text{hr}^3 \text{ft}^2 \text{F}^3} \cdot \frac{\text{lb}^2}{\text{ft}^6} \cdot \frac{\text{ft}}{\text{hr}^2}}{\frac{\text{lb}^2}{\text{hr}^2 \text{ft}^2}} \right)^{.333} \left(\frac{\frac{\text{lb}}{\text{hr ft}}}{\frac{\text{lb}}{\text{hr ft}}} \right)^{-.222}$$

$$\frac{\text{BTU}}{\text{hr ft}^2 \text{F}} = \left(\frac{\text{BTU}^3}{\text{hr}^3 \text{ft}^6 \text{F}^3} \right)^{.333}$$

which can be rewritten as:

$$h_o = 0.606 \left(\frac{k^3 \rho^2 g}{\mu^2} \right)^{0.333} \left(\frac{\Gamma}{\mu} \right)^{-0.22} \left(4 \right)^{-0.22} \left(\frac{1}{4} \right)^{-0.22}$$

$$= 0.822 \left(\frac{k^3 \rho^2 g}{\mu^2} \right)^{0.333} N_{Re}^{-0.22} \text{ where } N_{Re} = \left(\frac{4\Gamma}{\mu} \right)$$

For turbulent flow, Chun and Seban approximate the outside heat transfer coefficient with the following relation:

$$h_o = 0.0038 \left(\frac{k^3 \rho^2 g}{\mu^2} \right)^{0.333} N_{Re}^{0.4} N_{Pr}^{0.65}$$

These relationships were developed from experiments of water films flowing along the outside surface of vertical tubes. Kern [54] found that when vertical tube correlations were extended to horizontal tubes, the following relationship applies:

$$\Gamma = \frac{m}{L N_t^{0.667}} \quad \frac{\text{lb}}{\text{hr ft}} = \frac{\frac{\text{lb}}{\text{hr}}}{\text{ft}}$$

where m = mass flow rate in lb/hr

L = length of tube in ft

N_t = number of tubes

Condenser

For the outside (shell side) coefficient, McCabe and Smith [55] and McAdams [56] approximate the heat transfer coefficient for film-type condensation on horizontal tubes by the Nusselt theoretical equation. All fluid properties are evaluated at the arithmetic mean film temperature:

$$h_o = 1.51 \left(\frac{4\Gamma}{\mu} \right)^{-0.333} \left(\frac{k^3 \rho^2 g}{\mu^2} \right)^{.333}$$

where Γ is approximated by Kern as:

$$\Gamma = \frac{m}{L N_t^{0.667}}$$

$$\frac{\text{BTU}}{h \cdot \text{ft}^2 \cdot \text{F}} = 1.51 \left(\frac{4 \cdot \frac{\text{lb}}{\text{hr ft}}}{\frac{\text{lb}}{\text{hr ft}}} \right)^{-0.333} \left(\begin{array}{l} \text{Same evaluation} \\ \text{as in Chun + Seban} \\ \text{correlation in evaporator code} \end{array} \right)$$

All Components

Surface Area is calculated as follows:

$$A = (2\pi r) L N_t \quad \text{ft}^2 = (2\pi \text{ ft}) (\text{ft})$$

where A = surface area in ft^2
 r = radius of tube in ft
 L = Length of tube in ft
 N_t = Number of tubes in bundle

Velocity is calculated as follows:

$$v = \frac{m}{A \rho (N_t N_p)} \quad \frac{\text{ft}}{\text{hr}} = \frac{\frac{\text{lb}}{\text{hr}}}{\text{ft}^2 \cdot \frac{\text{lb}}{\text{ft}^3}}$$

where v = velocity in ft/hr
 m = mass flow rate in lb/hr
 A = cross-sectional area (πr^2)
 ρ = density in lb/ft^3
 N_t = number of tubes in bundle
 N_p = number of passes

Appendix 3. Component Fortran Codes

Appendix 3.1 Evaporator Code

Appendix 3.2 Condenser Code

Appendix 3.3 Absorber Code

Appendix 3.4 Generator Code

Appendix 3.1 Evaporator Code

CC EVAPORATOR

```
CC This code was written for the love of science.      ***
CC   - This code will query the user for input variables
CC   - It will accept input variables
CC   - It will print the answers to screen
CC   - Trish, Sept. 1994.

REAL*8 num_tubes,tube_thickE
REAL*8 temp_chilw,temp_chilwE,num_passes
REAL*8 temp_evapE,temp_evap
REAL*8 tube_lengthE,tube_diamE
REAL*8 ratio_surf,fac_foul,area_inner
REAL*8 mass_flowrE,therm_cond
REAL*8 pi,REV
REAL*8 vel,vel_ref,value,chilw_flowE
REAL*8 HIEV,U,T,HOE,THCL,DENL,CPLIQ,MUL,NPR,NRE

character*5 prompt
REAL*8 item
```

cc ***** Initial Values*****

```
data num_tubes      /      312/
data tube_lengthE /      6.858/
data tube_diamE     /      .0474/
data tube_thickE    /      .028/
data num_passes     /        3/
data ratio_surf     /      3.33/
data fac_foul       /     4000/
data temp_chilwE    /       50/
data temp_evapE     /       42/
data mass_flowrE    /     4380/
data therm_cond     /     2700/
data chilw_flowE    /     7485/
```

```
CC ***** Rudimentary clearsreen *****
100 WRITE (*,9600)
```

CC***** Data Echo Section*****

```
WRITE (*,2000) num_tubes
WRITE (*,2010) tube_lengthE
WRITE (*,2020) tube_diamE
WRITE (*,2030) tube_thickE
WRITE (*,2050) num_passes
WRITE (*,2070) ratio_surf
```

```

WRITE (*,2080) fac_foul
WRITE (*,2100) temp_chilwE
WRITE (*,2110) temp_evapE
WRITE (*,2120) mass_flowrE
WRITE (*,2130) therm_cond
WRITE (*,2140) chilw_flowE

```

```

CC**** Data Input Prompting: ALL, item number or blank? ****
write (*,2180)
read(*,'(a)') prompt

```

```

if (index(prompt,'A')+index(prompt,'a') .ne. 0) then

```

```

CC**** Data Input Section****

```

```

WRITE (*,2000) num_tubes ,': '
READ (*,*) num_tubes
WRITE (*,2010) tube_lengthE ,': '
READ (*,*) tube_lengthE
WRITE (*,2020) tube_diamE ,': '
READ (*,*) tube_diamE
WRITE (*,2030) tube_thickeE ,': '
READ (*,*) tube_thickeE
WRITE (*,2050) num_passes ,': '
READ (*,*) num_passes
WRITE (*,2070) ratio_surf ,': '
READ (*,*) ratio_surf
WRITE (*,2080) fac_foul ,': '
READ (*,*) fac_foul
WRITE (*,2100) temp_chilwE ,': '
READ (*,*) temp_chilwE
WRITE (*,2110) temp_evapE ,': '
READ (*,*) temp_evapE
WRITE (*,2120) mass_flowrE ,': '
READ (*,*) mass_flowrE
WRITE (*,2130) therm_cond ,': '
READ (*,*) therm_cond
WRITE (*,2140) chilw_flowE ,': '
READ (*,*) chilw_flowE

```

```

elseif (prompt .ne. ' ') then

```

```

CC**** Data Input Section: One Item at a Time ****

```

```

1000 read(prompt,'(f5.0)',err=1010) item
      go to 1020
1010 item=0
1020 if (item .eq. 1) then
      WRITE (*,2000) num_tubes ,': '
      READ (*,*) num_tubes

```

```

elseif (item .eq. 2) then
  WRITE (*,2010) tube_lengthE,': '
  READ (*,*) tube_lengthE
elseif (item .eq. 3) then
  WRITE (*,2020) tube_diamE ,': '
  READ (*,*) tube_diamE
elseif (item .eq. 4) then
  WRITE (*,2030) tube_thickE ,': '
  READ (*,*) tube_thickE
elseif (item .eq. 5) then
  WRITE (*,2050) num_passes ,': '
  READ (*,*) num_passes
elseif (item .eq. 6) then
  WRITE (*,2070) ratio_surf ,': '
  READ (*,*) ratio_surf
elseif (item .eq. 7) then
  WRITE (*,2080) fac_foul ,': '
  READ (*,*) fac_foul
elseif (item .eq. 8) then
  WRITE (*,2100) temp_chilwE ,': '
  READ (*,*) temp_chilwE
elseif (item .eq. 9) then
  WRITE (*,2110) temp_evapE ,': '
  READ (*,*) temp_evapE
elseif (item .eq.10) then
  WRITE (*,2120) mass_flowrE ,': '
  READ (*,*) mass_flowrE
elseif (item .eq.11) then
  WRITE (*,2130) therm_cond ,': '
  READ (*,*) therm_cond
elseif (item .eq.12) then
  WRITE (*,2140) chilw_flowE ,': '
  READ (*,*) chilw_flowE
else
  write(*,'(a)') ' Invalid item number'
endif

1050 write(*,2190)
  read(*,'(a)') prompt
  if (prompt .ne. ' ') go to 1000
endif

pi = 3.141592653

2000 FORMAT(' 1 - Number of tubes (1 or more):      ',
+          f10.0,a)
2010 FORMAT(' 2 - Tube length (ft):                  ',
+          f10.2,a)

```

```

2020 FORMAT(' 3 - Tube inside diameter (feet):      ',
+      f10.3,a)
2030 FORMAT(' 4 - Tube wall thickness (in):          ',
+      f10.3,a)
2050 FORMAT(' 5 - The number of circuits:           ',
+      f10.0,a)
2070 FORMAT(' 6 - The outside to inside area ratio:   ',
+      f10.2,a)
2080 FORMAT(' 7 - The fouling factor (BTU/ft2-F-hr):  ',
+      f10.0,a)
2100 FORMAT(' 8 - Tube chilled water average temperature (F): ',
+      f10.1,a)
2110 FORMAT(' 9 - The evaporator temperature (F):    ',
+      f10.1,a)
2120 FORMAT(' 10 - The mass flow of refrigerant (lb/hr): ',
+      f10.1,a)
2130 FORMAT(' 11 - The thermal conductivity of the tube matr'/
+      ' (BTU/hr-ft-F/in): ',
+      f10.1,a)
2140 FORMAT(' 12 - The chilled water flowrate (lb/min): ',
+      f10.1,a)
2180 format(' Enter item number to change (blank for no changes,',
+      ' ALL for all items): \')
2190 format(' Enter item number (blank when finished): \')

```

C ****Convert quantities to S.I. units****

```

temp_chilw=(temp_chilwE - 32) / 1.8
temp_evap=(temp_evapE - 32)/ 1.8

```

C This subroutine calculates the inside surface area (A, ft2)
C of a tube bundle

```

area_inner=num_tubes*tube_lengthE*tube_diamE*pi

```

CC Calculate the tubeside, turbulent, heat transfer

CC coefficient for the evaporator based on the Seider-Tate Equation.

```

T=temp_chilw

```

```

CALL Properties (T, THCL, DENL, CPLIQ, MUL)

```

```

NPR = CPLIQ * MUL /THCL

```

```

vel = chilw_flowE*60/(pi*tube_diamE**2/4)/DENL/
+      (num_tubes/num_passes)
NRE = tube_diamE*vel*DENL/MUL

```

```

HIEV = 0.023 * NRE**.8 * NPR**.3 * THCL/tube_diamE

```

C Calculate HFEV by subroutines based on dimensionless numbers.
 C This subroutine calculates the outside film coefficient for the
 C refrigerant evaporating outside of horizontal tube bundle.
 C First, the Reynolds Number (NRE) and Prandtl (NPR) numbers are calculated
 C and tested for turbulence.

T=temp_evap

CALL Properties (T, THCL, DENL, CPLIQ, MUL)

vel_ref = mass_flowrE/tube_lengthE/(num_tubes**0.667)

NPR = CPLIQ * MUL / THCL

NRE = 4*vel_ref/MUL

C Calculate outside film coefficient HOE, for turbulent or laminar flow case.

value = 10000

IF (NRE.GT.value) THEN

HOE= .0038 * NRE ** .4 * NPR ** .65 *
 + (THCL**3 * 32.2*(3600**2) * DENL**2 / MUL ** 2) ** .333

ELSE

HOE = .822 * (THCL**3 * 32.2 *(3600**2)* DENL**2/MUL**2)** .333 *
 + NRE ** (-.22)

ENDIF

CC 1/UAi =1/hiAi + thickness/kAm + thickness/fouling*Am +1/hoAo

CC where Ao = (Ao/Ai)(Ai) = (ratio_surf)(area_inner)

REV = 1/HIEV + tube_thickE/therm_cond
 + + 1/fac_foul + 1/(HOE*ratio_surf)

U = 1 / REV

CC**** Display Input Data ****

WRITE (*,*) ('=', j = 1, 60)

WRITE (*,*) 'Input Data:'

WRITE (*,2000) num_tubes

WRITE (*,2010) tube_lengthE

WRITE (*,2020) tube_diamE

WRITE (*,2030) tube_thickE

WRITE (*,2050) num_passes

```

WRITE (*,2070) ratio_surf
WRITE (*,2080) fac_foul
WRITE (*,2100) temp_chilwE
WRITE (*,2110) temp_evapE
WRITE (*,2120) mass_flowrE
WRITE (*,2130) therm_cond
WRITE (*,2140) chilw_flowE

```

CC**** Display Results ****

```

WRITE (*,9200)
WRITE (*,*) ('=', j = 1, 60)
WRITE (*,9210) HIEV
WRITE (*,9220) HOE
WRITE (*,9300) U
WRITE (*,*) ('=', j = 1, 60)

```

```

9200 FORMAT( //' Heat Transfer Information' , ': ', 'ORNL, ',
+ ' ABSIM module, *PWA*' )
9210 FORMAT( ' Inside resistance           =', F12.4,
+ ' BTU/hr-ft2-F' )
9220 FORMAT( ' Outside resistance          =', F12.4,
+ ' BTU/hr-ft2-F' )
9300 FORMAT( ' Overall heat transfer coefficient (U)   =', F12.2,
+ ' BTU/hr-ft2-F' )
9600 FORMAT( // )

```

CC**** Another run? ****

```

write (*,9990)
9990 format(' Do you want to do another run (Y/N)? \')
read(*,'(a)') prompt
if (index(prompt,'Y')+index(prompt,'y') .ne. 0) GO TO 100
STOP
END

```

C This subroutine calculates the properties of steam and water.

```

SUBROUTINE Properties (T, THCL, DENL, CPLIQ, MUL)
REAL*8      T,THCL,DENL,CPLIQ,MUL,ETA
REAL*8      THCL_S,DENL_S,CPLIQ_S,MUL_S

```

C THCL is thermal conductivity in BTU/hr-ft-R at T C.

```

THCL_S = (.5612 + .0019204 * T - 6.217E-06 * T ** 2 - 9E-09
+ * T ** 3) / 1000
THCL = THCL_S/1.8*56.8699*60/3.28084

```

C DENL is density in lb/ft3 at T C.

```

DENL_S = (1.0009 - .0000839 * T - 3.46E-06 * T ** 2) * 1000

```

```

DENL = DENL_S*2.20462/35.3147
C is the specific heat capacity in BTU/lb R at T C.
CPLIQ_S = 4.2177 - .0016178 * T + 1.6006E-05 * T ** 2
CPLIQ = CPLIQ_S*.947831/1.8/2.20462
C MUL is the absolute viscosity in lb/ft-hr at T C.
IF (T.GT.20) THEN
  ETA = -1.99916 + (1.3272 * (20 - T) - .001053 *
+      (T - 20) ** 2) / (T + 105)
  MUL_S = (EXP(ETA * 2.303)) / 10
ELSE
  ETA = 1301 / (998.338 + 8.1855 * (T - 20) + .00585
+      * (T - 20) ** 2) - 3.30233
  MUL_S = (EXP(ETA * 2.303)) / 10
END IF
MUL = MUL_S*2.20462*3600/3.28084

RETURN
END

```

Appendix 3.2 Condenser Code

CC CONDENSER CODE

```
CC This code was written for the love of science. ***
CC - This code will query the user for input variables
CC - It will accept input variables
CC - It will print the answers to screen
CC - Trish, Sept. 1994.

REAL*8 numC_tubes,tubeC_thicke
REAL*8 tempC_chilw,tempC_chilwE,numC_passes
REAL*8 tempC_condE,tempC_cond
REAL*8 tubeC_lengthE,tubeC_diamE
REAL*8 ratioC_surf,facC_foul,areaC_inner
REAL*8 massC_flowrE,thermC_cond
REAL*8 pi,REV
REAL*8 velC_ref,chilwC_flowE
REAL*8 HICO,U,T,HOC,THCL,DENL,CPLIQ,MUL,NPR,NRE

character*5 prompt
REAL*8 item
```

```
cc **** Initial Values****
```

```
cc** Insert initial values at Question Marks**
```

```
data numC_tubes / ?/
data tubeC_lengthE / ?/
data tubeC_diamE / ?/
data tubeC_thicke / ?/
data numC_passes / ?/
data ratioC_surf / ?/
data facC_foul / ?/
data tempC_chilwE / ?/
data tempC_condE / ?/
data massC_flowrE / ?/
data thermC_cond / ?/
data chilwC_flowE / ?/
```

```
CC **** Rudimentary clearsreen ****
```

```
100 WRITE (*,9600)
```

```
CC**** Data Echo Section****
```

```
WRITE (*,2000) numC_tubes
WRITE (*,2010) tubeC_lengthE
WRITE (*,2020) tubeC_diamE
WRITE (*,2030) tubeC_thicke
WRITE (*,2050) numC_passes
```

```

WRITE (*,2070) ratioC_surf
WRITE (*,2080) facC_foul
WRITE (*,2100) tempC_chilwE
WRITE (*,2110) tempC_condE
WRITE (*,2120) massC_flowrE
WRITE (*,2130) thermC_cond
WRITE (*,2140) chilwC_flowE

```

CC**** Data Input Prompting: ALL, item number or blank? ****

```

write (*,2180)
read(*,'(a)') prompt

```

```

if (index(prompt,'A')+index(prompt,'a') .ne. 0) then

```

CC**** Data Input Section****

```

WRITE (*,2000) numC_tubes ,': '
READ (*,*) numC_tubes
WRITE (*,2010) tubeC_lengthE ,': '
READ (*,*) tubeC_lengthE
WRITE (*,2020) tubeC_diamE ,': '
READ (*,*) tubeC_diamE
WRITE (*,2030) tubeC_thickE ,': '
READ (*,*) tubeC_thickE
WRITE (*,2050) numC_passes ,': '
READ (*,*) numC_passes
WRITE (*,2070) ratioC_surf ,': '
READ (*,*) ratioC_surf
WRITE (*,2080) facC_foul ,': '
READ (*,*) facC_foul
WRITE (*,2100) tempC_chilwE ,': '
READ (*,*) tempC_chilwE
WRITE (*,2110) tempC_condE ,': '
READ (*,*) tempC_condE
WRITE (*,2120) massC_flowrE ,': '
READ (*,*) massC_flowrE
WRITE (*,2130) thermC_cond ,': '
READ (*,*) thermC_cond
WRITE (*,2140) chilwC_flowE ,': '
READ (*,*) chilwC_flowE

```

```

elseif (prompt .ne. ' ') then

```

CC**** Data Input Section: One Item at a Time ****

```

1000 read(prompt,'(f5.0)',err=1010) item
      go to 1020
1010 item=0
1020 if (item .eq. 1) then
      WRITE (*,2000) numC_tubes ,': '

```

```

        READ (*,*) numC_tubes
    elseif (item .eq. 2) then
        WRITE (*,2010) tubeC_lengthE ,':'
        READ (*,*) tubeC_lengthE
    elseif (item .eq. 3) then
        WRITE (*,2020) tubeC_diamE ,':'
        READ (*,*) tubeC_diamE
    elseif (item .eq. 4) then
        WRITE (*,2030) tubeC_thickE ,':'
        READ (*,*) tubeC_thickE
    elseif (item .eq. 5) then
        WRITE (*,2050) numC_passes ,':'
        READ (*,*) numC_passes
    elseif (item .eq. 6) then
        WRITE (*,2070) ratioC_surf ,':'
        READ (*,*) ratioC_surf
    elseif (item .eq. 7) then
        WRITE (*,2080) facC_foul ,':'
        READ (*,*) facC_foul
    elseif (item .eq. 8) then
        WRITE (*,2100) tempC_chilwE ,':'
        READ (*,*) tempC_chilwE
    elseif (item .eq. 9) then
        WRITE (*,2110) tempC_condE ,':'
        READ (*,*) tempC_condE
    elseif (item .eq.10) then
        WRITE (*,2120) massC_flowrE ,':'
        READ (*,*) massC_flowrE
    elseif (item .eq.11) then
        WRITE (*,2130) thermC_cond ,':'
        READ (*,*) thermC_cond
    elseif (item .eq.12) then
        WRITE (*,2140) chilwC_flowE ,':'
        READ (*,*) chilwC_flowE
    else
        write(*,'(a)') ' Invalid item number'
    endif

1050  write(*,2190)
      read(*,'(a)') prompt
      if (prompt .ne. ' ') go to 1000
    endif

    pi = 3.141592653

2000 FORMAT(' 1 - Number of tubes (1 or more):',
+          f10.0,a)
2010 FORMAT(' 2 - Tube length (ft):',

```

```

+      f10.2,a)
2020 FORMAT(' 3 - Tube inside diameter (ft):      ',
+      f10.3,a)
2030 FORMAT(' 4 - Tube wall thickness (in):      ',
+      f10.3,a)
2050 FORMAT(' 5 - The number of circuits:      ',
+      f10.0,a)
2070 FORMAT(' 6 - The outside to inside area ratio:      ',
+      f10.2,a)
2080 FORMAT(' 7 - The fouling factor (BTU/hr-ft2-F):      ',
+      f10.0,a)
2100 FORMAT(' 8 - Cooling tower water input temperature (F): ',
+      f10.0,a)
2110 FORMAT(' 9 - The condenser temperature (F):      ',
+      f10.1,a)
2120 FORMAT(' 10 - The mass flow of refrigerant (lb/min):      ',
+      f10.1,a)
2130 FORMAT(' 11 - The thermal conductivity of the tube matrl'/
+      ' (BTU/hr-ft2-F/in):      ',
+      f10.0,a)
2140 FORMAT(' 12 - The chilled water flowrate (lb/hr):      ',
+      f10.1,a)

2180 format('/ Enter item number to change (blank for no changes,',
+      ' ALL for all items): ')
2190 format(' Enter item number (blank when finished): ')

```

C *****Convert quantities to S.I. units*****

```

tempC_chilw=(tempC_chilwE - 32) / 1.8
tempC_cond=(tempC_condE - 32)/ 1.8

```

C This subroutine calculates inside surface area (A, ft2) of a tube bundle

```

areaC_inner=numC_tubes*tubeC_lengthE*tubeC_diamE*pi

```

CC Calculate the tubeside, turbulent, heat transfer

CC coefficient for the condenser based on the Sieder-Tate Equation.

```

T=tempC_chilw
CALL Properties (T, THCL, DENL, CPLIQ, MUL)

```

```

NPR = CPLIQ * MUL /THCL
vel = chilwC_flowE*60/(pi*tubeC_diamE**2/4)/DENL/
+ (numC_tubes/numC_passes)
NRE = tubeC_diamE*vel*DENL/MUL

```

```

HICO = 0.023 * NRE**.8 * NPR**.4 * THCL/tubeC_diamE

```

C Calculate HFCO by correlation developed by McCabe and Smith or McAdams.
 C This subroutine calculates the outside film coefficient for the
 C refrigerant condensing outside of horizontal tube bundle.

T=tempC_cond

CALL Properties (T, THCL, DENL, CPLIQ, MUL)

velC_ref = massC_flowrE/(tubeC_lengthE*numC_tubes**0.667)

HOC=1.51*(4*velC_ref/MUL)**(-.33)*
 + (THCL**3*DENL**2*32.2*(3600**2)/MUL**2)**0.333

CC 1/U_{ai} = 1/h_i_{ai} + thickness/k_{Am} + thickness/fouling*_{Am} + 1/h_o_{ao}

CC where $A_o = (A_o/A_i)(A_i) = (\text{ratioC_surf})(\text{areaC_inner})$

REV=1/HICO + tubeC_thickE/thermC_cond
 + 1/facC_foul + 1/(HOC*ratioC_surf)

U = 1 / REV

CC**** Display Input Data ****

WRITE (*,*) ('=', j = 1, 60)
 WRITE (*,*) 'Input Data:'
 WRITE (*,2000) numC_tubes
 WRITE (*,2010) tubeC_lengthE
 WRITE (*,2020) tubeC_diamE
 WRITE (*,2030) tubeC_thickE
 WRITE (*,2050) numC_passes
 WRITE (*,2070) ratioC_surf
 WRITE (*,2080) facC_foul
 WRITE (*,2100) tempC_chilwE
 WRITE (*,2110) tempC_condE
 WRITE (*,2120) massC_flowrE
 WRITE (*,2130) thermC_cond
 WRITE (*,2140) chilwC_flowE

CC**** Display Results ****

WRITE (*,9200)
 WRITE (*,*) ('=', j = 1, 60)
 WRITE (*,9210) HICO
 WRITE (*,9220) HOC
 WRITE (*,9300) U
 WRITE (*,*) ('=', j = 1, 60)

9200 FORMAT(// 'Heat Transfer Information' , ': ', 'ORNL', '

```

+ ' ABSIM module, *PWA*' )
9210 FORMAT( ' Inside resistance           =', F12.4,
+           ' BTU/hrft2F' )
9220 FORMAT( ' Outside resistance         =', F12.4,
+           ' BTU/hrft2F' )
9300 FORMAT( ' Overall heat transfer coefficient (U)   =', F12.2,
+           ' BTU/hrmin2F' )
9600 FORMAT( // )

```

CC**** Another run? ****

```

write (*,9990)
9990 format(' Do you want to do another run (Y/N)? \')
read(*,'(a)') prompt
if (index(prompt,'Y')+index(prompt,'y') .ne. 0) GO TO 100
STOP
END

```

C This subroutine calculates the properties of steam and water.

```

SUBROUTINE Properties (T, THCL, DENL, CPLIQ, MUL)
REAL*8      T,THCL,DENL,CPLIQ,MUL,ETA
REAL*8      THCL_S,DENL_S,CPLIQ_S,MUL_S

C THCL is thermal conductivity in BTU/hr-ft-R at T C.
THCL_S = (.5612 + .0019204 * T - 6.217E-06 * T ** 2 - 9E-09
+         * T ** 3) / 1000
THCL = THCL_S/1.8*56.8699*60/3.28084

C DENL is density in lb/ft3 at T C.
DENL_S = (1.0009 - .0000839 * T - 3.46E-06 * T ** 2) * 1000
DENL = DENL_S*2.20462/35.3147

C is the specific heat capacity in BTU/lb R at T C.
CPLIQ_S = 4.2177 - .0016178 * T + 1.6006E-05 * T ** 2
CPLIQ = CPLIQ_S*.947831/1.8/2.20462

C MUL is the absolute viscosity in lb/ft-hr at T C.
IF (T.GT.20) THEN
ETA = -1.99916 + (1.3272 * (20 - T) - .001053 *
+ (T - 20) ** 2) / (T + 105)
MUL_S = (EXP(ETA * 2.303)) / 10
ELSE
ETA = 1301 / (998.338 + 8.1855 * (T - 20) + .00585
+ * (T - 20) ** 2) - 3.30233
MUL_S = (EXP(ETA * 2.303)) / 10
END IF
MUL = MUL_S*2.20462*3600/3.28084

RETURN
END

```

Appendix 3.3 Absorber Code

CC ABSORBER CODE

```
CC This code was written for the love of science. ***
CC - This code will query the user for input variables
CC - It will accept input variables
CC - It will print the answers to screen
CC - Trish, Sept. 1994.

REAL*8    numA_tubes,capA_load,tubeA_thickE
REAL*8    tempA_chilw,tempA_chilwE,numA_passes
REAL*8    tubeA_lengthE,tubeA_diamE
REAL*8    ratioA_surf, facA_foul, areaA_inner
REAL*8    thermA_cond
REAL*8    pi,REV
REAL*8    velA_ref,chilwA_flowE
REAL*8    HIAB,U,T,HOAB,THCL,DENL,CPLIQ,MUL,NPR,NRE

character*5 prompt
REAL*8    item
```

cc **** Initial Values****

cc**Insert initial values at question marks**

```
data numA_tubes / ?/
data tubeA_lengthE / ?/
data tubeA_diamE / ?/
data tubeA_thickE / ?/
data numA_passes / ?/
data ratioA_surf / ?/
data facA_foul / ?/
data capA_load / ?/
data tempA_chilwE / ?/
data thermA_cond / ?/
data chilwA_flowE / ?/
```

CC **** Rudimentary clearscreen ****

100 WRITE (*,9600)

CC**** Data Echo Section****

```
WRITE (*,2000) numA_tubes
WRITE (*,2010) tubeA_lengthE
WRITE (*,2020) tubeA_diamE
WRITE (*,2030) tubeA_thickE
WRITE (*,2050) numA_passes
WRITE (*,2070) ratioA_surf
WRITE (*,2080) facA_foul
```

```

WRITE (*,2090) capA_load
WRITE (*,2100) tempA_chilwE
WRITE (*,2130) thermA_cond
WRITE (*,2140) chilwA_flowE

```

CC**** Data Input Prompting: ALL, item number or blank? ****

```

write (*,2180)
read(*,'(a)') prompt

```

```

if (index(prompt,'A')+index(prompt,'a') .ne. 0) then

```

CC**** Data Input Section****

```

WRITE (*,2000) numA_tubes  ,': '
  READ (*,*)  numA_tubes
WRITE (*,2010) tubeA_lengthE ,': '
  READ (*,*)  tubeA_lengthE
WRITE (*,2020) tubeA_diamE   ,': '
  READ (*,*)  tubeA_diamE
WRITE (*,2030) tubeA_thickeE ,': '
  READ (*,*)  tubeA_thickeE
WRITE (*,2050) numA_passes   ,': '
  READ (*,*)  numA_passes
WRITE (*,2070) ratioA_surf   ,': '
  READ (*,*)  ratioA_surf
WRITE (*,2080) facA_foul     ,': '
  READ (*,*)  facA_foul
WRITE (*,2090) capA_load     ,': '
  READ (*,*)  capA_load
WRITE (*,2100) tempA_chilwE  ,': '
  READ (*,*)  tempA_chilwE
WRITE (*,2130) thermA_cond   ,': '
  READ (*,*)  thermA_cond
WRITE (*,2140) chilwA_flowE  ,': '
  READ (*,*)  chilwA_flowE

```

```

elseif (prompt .ne. ' ') then

```

CC**** Data Input Section: One Item at a Time ****

```

1000  read(prompt,'(f5.0)',err=1010) item
      go to 1020
1010  item=0
1020  if (item .eq. 1) then
      WRITE (*,2000) numA_tubes  ,': '
      READ (*,*)  numA_tubes
    elseif (item .eq. 2) then
      WRITE (*,2010) tubeA_lengthE ,': '
      READ (*,*)  tubeA_lengthE
    elseif (item .eq. 3) then

```

```

WRITE (*,2020) tubeA_diamE  ','
READ (*,*) tubeA_diamE
elseif (item .eq. 4) then
WRITE (*,2030) tubeA_thickE  ','
READ (*,*) tubeA_thickE
elseif (item .eq. 5) then
WRITE (*,2050) numA_passes  ','
READ (*,*) numA_passes
elseif (item .eq. 6) then
WRITE (*,2070) ratioA_surf  ','
READ (*,*) ratioA_surf
elseif (item .eq. 7) then
WRITE (*,2080) facA_foul  ','
READ (*,*) facA_foul
elseif (item .eq. 8) then
WRITE (*,2090) capA_load  ','
READ (*,*) capA_load
elseif (item .eq. 9) then
WRITE (*,2100) tempA_chilwE  ','
READ (*,*) tempA_chilwE
elseif (item .eq.12) then
WRITE (*,2130) thermA_cond  ','
READ (*,*) thermA_cond
elseif (item .eq.13) then
WRITE (*,2140) chilwA_flowE  ','
READ (*,*) chilwA_flowE
else
write(*,'(a)') ' Invalid item number'
endif

```

```

1050 write(*,2190)
read(*,'(a)') prompt
if (prompt .ne. ' ') go to 1000
endif

```

pi = 3.141592653

```

2000 FORMAT(' 1 - Number of tubes (1 or more):',
+ f10.0,a\),
2010 FORMAT(' 2 - Tube length (ft):',
+ f10.2,a\),
2020 FORMAT(' 3 - Tube inside diameter (ft):',
+ f10.3,a\),
2030 FORMAT(' 4 - Tube wall thickness (in):',
+ f10.3,a\),
2050 FORMAT(' 5 - The number of circuits:',
+ f10.0,a\),

```

```

2070 FORMAT(' 6 - The outside to inside area ratio:      ',
+      f10.2,a)
2080 FORMAT(' 7 - The fouling factor (BTU/hr-ft2-F):      ',
+      f10.0,a)
2090 FORMAT(' 8 - The absorber load (BTU/min):            ',
+      f10.1,a)
2100 FORMAT(' 9 - Cooling tower water average temperature (F): ',
+      f10.1,a)
2130 FORMAT(' 12 - The thermal conductivity of the tube matr'/
+      ' (BTU/hr-ft2-F/in):            ',
+      f10.0,a)
2140 FORMAT(' 13 - The cooling tower water flowrate (lb/min): ',
+      f10.1,a)
2150 FORMAT(' 14 - The fouling thickness (in):            ',
+      f10.3,a)
2180 format('/ Enter item number to change (blank for no changes,',
+      ' ALL for all items): \')
2190 format(' Enter item number (blank when finished): \')

```

C *****Convert quantities to S.I. units*****

tempA_chilw=(tempA_chilwE - 32) / 1.8

C This subroutine calculates inside surface area (A, ft2) of a tube bundle

areaA_inner=numA_tubes*tubeA_lengthE*tubeA_diamE*pi

CC Calculate the tubeside, turbulent, heat transfer

CC coefficient for the absorber based on the Sieder-Tate Equation.

T=tempA_chilw

CALL Properties (T, THCL, DENL, CPLIQ, MUL)

NPR = CPLIQ * MUL /THCL

velA_ref = chilwA_flowE*60/(pi*tubeA_diamE**2/4)/DENL/

+ (numA_tubes/numA_passes)

NRE = tubeA_diamE*velA_ref*DENL/MUL

HIAB = 0.023 * NRE**.8 * NPR**.4 * THCL/tubeA_diamE

C Calculate HFAB by empirical correlations.

HOAB = 273.3 + 0.00777*capA_load*60/areaA_inner

C where the Load is in BTU/hr and the Area is in ft2

CC $1/U_{Ai} = 1/h_{iAi} + \text{thickness}/k_{Am} + \text{thickness}/\text{fouling} * A_m + 1/h_{oAo}$

$$REV = 1/HIAB + tubeA_thickE/thermA_cond$$

$$+ 1/facA_foul + 1/(HOAB*ratioA_surf)$$

$$U = 1 / REV$$

CC**** Display Input Data ****

```
WRITE (*,*) ('=', j = 1, 60)
WRITE (*,*) 'Input Data'
WRITE (*,2000) numA_tubes
WRITE (*,2010) tubeA_lengthE
WRITE (*,2020) tubeA_diamE
WRITE (*,2030) tubeA_thickE
WRITE (*,2050) numA_passes
WRITE (*,2070) ratioA_surf
WRITE (*,2080) facA_foul
WRITE (*,2090) capA_load
WRITE (*,2100) tempA_chilwE
WRITE (*,2130) thermA_cond
WRITE (*,2140) chilwA_flowE
```

CC**** Display Results ****

```
WRITE (*,9200)
WRITE (*,*) ('=', j = 1, 60)
WRITE (*,9210) HIAB
WRITE (*,9220) HOAB
WRITE (*,9300) U
WRITE (*,*) ('=', j = 1, 60)
```

```
9200 FORMAT( // 'Heat Transfer Information' , ': ', 'ORNL', ' ',
+ ' ABSIM module, *PWA*' )
9210 FORMAT( ' Inside resistance           =', F12.4,
+ ' BTU/hrft2F' )
9220 FORMAT( ' Outside resistance          =', F12.4,
+ ' BTU/hrft2F' )
9300 FORMAT( ' Overall heat transfer coefficient (U)   =', F12.2,
+ ' BTU/hrft2F' )
9600 FORMAT( // )
```

CC**** Another run? ****

```
write (*,9990)
9990 format(' Do you want to do another run (Y/N)? \')
read(*,'(a)') prompt
if (index(prompt,'Y')+index(prompt,'y') .ne. 0) GO TO 100
STOP
END
```

C This subroutine calculates the properties of steam and water.

```

SUBROUTINE Properties (T, THCL, DENL, CPLIQ, MUL)
REAL*8      T,THCL,DENL,CPLIQ,MUL,ETA
REAL*8      THCL_S,DENL_S,CPLIQ_S,MUL_S

C THCL is thermal conductivity in BTU/hr-ft-R at T C.
  THCL_S = (.5612 + .0019204 * T - 6.217E-06 * T ** 2 - 9E-09
+           * T ** 3) / 1000
  THCL = THCL_S/1.8*56.8699*60/3.28084
C DENL is density in lb/ft3 at T C.
  DENL_S = (1.0009 - .0000839 * T - 3.46E-06 * T ** 2) * 1000
  DENL = DENL_S*2.20462/35.3147
C is the specific heat capacity in BTU/lb R at T C.
  CPLIQ_S = 4.2177 - .0016178 * T + 1.6006E-05 * T ** 2
  CPLIQ = CPLIQ_S*.947831/1.8/2.20462
C MUL is the absolute viscosity in lb/ft-hr at T C.
  IF (T.GT.20) THEN
    ETA = -1.99916 + (1.3272 * (20 - T) - .001053 *
+      (T - 20) ** 2) / (T + 105)
    MUL_S = (EXP(ETA * 2.303)) / 10
  ELSE
    ETA = 1301 / (998.338 + 8.1855 * (T - 20) + .00585
+      * (T - 20) ** 2) - 3.30233
    MUL_S = (EXP(ETA * 2.303)) / 10
  END IF
  MUL = MUL_S*2.20462*3600/3.28084

RETURN
END

```

Appendix 3.4 Generator Code

CC GENERATOR CODE

```
CC This code was written for the love of science. ***
CC - This code will query the user for input variables
CC - It will accept input variables
CC - It will print the answers to screen
CC - Trish, Sept. 1994.
```

```
REAL*8 numG_tubes,tubeG_thickE,capG_load
REAL*8 tempG_chilw,tempG_chilwE,numG_passes
REAL*8 tubeG_lengthE,tubeG_diamE
REAL*8 ratioG_surf, facG_foul, areaG_inner
REAL*8 thermG_cond
REAL*8 pi,REV
REAL*8 velG_ref,chilwG_flowE
REAL*8 HIG,U,HOG,THCL,DENL,T,CPLIQ,MUL
```

```
character*5 prompt
real*8 item
```

```
cc **** Initial Values****
```

```
cc**Insert initial values at question marks**
```

```
data numG_tubes / ?/
data tubeG_lengthE/ ?/
data tubeG_diamE/ ?/
data tubeG_thickE / ?/
data numG_passes/ ?/
data ratioG_surf / ?/
data facG_foul / ?/
data capG_load / ?/
data tempG_chilwE / ?/
data thermG_cond/ ?/
data chilwG_flowE / ?/
```

```
CC **** Rudimentary clearsreen ****
```

```
100 WRITE (*,9600)
```

```
CC**** Data Echo Section****
```

```
WRITE (*,2000) numG_tubes
WRITE (*,2010) tubeG_lengthE
WRITE (*,2020) tubeG_diamE
WRITE (*,2030) tubeG_thickE
WRITE (*,2050) numG_passes
WRITE (*,2070) ratioG_surf
```

```

WRITE (*,2080) facG_foul
WRITE (*,2090) capG_load
WRITE (*,2100) tempG_chilwE
WRITE (*,2130) thermG_cond
WRITE (*,2140) chilwG_flowE

```

CC**** Data Input Prompting: ALL, item number or blank? ****

```

write (*,2180)
read(*,'(a)') prompt

```

```

if (index(prompt,'A')+index(prompt,'a') .ne. 0) then

```

CC**** Data Input Section****

```

WRITE (*,2000) numG_tubes ,': '
READ (*,*) numG_tubes
WRITE (*,2010) tubeG_lengthE ,': '
READ (*,*) tubeG_lengthE
WRITE (*,2020) tubeG_diamE ,': '
READ (*,*) tubeG_diamE
WRITE (*,2030) tubeG_thicke ,': '
READ (*,*) tubeG_thicke
WRITE (*,2050) numG_passes ,': '
READ (*,*) numG_passes
WRITE (*,2070) ratioG_surf ,': '
READ (*,*) ratioG_surf
WRITE (*,2080) facG_foul ,': '
READ (*,*) facG_foul
WRITE (*,2090) capG_load ,': '
READ (*,*) capG_load
WRITE (*,2100) tempG_chilwE ,': '
READ (*,*) tempG_chilwE
WRITE (*,2130) thermG_cond ,': '
READ (*,*) thermG_cond
WRITE (*,2140) chilwG_flowE ,': '
READ (*,*) chilwG_flowE

```

```

elseif (prompt .ne. '') then

```

CC**** Data Input Section: One Item at a Time ****

```

1000 read(prompt,'(f5.0)',err=1010) item
go to 1020
1010 item=0
1020 if (item .eq. 1) then
WRITE (*,2000) numG_tubes ,': '
READ (*,*) numG_tubes
elseif (item .eq. 2) then
WRITE (*,2010) tubeG_lengthE ,': '
READ (*,*) tubeG_lengthE

```

```

elseif (item .eq. 3) then
  WRITE (*,2020) tubeG_diamE  ,': '
  READ (*,*) tubeG_diamE
elseif (item .eq. 4) then
  WRITE (*,2030) tubeG_thickE  ,': '
  READ (*,*) tubeG_thickE
elseif (item .eq. 5) then
  WRITE (*,2050) numG_passes  ,': '
  READ (*,*) numG_passes
elseif (item .eq. 6) then
  WRITE (*,2070) ratioG_surf  ,': '
  READ (*,*) ratioG_surf
elseif (item .eq. 7) then
  WRITE (*,2080) facG_foul  ,': '
  READ (*,*) facG_foul
elseif (item .eq. 8) then
  WRITE (*,2090) capG_load  ,': '
  READ (*,*) capG_load
elseif (item .eq. 9) then
  WRITE (*,2100) tempG_chilwE  ,': '
  READ (*,*) tempG_chilwE
elseif (item .eq.12) then
  WRITE (*,2130) thermG_cond  ,': '
  READ (*,*) thermG_cond
elseif (item .eq.13) then
  WRITE (*,2140) chilwG_flowE  ,': '
  READ (*,*) chilwG_flowE
else
  write(*,'(a)') ' Invalid item number'
endif

1050 write(*,2190)
      read(*,'(a)') prompt
      if (prompt .ne. ' ') go to 1000
endif

pi = 3.141592653

2000 FORMAT(' 1 - Number of tubes (1 or more):',
+          f10.0,a\),
2010 FORMAT(' 2 - Tube length (ft):',
+          f10.2,a\),
2020 FORMAT(' 3 - Tube inside diameter (ft):',
+          f10.3,a\),
2030 FORMAT(' 4 - Tube wall thickness (in):',
+          f10.3,a\),
2050 FORMAT(' 5 - The number of circuits:',
+          f10.0,a\)

```

```

2070 FORMAT(' 6 - Outside to inside area ratio:      ',
+          f10.2,a)
2080 FORMAT(' 7 - The fouling factor (BTU/hr-ft2-F):   ',
+          f10.0,a)
2090 FORMAT(' 8 - The generator load (BTU/min):        ',
+          f10.1,a)
2100 FORMAT(' 9 - Tube chilled water average temperature (F): ',
+          f10.1,a)
2110 FORMAT(' 10 - The evaporator temperature (F):      ',
+          f10.1,a)
2120 FORMAT(' 11 - The mass flow of refrigerant (lb/hr):  ',
+          f10.1,a)
2130 FORMAT(' 12 - The thermal conductivity of the tube matrl//
+          ' (BTU/hr-ft2-F/in):      ',
+          f10.0,a)
2140 FORMAT(' 13 - The chilled water flowrate (lb/min):   ',
+          f10.1,a)
2150 FORMAT(' 14 - The fouling thickness (in):           ',
+          f10.3,a)
2180 format('/ Enter item number to change (blank for no changes,',
+          ' ALL for all items): \)
2190 format(' Enter item number (blank when finished): \)

```

C ****Convert quantities to S.I. units****

$$\text{tempG_chilw} = (\text{tempG_chilwE} - 32) / 1.8$$

C This subroutine calculates inside surface area (A, ft2) of a tube bundle

$$\text{areaG_inner} = \text{numG_tubes} * \text{tubeG_lengthE} * \text{tubeG_diamE} * \pi$$

CC Calculate the tubeside, turbulent, heat transfer

CC coefficient from the Chun and Seban relationship.

T=TempG_chilw

CALL Properties (T, THCL, DENL, CPLIQ, MUL)

$$\text{velG_ref} = \text{chilwG_flowE} * 60 / (\text{tubeG_lengthE} * \text{numG_tubes} ** 0.667)$$

$$\text{HIG} = 1.51 * (4 * \text{velG_ref} / \text{MUL}) ** (-.33) * \\ + (\text{THCL} ** 3 * \text{DENL} ** 2 * 9.78 / \text{MUL} ** 2) ** .333$$

CC HOG is calculated using an empirical correlation:

$$\text{HOG} = 176.1 + 0.04945 * \text{capG_load} * 60 / \text{areaG_inner}$$

C where Load is in BTU/hr and Area is in ft2.

$$\text{CC } 1/\text{UAi} = 1/\text{hiAi} + \text{thickness}/\text{kAm} + \text{thickness}/\text{fouling} * \text{Am} + 1/\text{hoAo}$$

$$REV = 1/HIG + tubeG_thickE/thermG_cond$$

$$+ 1/facG_foul + 1/(HOG*ratioG_surf)$$

$$U = 1 / REV$$

CC**** Display Input Data ****

```
WRITE (*,*) ('=', j = 1, 60)
WRITE (*,*) 'Input Data:'
WRITE (*,2000) numG_tubes
WRITE (*,2010) tubeG_lengthE
WRITE (*,2020) tubeG_diamE
WRITE (*,2030) tubeG_thickE
WRITE (*,2050) numG_passes
WRITE (*,2070) ratioG_surf
WRITE (*,2080) facG_foul
WRITE (*,2090) capG_load
WRITE (*,2100) tempG_chilwE
WRITE (*,2130) thermG_cond
WRITE (*,2140) chilwG_flowE
```

CC**** Display Results ****

```
WRITE (*,9200)
WRITE (*,*) ('=', j = 1, 60)
WRITE (*,9210) HIG
WRITE (*,9220) HOG
WRITE (*,9300) U
WRITE (*,*) ('=', j = 1, 60)
```

```
9200 FORMAT( // 'Heat Transfer Information' , ': ' , 'ORNL' , ' ,
+ ' ABSIM module, *PWA*' )
9210 FORMAT( ' Inside resistance                =', F12.4,
+          ' BTU/hrft2F' )
9220 FORMAT( ' Outside resistance                =', F12.4,
+          ' BTU/hrft2F' )
9300 FORMAT( ' Overall heat transfer coefficient (U)    =', F12.2,
+          ' BTU/hr-ft2-F' )
9600 FORMAT( '////////////////////////////////' )
```

CC**** Another run? ****

```
write (*,9990)
9990 format(' Do you want to do another run (Y/N)? ')
read(*,'(a)') prompt
if (index(prompt,'Y')+index(prompt,'y') .ne. 0) GO TO 100
STOP
END
```

C This subroutine calculates the properties of steam and water.

```

SUBROUTINE Properties (T, THCL, DENL, CPLIQ, MUL)
REAL*8      T,THCL,DENL,CPLIQ,MUL,ETA
REAL*8      THCL_S,DENL_S,CPLIQ_S,MUL_S

C THCL is thermal conductivity in BTU/hr-ft-R at T C.
  THCL_S = (.5612 + .0019204 * T - 6.217E-06 * T ** 2 - 9E-09
+           * T ** 3) / 1000
  THCL = THCL_S/1.8*56.8699*60/3.28084
C DENL is density in lb/ft3 at T C.
  DENL_S = (1.0009 - .0000839 * T - 3.46E-06 * T ** 2) * 1000
  DENL = DENL_S*2.20462/35.3147
C is the specific heat capacity in BTU/lb R at T C.
  CPLIQ_S = 4.2177 - .0016178 * T + 1.6006E-05 * T ** 2
  CPLIQ = CPLIQ_S*.947831/1.8/2.20462
C MUL is the absolute viscosity in lb/ft-hr at T C.
  IF (T.GT.20) THEN
    ETA = -1.99916 + (1.3272 * (20 - T) - .001053 *
+      (T - 20) ** 2) / (T + 105)
    MUL_S = (EXP(ETA * 2.303)) / 10
  ELSE
    ETA = 1301 / (998.338 + 8.1855 * (T - 20) + .00585
+      * (T - 20) ** 2) - 3.30233
    MUL_S = (EXP(ETA * 2.303)) / 10
  END IF
  MUL = MUL_S*2.20462*3600/3.28084

RETURN
END

```

Appendix 4. Sample Input/Output File for Component Codes

Enter item number to change (blank for no changes, ALL for all items):
=====

Input Data:

1 - Number of tubes (1 or more):	312.
2 - Tube length (ft):	6.86
3 - Tube inside diameter (feet):	.047
4 - Tube wall thickness (in):	.028
5 - The number of circuits:	3.
6 - The outside to inside area ratio:	3.33
7 - The fouling factor (BTU/ft ² -F-hr):	4000.
8 - Tube chilled water average temperature (F):	50.0
9 - The evaporator temperature (F):	42.0
10 - The mass flow of refrigerant (lb/hr):	4380.0
11 - The thermal conductivity of the tube matrl (BTU/hr-ft-F/in):	2700.0
12 - The chilled water flowrate (lb/min):	7485.0

Heat Transfer Information: ORNL, ABSIM module, *PWA*

=====

Inside resistance	=	1431.2620 BTU/hr-ft ² -F
Outside resistance	=	738.8974 BTU/hr-ft ² -F
Overall heat transfer coefficient (U)	=	732.35 BTU/hr-ft ² -F

=====

Do you want to do another run (Y/N)?

Appendix 5. Sample Input/Output File for Cycle Analysis

LiBr-H2O 3-EFFECT CHILLER, PARALLEL FLOW

INPUT AND OUTPUT IN BRITISH UNITS

BRITISH UNITS ARE :

TEMPERATURES (T) IN DEG F

MASS FLOWRATES (F) IN LBS/MIN

CONCENTRATIONS (C) IN WEIGHT %

ENTHALPIES (H) IN BTU/LB

PRESSURES (P) IN PSIA

HEAT QUANTITIES (Q) IN BTU/MIN

TOLERANCES IN F, X: 1.0D-05 1.0D-08

NO. OF UNITS: 20

NO. OF STATE POINTS: 45

UNIT INPUT

```
1 51 1.1066D+05 0 .0000D+00 .0000D+00 1
26 2 28 29 30 0 0
2 11 1.4093D+04 0 .0000D+00 .0000D+00 0
43 2 3 4 5 6 0
3 21 1.2516D+04 0 .0000D+00 .0000D+00 0
37 9 10 11 12 31 0
4 21 1.2516D+04 0 .0000D+00 .0000D+00 0
40 14 15 16 32 18 0
5 41 1.2845D+04 0 .0000D+00 .0000D+00 0
20 22 23 24 25 0 0
6 61 0.0000D+00 0 .0000D+00 .0000D+00 0
16 21 0 0 0 0 0
7 33 1.1059D+04 0 .0000D+00 .0000D+00 0
44 8 7 1 0 0 0
8 33 1.2140D+04 0 .0000D+00 .0000D+00 0
38 13 17 39 0 0 0
9 72 0.0000D+00 0 .0000D+00 .0000D+00 0
19 9 20 0 0 0 0
10 61 0.0000D+00 0 .0000D+00 .0000D+00 0
11 19 0 0 0 0 0
11 61 0.0000D+00 0 .0000D+00 .0000D+00 0
22 26 0 0 0 0 0
12 33 1.2748D+04 0 .0000D+00 .0000D+00 0
41 33 27 42 0 0 0
13 21 1.2516D+04 0 .0000D+00 .0000D+00 -1
33 15 35 36 27 34 0
14 72 0.0000D+00 0 .0000D+00 .0000D+00 0
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21 14 10 0 0 0 0
15 72 0.0000D+00 0 .0000D+00 .0000D+00 0
12 39 7 0 0 0 0
16 72 0.0000D+00 0 .0000D+00 .0000D+00 0
32 42 17 0 0 0 0
17 82 0.0000D+00 0 .0000D+00 .0000D+00 0
40 41 13 0 0 0 0
18 82 0.0000D+00 0 .0000D+00 .0000D+00 0
37 38 8 0 0 0 0
19 72 0.0000D+00 0 .0000D+00 .0000D+00 0
45 1 43 0 0 0 0
20 82 0.0000D+00 0 .0000D+00 .0000D+00 0
45 44 5 0 0 0 0

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STATE POINT INPUT - STARTING VALUES

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1 1 1 .1260D+03 5 .1349D+04 2 .6131D+02 4 .1311D+00 0 .0000D+00
2 3 2 .4191D+02 2 .7279D+02 0 .0000D+00 4 .1311D+00 0 .1000D+01
3 3 0 .8500D+02 9 .1390D+05 0 .0000D+00 0 .0000D+00 0 .0000D+00
4 3 0 .9200D+02 9 .1390D+05 0 .0000D+00 0 .0000D+00 0 .0000D+00
5 1 10 .1062D+03 0 .1422D+04 4 .5817D+02 4 .1311D+00 0 .0000D+00
6 1 1 .1182D+03 0 .1345D+04 1 .6150D+02 4 .1311D+00 0 .0000D+00
7 1 1 .1634D+03 5 .1349D+04 2 .6131D+02 2 .8665D+00 0 .0000D+00
8 1 8 .1401D+03 0 .1422D+04 4 .5817D+02 2 .8665D+00 0 .0000D+00
9 3 3 .1812D+03 3 .1653D+02 0 .0000D+00 2 .8665D+00 0 .1000D+01
10 3 5 .1862D+03 4 .5627D+02 0 .0000D+00 3 .8597D+01 1 .4697D+00
11 3 5 .1862D+03 4 .5627D+02 0 .0000D+00 3 .8597D+01 0 .0000D+00
12 1 3 .1812D+03 1 .3390D+03 1 .6101D+02 2 .8665D+00 0 .0000D+00
13 1 9 .2327D+03 0 .1067D+04 4 .5817D+02 3 .8597D+01 0 .0000D+00
14 3 4 .2904D+03 1 .2148D+02 0 .0000D+00 3 .8597D+01 0 .1000D+01
15 3 7 .4158D+03 8 .3478D+02 0 .0000D+00 5 .6294D+02 0 .1000D+01
16 3 1 .2959D+03 8 .3478D+02 0 .0000D+00 5 .6294D+02 0 .0000D+00
17 1 1 .2599D+03 6 .1010D+04 3 .6141D+02 3 .8597D+01 0 .0000D+00
18 1 1 .2691D+03 0 .3610D+03 1 .5728D+02 3 .8597D+01 0 .0000D+00
19 3 6 .9702D+02 4 .5627D+02 0 .0000D+00 2 .8665D+00 1 .8599D-01
20 3 6 .9702D+02 2 .7279D+02 0 .0000D+00 2 .8665D+00 3 .3018D+00
21 3 5 .1862D+03 8 .3478D+02 0 .0000D+00 3 .8597D+01 1 .1129D+00
22 3 6 .9702D+02 2 .7279D+02 0 .0000D+00 2 .8665D+00 0 .0000D+00
23 3 0 .8500D+02 11 .3277D+04 0 .0000D+00 0 .0000D+00 0 .0000D+00
24 3 0 .9200D+02 11 .3277D+04 0 .0000D+00 0 .0000D+00 0 .0000D+00
25 3 6 .9702D+02 2 .7279D+02 0 .0000D+00 2 .8665D+00 3 .3018D+00
26 3 2 .4191D+02 2 .7279D+02 0 .0000D+00 4 .1311D+00 2 .5112D-01
27 1 7 .4158D+03 7 .6762D+03 5 .6117D+02 5 .6294D+02 0 .0000D+00
28 3 0 .5500D+02 10 .7467D+04 0 .0000D+00 0 .0000D+00 0 .0000D+00
29 3 0 .4500D+02 10 .7467D+04 0 .0000D+00 0 .0000D+00 0 .0000D+00
30 3 2 .4191D+02 2 .7279D+02 0 .0000D+00 4 .1311D+00 2 .5112D-01
31 1 1 .1674D+03 0 .3595D+03 1 .5753D+02 2 .8665D+00 0 .0000D+00
32 1 4 .2904D+03 1 .3340D+03 1 .6192D+02 3 .8597D+01 0 .0000D+00
33 1 1 .3879D+03 0 .6399D+03 4 .5817D+02 5 .6294D+02 0 .0000D+00

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34	1	1	.3983D+03	0	.7144D+03	1	.5790D+02	5	.6294D+02	0	.0000D+00
35	3	0	.4266D+03	12	.5362D+02	0	.0000D+00	0	.0000D+00	0	.1000D+01
36	3	0	.4266D+03	12	.5362D+02	0	.0000D+00	0	.0000D+00	0	.0000D+00
37	1	8	.1401D+03	0	.3555D+03	4	.5817D+02	2	.8665D+00	0	.0000D+00
38	1	8	.1401D+03	0	.1067D+04	4	.5817D+02	2	.8665D+00	0	.0000D+00
39	1	1	.1573D+03	6	.1010D+04	3	.6141D+02	2	.8665D+00	0	.0000D+00
40	1	9	.2327D+03	0	.4266D+03	4	.5817D+02	3	.8597D+01	0	.0000D+00
41	1	9	.2327D+03	0	.6399D+03	4	.5817D+02	3	.8597D+01	0	.0000D+00
42	1	1	.2450D+03	7	.6762D+03	5	.6117D+02	3	.8597D+01	0	.0000D+00
43	1	1	.1260D+03	1	.1349D+04	1	.6131D+02	4	.1311D+00	0	.0000D+00
44	1	10	.1062D+03	0	.1422D+04	4	.5817D+02	4	.1311D+00	0	.0000D+00
45	1	10	.1062D+03	0	.0000D+00	4	.5817D+02	4	.1311D+00	0	.0000D+00

STARTING TEMPERATURES AFTER APPLYING CONSTRAINTS

1	126.0	1
2	41.9	2
3	85.0	0
4	92.0	0
5	106.2	10
6	118.2	1
7	163.4	1
8	140.1	8
9	181.2	3
10	186.2	5
11	186.2	5
12	181.2	3
13	232.7	9
14	290.4	4
15	415.8	7
16	295.9	1
17	259.9	1
18	269.1	1
19	97.0	6
20	97.0	6
21	186.2	5
22	97.0	6
23	85.0	0
24	92.0	0
25	97.0	6
26	41.9	2
27	415.8	7
28	55.0	0
29	45.0	0
30	41.9	2
31	167.4	1
32	290.4	4
33	387.9	1
34	398.3	1

35 426.6 0
 36 426.6 0
 37 140.1 8
 38 140.1 8
 39 157.3 1
 40 232.7 9
 41 232.7 9
 42 245.0 1
 43 126.0 1
 44 106.2 10
 45 106.2 10

NO. OF VARIABLES 56
 NO. OF TEMPERATURES 21
 NO. OF CONCENTRATIONS 11
 NO. OF FLOWS 15
 NO. OF PRESSURES 4
 NO. OF VAPOR FRACTIONS 5

TOTAL NO. OF EQUATIONS 60
 NO. OF NONLINEAR EQUATIONS 44
 NO. OF LINEAR EQUATIONS 16
 NO. OF REDUNDANT EQUATIONS 4

LAST ITERATION NO. IS 67

IER = 1

EUCLIDEAN NORM OF FUN IS LESS THAN FTOL

NO. VARIABLE RESIDUAL EQUATION

1	2.7783D-01	3.7880D-09	HEAT BALANCE,	UNIT NO. 1
2	6.0014D-03	1.2460D-07	HEAT TRANSFER,	UNIT NO. 1
3	2.1401D-01	-7.2308D-09	T5=T5E(P2,C5)+DEVL	ABSORBER NO. 2
4	2.5304D-01	-7.7519D-09	T6=T6E(P2,C6)	ABSORBER NO. 2
5	3.9847D-01	-2.6826D-08	ADIABATIC EQB.	ABSORBER NO. 2
6	3.2321D-01	1.1642D-08	HEAT BALANCE,	UNIT NO. 2
7	4.5575D-01	1.5711D-08	HEAT TRANSFER,	UNIT NO. 2
8	4.7163D-01	-1.2542D-07	T5=T5E(P2,C5)+DEVL	DESORBER NO. 3
9	6.2411D-01	-1.3861D-07	T6=T6E(P2,C6)	DESORBER NO. 3
10	7.9802D-01	-1.3695D-08	ADIABATIC EQB.	DESORBER NO. 3
11	1.2147D+00	2.6175D-07	T4=T4E(P3,C3)	DESORBER NO. 3
12	8.1951D-01	-3.4636D-07	HEAT BALANCE,	UNIT NO. 3
13	7.1290D-01	-1.7667D-07	HEAT TRANSFER,	UNIT NO. 3
14	7.3996D-01	-6.1433D-06	T5=T5E(P2,C5)+DEVL	DESORBER NO. 4
15	1.8385D-01	8.1194D-08	T6=T6E(P2,C6)	DESORBER NO. 4
16	4.1160D-01	2.1522D-07	ADIABATIC EQB.	DESORBER NO. 4
17	1.1252D+00	8.8735D-07	T4=T4E(P3,C3)	DESORBER NO. 4
18	1.1510D+00	1.3370D-06	HEAT BALANCE,	UNIT NO. 4

19	3.7906D-01	6.2123D-06	HEAT TRANSFER,	UNIT NO. 4
20	6.5544D-01	-1.7655D-08	T2=T2E(P1,C2)+DEVL CONDNSER NO. 5	
21	2.7783D-01	-1.5605D-07	HEAT BALANCE,	UNIT NO. 5
22	6.1391D-01	-8.2959D-08	HEAT TRANSFER,	UNIT NO. 5
23	5.8223D-01	2.6175D-07	T2=T2E(P2,C2)	VALVE NO. 6
24	6.1574D-01	-4.8957D-07	H(IN)=H(OUT)	VALVE NO. 6
25	6.1037D-01	-2.4034D-09	HEAT BALANCE,	UNIT NO. 7
26	6.1510D-01	-1.7258D-09	HEAT TRANSFER,	UNIT NO. 7
27	5.7342D-01	-1.8344D-08	HEAT BALANCE,	UNIT NO. 8
28	6.1661D-01	-2.9783D-09	HEAT TRANSFER,	UNIT NO. 8
29	5.7576D-01	-1.7754D-07	ENERGY BALANCE	MIXER NO. 9
30	6.1289D-01	-1.7655D-08	T2=T2E(P2,C2)	VALVE NO.10
31	5.8012D-01	3.5740D-09	H(IN)=H(OUT)	VALVE NO.10
32	6.1391D-01	4.2950D-08	T2=T2E(P2,C2)	VALVE NO.11
33	9.7135D-02	-4.2368D-09	H(IN)=H(OUT)	VALVE NO.11
34	5.2850D-03	1.9561D-07	HEAT BALANCE,	UNIT NO.12
35	1.0081D+00	-1.3510D-06	HEAT TRANSFER,	UNIT NO.12
36	1.1802D-03	2.1144D-06	T5=T5E(P2,C5)+DEVL DESORBER NO.13	
37	4.1047D-03	6.9196D-07	T6=T6E(P2,C6)	DESORBER NO.13
38	2.4425D-02	9.7969D-08	ADIABATIC EQB.	DESORBER NO.13
39	1.5351D-03	-4.2892D-07	HEAT BALANCE,	UNIT NO.13
40	2.5696D-03	-1.4167D-06	HEAT TRANSFER,	UNIT NO.13
41	7.2710D-02	-2.6297D-07	ENERGY BALANCE	MIXER NO.14
42	2.3520D-01	-7.3208D-09	ENERGY BALANCE	MIXER NO.15
43	4.3519D-02	1.0425D-06	ENERGY BALANCE	MIXER NO.16
44	5.4213D-01	2.6951D-14	ENERGY BALANCE	MIXER NO.19
45	2.9191D-02	4.9222D-14	TOTAL MASS BALANCE,ABSORBER NO. 2	
46	3.8630D-03	1.1215D-08	ABS. MASS BALANCE, ABSORBER NO. 2	
47	9.7135D-02	-9.1479D-15	TOTAL MASS BALANCE,DESORBER NO. 3	
48	4.3606D-03	5.9745D-10	ABS. MASS BALANCE, DESORBER NO. 3	
49	2.8862D-02	-6.9974D-15	TOTAL MASS BALANCE,DESORBER NO. 4	
50	2.8682D-01	-1.0773D-06	ABS. MASS BALANCE, DESORBER NO. 4	
51	2.0411D+00	-3.9099D-16	TOTAL MASS BALANCE, MIXER NO. 9	
52	4.6074D-01	3.9515D-14	TOTAL MASS BALANCE,DESORBER NO.13	
53	8.6055D-02	7.2790D-08	ABS. MASS BALANCE, DESORBER NO.13	
54	2.9832D-01	-4.5949D-14	TOTAL MASS BALANCE, MIXER NO.14	
55	1.1099D-01	2.9396D-15	TOTAL MASS BALANCE, MIXER NO.15	
56	5.1135D-02	4.2172D-10	ABS. MASS BALANCE, MIXER NO.15	
57	-2.8656D+03	-2.9208D-14	TOTAL MASS BALANCE, MIXER NO.16	
58	4.1028D+03	9.9227D-07	ABS. MASS BALANCE, MIXER NO.16	
59	9.9527D+02	1.6377D-17	TOTAL MASS BALANCE, MIXER NO.19	
60	9.2913D+02	-1.3013D-15	ABS. MASS BALANCE, MIXER NO.19	

STATE TEMPER. ENTHALPY FLOW RATE CONCENTR. PRESSURE VAPOR FR.
POINT

1	1.2613D+02	6.3096D+01	1.3486D+03	6.1391D+01	1.3082D-01	0.0000D+00
2	4.1860D+01	1.0794D+03	7.3376D+01	0.0000D+00	1.3082D-01	1.0000D+00
3	8.5000D+01	5.2890D+01	1.3997D+04	0.0000D+00	0.0000D+00	0.0000D+00

4	9.2000D+01	5.9850D+01	1.3997D+04	0.0000D+00	0.0000D+00	0.0000D+00
5	1.0634D+02	4.7029D+01	1.4220D+03	5.8223D+01	1.3082D-01	0.0000D+00
6	1.1844D+02	6.0065D+01	1.3446D+03	6.1574D+01	1.3082D-01	0.0000D+00
7	1.6353D+02	7.9987D+01	1.3486D+03	6.1391D+01	8.6585D-01	0.0000D+00
8	1.4020D+02	6.3048D+01	1.4220D+03	5.8223D+01	8.6585D-01	0.0000D+00
9	1.8128D+02	1.1411D+03	1.6386D+01	0.0000D+00	8.6585D-01	1.0000D+00
10	1.8621D+02	6.0860D+02	5.6990D+01	0.0000D+00	8.6047D+00	4.6074D-01
11	1.8621D+02	1.5417D+02	5.6990D+01	0.0000D+00	8.6047D+00	0.0000D+00
12	1.8128D+02	8.7323D+01	3.3911D+02	6.1037D+01	8.6585D-01	0.0000D+00
13	2.3347D+02	1.0725D+02	1.0665D+03	5.8223D+01	8.6047D+00	0.0000D+00
14	2.8739D+02	1.1860D+03	2.1313D+01	0.0000D+00	8.6047D+00	1.0000D+00
15	4.1656D+02	1.2331D+03	3.5677D+01	0.0000D+00	6.1233D+01	1.0000D+00
16	2.9405D+02	2.6364D+02	3.5677D+01	0.0000D+00	6.1233D+01	0.0000D+00
17	2.6100D+02	1.2422D+02	1.0095D+03	6.1510D+01	8.6047D+00	0.0000D+00
18	2.6939D+02	1.2357D+02	4.3316D+02	5.7342D+01	8.6047D+00	0.0000D+00
19	9.6994D+01	1.5417D+02	5.6990D+01	0.0000D+00	8.6585D-01	8.6055D-02
20	9.6994D+01	3.7457D+02	7.3376D+01	0.0000D+00	8.6585D-01	2.9832D-01
21	1.8621D+02	2.6364D+02	3.5677D+01	0.0000D+00	8.6047D+00	1.1099D-01
22	9.6994D+01	6.4819D+01	7.3376D+01	0.0000D+00	8.6585D-01	0.0000D+00
23	8.5000D+01	5.2890D+01	3.2655D+03	0.0000D+00	0.0000D+00	0.0000D+00
24	9.2000D+01	5.9850D+01	3.2655D+03	0.0000D+00	0.0000D+00	0.0000D+00
25	9.6994D+01	3.7457D+02	7.3376D+01	0.0000D+00	8.6585D-01	2.9832D-01
26	4.1860D+01	6.4819D+01	7.3376D+01	0.0000D+00	1.3082D-01	5.1135D-02
27	4.1656D+02	1.9457D+02	6.0422D+02	6.1661D+01	6.1233D+01	0.0000D+00
28	5.5000D+01	2.3137D+01	7.5270D+03	0.0000D+00	0.0000D+00	0.0000D+00
29	4.5000D+01	1.3247D+01	7.5270D+03	0.0000D+00	0.0000D+00	0.0000D+00
30	4.1860D+01	6.4819D+01	7.3376D+01	0.0000D+00	1.3082D-01	5.1135D-02
31	1.6759D+02	7.5024D+01	3.5949D+02	5.7576D+01	8.6585D-01	0.0000D+00
32	2.8739D+02	1.3586D+02	4.0529D+02	6.1289D+01	8.6047D+00	0.0000D+00
33	3.8881D+02	1.8105D+02	6.3990D+02	5.8223D+01	6.1233D+01	0.0000D+00
34	3.9680D+02	1.8487D+02	6.4223D+02	5.8012D+01	6.1233D+01	0.0000D+00
35	4.2660D+02	1.2525D+03	5.3634D+01	0.0000D+00	0.0000D+00	1.0000D+00
36	4.2660D+02	4.0037D+02	5.3634D+01	0.0000D+00	0.0000D+00	0.0000D+00
37	1.4020D+02	6.3048D+01	3.5550D+02	5.8223D+01	8.6585D-01	0.0000D+00
38	1.4020D+02	6.3048D+01	1.0665D+03	5.8223D+01	8.6585D-01	0.0000D+00
39	1.5751D+02	7.7522D+01	1.0095D+03	6.1510D+01	8.6585D-01	0.0000D+00
40	2.3347D+02	1.0725D+02	4.2660D+02	5.8223D+01	8.6047D+00	0.0000D+00
41	2.3347D+02	1.0725D+02	6.3990D+02	5.8223D+01	8.6047D+00	0.0000D+00
42	2.4319D+02	1.1641D+02	6.0422D+02	6.1661D+01	8.6047D+00	0.0000D+00
43	1.2613D+02	6.3096D+01	1.3486D+03	6.1391D+01	1.3082D-01	0.0000D+00
44	1.0634D+02	4.7029D+01	1.4220D+03	5.8223D+01	1.3082D-01	0.0000D+00
45	1.0634D+02	4.7029D+01	0.0000D+00	5.8223D+01	1.3082D-01	0.0000D+00

NO.	TYPE	UA	EFF	CAT	LMTD
1	EVAPORATOR	1.0657D+04	7.6106D-01	3.1396D+00	6.9855D+00
2	ABSORBER	4.0930D+03	3.6184D-01	2.1342D+01	2.3801D+01
3	DESORBER	2.5161D+03	7.3552D-01	4.9224D+00	1.0293D+01
4	DESORBER	2.5152D+03	7.2988D-01	6.6614D+00	1.3752D+01

5	CONDENSER	2.8450D+03	5.8365D-01	4.9935D+00	7.9888D+00
6	VALVE	0.0000D+00	0.0000D+00	0.0000D+00	0.0000D+00
7	HEAT EXGER	1.0590D+03	6.5399D-01	1.9786D+01	2.1510D+01
8	HEAT EXGER	2.1400D+03	8.5668D-01	1.7314D+01	2.2027D+01
9	MIXER	0.0000D+00	0.0000D+00	0.0000D+00	0.0000D+00
10	VALVE	0.0000D+00	0.0000D+00	0.0000D+00	0.0000D+00
11	VALVE	0.0000D+00	0.0000D+00	0.0000D+00	0.0000D+00
12	HEAT EXGER	2.7484D+03	9.4695D-01	9.7136D+00	1.7183D+01
13	DESORBER	2.5160D+03	6.6306D-01	1.0041D+01	1.8164D+01
14	MIXER	0.0000D+00	0.0000D+00	0.0000D+00	0.0000D+00
15	MIXER	0.0000D+00	0.0000D+00	0.0000D+00	0.0000D+00
16	MIXER	0.0000D+00	0.0000D+00	0.0000D+00	0.0000D+00
17	SPLITTER	0.0000D+00	0.0000D+00	0.0000D+00	0.0000D+00
18	SPLITTER	0.0000D+00	0.0000D+00	0.0000D+00	0.0000D+00
19	MIXER	0.0000D+00	0.0000D+00	0.0000D+00	0.0000D+00
20	SPLITTER	0.0000D+00	0.0000D+00	0.0000D+00	0.0000D+00

NO.	TYPE	HEAT TRANSFER	DEVL	DEVG	IPINCH
1	EVAPORATOR	7.4444D+04	0.000D+00	0.000D+00	-1
2	ABSORBER	9.7418D+04	0.000D+00	0.000D+00	-1
3	DESORBER	2.5897D+04	0.000D+00	0.000D+00	1
4	DESORBER	3.4588D+04	0.000D+00	0.000D+00	1
5	CONDENSER	2.2729D+04	0.000D+00	0.000D+00	1
6	VALVE	0.0000D+00	0.000D+00	0.000D+00	0
7	HEAT EXGER	2.2779D+04	0.000D+00	0.000D+00	-1
8	HEAT EXGER	4.7137D+04	0.000D+00	0.000D+00	-1
9	MIXER	0.0000D+00	0.000D+00	0.000D+00	0
10	VALVE	0.0000D+00	0.000D+00	0.000D+00	0
11	VALVE	0.0000D+00	0.000D+00	0.000D+00	0
12	HEAT EXGER	4.7226D+04	0.000D+00	0.000D+00	-1
13	DESORBER	4.5702D+04	0.000D+00	0.000D+00	1
14	MIXER	0.0000D+00	0.000D+00	0.000D+00	0
15	MIXER	0.0000D+00	0.000D+00	0.000D+00	0
16	MIXER	0.0000D+00	0.000D+00	0.000D+00	0
17	SPLITTER	0.0000D+00	0.000D+00	0.000D+00	0
18	SPLITTER	0.0000D+00	0.000D+00	0.000D+00	0
19	MIXER	0.0000D+00	0.000D+00	0.000D+00	0
20	SPLITTER	0.0000D+00	0.000D+00	0.000D+00	0

COP = 1.6289

Vita

Patti Welesko Adcock grew up in Pittsburgh, Pennsylvania. She attended Saint Robert Bellarmine Catholic Grade School and graduated from East Allegheny High School in 1979. In 1983, Patti graduated from Carnegie-Mellon University in Pittsburgh, Pennsylvania, with a Bachelor of Science Degree in Chemical Engineering.

After graduation, Patti moved to Knoxville, Tennessee, to work for what was then the Union Carbide Corporation, Nuclear Division, and is now Martin Marietta Energy Systems, Incorporated. During her 11 years with Energy Systems, Patti has held several positions, including: process support engineer for the Gaseous Diffusion Plant, lead engineer in the Integrated Process Demonstration Facility in the Fuel Recycle Division, laboratory research and development engineer for the AVLIS Feed Reprocessing Program, and Section Manager and Conduct of Operations Program Manager for the K-25 Site.

Patti now works in the Energy Division at the Oak Ridge National Laboratory, which is managed by Martin Marietta Energy Systems for the United States Department of Energy. Currently, she is a technical monitor on several projects including a large commercial chiller contract to York International and a residential heat pump contract to United Technologies Research Center/Carrier Corporation.

Patti is currently a director of the Knoxville/Oak Ridge Section of the American Institute of Chemical Engineers, a past-president of the Smoky Mountain Section of the Society of Women Engineers, and a member of the American Society of Heating Refrigeration and Air Conditioning Engineers. She is on the Board of Directors of WATTec, Inc. and is currently serving as the organization's Treasurer.