Advanced Exergy Analysis for a Solar Double Stage Absorption Chiller

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Advanced Exergy Analysis for a Solar Double Stage Absorption Chiller

A Thesis Submitted in Partial Fulfillment of the Requirements for the Degree of Doctor of Philosophy in Building Performance and Diagnostics in the School of Architecture Carnegie Mellon University Pittsburgh, PA

By

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Abstract

Solar driven absorption chiller technology as an alternative mechanism for cooling has been the focus of tremendous recent interest due to its potential advantages for energy conservation and the environment benefits. However, the comparatively lower coefficient of performance (COP) of these systems compared to the COP of the electric chillers has prevented more widespread applications.

In this dissertation, a novel comparison between an electric and an absorption chiller will be presented (Chapter 3), including a method for calculating the true solar energy requirement for fossil fuel (coal) formation. The same comparison is then extended to the exergy domain. Compared to an electric chiller, in order to provide the same amount of cooling, a solar driven absorption chiller actually consumes a much smaller amount of both solar energy and exergy.

Beyond demonstrating this lower level of actual energy/exergy consumption, it is still important to increase the efficiency of the solar chiller system. Therefore, a detailed exergy destruction analysis (Chapter 4 and 5) is performed in this dissertation, both for the solar collector, which includes pumping exergy loss, and for a two-stage lithium bromide (LiBr)-water absorption chiller. These analyses allow for a better understanding of the exergy destruction due to a component's own inefficiency and/or due to the remaining components' inefficiencies. Given the limits of current technology, the results show that the COP of a two-stage LiBr-water absorption chiller can be improved from 1.1 to 1.38, a 25% efficiency increase, by recovering the avoidable exergy destruction.

Finally, based on the contributions described above and the conclusions that can be drawn from them, a range of potential future work is presented. Other comparative systems are discussed, and the challenges in properly assessing the performance of those systems are described.

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Nomenclature

Abbreviation

ABS Absorber							
ARI	American Refrigeration Institute						
AV	Avoidable part of exergy destruction						
CON	Condenser						
COP	Coefficient of Performance						
CPC	Compound parabolic Concentrator						
Des Exergy destruction							
DOE Department of Energy							
EES	Engineering equation solver						
EN	Endogenous part of exergy destruction						
EVP	Evaporator						
Ex	Exergy						
EX	Exogenous part of exergy destruction						
HTHX	High temperature heat exchanger						
HTRG High temperature regenerator							
IECC	International Energy Conservation Code						
IW	IW Intelligent Workplace						
LiBr	Lithium Bromide						
LMTD Log mean temperature difference							
LTHX Low temperature heat exchanger							
LTRG Low temperature regenerator							
NPP	NPP Net primary productivity						
PAR	PAR Photo-synthetically active radiation						

PF Preservation factor

- PV Photovoltaic
- UN Unavoidable part of exergy destruction
- VAV Variable air volume

Parameter

- ΔP Pressure drop in solar thermal collector
- ε Heat exchanger effectiveness
- η_0 Solar thermal collector optical efficiency
- $\eta_{abs,ex}$ Absorption chiller exergetic efficiency
- $\eta_{collector}$ Solar thermal collector energy efficiency
- $\eta_{collector,ex}$ Solar thermal collector exergetic efficiency
- $\eta_{elec,ex}~$ Electric chiller exergetic efficiency
- η_{fs} Fuel synthesis energy efficiency
- $\eta_{fs,ex}$ Fuel synthesis exergetic efficiency
- η_{pp} Power plant energy efficiency
- $\eta_{pp,ex}$ Power plant exegetic efficiency
- $\eta_{ps,PAR}$ Photosynthesis energy efficiency based on Photo-synthetically active radiation
- $\eta_{ps,total}$ Photosynthesis energy efficiency based on overall solar radiation on leaf
- $\eta_{ps,total,ex}$ Photosynthesis exergetic efficiency based on overall solar radiation on leaf
- $\eta_{total,abs}$ Life cycle total energy efficiency for converting sunlight to cooling for absorption chiller
- $\eta_{total,abs,ex}$ Life cycle total exergetic efficiency for converting sunlight to cooling for absorption chiller
- $\eta_{total,elec}$ Life cycle total energy efficiency for converting sunlight to cooling for electric driven chiller
- $\eta_{total,elec,ex}$ Life cycle total exergetic efficiency for converting sunlight to cooling for electric driven chiller
- ρ Working fluid density
- σ Entropy production

- ϕ Specific flow exergy
- Cp working fluid specific heat
- D Diameter of the solar thermal collector
- E Internal energy
- f friction coefficient
- g Gravity constant
- h Specific enthalpy
- H_{DNI} Direct normal solar radiation
- k_{loss} $\,$ Thermal loss coefficient for solar thermal collector $\,$
- L Length of the solar thermal collector
- *m* Mass flow rate
- **Q** Heat transfer rate
- Re Reynold number
- s Specific entropy
- t Time
- T Temperature
- V Velocity
- W Net rate of work
- X Mass fraction
- z Height

Subscript

- a Absorption chiller
- ambient Ambient condition
- cv Control volume
- double Double stage absorption chiller
- e Electricity chiller
- i Inlet
- in Input

- $j = j^{th}$ control volume boundary
- K kth component in an energy system
- o Outlet
- r Receiver
- rev Reversible process
- single Single stage absorption chiller

sun Sun

Chapter 1: Introduction

1.1 Background

Economic development and occupancy requests have been pushing building thermal comfort to higher levels during the last 20 years. The Building Energy Data Book (DOE, 2009) shows that 39% of the United States' primary energy is used in buildings, and more than 60% of that energy is provided for building cooling and heating. Traditional energy sources for building cooling systems are electricity and natural gas. For example, in a Variable Air Volume (VAV) system, electricity provides the power to drive the fans, pumps, valves, outdoor air economizer, etc., while natural gas provides the thermal energy to the heating coils and the necessary terminal reheat. In an electric driven chiller, electricity runs the compressor, fans, pumps and cooling towers.

From the same report (DOE 2009), it is found that buildings utilize 73% of the overall electricity production in the United States, and that building cooling systems are the major electricity consumers. The negative energy, environmental and economical impacts by using traditional electrical cooling systems include:

• Peak demand increase and electricity blackout: Peak demand refers to a period in which electricity is expected to be provided at a significantly higher than average supply level. Peak demand often occurs in the hot summer, when the electricity based cooling systems, such as compressor and fans, raise the rate of energy consumption significantly. If the power stations cannot meet the increasing peak demand, an electricity shortage or even a blackout could occur. A blackout is a total loss of power to an area, which is the most severe power outage. The current approach to meet this peak demand increase is to build more peaking power stations. However, since those stations mainly generate electricity during the summer seasons, the economic returns and long term operating efficiencies are limited.

- Carbon dioxide emissions: Electrical cooling systems are the major sources for carbon dioxide (CO₂) emissions from the building sector. As discussed, to condition a building in the summer time, the current cooling systems contribute to high level of electricity consumption, and frequently result in an increase for peak demand. CO₂ emissions are strongly associated with the power production for building cooling, and lead to global warming and climate change.
- Air pollution and human health damage: Dr. Jared L. Cohon, the president of Carnegie Mellon University, has chaired and co-authored a newly published report, "Hidden Costs of Energy: Unpriced Consequences of Energy Production and Use" (National Research Council, 2010). In the report, the "external effects" for energy production, distribution and utilization are fully discussed. The "external effects" refer to the beneficial or negative effects that are not reflected in energy market prices. To provide information in assisting energy related regulations and incentives, the authors monetized the environmental and human health damage impacts. It was found that for electricity generation in the U.S., the aggregate damages associated with emissions of SO₂, NO_x, and PM from coal-fired facilities in 2005 were approximately \$62 billion, or \$156 million on average per plant. The current electrical cooling systems significantly contribute to this air pollution and human health damage.

Given the effect that current building cooling systems have with respect to environmental and economical impacts in terms of peak power demand increases, power blackouts, CO_2 emissions, and air pollution, this dissertation will instead focus on a novel building cooling technology, a solar driven absorption chiller, which can greatly reduce operational electricity consumption and these negative "external effects."

1.2 Solar Driven Absorption Chiller

Solar driven absorption chiller technology as an alternative cooling technology has been the focus of tremendous recent interest due to the potential energy and environmental advantages, as shown in Figure 1.1.



Figure 1.1 Schematic diagram for a solar driven absorption chiller

The main driving force in an electric driven chiller is the electricity used to power the compressor. In a solar driven absorption chiller, the bulk of this electricity is replaced by thermal energy, provided by the sun through solar collectors. To replace the electric energy with heat energy, in the absorption chiller, the compressor is replaced by an absorber and a regenerator, and the traditional refrigerant is replaced by a refrigerant-absorbent fluid working pair. The most attractive feature of an absorption chiller system is that, by using heat, with different volatile temperatures between the refrigerant and absorbent, the refrigerant will evaporate from this fluid working pair in the regenerator and be reabsorbed into the fluid working pair in the absorber. Instead of using a compressor with a high energy consumption, an absorption chiller only uses pumps to transport the solution in the cycle, thus greatly reducing the chiller electricity consumption. Herold et al. (1996) showed that the main disadvantage for absorption chiller technology is its low coefficient of performance (COP) compared to a traditional electric driven chiller.

Following the American Refrigeration Institute (ARI) 550/590 testing procedure, a series of water chiller packages with minimum efficiency requirements are specified in the International Energy Conservation Code (IECC 2006). The code shows a traditional

comparison between an absorption chiller and electric chiller, in terms of the COP, which equals the energy output over the energy input. An average COP = 5.0 is listed for an electric chiller, while an average COP = 1.0 is given for an absorption chiller.

The code and traditional comparison method ignore an important issue, though: the denominators are different. Electricity is a high quality form of energy, while the thermal energy's quality is associated with its absolute temperature. In order to make a meaningful comparison, the high quality electricity has to be extended to the primary energy. Even if the primary energy is included, the comparison is still biased due to the different types of driving energy; i.e., solar thermal vs. fossil fuel. The fossil fuel was originally generated by solar energy, accumulated and conserved within a physical format, as a form of "ancient solar energy" (Dukes, 2003). The traditional COP defined by thermodynamics textbooks and codes neither reflects the primary energy used in the power plant, nor the actual ancient solar energy requirement for fossil fuel formation. The traditional COP comparison creates confusion and ignores the overall energy inefficiency in the electric cooling process. Some citations in the literature have tried to calculate the photosynthesis efficiency and fuel synthesis efficiency based on this original conversion efficiency. However, no research so far has been performed that combines this type of analysis with an overall solar, coal, and electricity cooling process investigation.

The comparison described above is in the energy domain. An energy analysis is the traditional method of assessing the way energy is used in an operation. Based on the first law of thermodynamics, the energy analysis and energy efficiency calculations are performed via an energy balance. However, an energy balance provides no information on the degradation of energy or resources during a process and does not quantify the usefulness or quality of the various energy and material streams flowing through a system and exiting as products and wastes (Diner and Rosen, 2007). The concept of exergy combines the first law and second law of thermodynamics. The exergy analysis clearly indicates the locations of energy degradation in a process and can therefore lead to improved system operation or component technology (Diner and Rosen, 2007). A main aim of exergy analysis is to identify meaningful (exergy) efficiencies and the causes and

true magnitudes of exergy losses (Diner and Rosen, 2007). Therefore, it is important also to introduce an exergy analysis in this thesis.

There are several papers focused on the energy performance of a parabolic trough solar collector (which, due to its performance, is the chosen solar collector for this research), while there is a lack of knowledge on the exergy analysis. Traditional energy fraction analyses cannot reveal the internal and external losses in the solar collector. Some papers focus on the exergy study of solar collectors, but all of them are applied to flat plate solar collectors, where the heat transfer coefficients for the heat loss are constant values.

In order to enhance the absorption chiller system performance, a series of conventional exergy analyses used in identifying the largest exergy destruction component can be found in the literature, but few of these papers have ever touched the component level exergy destruction together with the exergy destruction caused by the remaining components. The advanced exergy method, which was recently developed in Europe, further divides the exergy destruction in each component in terms of endogenous and exogenous parts, and also unavoidable and avoidable parts. The endogenous part of the exergy destruction is associated only with the irreversibility occurring when all of the other components operate in an ideal way and the component being considered operates with its current efficiency, while the exogenous part of exergy destruction is caused by the irreversibilities that occur in the remaining components. For an effective improvement in the overall system efficiency, the exergy subdivided into endogenous and exogenous destruction contributions is helpful for engineers to make decisions on whether they should focus on the component being considered or on the remaining system components. At the same time, only part of the exergy destruction can be avoided to improve system efficiency, so the subdivided unavoidable and avoidable exergy destruction provides engineers with information on how much avoidable exergy destruction potential is in the component and how much unavoidable exergy destruction is due to technological limitations, such as availability and cost of materials (Morosuk and Tsatsaronis, 2008).

1.3 Literature Review

The literature review is organized into four parts, to provide an overview of the scope of the existing work in the field, and to show the areas where more research is needed:

- 1. Solar energy and exergy requirements for fossil fuel formation;
- 2. Energy and exergy analysis for solar thermal collectors;
- 3. Energy and exergy analysis for absorption chiller systems; and
- 4. Advanced exergy analysis for energy systems.

1.3.1 Solar Energy and Exergy Requirements for Fossil Fuel Formation

In order to calculate the true solar energy and exergy requirements for fossil fuel formation, it is necessary to show the links and efficiencies for each stage of the process that transforms sunlight into fossil fuels. The photosynthesis efficiency and fuel synthesis efficiency are critical values in accurately delineating that calculation.

Photosynthesis efficiency

Depending on whether the solar radiation is considered as the net utilized solar radiation or as the total radiation arriving on a leaf surface, there exist two efficiencies. If the chemical energy of sugar generated in the plant is only related to the net utilized solar radiation, the relevant parameter is the energy of photo-synthetically active radiation (PAR). Potter et al. (1993) suggested a mean global efficiency of 0.39 g C MJ-1 PAR, which is an approximately 1.7% photosynthesis efficiency. A detailed discussion on the basis of this value is shown in Chapter 3 (Subsection 3.2.1).

Fuel synthesis efficiency

Dukes (2003) defined a preservation factor (PF) as the fraction of carbon that remains at the end of a transition from one fossil fuel precursor to the next, "such as from plant

matter to peat, on the path of coal formation." Dukes showed that the PF for the organic carbon in peat is 15.6%, and for the brown coal and hard coal is approximately 95.5% and 69%, respectively. The fuel synthesis efficiency is based on the multiplication of the two PFs and mining efficiencies. A detailed discussion of this calculation is shown in Chapter 3 (Subsection 3.2.2).

1.3.2 Energy and Exergy Analysis for Solar Thermal Collectors

In recent years, there has been increased interest in better designs for solar thermal collectors for medium/high temperature applications, together with double stage absorption chiller research and development. The solar thermal double stage absorption technology can provide cooling to the building based on process heat at a medium temperature $(150 \, \mathbb{C}\mbox{-}180 \, \mathbb{C})$ with a coefficient of performance (COP) of 1.2 and minimum electricity consumption. Solar thermal energy with a medium/high temperature heat source can also be efficiently designed to provide space cooling, space heating, and domestic hot water heating to a building through an energy cascading design, which is shown in Figure 1.2 (Hartkopf et al. 2004). These solar thermal technologies can greatly reduce the electricity and natural gas consumption in building operation, and can even potentially be coupled with other renewable energy applications to lead to net zero energy building system design.

A traditional flat plate solar collector can only provide heat at a temperature range from $60 \ C$ to $100 \ C$, which makes it suitable for space heating, as the input for a one stage absorption chiller for cooling, and for domestic hot water. In order to achieve solar thermal double stage absorption technology and other energy cascading designs, a medium temperature (150 \Capcalcolor) output is needed. A parabolic trough solar collector suitable for this level of output was designed, installed and studied by researchers in Center for Building Performance and Diagnostics with the results published in Qu (2008).

The traditional purpose for solar thermal collector optimization is to maximize the collector efficiency, defined in Duffle and Beckman (1974) as the ratio of the useful

energy gain to the incident solar energy during the same time period. However, this method does not consider the pumping power required to push the fluid through the collector. In 1981, Winn et al. proposed a criterion used for optimization to maximize the difference between the collected thermal energy and the required pumping power, but this criterion assumes the equality of mechanical energy and heat. Bejan (1982) presented the concept of the maximization of exergy delivery as the fundamental problem in solar collector thermal design.



Figure 1.2 Conceptual scheme for building-integrated energy cascading mechanisms (Hartkopf et al., 2004)

1.3.3 Energy and Exergy Analysis for Absorption Chiller Systems

There are a large number of publications on absorption cycles, but only the research most germane to this thesis work is cited below. Herold et al. (1996) summarized the technology of absorption chillers from fundamental knowledge, to field applications, to energy system modeling. Arun (2001) analyzed a double stage parallel flow absorption refrigeration cycle with water–lithium bromide as the working fluid based on the concept of an equilibrium temperature at the low pressure generator. Yin (2006) carried out an

energy study for a double stage steam driven absorption cycle at Carnegie Mellon University. Arora and Aushik (2009) developed a computational model for the parametric investigation of single-effect and series flow double stage LiBr/H₂O absorption refrigeration systems. The effects of the generator, absorber, condenser, evaporator and dead state temperatures are examined on the performance of these systems. The Engineering Equation Solver (EES) software program is frequently used for absorption chiller modeling due to comprehensive internal/external libraries for LiBr/H₂O and Ammonia/Water properties.

1.3.4 Advanced Exergy Analysis for Energy Systems

Advanced exergy analysis is relatively new to energy system engineering. The exergy destruction can be divided into endogenous/exogenous and unavoidable/avoidable parts. This will give engineers a better understanding about the exergy destruction from a single component caused by its own inefficiency, or by the remaining inefficient components in the system and the interactions among system components. Kelly et al. (2009) discussed four different approaches for calculating the endogenous part of the exergy destruction. In the paper, the authors presented the advantages, disadvantages and restrictions for applications associated with each approach and concluded that only structural theory did not give reasonable results. Morosuk and Tsatsaronis (2009) applied this concept to vapor-compression refrigeration machines using different refrigeration fluids. The paper aims to demonstrate the effect of different refrigerant properties on the results of advanced exergy analysis. Petrakopoulou et al. (2010) presented the application of an advanced exergoeconomic analysis to a combined cycle power plant. The advanced exergy analysis in this case further split the investment cost for each component in the combined cycle power plant into avoidable/unavoidable and endogenous/exogenous costs. By focusing on the endogenous/exogenous avoidable exergy destruction and avoidable cost, the engineers can perform an exergoeconomic analysis based on the advanced exergy analysis to find the investment optimization point. Only one paper in the exergy literature has been found that relates to an absorption chiller analysis, when Morosuk and Tsatsaronis (2008) applied this method to a theoretical single stage Ammonia/Water

absorption refrigeration machine.

1.3.5 Summary of the Literature Review

The summarized previous research work and new thesis work on an advanced exergy analysis for solar absorption chiller system are shown in Table 1.1. Based on the literature review, the current issues within the scope of work are:

- The traditional COP comparison between solar absorption cooling and electric cooling creates confusion and bias. There is a need to compare the two systems in terms of the total solar energy requirement.
- Extensive thermal collector energy performance studies have been done, while the few exergy analyses for the thermal collector mainly focus on flat plat collectors, not concentrating collectors.
- No research has been done for the detailed exergy analysis of a parabolic trough solar collector in terms of solar exergy input, useful exergy output, optical exergy loss, absorption exergy loss, thermal exergy loss, heat conduction exergy loss, and friction exergy loss.
- Extensive absorption chiller energy performance studies have been done, but only a few theoretical traditional exergy studies have been completed for the absorption chiller.
- Advanced exergy analysis is relative new. Most of the current papers only demonstrate the energy system calculation method. There is a need to apply this method to the analysis of a double stage absorption chiller, which has a more complex system configuration compared to a single stage chiller.

Table 1.1 Summary of related research work

Author	Fossil Fuel Formation with Ancient Solar Energy		Energy and Exergy analysis for solar thermal collectors		Energy and Exergy analysis for absorption chillers		Advanced Exergy analysis	
	Photosynthesis	Fuelsynthesis	Energy	Exergy	Energy	Exergy	Single Stage	Double Stage
			Performance	Performance	Performance	Performance	Ab Chiller	Ab Chiller
Potter (1993)	\checkmark							
Dukes (2003)								
Qu			\checkmark					
(CMU, 2008)								
Suzuki (1988)				\checkmark				
Yin								
(CMU, 2006)								
Arora (2009)					\checkmark			
Morosuk					\checkmark			
(2008)								
Hu	\checkmark							
(CMU, 2012)								

1.4 Hypothesis and Methodology

Based on the above discussion, the hypotheses of this dissertation are:

- A solar double stage absorption chiller (with an absorption chiller COP=1.2) is actually more efficient than an electric chiller (with a COP=5), in terms of the overall solar energy requirement in fossil fuel formation, energy efficiency and exergetic efficiency;
- The advanced exergy analysis method of dividing the exergy destruction into endogenous/exogenous parts and unavoidable/avoidable parts can help to better identify the order in which engineers should focus on the components, in order to thus boost the overall efficiency of the system.

This dissertation extends the traditional COP comparison between a solar absorption chiller and an electric chiller, and it also extends the comparison from the energy domain to the exergy domain. At the same time, in order to increase the system efficiency, a detailed exergy destruction analysis is performed for the solar collector, which includes pumping exergy loss. An advanced exergy destruction analysis is also conducted for an absorption chiller for better understanding the exergy destruction due to a component's own inefficiency and/or due to the remaining components' inefficiencies. A conceptual diagram of the dissertation is presented in Figure 1.3.



Figure 1.3 A conceptual diagram of the dissertation research

1.5 Thesis Chapter Overview

This thesis is comprised of six chapters:

This chapter, Chapter 1, **Introduction**, provides the background information and motivation for this dissertation. It also discusses the research gaps based on literature reviews, together with the hypotheses and detailed methodology.

Chapter 2, **Energy and Exergy Analysis**, provides the fundamental methodology for the energy analysis, exergy analysis, and advanced exergy analysis, which will then be applied in the subsequent chapters. By comparing the differences, the advantages of using the exergy analysis are discussed, together with the significance of using the advanced exergy analysis for multi-component energy systems.

Chapter 3, Efficiency Comparison for Electric and Solar Thermal Driven Chillers, provides a context for the scope of work by presenting a novel comparison between an electric and an absorption chiller. The new calculation includes a method for calculating the true solar energy requirement for fossil fuel (coal) formation. The same comparison is then extended to the exergy domain.

Chapter 4, **Conventional Exergy Analysis for the Parabolic Trough Solar Collector and the Double Stage Absorption Chiller**, applies the exergy analysis in evaluating the performance and exergy losses for both the solar collector and the double stage absorption chiller. In order to provide a framework for the exergy analysis, the energy analysis for the absorption chiller is also discussed.

Chapter 5, Advanced Exergy Analysis for a Double Stage Absorption Chiller, further splits the exergy destruction within a component into unavoidable/avoidable parts and endogenous/exogenous parts. The two splits are combined and the exergy destruction is finally divided into unavoidable endogenous, unavoidable exogenous, avoidable endogenous and avoidable exogenous parts.

Finally, Chapter 6, **Conclusions**, summaries the contributions and accomplishments achieved from this dissertation. It also points out the future research direction for exergy analysis and optimization in chiller systems.

Chapter 2: Energy and Exergy Analysis

This chapter provides a fundamental background for the development of energy, exergy, and advanced exergy analyses. By comparing the differences, the advantages of using the exergy analysis are discussed, together with the significance of using an advanced exergy analysis for multi-component energy systems.

2.1 Energy Analysis and the First Law of Thermodynamics

An energy analysis is a conventional method for assessing the way energy is used in any operation involving the physical or chemical processing of materials, and the transfer and/or conversion of energy (Dincer and Rosen, 2007). An energy analysis is based on the first law of thermodynamics.

The first law of thermodynamics is more commonly known as the law of energy conservation. It was first explicitly stated by Rudolf Clausiusin 1850:

"In all cases in which work is produced by the agency of heat, a quantity of heat is consumed which is proportional to the work done; and conversely, by the expenditure of an equal quantity of work an equal quantity of heat is produced."

The first law of thermodynamics indicates that energy can be neither created nor destroyed, and it can only change from one form to another form. This law defines internal energy as a state function, and provides a formal statement of the conservation of energy.

The most common energy systems, such as power generation and refrigeration systems, are open systems (systems in which mass flows through the various components). The typical components of power and refrigeration systems are boilers, turbines, evaporators,

condensers, etc., all of which have inlets and outlets. The expression of the first law of thermodynamics for open systems is:

$$\begin{bmatrix} \text{Rate of Internal} \\ \text{Energy Change} \\ \text{within Control Volume} \end{bmatrix} = \begin{bmatrix} \text{Net Rate of} \\ \text{Heat Addition} \end{bmatrix} - \begin{bmatrix} \text{Net Rate of} \\ \text{Work Out} \end{bmatrix} \\ + \begin{bmatrix} \text{Rate of Energy} \\ \text{Addition with Mass} \end{bmatrix} - \begin{bmatrix} \text{Rate of Energy} \\ \text{Removal with Mass} \end{bmatrix}$$
(2.1)

The mathematical equation for the first law of thermodynamics for an open system, or any component in an open system, is:

$$\frac{dE}{dt}|_{cv} = \dot{Q} - \dot{W}_{cv} + \sum \dot{m}_i \left(h_i + \frac{1}{2}V_i^2 + gz_i\right) - \sum \dot{m}_o \left(h_o + \frac{1}{2}V_o^2 + gz_o\right)$$
(2.2)

where, $\frac{dE}{dt}|_{cv}$ is the time rate of change the total energy stored within the given control volume; \dot{Q} is the net rate of heat addition; \dot{W}_{cv} is the net rate of the non-flow work out; and $\sum \dot{m} (h + \frac{1}{2}V^2 + gz)$ is the rate of the energy (h is the specific enthalpy, $\frac{1}{2}V^2$ is the specific kinetic energy, and gz is the specific potential energy) addition or removal due to mass flowing (\dot{m}) into (i) or out of (o) the given control volume.

For almost every typical component in an energy system, the velocity and height differences for the working fluid flows can be ignored, so that the rates of energy addition and removal are only associated with the enthalpy, such that:

 $\begin{bmatrix} Rate \ of \ Energy \\ Addition \ with \ Mass \end{bmatrix} = \sum \dot{m}_i \ h_i \qquad \text{and} \qquad$

$$\begin{bmatrix} \text{Rate of Energy} \\ \text{Removal with Mass} \end{bmatrix} = \sum \dot{m}_o h_o$$
(2.3)

An energy analysis is commonly used in evaluating the performance of a component or a system, and can be used to determine the first law efficiency (η_{th} , for a power production cycle) or the coefficient of performance (COP, for a heating or cooling cycle). The COP and η_{th} are both defined as the useful energy out, divided by the useful energy into the system.

However, an energy balance provides no information about the direction in which processes can spontaneously occur and/or the reversibility of the thermodynamic processes. The first law cannot provide information about the inability of any thermodynamic process to convert heat fully into mechanical work, or any insight into why mixtures cannot spontaneously separate themselves (Dincer and Rosen, 2007).

In the following section, the second law of thermodynamics will be introduced, and the means for determining energy quality degradation and irreversibility will be discussed.

2.2 The Second Law of Thermodynamics

The second law of thermodynamics emphasizes on the quality, rather than just the quantity, of different forms of energy, and explains the reasons that some spontaneous processes can only occur in one direction. Two well-known (and, ultimately, equivalent) statements of the second law of thermodynamics are cited below (Dincer and Rosen, 2007):

• Clausius statement:

"It is impossible for heat to move of itself from a lower-temperature reservoir to a higher temperature reservoir. That is, heat transfer can only occur spontaneously in the direction of temperature decrease."

• Kelvin–Planck statement:
"It is impossible for a system to receive a given amount of heat from a hightemperature reservoir and to provide an equal amount of work output. While a system converting work to an equivalent energy transfer as heat is possible, a device converting heat to an equivalent energy transfer as work is impossible."

The second law of thermodynamics defines entropy as the measurement of the randomness within a system. Because entropy is a state property, an expression of the second law of thermodynamics for an open system can be developed in a similar manner to that of the first law, as below:

$$\begin{bmatrix} \text{Rate of Entropy} \\ \text{Increase in} \\ \text{Control Volume} \end{bmatrix} = \begin{bmatrix} \text{Net Rate of} \\ \text{Entropy} \\ \text{Addition} \\ \text{by Heat} \end{bmatrix} + \begin{bmatrix} \text{Rate of} \\ \text{Entropy} \\ \text{Mass Flow In} \end{bmatrix} \\ - \begin{bmatrix} \text{Rate of} \\ \text{Entropy} \\ \text{Removal with} \\ \text{Mass Flow Out} \end{bmatrix} + \begin{bmatrix} \text{Rate of} \\ \text{Entropy} \\ \text{Production in} \\ \text{Control Volume} \end{bmatrix}$$
(2.4)

The mathematical equation for the second law of thermodynamics for an open system, or any component in an open system, is:

$$\frac{\mathrm{d}S}{\mathrm{d}t}|_{\mathrm{cv}} = \sum_{j} \frac{\dot{Q}}{\mathrm{T}_{j}} + \sum \dot{m}_{i} \, s_{i} - \sum \dot{m}_{o} \, s_{o} + \sigma_{\mathrm{cv}} \tag{2.5}$$

where, $\frac{dS}{dt}|_{cv}$ is the time rate of change of the entropy stored in the control volume; $\sum_{j} \frac{\dot{Q}}{T_{j}}$ is the net rate of the entropy addition due to heat transfer (T_j is the temperature at the jth control volume boundary, where the heat is crossing into the control volume that boundary); $\sum \dot{m}$ s is the rate of the entropy (s) addition (i) or removal (o) from the system due to mass flow (\dot{m}); and σ_{cv} is the entropy production in the control volume.

An ideal process is a process with no entropy generation/production, so that the system would operate at its best possible performance level. Therefore, compared with the actual

system performance, the second law efficiency (isentropic efficiency, η_t , for a power production process such as a turbine, and η_p , for a power consumption process such as a pump) can be determined. The η_t is defined as the actual turbine work out, divided by the ideal turbine work out, while the η_p is defined as the ideal pump work in, divided by the actual pump work in.

To better understand and quantify these restrictions on the flow and conversion of energy, it is necessary to combine the first law with the second law of thermodynamics. The exergy analysis is based on this combination and will be discussed in the next section.

2.3 Exergy Analysis

Exergy is a measure of a system's ability to do useful work as it is brought into thermalmechanical and chemical equilibria with the environment. It is derived from the combination of the first and second law of thermodynamics.

By definition, exergy is not simply a thermodynamic property, but is a property of both a system and the reference environment. For an example, a system in equilibrium with its environment has no exergy, and exergy is greater with a greater deviation from the environment. The environment is defined as the surroundings of the system, and the environment is large enough that the interactions with the system do not change its state. Therefore, the environment is usually defined at a fixed pressure, $P_{ambient}$, temperature, $T_{ambient}$, and chemical composition. Table 2.1, below, summarizes the differences between an energy and an exergy analysis.

Table 2.1 Comparison between energy and exergy analysis

Energy Analysis	Exergy Analysis		
Conserved in all Processes	Only Conserved in Reversible Processes		
Quantity of Energy	Quantity and Quality of Energy		
Independent of Environment	Depends on the Environment		

Units J,W	Units J,W		

An expression of an exergy analysis for an open system is:

$$\begin{bmatrix} \text{Rate of Exergy Change} \\ \text{Within Control Volume} \end{bmatrix} = \begin{bmatrix} \text{Net Exergy} \\ \text{Transfer by Heat} \end{bmatrix} - \begin{bmatrix} \text{Net Exergy} \\ \text{Transfer by Work} \end{bmatrix} \\ + \begin{bmatrix} \text{Rate of Exergy} \\ \text{Addition with Mass} \end{bmatrix} - \begin{bmatrix} \text{Rate of Exergy} \\ \text{Removal with Mass} \end{bmatrix} - \begin{bmatrix} \text{Exergy Destruction} \\ \text{Within Control Volume} \end{bmatrix} (2.6)$$

By combining the first and second law of thermodynamics, the mathematical equation of an exergy analysis for an open system, or any component in an open system, is:

$$\frac{dEx}{dt}|_{cv} = \sum_{j} (1 - \frac{Tambient}{T_{j}})\dot{Q} - \dot{W}_{cv} + \sum_{i} \dot{m}_{i} \phi_{i} - \sum_{i} \dot{m}_{o} \phi_{o} - T_{ambient} \sigma_{cv}$$
(2.7)

where $\frac{dEx}{dt}|_{cv}$ is the time rate of change of the exergy stored within the control volume; $\sum_{j}(1 - \frac{Tambient}{T_{j}})\dot{Q}$ is the net exergy change due to heat transfer(T_{j} is the temperature at the jth control volume boundary, where the heat is crossing into the control volume at that boundary); \dot{W}_{cv} is the net exergy transfer due to non-flow work out of the control volume; $\sum \dot{m} \dot{\phi}$ is the exergy addition (i) or removal (o)due to mass flow; and $T_{ambient}\sigma_{cv}$ is the exergy destruction within the control volume. The specific flow exergy, $\dot{\phi}_{i}$ (or ϕ_{o}), is a function of the ambient temperature, enthalpy, and entropy, and the inlet or outlet enthalpy and entropy:

$$\phi_i = (h_i - h_{ambient}) - T_{ambient}(s_i - s_{ambient})$$
, and

$$\phi_{o} = (h_{o} - h_{ambient}) - T_{ambient}(s_{o} - s_{ambient})$$
(2.8)

Based on the definition and equations, energy is never destroyed during a process; it only changes from one form to another. In contrast, exergy accounts for the irreversibility of a process due to the increase in entropy. Exergy is always destroyed when a process involves a temperature change. The system exergetic efficiency is defined as the useful exergy out from the system, divided by the overall exergy supplied to the system.

In order to understand the overall performance of a system, it is important to apply not only an energy analysis, but also an exergy analysis, since an exergy analysis can quantify the energy quality match between the supply and demand through the exergetic efficiency. In order to further elucidate this concept, Figure 2.1 is cited from Wall (1977). It shows an illustration for the energy flows, exergy flows, and the efficiencies for an oil furnace, an electric heater, an electric heat pump, and a combined power and heat plant.

The electric heater is a useful case for comparing the results of an energy analysis with the results of an exergy analysis. The figure shows that, in terms of energy efficiency, an electric heater is an efficient device, because 100% of the electricity is converted into thermal heat. However, from an energy quality or exergy point of view, the electric heater is far from ideal, since the exergetic efficiency is only 5%. The reason for that is because the electricity is a high quality form of energy, while space heating, at just above room temperature, is a low energy quality application. There is a mismatch between the supply and demand. It is important to apply an exergy analysis to address this match.

At the same time, since an exergy analysis can calculate the exergy destruction/entropy generation in each component within the system, it can assist in fundamentally understanding all of the possible losses in the system due to entropy generation. Therefore, in order to increase the system efficiency, engineering effort can focus on the components with the highest exergy destruction. However, energy systems commonly consist of multiple components, such as compressors, turbines, condensers, etc. In order to further analyze the exergy destruction and the interactions among the components, an advanced exergy analysis is needed. In the following section, an advanced exergy analysis is discussed.



Figure 2.1 Energy/Exergy analysis and efficiencies for an oil furnace, an electric heater, an electric heat pump, and a combined power and heat plant

(Wall 1977)

2.4 Advanced Exergy Analysis

In previous section, it was shown how a conventional exergy analysis can identify the components in a system with the highest levels of exergy destruction. The system efficiency can be improved by reducing the exergy destruction within those components. However, given the present technical limitations, such as the temperature differences in heat exchangers, material costs, etc., it is not always possible to recover all of the exergy destruction within a component. At the same time, for multi-component energy systems, part of the exergy destruction within the component is due to the remaining components' inefficiencies. Therefore, it may be better to improve the remaining components, and not just the component with the highest exergy destruction.

It is important to understand the subdivision of contributions to the exergy destruction within each component. Since a conventional exergy analysis cannot provide the required information, a better approach concerning this detailed exergy analysis is needed. One approach to accomplish this is known as an advanced exergy analysis (Tsatsaronis and Park, 2002). In an advanced exergy analysis, the exergy destruction within a component can be split into unavoidable and avoidable parts. The unavoidable exergy destruction is the exergy destruction that is not recoverable due to technological limitations, and the remaining part is the avoidable exergy destruction, which is recoverable and should be focused on by engineers to improve system efficiency. Additionally, the exergy destruction within a component can also be split into endogenous and exogenous parts. The endogenous exergy destruction is the exergy destruction that is calculated by only considering the inefficiency in the component of interest, while all the remaining components work ideally. The remaining part is exogenous exergy destruction, which is the exergy destruction, which is the exergy destruction caused by the irreversibility in the remaining components. This split can be expressed in the following equations.

$$ExDes_{k} = ExDes_{k}^{UN} + ExDes_{k}^{AV}$$
(2.9)

$$ExDes_{k} = ExDes_{k}^{EN} + ExDes_{k}^{EX}$$
(2.10)

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where $ExDes_k$ is the exergy destruction within k^{th} component; $ExDes_k^{UN}$ is the unavoidable exergy destruction within k^{th} component; $ExDes_k^{AV}$ is the avoidable exergy destruction within k^{th} component; $ExDes_k^{EN}$ is the endogenous exergy destruction within k^{th} component; and $ExDes_k^{EX}$ is the exogenous exergy destruction within k^{th} component.

Once the individual calculations are performed, a combined analysis is required to synthesize the results from the above two equations. This combined analysis produces a new term of interest, the unavoidable endogenous exergy destruction. This is the exergy destruction within a component which is not recoverable, and which is fully due to the component inefficiency while the remaining components work ideally. Based on the unavoidable endogenous exergy destruction, the overall exergy destruction can then be split into unavoidable endogenous exergy destruction, unavoidable exogenous exergy destruction, and avoidable exogenous exergy destruction. The mathematical expressions are shown below:

$$ExDes_{k} = ExDes_{k}^{UN, EN} + ExDes_{k}^{UN, EX} + ExDes_{k}^{AV, EN} + ExDes_{k}^{AV, EX}$$
(2.11)

$$ExDes_{k}^{UN,EX} = ExDes_{k}^{UN} - ExDes_{k}^{UN,EN}$$
(2.12)

$$ExDes_{k}^{AV,EN} = ExDes_{k}^{EN} - ExDes_{k}^{UN,EN}$$
(2.13)

$$ExDes_{k}^{AV,EX} = ExDes_{k}^{AV} - ExDes_{k}^{AV,EN}$$
(2.14)

where $ExDes_k$ is the exergy destruction within k^{th} component; $ExDes_k^{UN,EN}$ is the unavoidable endogenous exergy destruction within k^{th} component; $ExDes_k^{UN,EX}$ is the unavoidable exogenous exergy destruction within k^{th} component; $ExDes_k^{AV,EN}$ is the avoidable endogenous exergy destruction within k^{th} component; $ExDes_k^{AV,EN}$ is the avoidable exogenous exergy destruction within k^{th} component; $ExDes_k^{AV,EN}$ is the avoidable exogenous exergy destruction within k^{th} component; $ExDes_k^{AV,EN}$ is the

Based on these divisions and the definition for each term, an advanced exergy analysis provides detailed information on the potential improvement to system efficiency by focusing on the avoidable endogenous exergy destruction within a component. Compared to a conventional exergy analysis, this analysis based on the avoidable endogenous exergy destruction is more precise.

Chapter 3: Efficiency Comparison for Electric and Solar Thermal Driven Chillers

3.1 Conventional Chiller Efficiency Comparisons

Before presenting the energy, exergy, and advanced exergy analyses for the solar driven absorption chiller, it is useful to first place the performance of that system in context. As discussed in Chapter 1, a solar driven absorption chiller would most typically be used to replace an electric driven chiller. In the traditional comparison of COPs or efficiencies of these two systems, the site energy consumption (electricity consumption) is considered most frequently, while the primary energy (fossil fuel consumption in the power plant) is not often taken into account. Even if the primary fossil fuel is considered in the comparison (which, in and of itself, provides a more favorable comparison of the COPs), there are only few citations in the literature that move beyond that level to calculate the original energy required through fossil fuel formation. Fossil fuel was first generated by solar energy, accumulated and conserved within a physical format. As a result of this, some scientists have named fossil fuel the "ancient solar energy" (Dukes, 2003).

This chapter provides a new way to compare a solar driven absorption chiller and a traditional electric driven chiller for building cooling. Coal is chosen as a representative fossil fuel, and calculations are made for the solar energy used in generating the coal, based on data in the existing literature. This is then compared with the solar energy requirement of a solar absorption chiller system, given the same amount of cooling requirement. In addition to this energy comparison, this chapter also includes an exergy analysis of the systems. The exergy analysis, combining the first law and second law of thermodynamics, shows a clear picture of the entropy generation in the whole process, and the potential sub-system components to be improved to enhance the overall efficiency.

3.2 Energy Pathways and Efficiencies for the Two Cooling Processes

The context for this comparison can be summarized as follows. Two cooling processes are examined, as illustrated in Figure 3.1. One is a solar absorption cooling process, in which the solar energy is first converted into thermal energy, and then the thermal energy is used to drive the absorption chiller to provide cooling. The other one is a traditional electric chiller system used to provide cooling, in which, for simplicity, the electricity is assumed to be generated from a coal-fired power plant. The coal, as the only fossil fuel under consideration, is transformed from ancient plant matter, which stored ancient solar energy via photosynthesis. By taking into consideration the efficiency for the coal formation from solar energy, the two processes share the same energy provider, the sun.

There are several citations in the literature on the energy efficiencies for the different subprocesses. The efficiency of each process will be discussed in detail in the following subsections.

3.2.1 Photosynthesis Energy Efficiency

There are two different photosynthesis efficiencies that could be used, depending on whether the solar radiation is considered as the net utilized solar radiation or as the total radiation arriving on the leaf surface. If the chemical energy of sugar generated in the plant is only related to the net utilized solar radiation, this is the energy of photosynthetically active radiation (PAR). A mean global efficiency of 0.39 g C MJ-1 PAR is suggested (Potter et al., 1993), which is an approximately 1.7% photosynthesis efficiency. Researchers found that the photosynthesis efficiency is 2.4% in four crop species (Monteith, 1977). Petela (2008) analyzed photosynthesis using classical thermodynamics. In that paper, a simplified mathematical diffusive model of photosynthesis, shown in Figure 3.2, was used to analyze key aspects of the problem. The author found the energy efficiency, through the ratio of the values for the generated sugar and the absorbed radiation (given the assumptions in the paper), to be 2.23%.



Figure 3.1 Process diagram for two cooling technologies: (a) an electric driven chiller and (b) a solar driven absorption chiller



Figure 3.2 Petela (2008)'s simplified scheme of substances and radiation fluxes in photosynthesis

If the chemical energy of sugar generated is related to the total solar radiation on the plant, then when considering the radiation reflected from the leaf, the efficiency will be much lower. Rosen and Scott (2003) considered the Earth as a whole, and then considered the biosphere, people and civilization's energy systems. By calculating the energy input rate to the biosphere with respect to the net solar input, the photosynthesis efficiency is only 0.08%. This lower level of efficiency is also verified by (Encyclopedia Britannica CD 99), which states that the energy efficiency of photosynthesis is much less than 1% based on total solar energy input.

Therefore, this chapter takes the photosynthesis energy efficiencies as $\eta_{ps,PAR} = 2.11\%$ and $\eta_{ps,total} = 0.08\%$, with the subscripts of PAR and total indicating the absorbed solar energy and the total solar energy arriving on the leaf, as outlined in Figure 3.1.

3.2.2 Fuel Synthesis Energy Efficiency

Dukes (2003) reviewed the available literature and found that the net primary productivity (NPP) approach only focused on the consumption of the potential photosynthetic energy captured in modern times, which ignored the energy required through fossil fuel consumption. Dukes (2003) defined a preservation factor (PF) as the fraction of carbon that remains at the end of a transition from one fossil fuel precursor to the next, "such as from plant matter to peat, on the path of coal formation." Firstly, Dukes estimated the productivity of forests on three raised peat deposits based on the available literature. By dividing the mean estimate of annual carbon accumulation by the mean estimate for annual carbon input, the PF for the organic carbon in peat is 15.6%. Then, with time (millions of years), heat, and pressure, peat undergoes the processes needed for coal formation. The resulting PFs of brown coal and hard coal are approximately 95.5% and 69%. Finally, assuming that 50% of the world coal reserve is brown coal and the other 50% is hard coal, the efficiency of the fuel synthesis for each coal type without considering mining is the multiplicative of the PF for organic carbon in peat, the PF for a given type of coal, and the percentage of coal comprised of that type.

Dukes (2003) also discussed mining technology and mining efficiencies. In Arioglu (1994), on average, 62% of underground deposits and 82% of surface deposits are extractable using current mining technologies. With two-thirds of the world's hard coal production coming from underground mines (World Coal Institute, 2000), the mining efficiency is 62%*2/3+82%*1/3= 68.6%. So in this analysis, the fuel synthesis with mining efficiency is $\eta_{fs} = (15.6\%*95.5\%*50\%+15.6\%*69\%*50\%)*68.6\% = 8.8\%$.

3.2.3 Coal-Fired Power Plant Energy Efficiency

There are multiple works in the literature that focus on determining the energy efficiency for coal fired power plants. However, since it is important in this analysis to also consider the exergetic efficiency of a given plant (as will be discussed later), one must obtain the energy and exergetic efficiencies from the same type of power plant, rather than taking the average values. As an example, Rosen and Dincer (1997) took into consideration a coal fired power plant in the Nanticoke generating station, and calculated both the energy and exergetic efficiencies. Individual units of the station each have net electrical outputs of approximately 500MW, one of which is shown in Figure 3.3. It was found that in the overall plant, the energy efficiency, defined as the ratio of net electrical energy output to coal energy input, is 37%.



Figure 3.3 Rosen and Dincer (1997) breakdown of one unit in the coal fired power plant

Similarly, in Erdem et al. (2009), the authors analyzed the performance of nine thermal power plants under the control of governmental bodies in Turkey, from both an energetic and exergetic viewpoint. The considered power plants are mostly conventional reheat steam power plants fed by low quality coal. It was found that the averaged energy efficiency of the nine power plants is 38.6%. Based on the above two values, in this analysis, the coal fired power plant energy efficiency is taken as $\eta_{pp} = 37.8\%$.

3.2.4 Electric Driven Chiller Energy Efficiency

In the American Refrigerant Institute (ARI) 550/590 testing procedure, a series of water chiller packages with minimum efficiency requirements are specified in the International Energy Conservation Code (IECC 2006). Based on these values, an average COP_e of around 5.0 is used in this chapter.

3.2.5 Solar Thermal Collector and Absorption Chiller Energy Efficiencies

The energy efficiency for the solar thermal collector is defined as the useful energy obtained from the thermal fluid loop over the total solar radiation arriving on the collector. The author proposed and analyzed an integrated compound parabolic concentrating collector (CPC) and two-stage hot water absorption chiller in (Hu et al., 2011). The whole system energy flow diagram is shown in Figure 3.4. In the paper, the absorption chiller COP_a was found to be 1.16 and the CPC collector efficiency with a 156 °C collector inlet temperature was $\eta_{collector} = 58.7\%$. This leads to an overall system efficiency of 68% (58.7% multiplied by 1.16).



Figure 3.4 Energy flow and energy conservation for XCPC integrating with absorption chiller (Hu et al., 2011)

(Absorption chiller COP = 1.16, overall system energy efficiency = 58.7% * 1.16 = 68%)

3.3 Exergy Pathways and Exergetic Efficiencies for the Two Cooling Processes

The energy analysis described above is the traditional method of assessing the way energy is used in a process or system. Based on the first law of thermodynamics, the energy analysis and energy efficiency calculations are performed via an energy balance. However, an energy balance provides no information on the degradation of energy or resources during a process and does not quantify the usefulness or quality of the various energy and material streams flowing through a system and exiting as products and wastes (Dincer and Rosen, 2007). As described in more detail in Chapter 2, the concept of exergy combines the first law and second law of thermodynamics. An exergy analysis clearly indicates the locations of energy degradation in a process and can therefore lead to improved operation or technology (Dincer and Rosen, 2007). A main aim of an exergy analysis is to identify meaningful (exergy) efficiencies and the causes and true magnitudes of exergy losses (Dincer and Rosen, 2007). As in the review of energy efficiencies, above, there exist several exergetic efficiencies for the sub-processes in the current literature.

3.3.1 Photosynthesis Exergetic Efficiency

For the photosynthesis exergetic efficiency with respect to total solar radiation on the leaf, Szargut and Peleta (1965) first introduced an exergy analysis of the plants. In their study, an exergy balance was carried out for a one-hectare area of forest over a one year period. It was found that the exergetic efficiency is relatively very small, $\eta_{ps,total,ex} = 0.033\%$, with the growth of wood relating to the total radiation, not only to the PAR part of the whole spectrum. If the photosynthesis exergetic efficiency is only with respect to the absorbed PAR, in the work of (Petela, 2008), Petela found the exergetic efficiency, when only considering the absorbed radiation and neglecting the radiation from the leaf, to be 2.74%. He also indicated that under this efficiency definition, the exergetic efficiency is always larger than the energy efficiency due to the fact that the denominator in the exergetic is, by definition, larger than the denominator in the exergetic

efficiency, whereas the numerator in the energy efficiency is smaller than the numerator in the exergetic efficiency (Petela, 2008).

3.3.2 Fuel Synthesis Exergetic Efficiency

The fuel synthesis exergetic efficiency remains the same as the value calculated in (Dukes, 2003), $\eta_{fs} = \eta_{fs,ex} = 8.8\%$, with the assumption that the chemical exergy contained in the plant shares the same conversion efficiency as the energy in the plant-peat-coalmining conversion.

3.3.3 Coal-Fired Power Plant Exergetic Efficiency

In order to provide an accurate assessment of the exergetic efficiencies for coal-fired power plants, the same literature sources for the energy efficiencies are used, as discussed above. Rosen and Dincer (1997) calculated the exergetic efficiency for the coal fired power plant to be 36%. In the work of Erdem et al. (2009), the averaged exergetic efficiency for the nine coal fired power plants is 33.16%. In this chapter, the coal fired power plant exergetic efficiency is therefore assumed to be $\eta_{pp,ex}=34.58\%$.

3.3.4 Electric Driven Chiller Exergetic Efficiency

In Dincer and Moran (2007), the authors use one chapter to discuss cogeneration and district energy systems. With reference to the exergetic efficiency of an electric driven chiller, it is shown that a centrifugal chiller with a COP of 5.32 has an exergetic efficiency of 21%. Therefore, $\eta_{elec,ex} = 21\%$ is used in this chapter.

3.3.5 Solar Thermal Collector and Absorption Chiller Exegetic Efficiencies

In Hu et al. (2011), the authors calculated the exergetic efficiencies for both a compound parabolic concentrating collector and an absorption chiller, together with the energy efficiencies. The results showed that the exergetic efficiency for the two stage absorption

chiller is $\eta_{abs,ex} = 9.96\%$ with a 45 °C condensing temperature and 4 °C chiller water supply temperature. At the same time, the CPC collector exergetic efficiency is $\eta_{collector,ex}$ = 19.7% with a collector inlet temperature of 156 °C and an outlet temperature of 165 °C. This leads to an overall system exergetic efficiency of 1.96% (9.96% multiplied by 19.7%). The exergy flow and exergy destruction diagram is shown in Figure 3.5.



Figure 3.5 Exergy flow and exergy destruction for XCPC integrating with absorption chiller (Hu et al., 2011) (Absorption chiller exergetic efficiency = 9.96%, overall system exergetic efficiency = 9.96% * 19.7% = 1.96%)

3.4 System Comparison

Now that the energy and exergetic efficiencies of each process have been defined, the conventional electric driven chiller system and solar driven absorption chiller system can be compared. Returning to Figure 3.1, the total efficiency for converting sunlight to cooling for the electric driven chiller is:

$$\eta_{\text{total,elec}} = \eta_{\text{ps,total}} * \eta_{\text{fs}} * \eta_{\text{pp}} * \text{COP}_{e}$$
(3.1)

which is the efficiency of the photosynthesis process, multiplied by the fuel synthesis, multiplied by the power plant efficiency, and finally multiplied by the coefficient of performance of the electric driven chiller. Therefore, the total efficiency of the electric driven chiller is:

$$\eta_{\text{total,elec}} = 0.08\% * 8.8\% * 37.8\% * 5 = 0.000133 = 0.0133\%$$
 (3.2)

In a similar fashion, the total efficiency for converting sunlight to cooling for the solar driven absorption chiller is:

$$\eta_{\text{total ,abs}} = \eta_{\text{collector}} * \text{COP}_{a}$$
(3.3)

which is the efficiency of the solar collector multiplied by the coefficient of performance of the absorption chiller. This calculation results in a total efficiency of:

$$\eta_{\text{total ,abs}} = 58.7\% * 1.16 = 0.681 = 68.1\%$$
 (3.4)

The two energetic efficiencies can then be compared:

$$\text{Ratio}_{\text{SolarEnergyConversion}} = \frac{\eta_{\text{total ,abs}}}{\eta_{\text{total ,elec}}} = \frac{68.1}{0.0133}$$
(3.5)

where it can be seen that the solar driven absorption chiller is approximately 5110 times more efficient in converting solar energy input to cooling.

The exergetic efficiencies for both systems are constructed in an analogous way. The total exergetic efficiency of the electric driven chiller is:

$$\eta_{\text{total ,elec ,ex}} = \eta_{\text{ps ,total ,ex}} * \eta_{\text{fs ,ex}} * \eta_{\text{pp ,ex}} * \eta_{\text{elec ,ex}}$$
(3.6)

$$\eta_{\text{total,elec,ex}} = 0.033\% * 8.8\% * 34.58\% * 21\% = 0.00021\%$$
(3.7)

while the total exergetic efficiency for the solar driven chiller is:

$$\eta_{\text{total ,abs,ex}} = \eta_{\text{collector ,ex}} * \eta_{\text{abs,ex}}$$
(3.8)

$$\eta_{\text{total,abs,ex}} = 19.7\% * 9.96\% = 1.96\%$$
 (3.9)

The two exergetic efficiencies can then be compared:

$$Ratio_{SolarExergyConversion} = \frac{\eta_{total,abs,ex}}{\eta_{total,elec,ex}} = \frac{1.96}{0.00021}$$
(3.10)

The ratio of the two exergetic efficiencies is found to be equal to 9,294.

3.5 Discussion

Obviously, the magnitude of these numbers is not of immediately practical application for building owners or operators. However, it is important to note that the traditional COPs used to compare electric driven and heat driven systems are equally prone to fallacies of interpretation. In comparing the COP of an electric driven chiller (even including plant efficiency) to the COP of a solar driven absorption chiller, those calculations do not account for the ultimately limited availability of the fossil fuels used to produce that electricity. The goal of this chapter, then, is to provide a possible step in that direction. It should also be further noted that even the calculations provided in this analysis do not account for the necessary temperature, pressure, and time that are needed to form the fossil fuels. For example, the formation time for coal is in millions of years, while the formation time for a CPC collector is from hours to half a day based on design, installation and local solar radiation. Dukes (2003) showed a calculation that the fossil fuels burned in a whole year (1997) were created from organic matter containing 44* 1018 grams of carbon, which is 400 times larger than the net primary productivity of the planet's current biota.

In addition to the quantity of energy required, the quality of the energy is equally relevant. An important point to be made is that the occupancy thermal comfort zone is very close to the reference environment, compared to the temperature and pressure in a coal-based power plant. If one can use low quality energy (for example, thermal energy from CPC collectors) to meet the cooling requirement, a better supply-demand condition is met, and a higher exergetic efficiency system is designed. At the same time, generated electricity can be saved to serve applications with high quality energy requirements, such as personal electronics or computers. The current electricity grid system is under increasing stress from society's higher power demands, and better matching exergy levels can help to alleviate this coming crisis.

Of course, while there are many advantages, as mentioned above, for a solar thermal cooling system, there are also disadvantages and application limitations. Based on current efficiencies and a typical building cooling load, the area required for a solar collector is large, which makes its application impossible for a high density urban environment. The ideal application areas will be for low density communities, like shopping malls. For high density urban environments, Mazdar city in Abu Dhabi presents an example plan for district cooling (Hope and Stanton, 2008). By combining them with a district cooling system, the solar thermal collectors may not only be installed on the roof, but can also be

installed in series/parallel groups on the ground to provide thermal energy to the chiller for high density urban environment cooling.

This analysis could, of course, be extended further. For example, no calculations of the environmental impact of the creation and fuel consumption of the respective systems were included. Moreover, all fossil fuel-based electricity generation does not originate with coal. A comparative analysis using, for example, natural gas for power generation would also be instructive. Finally, the possibility of using a solar photovoltaic system to generate electricity was also not considered. Compared to a solar thermal application, a solar photovoltaic system would have a higher price, and a higher embedded energy, but might still result in a more favorable energy and exergetic efficiency comparison. Nonetheless, even aside from these future considerations, with a discussion about the pros and cons for the two given systems in terms of efficiency, this chapter is useful in understanding the true fundamental life cycle energy and exergy costs in chiller applications for building cooling.

Chapter 4: Conventional Exergy Analysis for the Parabolic Trough Solar Collector and the Double Stage Absorption Chiller

Chapter 3 discussed a novel comparison between an electric and an absorption chiller, including a method for calculating the true solar energy requirement for fossil fuel (coal) formation. The same comparison was extended to the exergy domain. Compared to an electric chiller, in order to provide the same amount of cooling, a solar driven absorption chiller actually consumes a much smaller amount of both solar energy and exergy.

However, beyond demonstrating this lower level of actual energy/exergy consumption, it is important to increase the efficiency of the solar chiller system. Therefore, in this chapter, a detailed exergy destruction analysis is performed both for the solar collector and the double stage absorption chiller.

4.1 Experimental Setup for the Parabolic Trough Solar Collector

The parabolic trough solar collector that will be analyzed is the Broad BJ16A model, which is installed in the Robert L. Preger Intelligent Workplace (IW), an office space for multiple uses (classrooms, laboratories, meeting spaces, and offices for faculty and students) at Carnegie Mellon University.

To satisfy the requirement of the double effect absorption chiller, 52 m^2 of linear parabolic trough solar thermal receivers were installed, including a circulating propylene glycol water mixture; instrumentation for flow, temperature, pressure, and direct normal solar radiation; circulation pumps; an expansion/pressure tank; and a drain/filling

apparatus. The process instrumentation diagram of the test solar absorption cooling and heating system is shown in Figure 4.1.



Figure 4.1 Processes and instrumentation diagram of the test solar absorption cooling and heating system (Qu, 2008)

This experimental setup is thoroughly discussed in Qu (2008). The parabolic trough solar collector is a single axis tracking solar collector, which tracks the solar altitude from east to west during the day to ensure that the solar radiation is fully redirected and focused on the linear receiver.

A module of the Broad parabolic trough solar collector receiver weighs 200 kg. It is designed to withstand a 31 m/sec wind load. The 13.34 m² aperture area and 0.68 m² receiver area corresponds to a 19.6concentration ratio. The Broad parabolic trough solar collector tracking drive is a large semi circular gear engaged with a small gear powered by a 24 Volt servo motor. The picture and the characteristics of the parabolic trough solar collector installed in the IW are addressed in Figure 4.2 and Table 4.1 (Qu, 2008).



Figure 4.2 Broad BJ16A parabolic trough solar collectors installed on the roof of Intelligent Workplace (Qu, 2008)

Table 4.1 Broad BJ16A	parabolic trough solar collector	specifications ((Qu, 2008)
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BJ16A Parabolic Trough Solar Collector				
Manufacturer	Broad Air Conditioning Broad Town, Changsha, Hu Nan, China			
Operating Temperature	60-180 °C			
Module Size	5.75 m * 2.3m, 13.225 m ²			
Module Operating Weight	200 kg			
Drive Group Size	2 modules; $26.45m^2$			
Delta-T Loop Size	2 drive groups; 52.9 m^2			
Rim angle	73 °			
Reflectors	Typical reflectance 0.8			
Focal length	81.8 cm			

	Absorber OD: 3.8 cm		
Receiver	Base Material: Stainless steel 304L		
	Coating: Black nickel		
	Typical absorptivity: 0.96		
	Typical emittance: 0.14@ 100 ℃		
	Pyrex glass cover OD: 10.2 cm		
	Transmissivity: 0.91		
	Vacuum in the annular space		
Sun Tracking	Single-axis elevation tracking based on the calculated sun position		
Tracking Drive	24 V Powered		
System	Servo motor		
WindLoada	16 m/sec (tracking)		
wind Loads	31 m/sec (stowed)		

4.2 Experimental Results for the Parabolic Trough Solar Collector

Qu (2008) reported experimental results for operation of the installed parabolic trough solar collector in the Intelligent Workplace. The results showed that the optical efficiency is 0.634 and the linear coefficient of thermal losses is $1.4 \text{ W/K} \cdot \text{m}^2$. Therefore, the parabolic trough solar collector energy performance equation can be expressed as:

$$\eta = \eta_0 - k_{\text{loss}} * \left(\frac{T_r - T_{\text{ambient}}}{H_{\text{DNI}}}\right)$$
(4.1)

where η_0 is the optical efficiency (0.634), T_r is the receiver temperature, T_b is the ambient temperature (30 °C), k_{loss} is the thermal loss coefficient (1.4 W/K•m²), and H_{DNI} is the direct normal irradiance (DNI) on the collector.

Figure 4.3 can be used to visualize how equation (4.1), the collector's thermal efficiency, is related to the total solar radiation on the collector, the optical efficiency and the thermal loss. For this installation, the measured data show a 63.4% optical efficiency, which

means, given a 100% solar radiation shining on the parabolic trough solar collector, only 63.4% can be absorbed in the collector. As the collector thermal fluid heats up, the thermal loss is proportional to $\frac{T_r - T_{ambient}}{H_{DNI}}$; in this case, the proportionality constant is 1.4.

The energy analysis method is a useful technique for providing one measure of the performance of the system, but cannot provide all of the required information to fully understand the energy quality losses for the parabolic trough solar collector. Sometimes, it is even misleading for engineers to try to improve the system efficiency. For an example, based on the energy analysis, the only two ways to increase the system efficiency are 1) increase the collector optical efficiency, and 2) reduce the collector thermal loss. However, increasing the optical efficiency can cause changes in the collector's configuration that lead to unintended consequences in its operation. In the next section, a detailed exergy destruction analysis method is introduced for the same collector given the same inlet/outlet and ambient conditions. This can provide a clear picture of all of the energy quality losses in operating the system, in order to better target ways to improve the system efficiency.



Figure 4.3 Parabolic trough solar collector energy analysis

4.3 Exergy Modeling for the Parabolic Trough Solar Collector

In the exergy domain, all the of necessary entropy generation mechanisms are evaluated. For a parabolic trough solar collector, based on Suzuki (1988), the exergy losses not only include optical loss and thermal loss to the ambient environment, but also include the exergy destruction inside the parabolic trough solar collector such as:

- Absorption exergy destruction: exergy destruction due to the temperature difference between the collector receiver and flowing fluid;
- Heat conduction destruction: exergy destruction due to the axis temperature difference; and
- Friction exergy destruction: exergy destruction due to fluid friction.

A general balance equation can be written as follows for the steady state condition:

$$Ex_{in} = Ex_{optical} + Ex_{thermal} + Ex_{absorption} + Ex_{conduction} + Ex_{friction} + Ex_{useful}$$

$$(4.2)$$



This equation is illustrated in Figure 4.4.

Figure 4.4 Parabolic trough solar collector exergy analysis

Given the thermal efficiency equation, $\eta = \eta_0 - k_{loss} *(\frac{Tr - Tambient}{H_{DNI}})$, one can find that the exergy of the total solar radiation on the parabolic trough solar collector per unit area is:

$$Ex_{in} = H_{DNI}(1 - \frac{T_{ambient}}{T_{sun}})$$
(4.3)

where T_{sun} is the apparent sun temperature and equals 75% of the blackbody temperature of the sun, which is 4500K (Bejan et al. 1981).

Therefore, the optical exergy loss is:

$$Ex_{optical} = (1 - \eta_0) H_{DNI} \left(1 - \frac{T_{ambient}}{T_{sun}}\right)$$
(4.4)

The thermal exergy loss to the environment is:

$$Ex_{thermal} = k_{loss} (T_r - T_{ambient})(1 - \frac{T_{ambient}}{T_r})$$
(4.5)

The absorption exergy destruction is:

$$Ex_{absorption} = \eta_0 H_{DNI} \left[\left(1 - \frac{T_{ambient}}{T_{sun}} \right) - \left(1 - \frac{T_{ambient}}{T_r} \right) \right]$$
(4.6)

The heat conduction exergy destruction per unit area is given by Suzuki (1988):

$$Ex_{conduction} = T_{ambient} \Delta S_{conduction}$$
(4.7)

$$\Delta S_{\text{conduction}} = \int_{T_i}^{T_o} \frac{c_p \, \text{mdT}}{T} - \frac{1}{T_p} \int_{T_i}^{T_o} c_p \, \text{mdT}$$
$$= c_p m \left[\ln(\frac{T_o}{T_i}) - \frac{T_o - T_i}{T_r} \right]$$
$$= m \left[(s_o - s_i) - \frac{h_o - h_i}{T_r} \right]$$
(4.8)

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where c_p is the working fluid specific heat, h is the working fluid specific enthalpy, and s is the working fluid specific entropy.

The useful exergy gain is:

$$Ex_{useful} = c_p m \left[T_o - T_i - T_{ambient} ln \left(\frac{T_o}{T_i} \right) \right]$$

= m[H_o - H_i-T_{ambient} (S_o-S_i)] (4.9)

If the friction exergy loss is considered, the useful exergy gain is then:

$$Ex_{useful, friction} = Ex_{useful} - Ex_{friction}$$
(4.10)

where the friction exergy loss is given by Bejan (1982):

$$Ex_{friction} = \frac{mT_{ambient} \ \Delta P}{\rho T_i}$$
(4.11)

The pressure drop in the collector tubes is mainly considered in Kar (1985):

$$\Delta \mathbf{P} = \mathbf{f} \rho \mathbf{L} \frac{\mathbf{V}^2}{2\mathbf{D}} \tag{4.12}$$

and the friction coefficient f is given by:

$$f = \frac{64}{Re} for Re \le 2200$$

f = 0.316Re^{-0.25} for Re>2200 (4.13)

where Re is the Reynolds number, and is a measure of the velocity, density, viscosity, and geometry of the flow. With equations (4.2) to (4.13), a comprehensive exergy analysis for parabolic trough solar collector is provided.

In order to show the importance of considering the friction exergy loss, two cases are presented.

a) Exergy analysis without considering the friction exergy loss

If the friction exergy loss is not considered, one can obtain an exergy efficiency plot that is similar to a thermal efficiency plot. Figures 4.5 and 4.6 show the thermal and exergy efficiency fractions for a total solar radiation of 100%, while Tables 4.2 and 4.3 show the quantitative breakdown of the energy/exergy and losses. As can be seen from Figures 4.5 and 4.6, the energy efficiency for the parabolic trough solar collector at 160 $\$ is 38%, while the exergetic efficiency is 12%.



Figure 4.5 Stacked energy fractions

Tr – Tambient	Useful	Thermal	Optical	
HDNI	Energy	Loss	Loss	
IIDNI	(W)	(W)	(W)	
0.1	359.138	101.78	266.082	
0.11	348.96	111.958	266.082	
0.12	338.782	122.136	266.082	
0.13	328.604	132.314	266.082	
0.14	318.426	142.492	266.082	
0.15	308.248	152.67	266.082	
0.16	298.07	162.848	266.082	
0.17	287.892	173.026	266.082	
0.18	277.714	183.204	266.082	
0.19	267.536	193.382	266.082	

Table 4.2 Energy analysis for the parabolic trough solar collector (per unit area)



Figure 4.6 Stacked exergy fractions

Tr — Tambient HDNI	ExUseful (W)	ExDes Cond (W)	ExLoss Thermal (W)	ExDes Absorp (W)	ExLoss Optical (W)	ExSolar Input (W)
0.1	69.45	0.02308	19.69	340.7	248.1	678
0.11	72.83	0.01999	23.37	333.6	248.1	678
0.12	75.7	0.01729	27.3	326.8	248.1	678
0.13	78.09	0.01493	31.45	320.3	248.1	678
0.14	80.03	0.01286	35.82	314	248.1	678
0.15	81.55	0.01106	40.39	307.9	248.1	678
0.16	82.66	0.00949	45.16	302	248.1	678
0.17	83.38	0.008121	50.12	296.3	248.1	678
0.18	83.74	0.00693	55.24	290.8	248.1	678
0.19	83.75	0.005894	60.54	285.5	248.1	678

Table 4.3 Exergy analysis for the parabolic trough solar collector (per unit area)

In the energy analysis, it is obvious that the higher the receiver temperature, the higher the thermal energy loss, and thus the lower the energy efficiency. However, in the exergy analysis with the same temperature range, because the solar collector is leaking thermal energy to the environment at the collector receiver temperature, and the receiver temperature compared to the apparent sun temperature is smaller, the exergy losses due to heat leakage to the ambient environment is small, as shown in Figure 4.6, but the exergy efficiency increases with the increasing receiver temperature.

In the figure, the line for useful exergy and the line for exergy destruction due to axis heat conduction almost merge with each other. This means that the conduction exergy destruction is a very small quantity, which can be ignored for the analysis.

As mentioned above, if just an energy analysis is applied to the collector, the only ways to increase the collector efficiency are reducing the optical and thermal energy losses. However, this exergy analysis clearly shows all of the entropy generation (exergy destruction) in this parabolic trough solar collector, and it can be seen that, in order to increase the collector efficiency, it equally important to reduce the heat absorption exergy loss.

b) Exergy analysis considering the friction exergy loss

The previous analysis is useful as a baseline calculation, but the exergy fractions shown in Figure 4.6 do not reflect that exergy efficiency can change with a varying mass flow rate or friction exergy loss, which is an important parameter. A comparison between the thermal and exergy efficiency with changing mass flow rate is provided below. In this analysis, it is considered that the parabolic trough solar inlet temperature is constantly at $150 \,^{\circ}$ for a two stage absorption chiller application (normal operating temperatures range from $150 \,^{\circ}$ to $180 \,^{\circ}$). Figure 4.7 shows the temperature profiles with the changing mass flow rate, and Figure 4.8 shows the exergy and energy efficiency with the changing mass flow rate under this constant inlet temperature.



Figure 4.7 Temperature profiles vs. mass flow rate


Figure 4.8 Exergy and energy efficiency vs. mass flow rate

From Figures 4.7 and 4.8, it can be seen that the receiver temperature is lower at the higher mass flow rate. This is because with a higher mass flow rate, the absorber is better cooled down by the lower solar collector inlet temperature; thus, at a higher mass flow rate, the exergy efficiency is lower. However, the higher the mass flow rate, the lower the absorber temperature, and thus the lower the thermal loss to the ambient environment, so the higher the thermal energy efficiency. This demonstrates the opposite trends in energy and exergy efficiency with changing the mass flow rate. From Figure 4.8, with the mass flow rate changing from 0.005 kg/s to 0.02 kg/s, the exergy efficiency decreases from 12.4% to 12%, a 0.4% efficiency reduction, while the energy efficiency increases from 38.47% to 39.01%, a 0.54% efficiency increase.



Figure 4.9 Exergetic efficiency with/without friction exergy loss

Based on equations (4.10) to (4.13), the friction exergy loss is small at small mass flow rates. The exergetic efficiencies with and without friction exergy loss are plotted in Figure 4.9. From Figure 4.9, the highest limiting mass flow rate for the parabolic trough solar collector is 4.1 kg/s, at which the exergetic efficiency is 0.1%. Almost all of the useful exergy collected in the collector is offset by the friction exergy loss in the system. Based on Figures 4.8 and 4.9, the parabolic trough solar collector should ideally be operated at a lower mass flow rate to take advantage of the higher exergetic efficiency.

4.4 Experimental Setup for the Double Stage Absorption Chiller

The 52 m² parabolic trough solar collector described in Section 4.1 is integrated with a 16 kW double stage lithium bromide absorption chiller to provide cooling and heating to Intelligent Workplace. The Broad absorption chiller is driven either by hot water or by natural gas. This chiller incorporates a cooling tower to reject heat from its operation as required in the cooling cycle, as shown in Figure 4.10, together with the schematic system diagram.

The absorption chiller was thoroughly tested in Yin (2006) and Qu (2008). Some of the experimental data from these studies are cited for model validation purposes in the following sections.



Figure 4.10 Installation and schematic diagram for the double stage absorption chiller in its design condition

4.5 Ideal Cycles for Absorption Chillers

In order to quantify the potential for increasing the efficiency of absorption chiller cycles, it is important to understand the theoretical upper limit of the COP for a given set of ambient conditions. The theoretical upper limit of the COP is defined as the COP that could be attained from ideal thermal processes that have no entropy generation (no exergy destruction). The Carnot cycle is a theoretical thermodynamic cycle that is the most efficient cycle for converting a given amount of thermal energy into work (and the reverse), since no entropy is generated for any of the state change processes. Under this context, the Carnot and reverse Carnot cycles are used to define the ideal cycles for absorption chillers, and thus to calculate the theoretical upper limit of the COP.

4.5.1 Ideal Cycles for a Single Stage Absorption Chiller

Figure 4.11 shows an ideal cycle for a single stage absorption chiller. The cycles are a combination of a Carnot cycle and a reverse Carnot cycle. T_0 , T_1 and T_2 are the evaporating, condensing and regenerating temperatures, respectively (Herold, 1996). The Carnot cycle receives heat Q_2 at temperature T_2 , and dumps the heat Q_1 ' to the environment while providing work W (= $Q_2 - Q_1$ '). At the same time, the work W is used in the reverse Carnot cycle. The heat Q0 is absorbed from the space to be cooled at temperature T0, and the heat Q_1 '' (= Q_0 +W) is released into the environment.



Figure 4.11 Ideal cycles for a single stage absorption cycle

The total absorption cycle efficiency is the multiplication of the Carnot and the reverse Carnot efficiencies, as shown in equation (4.14), which are functions of the absolute temperatures between which the cycles operate. This theoretical efficiency is the highest absorption chiller COP for the given set of environmental conditions (the room, condensing, and regenerating temperatures). The theoretical COP, since there is no entropy generation, also signifies a100% system exergetic efficiency.

$$\text{COP}_{\text{rev, single}} = \frac{\text{T2}-\text{T1}}{\text{T2}} * \frac{\text{T0}}{\text{T1}-\text{T0}}$$
 (4.14)

It can be observed that the theoretical upper COP limit is only associated with T_0 , T_1 , and T_2 , and it is independent of the working fluids. For a typical single stage absorption chiller operating in the summer, if the regenerating temperature T_2 is 95°C (368 K), the condensing temperature T_1 is 30°C (303 K), and the evaporating temperature T_0 is 7°C (280 K), the theoretical COP is therefore 2.15. Compared with currently available single stage absorption chillers, for which the normal COP is around 0.6, this indicates that more than 70% of the useful work is destroyed due to entropy generation (exergy destruction) within the components and processes.

4.5.2 Ideal Cycles for a Double Stage Absorption Chiller

Based on the ideal cycles for a single stage absorption chiller, an additional Carnot cycle can be introduced to form the ideal cycles for a double stage absorption chiller. As shown in Figure 4.12, T_3 is the temperature in the low temperature regenerator, while T_0 , T_1 and T_2 remain the same as those in the cycles for a single stage absorption chiller.

The upper Carnot cycle receives heat Q_2 at temperature T_2 , and dumps the heat Q to the lower Carnot cycle while providing work W_1 (= $Q_2 - Q$). At the same time, the work W_1 is used in the reverse Carnot cycle. The lower Carnot cycle receives heat Q from the higher cycle, and dumps the heat Q_1 ' to the environment, while providing work W_2 (=Q- Q_1 '). The total work W=W_1+W_2 is used in the reverse Carnot cycle to provide cooling. The heat Q_0 is absorbed from the space to be cooled at temperature T_0 , and the heat Q_1 " (= Q_0 +W) is released into the environment.

The advantage of a double stage absorption chiller compared to a single stage absorption chiller is the high temperature regenerator can utilize the higher temperature, higher quality heat source more efficiently by recovering the thermal energy and powering a low temperature regenerator. From a thermodynamic point of view, double stage absorption chillers provide additional work (W_2) to run the reverse Carnot cycle for cooling.



Figure 4.12 Ideal cycles for a double stage absorption cycle

The total absorption cycle efficiency, then, is the addition of the multiplication of the Carnot and the reverse Carnot efficiencies for the high and low temperature regenerators, as shown in equation (4.15). Again, it is the theoretical COP, so there is no entropy generation, and thus, it also signifies a 100% system exergetic efficiency for the double stage absorption chiller.

$$COP_{rev, double} = \frac{T2 - T3}{T2} * \frac{T0}{T1 - T0} + \frac{T3 - T1}{T3} * \frac{T0}{T1 - T0}$$
(4.15)

In a similar manner as the single stage absorption chiller, the theoretical COP high limit is only associated with T_0 , T_1 , T_2 , and T_3 , and it is independent of the working fluids. Compared to the ambient temperatures, such as T_0 , T_1 , and T_2 , T_3 is an internal temperature which represents the low temperature regenerator, and thus T_3 can vary between the condensing temperature T_1 and the high temperature regenerator temperature T_2 , based on different chiller designs. Figure 4.13 shows the effect of varying T_3 between T_1 and T_2 , given a typical double stage absorption chiller operating in summer, if the regenerating temperature T_2 is 165°C (438K), the condensing temperature T_1 is 30°C (303K), and the evaporating temperature T_0 is 7°C (280K).



Figure 4.13 Theoretical COP for a double stage absorption chiller

It can be seen from Figure 4.13 that the theoretical COP for a double stage absorption chiller varies between 3.75 and 4.09. It peaks at 4.09, when T_3 is 91°C. Based on the theoretical upper COP limit of 4.09, compared with currently available double stage absorption chiller COPs of 1.1, it can again be seen that more than 70% of the useful work is destroyed due to entropy generation (exergy destruction).



Figure 4.14 COPs from current technology vs. COPs in ideal cycles for single and double stage absorption chillers

To sum up, as indicated in Figure 4.14, the current COPs for single and double stage absorption chillers are 0.6, and 1.1 respectively, while, given the assumptions for T_0 , T_1 , T_2 , and T_3 , the COPs for ideal cycles are 2.51 and 4.09. In both systems, more than 70% of the working potential for the thermal energy input is destroyed via energy conversions. The COP of 4.09 for double stage absorption chillers is the highest limit given the same set of ambient conditions. Better design, engineering and manufacturing will help to increase chiller COP towards this high limit. To aid in moving toward this direction, in the following sections, the methodology for an advanced exergy analysis will be discussed and then used in diagnosing the exergy destruction recovery potentials for increasing double stage absorption chiller COPs.

4.6 Energy and Exergy Modeling for the Double Stage Absorption Chiller

In order to understand the capacity for increasing the efficiency of a double stage absorption chiller, it is important to define the limitations on the various state points in the cycle. The following subsections define the state point assumptions, specify the relevant temperature differences at the exits and inlets of the heat exchangers, examine the effects of using lithium bromide, and then show how the exergy generation is calculated using those specifications.

4.6.1 State Point Assumptions for the External Loops

The external loops are, specifically, the solar loop into and out of the high temperature regenerator, the building loop into and out of the evaporator, and the condensing loop into and out of the condenser, as shown in the red color in Figure 4.15.

The solar loop uses water as the heat transfer media. In order to avoid flashing/steaming in the solar loop, the water has to be pressurized. In this study, the solar loop pressure is defined as the saturation pressure in the solar loop into the high temperature regenerator. The building loop and condensing loop are at ambient pressure.

The temperatures for the condensing loop inlets are 30°C for a typical summer operation, and the condensing loop outlet temperatures are calculated based on energy balances for the absorber and condenser. The temperature from the solar loop to the high temperature regenerator is 165°C based on experimental results (Qu, 2008). The temperature from the building loop to the evaporator is 12°C, and to the building loop after the evaporator is 7°C, based on manufacturer design conditions (Broad brochure 2008).

The mass flow rate for building loop is calculated based on 16 kW of cooling and the enthalpy difference between the evaporator inlet and outlet, while the mass flow rates in all of the other loops are assumed to be 1 kg/s (15.9 GPM water). All of the assumptions for the various state points in the external loops are summarized in Table 4.4.



Figure 4.15 A double stage absorption chiller with external loops

A double stage absorption chiller, as shown in Figure 4.15, has four external heat exchangers (high temperature regenerator, evaporator, absorber, and condenser) and three internal heat exchangers (low temperature regenerator, high temperature heat exchanger, and low temperature heat exchanger) working simultaneously to provide cooling from a given heat source. In order to reduce the exergy destruction inside the double stage absorption chiller, it is important to understand the heat transfer mechanisms and temperature differences within each heat exchanger. In the following subsections, five

heat exchanger T-Q diagrams are presented, together with the assumptions behind the resulting temperature differences in each one. Again, a summary of some of the various external state points is also presented in Table 4.4.

State Points	State Point Position	Temperature [°C]	Pressure [kPa]	Enthalpy [kJ/kg]	Entropy [kJ/kgK]	Mass Flow Rate [kg/s]
21	HTRG inlet	165	700.3	697.4	1.993	1
22	HTRG outlet	Calculated	700.3	Calculated	Calculated	1
23	ABS inlet	30	101.3	125.8	0.4365	1
24	ABS outlet	Calculated	101.3	Calculated	Calculated	1
25	CON inlet	30	101.3	125.8	0.4365	1
26	CON outlet	Calculated	101.3	Calculated	Calculated	1
27	EVP inlet	12	101.3	50.5	0.1804	Calculated
28	EVP outlet	7	101.3	29.5	0.1063	Calculated

Table 4.4 Assumptions for the state points in the external loops

High Temperature Regenerator

For the high temperature regenerator, from Figure 4.15, it can be seen that T21 and T22 are the temperatures for the solar loop inlet and outlet, T13 is the temperature of the dilute solution pumped into the HTRG, and T14 is the temperature of the strong solution leaving the HTRG. T17 is not shown in the figure, it is the temperature for the generated water vapor leaving the HTRG where the dilute solution gets heated and reaches its saturated condition. When water volatilizes from the dilute solution, the mass fraction for the solution will increase. Plotting this heat transfer process on a T-Q diagram results in Figure 4.16.

Based on this T-Q diagram, one can define a temperature difference:

$$\Delta T_{\rm HTRG} = T21 - T14$$
 (4.16)

Since T21 is fixed at 165°C from the solar array, a lower value of ΔT_{HTRG} indicates a more efficient transfer of heat, a higher strong solution outlet temperature (T14), and a lower level of exergy destruction in the high temperature regenerator.



Figure 4.16 T-Q diagram for high temperature regenerator

Low Temperature Regenerator

Figure 4.17 shows the T-Q diagram for the low temperature regenerator in a double stage absorption chiller.

Although the water vapor leaving the HTRG and entering the LTRG at state T17 is initially in a superheated state, it rapidly condenses to a saturated state. Therefore, for simplicity, it can be assumed that the high temperature stream is at the constant condensing temperature T18, so that one can define as a temperature difference:

$$\Delta T_{\rm LTRG} = T18 - T4 \tag{4.17}$$

As before, the lower the value of the ΔT_{LTRG} , the more efficient the heat transfer, and also the lower the exergy destruction.



Figure 4.17 T-Q diagram for low temperature regenerator

Condenser

Figure 4.18 shows the T-Q diagram for the condenser in a double stage absorption chiller.

T7 is the temperature for the water vapor in a superheated state. In the condenser, the superheated water vapor condenses to saturated water vapor, which is at a constant temperature T8. T25 is ambient temperature, 30°C, under typical summer conditions. T19 is the temperature for the water generated from high temperature regenerator after passing the expansion valve. T19 is in a two-phase condition. Since state point 19 has the same pressure as state point 8, so T19 equals T8. One can then define the temperature difference:

$$\Delta T_{\rm CON} = T8 - T25 \tag{4.18}$$

Since T25 is 30°C, the lower the value of ΔT_{CON} , the more efficient the heat transfer, and the lower the exergy destruction in the condenser.



Figure 4.18 T-Q diagram for condenser

Absorber

Figure 4.19 shows the T-Q diagram for the absorber in a double stage absorption chiller.

State points 23 and 24 are the entry and exit states of the condensing loop, whose purpose is to take away the heat generated from the solution and water vapor mixing in the absorber. State point 1 is the saturated lithium bromide solution leaving the absorber.

As before, the temperature difference is defined as:

$$\Delta T_{ABS} = T1 - T23 \tag{4.19}$$

where T23 is the ambient temperature (30°C). It can be observed that the lower the ΔT_{ABS} , the more efficient the heat transfer to cool down the fluids to state point 1, and the lower the exergy destruction in the absorber.



Figure 4.19 T-Q diagram for absorber

Evaporator

Figure 4.20 shows the T-Q diagram for the evaporator in a double stage absorption chiller.

T28 and T27 are design conditions for the building loop out of the evaporator (a 7°C temperature set point) and the building loop into the evaporator (12°C). T9 and T10 represent the entry and exit points for the water vaporizing inside the evaporator, and are therefore the same temperature. So, in order to provide cooling, T9 and T10 must be lower than 7°C, and a temperature difference can be defined as:

$$\Delta T_{\rm EVP} = T28 - T9 = 7 \,\rm C - T9 \tag{4.20}$$

It can be observed that the lower the value of ΔT_{EVP} , the lower the exergy destruction inside the evaporator.



Q, Heat Movement

Figure 4.20 T-Q diagram for evaporator

High Temperature and Low Temperature Heat Exchangers

The two internal heat exchangers are used to recover the heat from the strong solution flows leaving the regenerators to the dilute solution flows going into the regenerators. Therefore, the state points are not directly related to the temperatures of the external loops, and the heat exchanger effectiveness is used to define the efficiencies of the two heat exchangers.

$$\varepsilon_{\rm HTHX} = \frac{T15 - T14}{T12 - T14} \tag{4.21}$$

and

$$\varepsilon_{\text{LTHX}} = \frac{\text{T5}-\text{T4}}{\text{T2}-\text{T4}} \tag{4.22}$$

4.6.2 Lithium Bromide Enthalpy-Composition Diagram for Real Cycles

Figure 4.21 shows the state points of an actual double stage absorption chiller on a lithium bromide solution enthalpy-composition chart. As defined in the previous subsection, the temperature differences include ΔT_{HTRG} , ΔT_{LTRG} , ΔT_{ABS} , ΔT_{CON} , and ΔT_{EVP} . State point 1 is the saturated solution at T1 and low pressure. The solution pump increases the pressure to the intermediate pressure to move to subcooled state point 2. Since only a small amount of pump work is provided, the enthalpy of state point 2 is approximately the same as at state point 1. After the low temperature heat exchanger, state point 2 is heated to state point 3 by recovering the heat from the low temperature heat exchanger. In the low temperature regenerator, there are two separate loops; one loop is heated to state point 4, and superheated steam is generated at state point 7, and the other loop indirectly is heated to state point 11.

The solution leaving point 4 enters the low temperature heat exchanger to preheat the incoming fluid and reduce its temperature to state point 5. After an expansion valve, state point 5 reduces its pressure to the two-phase mixture region at state point 6. Since an

expansion value changes the state of a fluid at a constant enthalpy, the enthalpy for state point 6 stays the same as that of state point 5.



Figure 4.21 Thermodynamic representation for real cycles in a double stage absorption chiller with all the irreversibility

Another solution pump increases the pressure at state point 11, and thus reaches state point 12. The processes between 12 to 13, 13 to 14, 14 to 15, and 15 to 16, are the same as 2 to 3, 3 to 4, 4 to 5, and 5 to 6. The solution from state point 16 is then combined with state point 4.

As part of the process from 13 to 14, high temperature superheated steam is provided at state point 17. This steam then enters the low temperature regenerator to provide superheated steam state point 7 (as discussed in the previous paragraph). At the same time, superheated steam at state point 17 is condensed to saturated liquid water at high pressure, and exits at state point 18. A refrigerant expansion valve is used to reduce the pressure at state point 18 to the intermediate pressure, so state point 19 is in the two-phase region.

The vapor part of the flow at state point 19 combines with the superheated flow at state point 7, and is then condensed (in the condenser) to form saturated liquid water at state point 8. State point 8 is at the intermediate pressure, so another refrigerant expansion valve is used to decrease its pressure to state point 9, thus reducing the evaporation temperature to provide cooling to the building loops.

State point 9 is in the two-phase region, and the liquid part of the flow at state point 9 is heated by the building loop to saturated water vapor. This water vapor is combined with the vapor part of state point 9 to form state point 10. The water vapor state point 10 is absorbed by the strong solution from state point 6 to become the dilute solution in state point 1, thus completing the cycle.

All the temperature differences defined in 4.6.1 are shown in combination with the mixture composition state points described in this subsection in Figure 4.21. In the following subsection, these values are then used to calculate the thermodynamic properties at each of the state points, and to calculate the related exergy destruction.

4.6.3 Modeling Methodology

The components in the system are all defined as thermodynamic open systems, which allow mass to flow into/out of the system and exchange heat and/or work within the systems.

For any open system, the mass balance is,

$$\frac{\mathrm{dm}}{\mathrm{dt}}|_{\mathrm{cv}} = \sum \dot{m}_{\mathrm{i}} \cdot \sum \dot{m}_{\mathrm{o}} \tag{4.23}$$

Under steady state conditions,

$$\sum \dot{m}_i = \sum \dot{m}_o \tag{4.24}$$

and for a solute mass balance,

$$\sum \dot{m}_i x_i = \sum \dot{m}_e x_o \tag{4.25}$$

Equations (2.2), (2.7), and (4.23) - (4.25) form the mathematical models for each component in the system. They are shown below under steady state conditions.

Absorber

The mass balances for the absorber are:

$$m23 = m24$$
 (4.26)

$$m1 = m10 + m6$$
 (4.27)

while the solute mass balance for the absorber is:

$$m1*x1 = m6*x6$$
 (4.28)

The energy balances for the absorber are:

$$m10*h10 + h6*m6 - Q_{ABS} - m1*h1 = 0$$
 (4.29)

$$Q_{ABS} = m23*(h24 - h23)$$
 (4.30)

The exergy destruction in the absorber is:

$$ExDes_{ABS} = m23*[(h23 - h24) - T_{ambient}*(s23 - s24)] + m10*(h10 - T_{ambient}*s10) + m6*(h6) - T_{ambient}*s6) - m1*(h1 - T_{ambient}*s1)$$
(4.31)

while the exergy loss from the absorber is:

$$ExLoss_{ABS} = m23*[(h24 - h23) - T_{ambient}*(s24 - s23)]$$
(4.32)

Condenser

The mass balances for the condenser are:

$$m25 = m26$$
 (4.33)

$$m8 = m7 + m19$$
 (4.34)

The energy balances for the condenser are:

$$Q_{\text{CON}} = m7*h7 + m19*h19 - m8*h8$$
(4.35)

$$Q_{\text{CON}} = m25*(h26-h25)$$
 (4.36)

The exergy destruction in the condenser is:

$$ExDes_{CON} = m25*[(h25 - h26) - T_{ambient}*(s25 - s26)] + m7*(h7 - T_{ambient}*s7) + m19*(h19 - T_{ambient}*s19) - m8*(h8 - T_{ambient}*s8)$$
(4.37)

while the exergy loss from the condenser is:

$$ExLoss_{CON} = m25*[(h26-h25) - T_{ambient}*(s26-s25)]$$
(4.38)

Evaporator

The mass balances for the evaporator are:

$$m27 = m28$$
 (4.39)

$$m9 = m10$$
 (4.40)

The energy balances for the evaporator are:

$$Q_{EVP} = m9*(h10 - h9)$$
 (4.41)

$$Q_{EVP} = m27^*(h27 - h28) \tag{4.42}$$

The exergy destruction in the evaporator is:

$$ExDes_{EVP} = m27*[(h27 - h28) - T_{ambient}*(s27 - s28)] + m9*[(h9 - h10) - T_{ambient}*(s9-s10)]$$
(4.43)

while the useful exergy for the building cooling is:

$$Ex_{LOAD} = m27*[(h27 - h28) - T_{ambient}*(s27 - s28)]$$
(4.44)

High Temperature Regenerator

The mass balances for the high temperature regenerator are:

$$m21 = m22$$
 (4.45)

$$m13 = m14 + m17$$
 (4.46)

while the solute mass balance for the high temperature regenerator is:

$$m13*x13 = m14*x14 \tag{4.47}$$

The energy balances for the high temperature regenerator are:

$$Q_{\rm HTRG} = m17*h17 + m14*h14 - m13*h13$$
(4.48)

$$Q_{\rm HTRG} = m21^{*}(h21 - h22) \tag{4.49}$$

The exergy destruction in the high temperature regenerator is:

$$ExDes_{HTRG} = m21*[(h21 - h22) - T_{ambient}*(s21 - s22)] + m13*(h13 - T_{ambient}*s13) - m14*(h14 - T_{ambient}*s14) - m17*(h17 - T_{ambient}*s17)$$
(4.50)

The exergy supply from the parabolic trough solar collector is:

$$ExSupply_{HTRG} = m21*[(h21-h22) - T_{ambient}*(s21-s22)]$$
(4.51)

Low Temperature Regenerator

The mass balances for the low temperature regenerator are:

$$m17 = m18$$
 (4.52)

$$m3 + m16 = m4 + m11 + m7 \tag{4.53}$$

while the solute mass balance for the low temperature regenerator is:

$$m3^{*}x3 + m16^{*}x16 = m4^{*}x4 + m11^{*}x11$$
(4.54)

The energy balances for the low temperature regenerator are:

$$h3*m3 + m16*h16 + m17*h17$$

= h4*m4 + h11*m11 + m18*h18 + m7*h7 (4.55)

while the exergy destruction in the low temperature regenerator is:

$$ExDes_{LTRG} = m17*[(h17 - h18) - T_{ambient}*(s17 - s18))$$

+m3*(h3 - T_{ambient}*s3)+ m16*(h16 - T_{ambient}*s16)
-m11*(h11 - T_{ambient}*s11)- m4*(h4 - T_{ambient}*s4)
-m7*(h7 - T_{ambient}*s7) (4.56)

High Temperature Heat Exchanger

The mass balances for the high temperature heat exchanger are:

$$m12 = m13$$
 (4.57)

$$m14 = m15$$
 (4.58)

while the solute mass balances for the high temperature heat exchanger are:

$$m12*x12=m13*x13$$
 (4.59)

$$m14*x14 = m15*x15 \tag{4.60}$$

The energy balances for the high temperature heat exchanger is:

$$m11*(h13 - h12) = m14*(h14 - h15)$$
 (4.61)

while the exergy destruction in the high temperature heat exchanger is:

$$ExDes_{HTHX} = m12*[(h12 - h13) - T_{ambient}*(s12 - s13)] + m14*[(h14 - h15) - T_{ambient}*(s14 - s15)]$$
(4.62)

Low Temperature Heat Exchanger

The mass balances for the low temperature heat exchanger are:

$$m2 = m3$$
 (4.63)

$$m4 = m5$$
 (4.64)

while the solute mass balances for the low temperature heat exchanger are:

$$m2^*x2 = m3^*x3 \tag{4.65}$$

$$m4*x4 = m5*x5$$
 (4.66)

The energy balance for the low temperature heat exchanger is:

$$m1^{*}(h3 - h2) = m4^{*}(h4 - h5)$$
(4.67)

The exergy destruction in the low temperature heat exchanger is:

$$ExDes_{LTHX} = m2^{*}[(h2 - h3) - T_{ambient}^{*}(s2 - s3)] + m4^{*}[(h4 - h5) - T_{ambient}^{*}(s4 - s5)]$$
(4.68)

Lower Solution Expansion Valve

The mass balance for the lower solution expansion valve is:

$$m5 = m6$$
 (4.69)

while the solute mass balance for the lower solution expansion valve is:

$$m5*x5 = m6*x6$$
 (4.70)

The energy balance for the lower solution expansion valve is:

$$h5 = h6$$
 (4.71)

while the exergy destruction in the lower solution expansion valve is:

$$ExDes_{VALVE,L,S} = m5*T_{ambient}*(s6 - s5)$$
(4.72)

Upper Solution Expansion Valve

The mass balance for the upper solution expansion valve is:

$$m15 = m16$$
 (4.73)

while the solute mass balance for the upper solution expansion valve is:

$$m15*x15 = m16*x16 \tag{4.74}$$

The energy balance for the upper solution expansion valve is:

$$h15 = h16$$
 (4.75)

The exergy destruction in the upper solution expansion valve is:

$$ExDes_{VALVE,H,S} = m15*T_{ambient}*(s16 - s15)$$
 (4.76)

Lower Refrigerant Expansion Valve

The mass balance for the lower refrigerant expansion valve is:

$$m8 = m9$$
 (4.77)

while the solute mass balance for the lower refrigerant expansion valve is:

$$m8*x8 = m9*x9 \tag{4.78}$$

The energy balance for the lower refrigerant expansion valve is:

$$h8 = h9 \tag{4.79}$$

while the exergy destruction in the lower refrigerant expansion valve is:

$$ExDes_{VALVE,L,R} = m8*T_{ambient}*(s9 - s8)$$
(4.80)

Upper Refrigerant Expansion Valve

The mass balance for the upper refrigerant expansion valve is:

$$m18 = m19$$
 (4.81)

while the solute mass balance for the upper refrigerant expansion valve is:

$$m18*x18 = m19*x19 \tag{4.82}$$

The energy balance for the upper refrigerant expansion valve is:

$$h18 = h19$$
 (4.83)

while the exergy destruction in the upper refrigerant expansion valve is:

$$ExDes_{VALVE,H,R} = m18*T_{ambient}*(s19 - s18)$$
 (4.84)

Lower Pump

The mass balance for the lower pump is:

$$m1 = m2$$
 (4.85)

while the solute mass balance for the lower pump is:

$$m1^*x1 = m2^*x2 \tag{4.86}$$

The energy balance for the lower pump is:

$$Pump_l = m1*v1*(Pm-Pl)$$
 (4.87)

$$h2 = h1 + Pump_1/m1$$
 (4.88)

There is no exergy destruction for the lower pump when assuming 100% isentropic efficiency.

Upper Pump

The mass balance for the upper pump is:

$$m11 = m12$$
 (4.89)

The solute mass balance for the upper pump is:

$$m11*x11 = m12*x12 \tag{4.90}$$

The energy balance for the upper pump is:

$$Pump_{h} = m11*v11*(Ph-Pm)$$
(4.91)

$$h12 = h11 + Pump_h/m11$$
 (4.92)

There is also no exergy destruction for the upper pump when assuming 100% isentropic efficiency.

Overall System

Combining the equations from each component, the overall system exergy flow can be expressed as equation (4.93): the overall exergy supplied to the system (supplies to HTRG and two pumps) is equal to the summation of the useful exergy to the load, the exergy destruction in each component (ABS, CON, EVP, HTRG, LTRG, HTHX, LTHX, and VALVES), and the exergy losses to the environment (through ABS and CON):

$$\begin{split} ExSupply_{HTRG} + Pump_{h} &= ExDes_{ABS} + ExDes_{CON} + ExDes_{EVP} + ExDes_{HTRG} \\ &+ ExDes_{LTRG} + ExDes_{HTHX} + ExDes_{LTHX} + ExDes_{VALVE,L,S} \\ &+ ExDes_{VALVE,H,S} + ExDes_{VALVE,L,R} + ExDes_{VALVE,H,R} \\ &+ ExLoss_{ABS} + ExLoss_{CON} + Ex_{LOAD} \end{split}$$
(4.93)

Therefore, the system exergetic efficiency can be defined as

$$\eta_{ex,abs} = Ex_{LOAD} / (ExSupply_{HTRG} + Pump_l + Pump_h)$$
(4.94)

4.6.4 Results of the Real Cycle Exergy Analysis

For this complete cycle analysis, we assume that all of the defined temperature differences are set to 8 K, except in the evaporator, and that the two internal heat recovery heat exchanger efficiencies are equal to 50%. For the evaporator, the temperature difference is 3 K. There are two reasons that the evaporator has a lower temperature difference: 1) there is a refrigerant pump constantly spraying the liquid water onto the coil from building loop, and 2) to avoid ice formation in the evaporator. Combining equations (4.26) - (4.94), Table 4.4, and the new assumptions, the thermodynamic state point calculations are shown in Table 4.5. The programming code that produced these values is in the Appendix.

	Enthalpy	Mass	Pressure	Vapor	Temperature	Mass	Entropy
	[kJ/kg]	Flow Rate	[kPa]	Quality	[° C]	Fraction	[kJ/kgK]
		[kg/s]				[%]	
1	101.1	0.087	0.813	0	38	57.868	0.2179
2	101.1	0.087	6.63		38	57.868	0.2179
3	143.9	0.087	6.63		59.47	57.868	0.349
4	220.6	0.081	6.63	0	88.07	62.761	0.4777
5	174.2	0.081	6.63		63.03	62.761	0.3447
6	174.2	0.081	0.813	0.009	48.95	62.761	0.3461
7	2643.8	0.003	6.63	100	77.03	0	8.517
8	159.1	0.007	6.63	0	38	0	0.5455
9	159.1	0.007	0.813	0.057	4	0	0.5746
10	2507.9	0.007	0.813	1	4	0	9.049
11	178.8	0.05	6.63	0	77.03	57.868	0.4515
12	178.9	0.05	87.899		77.06	57.868	0.4517
13	247.2	0.05	87.899		111.32	57.868	0.6411
14	348.3	0.046	87.899	0	157	62.761	0.8114
15	274.2	0.046	87.899		117.03	62.761	0.6233
16	274.2	0.046	6.63	0.018	90.78	62.761	0.6235
17	2764.7	0.004	87.899	100	143.8	0	7.645
18	402.5	0.004	87.899	0	96.07	0	1.262
19	402.5	0.004	6.63	0.101	38	0	1.328
21	697.4	1	700.293		165		1.993
22	683	1	700.293		161.67		1.96
23	125.8	1	101.3		30		0.4365
24	148	1	101.3		35.33		0.5094

Table 4.5 Thermodynamic properties for state points in a real cycle

25	125.8	1	101.3	30	0.4365
26	133.9	1	101.3	31.96	0.4634
27	50.5	0.764	101.3	12	0.1804
28	29.5	0.764	101.3	7	0.1063

The chiller thermal COP is defined as the ratio of the cooling energy from the evaporator over the thermal energy provided in the high temperature regenerator.

$$COP_{thermal} = \frac{m27*(h27-h28)}{m21*(h21-h22)}$$
(4.95)

If the pumping electricity consumptions are also considered, the chiller overall COP can be defined as the ratio of the cooling energy from the evaporator over the summation of the thermal energy provided in the high temperature regenerator and pumping electricity consumptions.

$$COP_{overall} = \frac{m27*(h27-h28)}{m21*(h21-h22)+m1*(h2-h1)+m11*(h12-h11)}$$
(4.96)

From the values found in the analysis, above, the $\text{COP}_{\text{thermal}}$ is found to be 1.106 and the $\text{COP}_{\text{overall}}$ is 1.069. These values are validated with experimental results for this absorption chiller, a Broad BCT 16, found in Yin (2006). In Yin (2006), the same type of chiller was tested and the $\text{COP}_{\text{thermal}}$ was found to be 1.11, while $\text{COP}_{\text{overall}}$ was 1.04.

The total solar thermal energy supplied to the absorption chiller is $Q_{\text{thermal}} = m21*(h21-h22) = 14.467$ kW, and the cooling energy from the evaporator is $Q_{\text{cooling}} = m27*(h27-h28) = 16$ kW.

For the exergy analysis, the total exergy supplied from the solar loop is: ExSupply = m21*(h21 - h22 - Tambient*(s21 - s22)) = 4.426 kW,

the total exergy destruction within the absorption chiller system is: ExDes=ExSupply-Exloss-Exload=3.039 kW,

the exergy loss to the environment is:

Exloss = m25*(h26 - h25 - Tambient*(s26 - s25)) + m23*(h24 - h23 - Tambient*(s24 - s23)) = 0.235 kW,

and the exergy supplied for building cooling is:

Exload = m21*(h21 - h22 - Tambient*(s21 - s22)) = 1.152 kW.



Figure 4.22 Exergy destruction breakdown in a real cycle

Based on this analysis, it can be observed that more than 74% of the available solar thermal exergy is wasted within the double stage absorption chiller. The ideal COP with no entropy generation within the processes equals 1.069 divided by 26.03%, which is 4.1. This is close to the ideal COP of 4.09, as defined in Subsection 4.5.2, given the same solar loop, building loop and ambient loop conditions. The exergy destruction breakdown in the real cycle is shown in Figure 4.22. Together, the absorber, high temperature regenerator, and low temperature regenerator are responsible for more than 70% of the overall exergy destruction in the system. In order to improve the system COP, based on the exergy destruction values, research should first focus on the absorber, followed by (in order) the high temperature regenerator, low temperature regenerator, evaporator, high

temperature heat exchanger, low temperature heat exchanger, condenser, and expansion valves. Figure 4.23 illustrated both the energy and exergy flows through the cycle.



Figure 4.23 Energy and exergy analysis for the real cycle

4.7 Summary

To sum up, from the exergy analysis for the parabolic trough solar collector, one can find that as well as determining the ideal operating conditions, it is equally important to perform energy and exergy studies for the parabolic trough solar collector design. As shown in this analysis, the operating temperature for the receiver can only be obtained from the energy analysis and/or experiments, and the receiver temperature together with collector inlet and outlet temperatures are vital to the exergy analysis. The exergy analysis looks in more detail into the internal exergy destruction/loss, and thus it can aid in system improvement and optimum operation. Additionally, by incorporating the friction exergy loss, the exergy analysis can be used to set the limit for the highest desired mass flow rate for real operation. The exergy analysis for the double stage absorption chiller quantitatively evaluates the exergy destruction in each component in the system, which aids in determining that the proper order of research focus for engineering improvements to the system should be, in descending order, the absorber, high temperature regenerator, and low temperature regenerator, since they are responsible for more than 70% of the overall exergy destruction.
Chapter 5: Advanced Exergy Analysis for a Double Stage Absorption Chiller

In this chapter, an advanced exergy analysis is applied to the double stage absorption chiller. The same temperature conditions are applied as in the last chapter, and the exergy destruction is divided into unavoidable/avoidable parts (UN/AV) and endogenous/exogenous parts. (EN/EX). Quantifying the avoidable exergy destruction will help identify the potential for improvement for each sub-component, while, at the same time, quantifying the endogenous exergy destruction will create a clearer picture on whether or not the focus should be on a given component.

5.1 Avoidable/Unavoidable Energy Destruction Analysis

In order to simplify the analysis, one must identify which components have no realistic potential for improvement in the near future. For example, in the double stage absorption chiller, the entropy generation (exergy destruction) within the expansion valves is widely known to be unrecoverable. Therefore, in this section, the analyses for the expansion valves will be omitted. Additionally, it can be observed that the strong solution flows after the two internal heat exchangers (state points 5 and 15) are close to the LiBr crystallization line, shown in Figure 5.1. To avoid crystallization, a 50% efficiency is kept and used for the avoidable/unavoidable exergy destruction analysis for both the high and low temperature heat exchangers.



Figure 5.1 No improvement potential for HTHX/LTHX to avoid crystallization

As discussed in Chapter 2, the thermodynamic cycles that are considered to have unavoidable exergy destruction are the cycles in which all of the components are working at their minimum irreversibility. To approach that ideal, it is assumed that all of the temperature differences defined in Subsection 4.6.1 are set to a value ten times smaller than the temperature differences used in a real cycle calculation. Specifically, except for the evaporator, a 0.8 K temperature difference is specified for all of the components. For the evaporator, the temperature difference is 0.3 K. This newly defined temperature difference is called Δ Tmin to indicate the technology limit (in the near term) for the lowest possible temperature difference (and exergy destruction) to still transfer heat. Figure 5.2 shows the thermodynamic cycle state points with the unavoidable exergy destruction temperature differences. It is observed that the cycles are similar to the real cycles, but instead of using Δ T, they use Δ Tmin. The thermodynamic properties for the cycles with unavoidable exergy destruction are shown in Table 5.1. The coding for this analysis is in Appendix.



Figure 5.2 Thermodynamic representation for cycles in a double stage absorption chiller with unavoidable exergy destruction

	Enthalpy	thalpy Mass Pressure Vapor Te		Temperature	Mass	Entropy	
	[kJ/kg]	Flow Rate	[kPa]	Quality	[° C]	Fraction	[kJ/kgK]
		[kg/s]				[%]	
1	66.4	0.03	0.982	0	30.8	52.292	0.2017
2	66.4	0.03	4.444		30.8	52.292	0.2017
3	106.4	0.03	4.444		49.54	52.292	0.328
4	246.2	0.024	4.444	0	89.83	67.082	0.4608
5	194.8	0.024	4.444		60.32	67.082	0.3148
6	194.8	0.024	0.982	0	59.96	67.082	0.3148
7	2607.9	0.003	4.444	100	57.85	0	8.596
8	129	0.007	4.444	0	30.8	0	0.4476
9	129	0.007	0.982	0.041	6.7	0	0.4622
10	2512.8	0.007	0.982	1	6.7	0	8.98
11	124.2	0.017	4.444	0 57.85		52.292	0.3823
12	124.3	0.017	71.809		57.87	52.292	0.3824
13	196.4	0.017	71.809		91.49	52.292	0.5918
14	375.7	0.013	71.809	0	164.2	67.082	0.7921
15	283.1	0.013	71.809		111.04	67.082	0.5599
16	283.1	0.013	4.444	0.012	91.84	67.082	0.5621
17	2729.2	0.004	71.809	100	125.11	0	7.65
18	379.6	0.004	71.809	0	90.63	0	1.199
19	379.6	0.004	4.444	0.103	30.8	0	1.272
21	697.4	1	700.293		165		1.993
22	685.6	1	700.293		162.29		1.966
23	125.8	1	101.3		30		0.4365
24	145.2	1	101.3		34.65		0.5002
25	125.8	1	101.3		30		0.4365
26	134.1	1	101.3		31.99		0.4639
27	50.5	0.764	101.3		12		0.1804
28	29.5	0.764	101.3		7		0.1063

Table 5.1 Thermodynamic properties for state points in cycles with unavoidable exergy destruction



Figure 5.3 Exergy destruction breakdown into avoidable/unavoidable parts for a double stage absorption chiller

The resulting calculated exergy destruction in each component is the unavoidable exergy destruction. By comparing these values with Figure 4.22, the avoidable contribution to the exergy destruction can be obtained by subtracting the unavoidable exergy destruction from the overall exergy destruction. In Figure 5.3, the avoidable/unavoidable breakdown is shown.

In Figure 5.3, the overall length of each bar represents the exergy destruction in each component, which is identical to the previous Figure 4.22. The red part of the exergy destruction indicates the avoidable exergy destruction. (As stated previously, all of the exergy destruction amounts from the four expansion valves are unavoidable.) Based on current technology (the minimum heat transfer temperature difference), all of the avoidable parts of the exergy destruction could be recovered. If there was only unavoidable exergy destruction in the system, the exergy destruction in the double stage absorption chiller would be reduced to 2.33 kW. Given the same amount of exergy provided for cooling and lost through the condenser, the system exergetic efficiency can therefore be improved to 30.99%. This exergetic efficiency increase indicates an increase

in the system COP. The new COP (with pumping work) after recovering the avoidable exergy destruction is equal to the multiplication of the theoretical upper COP limit and the system exergetic efficiency, and is equal to 1.27, which is an increase of 19%. (The new COP without pump work is 1.38, an increase of 25%.)

Compared to the conventional exergy analysis, this breakdown provides further information on what the research focus priorities should be for improving the system COP. Rather than focusing on the total exergy destruction, engineers should instead focus on the quantities of the avoidable exergy destruction. Therefore, research should first focus on the condenser, followed by the evaporator, the low temperature heat exchanger, the high temperature heat exchanger, the absorber, the low temperature regenerator, and finally the high temperature regenerator. Of course, none of these components operate completely independently, so it is also important to understand how improving each of these components will affect the other components. To do this, in the following section, we will further refine the avoidable/unavoidable analysis by introducing the concept of endogenous/exogenous exergy destruction.

5.2 Endogenous/Exogenous Exergy Destruction Analysis

The endogenous exergy destruction is the exergy destruction for a given component that is operating irreversibly, while all other components are operating ideally. Conversely, the exogenous exergy destruction is the exergy destruction contributed from all other components operating irreversibly to a given component that is operating ideally. As discussed previously, all of the expansion valves and the two internal heat exchangers have no potential for improvement due to technology limitations and solution crystallization. In order to calculate the endogenous exergy destruction within other components, one has to define the ideal working conditions for the expansion valves and internal heat exchangers.

The ideal expansion valve is an expander that reduces the pressure of a fluid at a constant entropy rather than a constant enthalpy, as shown in Figure 5.4. Both the pressure and enthalpy of the fluid will be reduced at the outlet of the ideal expander. This ideal valve does not exist for the LiBr solution, because in order to reduce the pressure and keep the same entropy, the solution would undergo crystallization (a negative vapor fraction). However, since the endogenous/exogenous exergy destruction analysis uses the components' ideal operation, this crystallization effect is ignored for these theoretical calculations.



Figure 5.4 Real and Ideal expansion valves

Ideal high and low temperature heat exchangers would operate with 100% efficiency. However, this means that the strong solution at the two heat exchanger outlets (state points 5 and 15) would have the same temperature as state points 2 and 12, respectively. Crystallization would then occur at state points 5 and 15. As above, though, for the endogenous/exogenous analysis, the practical effects of this crystallization are neglected.

The ideal operation of the condenser also requires special considerations. From Figure 5.5, it can be seen that, for ideal operation, $\Delta T_{CON} = T8 - T25 = 0$. However, based on the energy balance, since the condenser will transfer the heat to an environmental fluid, T26 will always be higher than T25. This will then result in a negative log mean temperature difference (LMTD), which is physically impossible, as shown in Figure 5.5. Therefore, in this analysis, for the ideal operation of the condenser, ΔT_{CON} does not equal zero, but instead equals its minimum temperature difference based on the limitations of the technology (0.8 K).



Figure 5.5 Condenser with minimum temperature difference

5.2.1 Endogenous/Exogenous Exergy Destruction for the Absorber

The state points for the thermodynamic cycles for calculating the absorber's endogenous exergy destruction are shown in Figure 5.6. In this calculation, $\Delta T_{ABS} = 8$ K, $\Delta T_{CON, MIN} = 0.8$ K, $\Delta T_{EVP} = 0$ (T28 = T9), $\Delta T_{HTRG} = 0$ (T21=T14), $\Delta T_{LTRG} = 0$ (T18=T4), the LTHX efficiency is 100% (T2=T5), the HTHX efficiency is 100% (T12=T15), and the expansion valves operate ideally (S15 = S16, H15 > H16). The calculation results for these state points are shown in Table 5.2, and the programming code is in the Appendix.



Figure 5.6 Thermodynamic cycles for endogenous exergy destruction in the absorber

	Enthalpy	Mass	Pressure	Vapor	Temperature	Mass	Entropy
	[kJ/kg]	Flow Rate	[kPa]	Quality	[°C]	Fraction	[kJ/kgK]
		[kg/s]		-	[-]	[%]	-
1	02.5	0.04	1.002	0	20	56.005	0.2264
1	95.5	0.04	1.002	0	38	56.085	0.2204
2	95.5	0.04	4.444		30 75.16	56.085	0.2204
3	109.3	0.04	4.444	4.444 /5.16 56.08		50.085	0.4552
4	248.9	0.033	4.444	0	90.51	67.359	0.4625
5	157.8	0.033	4.444	0.010	38	67.359	0.196
6	157	0.033	1.002	-0.013	59.05	67.359	0.196
7	2621.6	0.003	4.444	100	65.1	0	8.637
8	129	0.007	4.444	0	30.8	0	0.4476
9	125	0.007	1.002	0.038	7	0	0.4476
10	2513.4	0.007	1.002	1	7	0	8.973
11	148.8	0.022	4.444	0	0 65.1 56.08		0.3952
12	148.8	0.022	71.495		65.12	56.085	0.3953
13	293.1	0.022	71.495		135.62	56.085	0.7903
14	378.1	0.019	71.495	0	165	67.359	0.7921
15	204.9	0.019	71.495		65.12	67.359	0.3383
16	203.4	0.019	4.444	-0.015	88.1	67.359	0.3383
17	2745.2	0.004	71.495	100	133.22	0	7.692
18	379.1	0.004	71.495	0	90.51	0	1.198
19	357.2	0.004	4.444	0.094	30.8	0	1.198
21	697.4	1	700.293		165		1.993
22	686.7	1	700.293		162.53		1.968
23	125.8	1	101.3		30		0.4365
24	144.1	1	101.3		34.38		0.4965
25	125.8	1	101.3		30		0.4365
26	134	1	101.3		31.97		0.4636
27	50.5	0.764	101.3		12		0.1804
28	29.5	0.764	101.3		7		0.1063

Table 5.2 Thermodynamic properties for state points in cycles for endogenous exergy destruction in the absorber

As shown in Table 5.2, the vapor qualities for state points 6 and 16 are negative leaving the ideal expansion valves, which is physically unrealistic. However, since this does not affect the overall energy balance calculation in the absorber and low temperature regenerator, the negative values are kept in the analysis.

The endogenous exergy destruction for the absorber is:

ENExDesABS = m23*(h23 - h24 - Tambient*(s23 - s24)) + m10*(h10 - Tambient*s10) + m6*(h6 - Tambient*s6) - m1*(h1 - Tambient*s1) = 0.7395 kW.

The exogenous exergy destruction is the overall exergy destruction in the absorber minus the endogenous exergy destruction, and is equal to -0.0154 kW. This negative value requires further clarification. The exergy destruction is related to the entropy generation in the system, and since entropy generation in a natural process is always positive, the exergy destruction must also be positive. Here, the negative exogenous exergy destruction seems contradictory to this fundamental theory. The consequences of this negative value will be discussed in Subsection5.2.8.

5.2.2 Endogenous/Exogenous Exergy Destruction for the Condenser

The state points for the thermodynamic cycles for the calculation of the condenser's endogenous exergy destruction are shown in Figure 5.7. In this calculation, $\Delta T_{CON} = 8$ K, $\Delta T_{ABS} = 0$ K (T1 = T23), $\Delta T_{EVP} = 0$ (T28 = T9), $\Delta T_{HTRG} = 0$ (T21=T14), $\Delta T_{LTRG} = 0$ (T18=T4), the LTHX efficiency is 100% (T2=T5), the HTHX efficiency is 100% (T12=T15), and the expansion valves operate ideally. The calculation results are shown in Table 5.3, and the programming code is in the Appendix. From these state points, we can find that the endogenous exergy destruction for the condenser is:

ENExDesCON = m25*(h25 - h26 - Tambient*(s25 - s26)) + m7*(h7 - Tambient*s7) + m19*(h19 - Tambient*s19) - m8*(h8 - Tambient*s8) = 0.1896 kW.

The exogenous exergy destruction is therefore 0.0067 kW (= 0.1963 kW - 0.1896 kW).



Figure 5.7 Thermodynamic cycles for endogenous exergy destruction in the condenser

	Enthalpy	thalpy Mass Pressure Vapor Temperature		Temperature	Mass	Entropy	
	[kJ/kg]	Flow Rate	[kPa]	Quality	[° C]	Fraction	[kJ/kgK]
		[kg/s]				[%]	
1	63.1	0.032	1.002	0.000	30.00	51.619	0.2008
2	63.1	0.032	6.630		30.00	51.619	0.2009
3	154.0	0.032	6.630		72.13	51.619	0.4803
4	248.0	0.025	6.630	0.000	95.17	65.665	0.495
5	132.4	0.025	6.630		30.00	65.665	0.1559
6	131.6	0.025	1.002	-0.016	55.18	65.665	0.1559
7	2620.5	0.003	6.630	100.000 64.72		0.000	8.449
8	159.1	0.007	6.630	0.000	38.00	0.000	0.5455
9	152.4	0.007	1.002	0.050	7.00	0.000	0.5455
10	2513.4	0.007	1.002	1.000	7.00	0.000	8.973
11	137.9	0.018	6.630	0.000 64.72 51		51.619	0.4331
12	138.0	0.018	85.050		64.74	51.619	0.4332
13	277.9	0.018	85.050		129.16	51.619	0.8184
14	372.0	0.014	85.050	0.000	165.00	65.665	0.8122
15	194.0	0.014	85.050		64.74	65.665	0.3427
16	191.2	0.014	6.630	-0.019	92.19	65.665	0.3427
17	2735.3	0.004	85.050	100.000	128.81	0.000	7.588
18	398.7	0.004	85.050	0.000	95.17	0.000	1.252
19	378.9	0.004	6.630	0.091	38.00	0.000	1.252
21	697.4	1.000	700.293		165.00		1.993
22	686.8	1.000	700.293		162.55		1.968
23	125.8	1.000	101.300		30.00		0.4365
24	144.1	1.000	101.300		34.38		0.4965
25	125.8	1.000	101.300		30.00		0.4365
26	133.9	1.000	101.300		31.95		0.4633
27	50.5	0.764	101.300		12.00		0.1804
28	29.5	0.764	101.300		7.00		0.1063

Table 5.3 Thermodynamic properties for state points in cycles for endogenous exergy destruction in the condenser

5.2.3 Endogenous/Exogenous Exergy Destruction for the Evaporator

The state points for the thermodynamic cycles for the evaporator's endogenous exergy destruction are shown in Figure 5.8. In this calculation, $\Delta T_{EVP} = 3 \text{ K}$, $\Delta T_{CON,MIN} = 0.8 \text{ K}$, $\Delta T_{ABS} = 0$ (T1 = T23), $\Delta T_{HTRG} = 0$ (T21=T14), $\Delta T_{LTRG} = 0$ (T18=T4), the LTHX efficiency is 100% (T2=T5), the HTHX efficiency is 100% (T12=T15), and the expansion valves operate ideally. The calculation results are shown in Table 5.4, and the programming code is in the Appendix. The endogenous exergy destruction for the evaporator is then:

ENExDesEVP = m27*(h27 - h28 - Tambient*(s27 - s28))+ m9*(h9 - h10 - Tambient*(s9 - s10)) = 0.34 kW

which is the same as the overall exergy destruction in evaporator. Therefore, there is no exogenous exergy destruction in the evaporator.

The reason for this lack of exogenous exergy destruction is that the temperatures and pressures in both ends of the building loop are fixed (T28 = 7°C and T27 = 12°C), which also fixes the values of the specific enthalpy and specific entropy. The mass flow rate for the building loop is also fixed, based on the specified 16 kW cooling capacity. Furthermore, as discussed previously, the flow at state point 10 is set to be saturated water vapor, and its properties are strongly related to the two-phase mixture at state point 9 (both state points are at the same pressure and temperature). Therefore, the exergy destruction in the evaporator is only a function of the conditions at state point 9, with the same equations and same state points for the calculations of both the exergy destruction and the endogenous exergy destruction in the evaporator.

This analysis of the evaporator illustrates the usefulness of an endogenous/exogenous exergy destruction breakdown. In assessing which component improvements will best improve the overall efficiency of the system, a researcher can be assured in his focus on the evaporator without considering the irreversibility from other components, since there is no exogenous exergy destruction in the evaporator.



Figure 5.8 Thermodynamic cycles for endogenous exergy destruction in the evaporator

	Enthalpy	Mass	Pressure	Vapor	Temperature	Mass	Entropy
	[kJ/kg]	Flow Rate	[kPa]	Quality	[° C]	Fraction	[kJ/kgK]
		[kg/s]			[-]	[%]	
1	68.2	0.033	0.813	0.000	30.00	53.571	0.188
2	68.2	0.033	4.444		30.00	53.571	0.188
3	151.6	0.033	4.444		69.61	53.571	0.4447
4	248.9	0.026	4.444	0.000	90.51	67.359	0.4625
5	144.0	0.026	4.444		30.00	67.359	0.1539
6	143.4	0.026	0.813	-0.016	55.05	67.359	0.1539
7	2612.4	0.003	4.444	100.000	60.21	0.000	8.609
8	129.0	0.007	4.444	0.000	30.80	0.000	0.4476
9	123.9	0.007	0.813	0.043	4.00	0.000	0.4476
10	2507.9	0.007	0.813	1.000	4.00	0.000	9.049
11	131.8	0.018	4.444	0.000	60.21	53.571	0.3858
12	131.9	0.018	71.495		60.23	53.571	0.386
13	276.4	0.018	71.495		128.51	53.571	0.7867
14	378.1	0.015	71.495	0.000	165.00	67.359	0.7921
15	196.4	0.015	71.495		60.23	67.359	0.3134
16	194.2	0.015	4.444	-0.018	87.63	67.359	0.3134
17	2734.2	0.004	71.495	100.000	127.66	0.000	7.665
18	379.1	0.004	71.495	0.000	90.51	0.000	1.198
19	357.2	0.004	4.444	0.094	30.80	0.000	1.198
21	697.4	1.000	700.293		165.00		1.993
22	686.7	1.000	700.293		162.54		1.968
23	125.8	1.000	101.300		30.00		0.4365
24	144.1	1.000	101.300		34.38		0.4965
25	125.8	1.000	101.300		30.00		0.4365
26	134.0	1.000	101.300		31.96		0.4635
27	50.5	0.764	101.300		12.00		0.1804
28	29.5	0.764	101.300		7.00		0.1063

 Table 5.4 Thermodynamic properties for state points in cycles for endogenous exergy destruction in the evaporator

5.2.4 Endogenous/Exogenous Exergy Destruction for the High Temperature Regenerator

The state points for the thermodynamic cycles for the calculation of the high temperature regenerator's endogenous exergy destruction are shown in Figure 5.9. In this calculation, $\Delta T_{HTRG} = 8 \text{ K}$, $\Delta T_{CON,MIN} = 0.8 \text{ K}$, $\Delta T_{ABS} = 0$ (T1 = T23), $\Delta T_{EVP} = 0$ (T9 =T28), $\Delta T_{LTRG} = 0$ (T18=T4), the LTHX efficiency is 100% (T2=T5), the HTHX efficiency is 100% (T12=T15), and the expansion valves operate ideally. The calculation results are shown in Table 5.5, and the programming code is in the Appendix. The endogenous exergy destruction for the high temperature regenerator is:

ENExDesHTRG = m21*(h21 - h22 - Tambient*(s21 - s22)) + m13*(h13 - Tambient*s13) - m14*(h14 - Tambient*s14)- m17*(h17 - Tambient*s17) = 0.5329 kW.

Therefore, the exogenous exergy destruction is 0.0694 kW (= 0.6023 kW - 0.5329 kW).



temperature regenerator

	Enthalpy	Mass	Pressure	Vapor	Temperature	Mass	Entropy
	[kJ/kg]	Flow Rate	[kPa]	Quality	[° C]	Fraction	[kJ/kgK]
		[kg/s]			L - J	[%]	
1	63.1	0.031	1.002	0.000	30.00	51.619	0.2008
2	63.1	0.031	4.444		30.00	51.619	0.2009
3	142.2	0.031	4.444		66.68	51.619	0.4457
4	236.7	0.024	4.444	0.000	87.44	66.107	0.4548
5	135.5	0.024	4.444		30.00	66.107	0.1556
6	134.6	0.024	1.002	-0.017	56.03	66.107	0.1556
7	2605.6	0.003	4.444	100.000	56.64	0.000	8.589
8	129.0	0.007	4.444	0.000	30.80	0.000	0.4476
9	125.0	0.007	1.002	0.038	7.00	0.000	0.4476
10	2513.4	0.007	1.002	1.000	7.00	0.000	8.973
11	120.5	0.017	4.444	0.000	56.64	51.619	0.3807
12	120.5	0.017	63.574		56.66	51.619	0.3808
13	258.7	0.017	63.574		120.38	51.619	0.7691
14	359.4	0.013	63.574	0.000	157.00	66.107	0.7729
15	182.4	0.013	63.574		56.66	66.107	0.299
16	179.9	0.013	4.444	-0.019	84.55	66.107	0.299
17	2720.2	0.004	63.574	100.000	120.16	0.000	7.683
18	366.2	0.004	63.574	0.000	87.44	0.000	1.163
19	346.3	0.004	4.444	0.090	30.80	0.000	1.163
21	697.4	1.000	700.293		165.00		1.993
22	687.0	1.000	700.293		162.59		1.969
23	125.8	1.000	101.300		30.00		0.4365
24	143.9	1.000	101.300		34.33		0.4958
25	125.8	1.000	101.300		30.00		0.4365
26	134.0	1.000	101.300		31.96		0.4635
27	50.5	0.764	101.300		12.00		0.1804
28	29.5	0.764	101.300		7.00		0.1063

Table 5.5 Thermodynamic properties for state points in cycles for endogenous exergy destruction in the high temperature regenerator

5.2.5 Endogenous/Exogenous Exergy Destruction for the Low Temperature Regenerator

The state points for the thermodynamic cycles for calculating the low temperature regenerator's endogenous exergy destruction are shown in Figure 5.10. In this calculation, $\Delta T_{LTRG} = 8 \text{ K}$, $\Delta T_{CON,MIN} = 0.8 \text{ K}$, $\Delta T_{ABS} = 0$ (T1 = T23), $\Delta T_{EVP} = 0$ (T9 =T28), $\Delta T_{HTRG} = 0$ (T14 =T21), the LTHX efficiency is 100% (T2=T5), the HTHX efficiency is 100% (T12=T15), and the expansion valves operate ideally. The calculation results are shown in Table 5.6, and the programming code is in the Appendix. The endogenous exergy destruction for the low temperature regenerator is:

ENExDesLTRG = m17*(h17 - h18 - Tambient*(s17 - s18))+ m3*(h3 - Tambient*s3)+ m16*(h16 - Tambient*s16)- m11*(h11- Tambient*s11)- m4*(h4 -Tambient*s4)- m7*(h7 - Tambient*s7) = 0.4961 kW.

The exogenous exergy destruction is then 0.0399 kW (= 0.536 kW - 0.4961 kW).



temperature regenerator

	Enthalpy	Mass	Pressure	Vapor	Temperature	Mass	Entropy
	[kJ/kg]	Flow Rate	[kPa]	Quality	[° C]	Fraction	[kJ/kgK]
		[kg/s]				[%]	
1	63.1	0.031	1.002	0.000	30.00	51.619	0.2008
2	63.1	0.031	4.444		30.00	51.619	0.2009
3	141.9	0.031	4.444		66.54	51.619	0.4448
4	233.9	0.024	4.444	0.000	86.75	65.820	0.453
5	133.5	0.024	4.444		30.00	65.820	0.1558
6	132.6	0.024	1.002	-0.016	55.48	65.820	0.1558
7	2605.6	0.003	4.444	100.000 56.64 0.0		0.000	8.589
8	129.0	0.007	4.444	0.000	30.80	0.000	0.4476
9	125.0	0.007	1.002	0.038	7.00	0.000	0.4476
10	2513.4	0.007	1.002	1.000	7.00	0.000	8.973
11	120.5	0.017	4.444	0.000	0.000 56.64 51.61		0.3807
12	120.5	0.017	83.743		56.67	51.619	0.3809
13	271.0	0.017	83.743		126.02	51.619	0.8008
14	372.5	0.014	83.743	0.000	165.00	65.820	0.8104
15	180.7	0.014	83.743		56.67	65.820	0.2999
16	178.2	0.014	4.444	-0.018	83.93	65.820	0.2999
17	2734.5	0.004	83.743	100.000	128.34	0.000	7.593
18	396.9	0.004	83.743	0.000	94.75	0.000	1.247
19	372.0	0.004	4.444	0.100	30.80	0.000	1.247
21	697.4	1.000	700.293		165.00		1.993
22	686.9	1.000	700.293		162.57		1.969
23	125.8	1.000	101.300		30.00		0.4365
24	143.9	1.000	101.300		34.33		0.4958
25	125.8	1.000	101.300		30.00		0.4365
26	134.0	1.000	101.300		31.98		0.4637
27	50.5	0.764	101.300		12.00		0.1804
28	29.5	0.764	101.300		7.00		0.1063

Table 5.6 Thermodynamic properties for state points in cycles for endogenous exergy destruction in the low temperature regenerator

5.2.6 Endogenous/Exogenous Exergy Destruction for the High Temperature Heat Exchanger

The state points for the thermodynamic cycles for the high temperature heat exchanger's endogenous exergy destruction are shown in Figure 5.11. In this calculation, the HTHX has an efficiency of 50%, $\Delta T_{CON,MIN} = 0.8 \text{ K}$, $\Delta T_{ABS} = 0$ (T1 = T23), $\Delta T_{EVP} = 0$ (T9 =T28), $\Delta T_{HTRG} = 0$ (T14 =T21), $\Delta T_{LTRG} = 0$ (T4 = T18), the LTHX efficiency is 100% (T2=T5), and the expansion valves operate ideally. The calculation results are shown in Table 5.7, and the programming code is in the Appendix. The calculated endogenous exergy destruction for the high temperature heat exchanger is:

ENExDesHTHX = m12*(h12 - h13 - Tambient*(s12 - s13)) + m14*(h14 - h15 - Tambient*(s14 - s15)) = 0.1339 kW,

So the exogenous exergy destruction is 0.1077 kW (= 0.2416 kW - 0.1339 kW).



Figure 5.11 Thermodynamic cycles for endogenous exergy destruction in the high temperature heat exchanger

	Enthalpy	Mass	Pressure	Vapor	Temperature	Mass	Entropy
	[kJ/kg]	Flow Rate	[kPa]	Quality	[° C]	Fraction	[kJ/kgK]
		[kg/s]				[%]	
1	63.1	0.029	1.002	0.000	30.00	51.619	0.2008
2	63.1	0.029	4.444		30.00	51.619	0.2009
3	143.5	0.029	4.444		67.31	51.619	0.4497
4	248.9	0.022	4.444	0.000	90.51	67.359	0.4625
5	144.0	0.022	4.444		30.00	67.359	0.1539
6	142.9	0.022	1.002	-0.018	58.42	67.359	0.1539
7	2605.6	0.003	4.444	100.000 56.64		0.000	8.589
8	129.0	0.007	4.444	0.000	30.80	0.000	0.4476
9	125.0	0.007	1.002	0.038	7.00	0.000	0.4476
10	2513.4	0.007	1.002	1.000	7.00	0.000	8.973
11	120.5	0.015	4.444	0.000	0.000 56.64 51.0		0.3807
12	120.5	0.015	71.495		56.66	51.619	0.3809
13	192.5	0.015	71.495		89.95	51.619	0.5906
14	378.1	0.011	71.495	0.000	165.00	67.359	0.7921
15	284.1	0.011	71.495		110.83	67.359	0.5569
16	283.7	0.011	4.444	0.011	92.42	67.359	0.5569
17	2726.2	0.004	71.495	100.000	123.60	0.000	7.645
18	379.1	0.004	71.495	0.000	90.51	0.000	1.198
19	357.2	0.004	4.444	0.094	30.80	0.000	1.198
21	697.4	1.000	700.293		165.00		1.993
22	686.4	1.000	700.293		162.47		1.967
23	125.8	1.000	101.300		30.00		0.4365
24	143.9	1.000	101.300		34.34		0.496
25	125.8	1.000	101.300		30.00		0.4365
26	134.5	1.000	101.300		32.08		0.4651
27	50.5	0.764	101.300		12.00		0.1804
28	29.5	0.764	101.300		7.00		0.1063

Table 5.7 Thermodynamic properties for state points in cycles for endogenous exergy destruction in the high temperature heat exchanger

5.2.7 Endogenous/Exogenous Exergy Destruction for the Low Temperature Heat Exchanger

The state points for the thermodynamic cycles for the low temperature heat exchanger's endogenous exergy destruction are shown in Figure 5.12. In this calculation, the LTHX has an efficiency of 50%, $\Delta T_{CON,MIN} = 0.8 \text{ K}$, $\Delta T_{ABS} = 0$ (T1 = T23), $\Delta T_{EVP} = 0$ (T9 =T28), $\Delta T_{HTRG} = 0$ (T14 =T21), $\Delta T_{LTRG} = 0$ (T4 = T18), the HTHX efficiency is 100% (T12=T15), and the expansion valves operate ideally. The calculation results are shown in Table 5.8, and the programming code is in the Appendix. The calculated endogenous exergy destruction for the low temperature heat exchanger is:

ENExDesLTHX = m2*(h2 - h3 - Tambient*(s2 - s3)) + m4*(h4 - h5 - Tambient*(s4 - s5))= 0.1137 kW,

So the exogenous exergy destruction is 0.1094 kW (= 0.2231 kW - 0.1137 kW).





	Enthalpy	Mass	Pressure	Vapor	Temperature	Mass	Entropy
	[kJ/kg]	Flow Rate	[kPa]	Quality	[° C]	Fraction	[kJ/kgK]
	_	[kg/s]		-	[-]	[%]	
1	63.1	0.029	1.002	0.000	30.00	51.619	0.2008
2	63.1	0.029	4.444		30.00	51.619	0.2009
3	103.3	0.029	4.444		48.68	51.619	0.3281
4	248.9	0.022	4.444	0.000	90.51	67.359	0.4625
5	196.4	0.022	4.444		60.26	67.359	0.3135
6	196.4	0.022	1.002	-0.000	60.83	67.359	0.3135
7	2605.6	0.003	4.444	100.000	56.64	0.000	8.589
8	129.0	0.007	4.444	0.000	30.80	0.000	0.4476
9	125.0	0.007	1.002	0.038	7.00	0.000	0.4476
10	2513.4	0.007	1.002	1.000	7.00	0.000	8.973
11	120.5	0.017	4.444	0.000	56.64	51.619	0.3807
12	120.5	0.017	71.495		56.66	51.619	0.3809
13	264.5	0.017	71.495		123.05	51.619	0.7842
14	378.1	0.013	71.495	0.000	165.00	67.359	0.7921
15	190.2	0.013	71.495		56.66	67.359	0.295
16	187.5	0.013	4.444	-0.020	87.29	67.359	0.295
17	2726.2	0.004	71.495	100.000	123.60	0.000	7.645
18	379.1	0.004	71.495	0.000	90.51	0.000	1.198
19	357.2	0.004	4.444	0.094	30.80	0.000	1.198
21	697.4	1.000	700.293		165.00		1.993
22	686.2	1.000	700.293		162.43		1.967
23	125.8	1.000	101.300		30.00		0.4365
24	145.1	1.000	101.300		34.62		0.4998
25	125.8	1.000	101.300		30.00		0.4365
26	133.5	1.000	101.300		31.84		0.4618
27	50.5	0.764	101.300		12.00		0.1804
28	29.5	0.764	101.300		7.00		0.1063

 Table 5.8 Thermodynamic properties for state points in cycles for endogenous exergy destruction in the low temperature heat exchanger

5.2.8 Summary of Endogenous/Exogenous Exergy Destruction for all Components

After calculating all of the individual component endogenous/exogenous exergy destruction values, it is useful to examine them in relation to each other. Figure 5.13 shows this breakdown.



Figure 5.13 Exergy destruction breakdown into endogenous/exogenous parts for a double stage absorption chiller

From the figure, it can be seen that most of the exergy destruction within a single component is an endogenous exergy destruction due to its own irreversibility, as would be expected. However, for the absorber, the value of the endogenous exergy destruction from other components is negative, and this negative exergy destruction conflicts with entropy generation theory. The explanations for the negative value are:

• The evaluation for the endogenous exergy destruction in the absorber assumes that there are ideal operations in the remaining components. However, those thermodynamic cycles do not exist in reality. • There is a trade off in reducing the exergy destruction for the absorber. If the efficiencies in the remaining components increase, due to the negative exogenous exergy destruction, the absorber's exergy destruction will not decrease, but actually increase by 0.02 kW.

5.3 Combined Analysis

Sections 5.1 and 5.2 introduce the division of exergy into avoidable/unavoidable exergy destruction contributions, and endogenous/exogenous exergy destruction contributions. Each of these individual divisions can provide useful data about the behavior of a given component within the overall system, but combining these analyses yields even more important information. This section, therefore, combines the previous two sections, and further divides the exergy destruction into unavoidable endogenous (ExDes^{UN,EN}), unavoidable exogenous (ExDes^{UN,EX}), avoidable endogenous (ExDes^{AV,EN}), and avoidable exogenous (ExDes^{AV,EX}) exergy destruction, to better guide research into system efficiency improvement.

To find each of these values, we must begin with calculating the unavoidable endogenous exergy destruction (ExDes^{UN,EN}). The calculation is similar to the endogenous exergy destruction calculation in Section 5.2, except that the irreversibility in the component of interest uses the minimum temperature difference rather than the real temperature difference. Once ExDes^{UN,EN} is obtained, the other three values can be found through calculations incorporating ExDes^{UN}, ExDes^{EN}, and ExDes^{AV}, as shown in the following equations:

$$ExDes^{UN,EX} = ExDes^{UN} - ExDes^{UN,EN}$$
(5.1)

$$ExDes^{AV,EN} = ExDes^{EN} - ExDes^{UN,EN}$$
(5.2)

$$ExDes^{AV,EX} = ExDes^{AV} - ExDes^{AV,EN}$$
(5.3)

Table 5.9 summarizes the irreversibilities (temperature differences) used in each calculation, and the calculation results are shown in the Table 5.10, which includes the real cycle exergy destruction, the breakdown of unavoidable/avoidable exergy destruction, the breakdown of endogenous/exogenous exergy destruction, and, finally, the breakdown of ExDes^{UN,EN}, ExDes^{UN,EN}, ExDes^{AV,EN}, and ExDes^{AV,EX}.

	Real Cycle	UN	EN	UN,EN
			$\Delta T_{ABS} = 8K, \ \Delta T_{HTRG} = 0, \ \Delta T_{LTRG} = 0,$	$\Delta T_{ABS} = 0.8K, \ \Delta T_{HTRG} = 0, \ \Delta T_{LTRG} = 0,$
ABS			$\Delta T_{EVP} = 0$, 100 % HTHX, 100 % LTHX,	$\Delta T_{\rm EVP}=0,\ 100\ \%\ HTHX,\ 100\ \%\ LTHX,$
			$\Delta T_{\rm CON} = 0.8$ K, ideal expansion values	$\Delta T_{\rm CON} = 0.8$ K, ideal expansion values
		$\Delta T_{ABS} = 0.8 \text{ K}$	$\Delta T_{ABS} = 0, \ \Delta T_{HTRG} = 8K, \ \Delta T_{LTRG} = 0,$	$\Delta T_{ABS}=0, \ \Delta T_{HTRG}=0.8K, \ \Delta T_{LTRG}=0,$
HTRG	$\Delta T_{ABS} = 8 \ K$	$\Delta T_{HTRG} = 0.8$	$\Delta T_{EVP} = 0$, 100 % HTHX, 100 % LTHX,	$\Delta T_{\rm EVP} = 0, \ 100 \ \% \ HTHX, \ 100 \ \% \ LTHX,$
	$\Delta T_{HTRG} = 8 \text{ K}$	К	$\Delta T_{\rm CON} = 0.8$ K, ideal expansion values	$\Delta T_{\rm CON} = 0.8$ K, ideal expansion values
	$\Delta T_{LTRG} = 8 \ K$	$\Delta T_{LTRG} = 0.8$	$\Delta T_{ABS} = 0, \ \Delta T_{HTRG} = 0, \ \Delta T_{LTRG} = 8K,$	$\Delta T_{ABS} = 0, \ \Delta T_{HTRG} = 0, \ \Delta T_{LTRG} = 0.8K,$
LTRG	$\Delta T_{EVP} = 3 \text{ K}$	K	$\Delta T_{EVP} = 0$, 100 % HTHX, 100 % LTHX,	$\Delta T_{\rm EVP}=0,\ 100\ \%\ HTHX,\ 100\ \%\ LTHX,$
	50% HTHX	$\Delta T_{EVP} = 0.3 \text{ K}$	$\Delta T_{\rm CON} = 0.8$ K, ideal expansion values	$\Delta T_{\rm CON} = 0.8$ K, ideal expansion values
	50% LTHX	50%	$\Delta T_{ABS} = 0, \ \Delta T_{HTRG} = 0, \ \Delta T_{LTRG} = 0,$	$\Delta T_{ABS} = 0, \ \Delta T_{HTRG} = 0, \ \Delta T_{LTRG} = 0,$
EVP	$\Delta T_{\rm CON}=8$	50%	$\Delta T_{EVP} = 8K$, 100 % HTHX, 100 % LTHX,	$\Delta T_{\rm EVP} = 0.8 K, 100\%$ HTHX, 100% LTHX,
		$\Delta T_{\rm CON}=0.8$	$\Delta T_{\rm CON} = 0.8$ K, ideal expansion values	$\Delta T_{\rm CON} = 0.8$ K, ideal expansion values
			$\Delta T_{ABS} = 0, \ \Delta T_{HTRG} = 0, \ \Delta T_{LTRG} = 0,$	The same as EN
HTHX			$\Delta T_{EVP} = 0,50$ % HTHX,100 % LTHX,	(Crystallization, no improvement
			$\Delta T_{\rm CON} = 0.8$ K, ideal expansion values	potential)

Table 5.9 Temperature differences summary for different cycle calculations

		$\Delta T_{ABS}=0, \ \Delta T_{HTRG}=0, \ \Delta T_{LTRG}=0,$	The same as EN
LTHX		$\Delta T_{\rm EVP}$ = 0, 100 % HTHX, 50 % LTHX,	(Crystallization, no improvement
		$\Delta T_{\text{CON}} = 0.8$ K, ideal expansion values	potential)
		$\Delta T_{ABS}=0, \ \Delta T_{HTRG}=0, \ \Delta T_{LTRG}=0,$	$\Delta T_{ABS}=0, \ \Delta T_{HTRG}=0, \ \Delta T_{LTRG}=0,$
CON		$\Delta T_{\rm EVP} = 0,100$ % HTHX,100 % LTHX,	$\Delta T_{EVP} = 0$, 100% HTHX,100% LTHX,
		$\Delta T_{\rm CON} = 8$ K, ideal expansion values	$\Delta T_{\rm CON} = 0.8$ K, ideal expansion values

	ExD	UN	AV	EN	EX	UN,EN	UN,EX	AV,EN	AV,EX
ABS	0.7241	0.6439	0.0802	0.7395	-0.0154	0.5598	0.0841	0.1797	-0.0995
HTRG	0.6023	0.5839	0.0184	0.5329	0.0694	0.4598	0.1241	0.0731	-0.0547
LTRG	0.536	0.4938	0.0422	0.4961	0.0399	0.4028	0.091	0.0933	-0.0511
EVP	0.34	0.1712	0.1688	0.34	0	0.1712	0	0.1688	0
HTHX	0.2416	0.1455	0.0961	0.1339	0.1077	0.1339	0.0116	0	0.0961
LTHX	0.2231	0.1148	0.1083	0.1137	0.1094	0.1137	0.0011	0	0.1083
CON	0.1963	0.001067	0.195233	0.1896	0.0067	0.000942	0.000125	0.188658	0.006575

Table 5.10 Overall advanced exergy analysis



Figure 5.14 Exergy destruction breakdown with UN,EN UN,EX AV,EN and AV,EX parts

Figure 5.14 shows the advanced exergy analysis breakdown for the double stage absorption chiller. Based on the advanced exergy analysis, the component that should be improved first is the condenser: The value of the condenser's endogenous avoidable exergy destruction (0.19 kW) is the highest among all components. At the same time, for the condenser, the endogenous exergy destruction is higher than the exogenous exergy destruction. This means that in order to decrease condenser exergy destruction, improving the condenser itself is more important than improving other components. The endogenous avoidable exergy destruction is 19 times that of the exogenous exergy destruction, so the total exergy destruction in the condenser can be reduced mainly through improvement in condenser.

The absorber is the second component that should be improved based on its endogenous avoidable exergy destruction of 0.18 kW. That value is higher than the avoidable exergy destruction of 0.08 kW in the absorber, which is due to -0.10 kW of exogenous exergy destruction contributed from the remaining components. If the remaining components are improved, the overall contribution will not decrease but increase the absorber's exergy destruction by 0.10 kW, so the endogenous exergy destruction also increases to 0.18 kW to meet the overall avoidable exergy destruction in the absorber as 0.08 kW (= 0.18 kW – 0.10 kW).

The evaporator is the third component to be improved. As discussed in the previous section, there is no exogenous exergy destruction in the evaporator, so the total exergy destruction in the evaporator can be reduced only through improvement in the evaporator itself. The endogenous exergy destruction is 0.17 kW, compared to a 0.34 kW total exergy destruction in the evaporator, so there is 50% improvement potential for the evaporator.

The low temperature and high temperature regenerators have 0.09 kW and 0.07 kW endogenous avoidable exergy destructions, respectively. They are the fourth and fifth components that should be considered for improvement. As in the absorber, the exogenous avoidable exergy destructions are negative.
For the high temperature heat exchanger and the low temperature heat exchanger, as discussed in previous section, there is no improvement potential due to solution crystallization. From the analysis, there is no endogenous avoidable exergy destruction for those two heat exchangers. So the total exergy destructions can only be reduced through improvement in the remaining components.

5.4 Sensitivity Analysis

The advanced exergy analysis presented in Sections 5.1 to 5.3 is based on the unique high temperature regenerator temperature (165°C) and condenser/absorber inlet temperature (30°C) in Table 4.4. In this section, a sensitivity analysis is performed by varying the condensing and regenerating temperature by 5%. The varying conditions are summarized in Table 5.11. The results for the base case are previously shown in Figure 5.14.

Case	State Points	Conditions [°C]
Basa casa	21	165*
(Sections 5.1 to 5.3)	23	30*
	25	30*
Sensitivity analysis	21	173.25
case 1	23	30*
(Regenerating temp + 5%)	25	30*
Sensitivity analysis	21	156.75
case 2	23	30*
(Regenerating temp - 5%)	25	30*
Sensitivity analysis	21	165*
case 3	23	31.5
(Condensing temp + 5%)	25	31.5

Table 5.11 Sensitivity analysis for the double stage absorption chiller

Sensitivity analysis	21	165*
case 4	23	28.5
(Condensing temp - 5%)	25	28.5

*Experimental Data

By using the same code (given in Appendices 1-9) and changing the conditions according to Table 5.11, the sensitivity analysis results are shown in Figures 5.15.

Based on the data from Figures 5.15, the sensitivity analysis shows that

- 1) Due to 5% variation in regenerating and condensing temperatures for each case, the values of the overall exergy destruction and subsets of the exergy destruction in each component are slightly different from each other and from the base case, but the order of the components from the highest to the lowest overall exergy destruction is the same compared to the base case. The order of the components, from highest to lowest overall exergy destruction, is the absorber, high temperature regenerator, low temperature regenerator, evaporator, high temperature heat exchanger, low temperature heat exchanger, and condenser;
- 2) The order of the components with the highest to the lowest avoidable, endogenous exergy destruction is the same for each case. The order of potential improvement among the components (from most to least important) is the same as that found in the base case: the condenser, absorber, evaporator, low temperature regenerator, and high temperature regenerator. So with 5% variations in regenerating and condensing temperatures, there is no effect on the ranking of the components to be examined for potential improvements.



Figure 5.15 Exergy destruction breakdown with UN, EN UN, EX AV, EN and AV, EX parts (sensitivity analysis)

5.5 Summary

In summary, from this chapter, the following conclusions can be drawn:

- The system COP is strongly related to the exergy destruction in the system. In order to increase the system COP, research and engineering efforts should focus on the equipment with the highest exergy destruction.
- 2) Based on a conventional exergy destruction analysis, research should focus on the various components in the following order of decreasing importance: the absorber, high temperature regenerator, low temperature regenerator, evaporator, high temperature heat exchanger, low temperature heat exchanger, condenser, and expansion valves. However, this analysis cannot reflect quantitatively how much of the exergy destruction in a given component is avoidable.
- 3) An avoidable/unavoidable exergy destruction analysis provides more information than the generation exergy destruction analysis on what the order of component improvement should be. The new research focus order (again from most to least significant) is found to be: the condenser, evaporator, low temperature heat exchanger, high temperature heat exchanger, absorber, low temperature regenerator, and high temperature regenerator. The expansion valves have no potential for improvement due to the limitations of the current technology. The new order listed above is clearly very different from that of the conventional exergy analysis; however, it should be noted that this avoidable exergy destruction analysis does not incorporate the potential contributions to or from other components by increasing the efficiency of a single component.
- 4) In order to include these inter-component interactions, we must further break down the exergy destruction into unavoidable endogenous, unavoidable exogenous, avoidable endogenous, and avoidable exogenous parts. This breakdown is by far the most comprehensive view of the exergy destruction in the

double stage absorption chiller. Through this combined analysis, the final order of potential improvement among the components (from most to least important) is: the condenser, absorber, evaporator, low temperature regenerator, and high temperature regenerator. Both the high and low temperature heat exchangers have no potential for potential due to limitations in the physical behavior of the working fluids (crystallization of the LiBr solution).

5) From the combined analysis, and based on current technology limits (as embodied by the lowest possible temperature differences and the LiBr solution behavior), the maximum possible increase in the COP for a double stage absorption chiller is 25%, for a final COP of 1.38.

Chapter 6: Conclusions

6.1 Contributions

Energy, exergy, and advanced exergy analyses were applied in this dissertation to evaluate the performance of a solar double stage absorption chiller, using the equipment in the Intelligent Workplace at Carnegie Mellon University as an example. The overall system was assessed in a novel manner, different from the typical energy analysis applied to such a system. The major contributions from this research are presented below:

1) Assessment of the true fundamental life cycle energy and exergy costs in chiller applications.

In the traditional comparison of COPs or efficiencies between electric chillers and thermal driven absorption chillers, only the site energy consumption (electricity and thermal energy consumption) is generally considered. However, this is not a fair comparison, because the driving forces are different types of energy with different energy qualities. This dissertation provides a new way to compare the two systems. Coal is chosen as a representative fossil fuel, and calculations are made for the solar energy used in generating the coal, based on data in the existing literature. This is then compared with the solar energy requirement of a solar absorption chiller system, given the same amount of cooling requirement. The analysis calculates the true fundamental life cycle solar energy and exergy consumption for the chiller operation. It is demonstrated that this is the only fair comparison, because both chillers are ultimately solar-powered chillers. This is also the most comprehensive comparison, including all of the possible losses during the solar energy and exergy conversions. Both the energy and exergy analyses show that the solar absorption chiller is the most efficient system. For the energy analysis, although the electric chiller has a higher COP, the efficiency of the photosynthesis and fuel synthesis processes greatly reduces the ultimate electric chiller efficiency. For the exergy analysis,

it can be shown that building cooling is a low exergy application. Therefore, in the electric chiller, there exists a huge mismatch between the high quality electricity supply and the low quality building cooling demand. The solar double stage absorption chiller has a higher system exergetic efficiency than the electric chiller, because the solar collector harvests the solar exergy and converts it into low exergy thermal fluids. It provides a good match between the low exergy supply and the low exergy building cooling application demands.

2) Establishment of energy and exergy analyses for the solar double stage absorption chiller and calibration of the models with experimental results.

An energy analysis is applied to the solar double stage absorption chiller system, and is then validated with experimental results from the literature. From the energy analysis alone, it appears that the only ways to increase the collector efficiency are reducing the optical and thermal losses to the environment. However, adding an exergy analysis to the energy analysis shows a more complete picture of all the energy quality losses due to entropy generation. In the exergy domain, the exergy losses not only include the optical and thermal exergy losses, but also include the absorption exergy destruction, heat conduction exergy destruction, and friction exergy destruction. It is found that the absorption exergy destruction is equally important, and should be reduced for collector efficiency improvement. The energy analysis for the double stage absorption chiller only evaluates the system COP, but cannot provide any information on the irreversibilities in the system. This information can be provided by the exergy analysis. Based on the quantity of the exergy destruction in each component, research into system efficiency improvement should focus on the various components in the following order of decreasing importance: absorber, high temperature regenerator, low temperature regenerator, evaporator, high temperature heat exchanger, low temperature heat exchanger, condenser, and expansion valves.

3) Application of the advanced exergy analysis for the double stage absorption chiller to aid engineers in improving the absorption chiller COP.

Although the basic exergy analysis provides more insight than the energy analysis into potential improvements to the system, still more information can be gained by applying an advanced exergy analysis. An advanced exergy analysis further breaks down the exergy destruction within a component into unavoidable endogenous, unavoidable exogenous, avoidable endogenous, and avoidable exogenous exergy destruction. These divisions provide, by far, the most comprehensive view of the exergy destruction in the double stage absorption chiller. Through this analysis, the final order of the potential improvement among the components (from most to least important) is: condenser, absorber, evaporator, low temperature regenerator, and high temperature regenerator. Both the high and low temperature heat exchangers have no potential for improvement due to limitations in the physical behavior of the working fluids (crystallization of the LiBr solution). Based on the current technology limits (as embodied by the lowest possible temperature differences and the LiBr solution behavior), the maximum possible increase in the COP for a double stage absorption chiller is 25%, for a final COP of 1.38.

6.2 Conclusions

In conclusion, the contributions outlined above address the hypotheses from Chapter 1. Those hypotheses are:

1. A solar double stage absorption chiller (with an absorption chiller COP=1.2) is actually more efficient than an electric chiller (with a COP=5), in terms of the overall solar energy requirement in fossil fuel formation, energy efficiency, and exergetic efficiency.

As shown in Chapter 3, given the same amount of cooling energy and exergy, the solar double stage absorption chiller is roughly 5,000 times more efficient than the electric chiller, and in the exergy domain, the solar absorption chiller is close to 10,000 times more efficient.

2. The advanced exergy analysis method of dividing the exergy destruction into endogenous/exogenous parts and unavoidable/avoidable parts can help to better identify the order in which engineers should focus on the components, in order to boost the overall efficiency of the system.

As discussed in Chapters 2, 4, and 5, an advanced exergy analysis can provide more information on the component improvement potential based on technology limitations, and diagnose the interactions between components within an energy system. A double stage absorption chiller is a multi-component energy system, which converts thermal energy into useful cooling energy. The advanced exergy analysis better assists engineers in improving the system efficiency, by directing the focus on the avoidable endogenous exergy destruction in each component. For example, in a traditional exergy analysis, improving the performance of the condenser is considered to be a low priority. However, by applying an advanced exergy analysis, it is instead demonstrated that the condenser holds the highest potential for improvement. Through the advanced exergy analysis, it was found that the double stage lithium bromide absorption chiller has the potential for a 25% improvement in COP, for a final COP of 1.38 (given the external conditions of 165 \mathbb{C} thermal fluids to high temperature regenerator, 7 \mathbb{C} for the chilled water supply, and 30 \mathbb{C} from the cooling tower in summer operation).

6.3 Future Work

This dissertation presents the true fundamental life cycle solar energy and exergy consumptions for different chillers, and applies exergy/advanced exergy analyses to a solar double stage absorption chiller. The research contributions point to future research directions:

1) Entropy generation minimization for efficiency improvement

The dissertation gives researchers and engineers a thorough understanding of all the losses within the system. This serves as the first step for the improvement of the energy

system efficiency. The components with the highest avoidable endogenous exergy destruction should be focused upon first, and the entropy generation minimization methodology should be applied to reduce the entropy generation in those components. That methodology includes, for example, temperature difference reduction for a heat exchanger, and friction reduction in piping, and is based on Bejan's work (1995).

2) Exergoeconomic optimization for the overall system

As shown in the advanced exergy analysis, the smaller the temperature difference in the heat exchanger, the smaller the entropy generation. To achieve this, though, given the same heat transfer rate, manufacturers need a larger heat transfer area, and potentially more material utilization. Exergoeconomic optimization can address this issue, because the optimization considers both the exergy destruction and the material/manufacturing costs. It is important to perform an exergoeconomic optimization based on the cost for each component, while a genetic algorithm can be used for an overall exergoeconomic optimization.

 Environmental impact comparisons among different types of primary energy for the chiller applications

The true life cycle energy and exergy analysis for the various chiller systems could be extended further. For example, all fossil fuel-based electricity generation does not originate with coal. A comparative analysis using, for example, natural gas for power generation would also be instructive. At the same time, the rapidly decreasing price for the photovoltaic panels provides an opportunity for PV integrated electric chiller operation. Future work should include the possibility of using a solar photovoltaic system to generate electricity, and then using this electricity for electric chillers.

4) Embodied energy analysis for different types of chillers

The energy consumption values discussed in the previous chapters are all associated with the chiller operation. In order to fully understand the total energy consumption, which includes the chiller manufacturing and transportation energy consumption, an embodied energy analysis should be performed.

References

Arioglu, E. (1994). Outlook for coal reserves, production, consumption, International Coal Trade, *Coal: Resources, Properties, Utilization, Pollution, Istanbul Technical University, Istanbul*, 183-195.

Bejan, A., (1995). *Entropy generation minimization: the method of thermodynamic optimization of finite-size systems and finite-time processes, CRC Press, 1 edition.*

Bejan, A., Keary, DW., and Kreith, F.(1981). Second law analysis and synthesis of solar collector system, *Journal of Solar Energy Engineering*, *103*:23-8.

Building energy data book. (2009). Department of Energy.

Broad Air Conditioning Brochure (2006)

Clausius, R. (1850). Ueber die bewegende Kraft der Wärme und die Gesetze, welche sich daraus für die Wärmelehre selbst ableiten lassen, Annalen der Physik und Chemie (Poggendorff, Leipzig), 155 (3): 368-394, particularly on page 373, translation taken from Truesdell, C.A. (1980). The Tragicomical History of Thermodynamics, 1822-1854, *Springer, New York, ISBN 0-387-90403-4*, pages 188-189.

Coal: Power for progress. (2000). World Coal Institute, London.

Dincer, I., and Rosen, M.A. (2007). Exergy: energy, environment and sustainable development, *Elsevier*.

Dukes, J.S. (2003).Burning buried sunshine: human consumption of ancient solar energy, *Climatic Change*, *61*, 31-44.

Energy efficiency of photosynthesis. (1999). Encyclopedia Britannica CD 99.

Erdem, H.H., Akkaya, A.V., Cetin, B., Dagdas, A., Sevilgen, S.H., Sahin, B., Teke, I., Gungor, C. and Atas, S. (2009). Comparative energetic and exergetic performance analyses for coal-fired thermal power plants in Turkey, *International Journal of Thermal Sciences*, *48*, 2179-2186.

Herold, K.E., Radermacher, R., &Klein, S.A. (1996). *Absorption chillers and heat pumps,* CRC Press LLC

Hope, B., and Stanton, C. (2008). Masdar eyes carbon credit bonus, *TheNational*, Retrieved from http://www.thenational.ae/news/worldwide/middle-east/masdar-eyescarbon-credit-bonus

Hu, Y., Schaefer, L., and Hartkopf, V. (2011). Energy and exergy analysis of integrating compound parabolic collectors (CPC) with Lithium Bromide (Li-Br) absorption chiller for building heating and cooling to achieve net zero buildings. *ASHRAE 2011 Annual Conference, Montreal*

International Energy Conservation Code, 2006

Monteith, J.L. (1977). Climate and efficiency of corp production in Britain, *Philosophical Transactions of the Royal Society of London*. *Series B, Biological Sciences* 281, 277-284.

National Research Council.(2010). *Hidden Costs of Energy: Unpriced Consequences of Energy Production and Use*, Committee on Health, Environmental, and Other External Costs and Benefits of Energy Production and Consumption, The National Academies Press.

Petela, R. (2008). An approach to the exergy analysis of photosynthesis, *Solar Energy* 82, 311-328.

Potter, C.S., Randerson, J.T., Field, C.B., Matson, P.A., Vitousek, P.M., Mooney, H.A., and Klooster, S.A.(1993). Terrestrial ecosystem production: a process model based on global satellite and surface data, *Global Biogeochem, Cycles 7*, 811-841.

Qu, M. (2007). Model Based Design and Performance Analysis of Solar Absorption Cooling and Heating System, *PhD Dissertation*, Carnegie Mellon University

Rosen, M.A. and Dincer, I. (1997). On exergy and environmental impact, *International Journal of Energy Research* 21, 643–654.

Rosen, M.A. andScott, D.S. (2003). Entropy production and exergy destruction: part Ihierarchy of earth's major constituencies, *International Journal of Hydrogen Energy* 28, 1307-1313.

Szargut, J., and Petela, R. (1965).Exergetic efficiency of plant vegetation.*Zesz. Nauk. Pol. SI. No. 129, Energ., z. 18,* 3-19 (in Polish)

Suzuki, A. 1988.General theory of exergy-balance analysis and application to solar collectors, *Energy, Vol. 13*, No.2,153-160.

Tsatsaronis, G., and Park, MH. (2002). On avoidable and unavoidable exergy destructions and investment costs in thermal systems. *Energy Conversion and management*, *43*, 1259-1270.

Wall, G.1977. Exergy-a useful concept within resource accounting, Retrieved from http:// http://www.exergy.se/goran/thesis/paper1/paper1%20old.html Yin, HX.(2006). An Absorption Chiller in a Micro BCHP Application: Model based Design and Performance Analysis, *PhD Dissertation*, Carnegie Mellon University

Appendix

Appendix 1 Codes for the Exergy Destruction Analysis in Real Cycles

{The state point assumptions for the cycles are based on Hu et al. (2011), Yin (2008), and Herold et al. (1996).}

{Units and Environmental References} SI=2 P0=101.3 [kPa] T0=303 [K]

{Heat Transfer Fluid Flow Rate Inputs} Qevap=16 [kW] m[21]=1 [kg/sec] m[25]=1 [kg/sec]

m[23]=1 [kg/sec]

{Heat Transfer Fluid Inlet Temperature Inputs}

T[21]=165 [C] T[25]=30 [C] T[27]=12 [C] T[28] = 7 [C] T[23]=30 [C]

{Defined Delta Temperature Inputs}

DeltaTabs= 8 [K] DeltaTcon= 8 [K] DeltaTevp= 3 [K] DeltaThtrg= 8 [K] DeltaTltrg = 8 [K] Eff_HX=0.5

{Absorber}

m[10]*h[10]+h[6]*m[6]-m[1]*h[1]=m[23]*(H[24]-H[23]) Qabs=m[23]*(H[24]-H[23]) T[1]=T[23]+DeltaTabs P[23]=P0 P[24]=P0 H[23]=Enthalpy(Water, T=T[23], P=P[23]) H[24]=Enthalpy(Water, T=T[24], P=P[24]) S[23]= Entropy(Water, T=T[24], P=P[23]) S[24]=Entropy(Water, T=T[24], P=P[24]) M[24]=M[23]

{Condenser} m[8]=m[7]+m[19] m[7]*h[7]+m[19]*h[19]-m[8]*h[8]=m[25]*(H[26]-H[25]) Qcond=m[25]*(H[26]-H[25]) T[8]=T[25]+DeltaTcon P[25]=P0 P[26]=P[25] H[25]=Enthalpy(Water, T=T[25], P=P[25]) H[26]=Enthalpy(Water, T=T[26], P=P[26]) S[25]=Entropy(Water, T=T[25], P=P[25]) S[26]=Entropy(Water, T=T[26], P=P[26]) M[26]=M[25]

{Evaporator}

m[9]*(h[10]-h[9])=m[27]*(H[27]-H[28]) Qevap=m[27]*(H[27]-H[28]) T[9]=7-DeltaTevp P[27]=P0 P[28]=P0 H[27]=Enthalpy(Water, T=T[27], P=P[27]) H[28]=Enthalpy(Water, T=T[28], P=P[28]) M[28]=M[27] S[27]=Entropy(Water, T=T[27], P=P[27]) S[28]=Entropy(Water, T=T[28], P=P[28])

{High TemperatureRegenerator}

 $\begin{array}{l} x[14]=X_LIBR(T[14],Ph,SI) \\ m[13]=m[14]+m[17] \\ m[13]*x[13]=m[14]*x[14] \\ m[17]*h[17]+m[14]*h[14]-m[13]*h[13]=m[21]*(H[21]-H[22]) \\ Qgenh=m[21]*(H[21]-H[22]) \\ T[14]=T[21]-DeltaThtrg \\ P[21]=Pressure(Water, T=T[21], X=0) \\ P[22]=P[21] \\ H[21]=Enthalpy(Water, T=T[21], X=0) \\ H[22]=Enthalpy(Water, T=T[21], X=0) \\ S[21]=Entropy(Water, T=T[21], X=0) \\ S[22]=Entropy(Water, T=T[22], P=P[22]) \\ S[22]=Entropy(Water, T=T[22], P=P[22]) \\ M[22]=M[21] \end{array}$

{Low Temperature Regenerator}

 $\begin{array}{l} x[11]=x[3] \\ x[4]=x[16] \\ m[3]+m[16]=m[4]+m[11]+m[7] \\ m[3]^xx[3]+m[16]^xx[16]=m[4]^xx[4]+m[11]^xx[11] \\ T[11]=T[7] \\ h[3]^xm[3]+m[16]^h[16]+m[17]^h[17]=h[4]^xm[4]+h[11]^xm[11]+m[18]^h[18]+m[7]^h[7] \\ Qcd=m[17]^x(h[17]-h[18]) \\ T[4]=T[18]-DeltaTItrg \\ \end{array}$

{Low Temperature Heat Exchanger} Eff_HX=(T[4]-T[5])/(T[4]-T[2]) m[1]*(h[3]-h[2])=m[4]*(h[4]-h[5]) Qhxl=m[4]*(h[4]-h[5])

{Refrigerant Expansion Valves} h[8]=h[9] h[18]=h[19]

{Solution Expansion Valves} h[6]=h[5] h[16]=h[15]

{Pump Calculation}

Pump1=m[1]*v1*(Pm-Pl)/1000 {kW} h[2]=h[1]+Pump1/m[1] Pump2=m[11]*v11*(Ph-Pm)/1000{kW} h[12]=h[11]+Pump2/m[11]

{COP}

COP=Qevap/Qgenh COPe=Qevap/(Qgenh+0.5)

{Mass Balances}

m[2]=m[1] m[3]=m[2] m[5]=m[4] m[6]=m[5] m[9]=m[8] m[10]=m[9] m[12]=m[11] m[13]=m[12] m[15]=m[14] m[16]=m[15] m[18]=m[17] m[19]=m[18]

{Solute Mass Balances}

x[2]=X[1] x[3]=x[2] x[5]=x[4] x[6]=x[5] x[12]=X[11] x[13]=x[12] x[15]=x[14] x[16]=x[15]

{Working fluid property relations}

$$\begin{split} &h[1]=H_LIBR(T[1],X[1],SI) \\ &x[1]=X_LIBR(T[1],PI,SI) \\ &v1=V_LIBR(T[1],x[1],SI) \\ &h[2]=H_LIBR(T[2],x[2],SI) \\ &h[3]=H_LIBR(T[3],x[3],SI) \\ &h[4]=H_LIBR(T[4],x[4],SI) \\ &T[4]=T_LIBR(Pm,x[4],SI) \\ &h[5]=H_LIBR(T[5],x[5],SI) \\ &CALL Q_LIBR(h[5],PI,x[5],2:q[6]*100,T[6],XI6,hI6,hv6) \\ &T[7]=T_LIBR(Pm,x[3],SI) \\ &h[7]=enthalpy(WATER,T=T[7],P=Pm) \\ &h[8]=enthalpy(WATER,T=T[8],x=q[8]) \\ &T[9]=temperature(WATER,h=h[9],P=PI) \\ &q[9]=quality(WATER,h=h[9],T=T[9]) \\ &h[10]=enthalpy(WATER,T=T[10],x=q[10]) \end{split}$$

v11=V_LIBR(T[11],x[1],SI) h[11]=H_LIBR(T[11],x[11],SI) h[12]=H_LIBR(T[12],x[12],SI) h[13]=H_LIBR(T[13],x[13],SI) h[14]=H_LIBR(T[14],x[14],SI) h[15]=H_LIBR(T[15],x[15],SI) CALL Q_LIBR(h[15],Pm,x[15],2:q[16]*100,T[16],XI16,hI16,hv16) T[17]=T_LIBR(Ph,x[13],SI) h[17]=enthalpy(WATER,T=T[17],P=Ph) h[18]=enthalpy(WATER,T=T[18],x=q[18]) T[19]=temperature(WATER,h=h[19],P=Pm) q[19]=quality(WATER,h=h[19],T=T[19])

{Pure Water, 0 mass fraction}

x[7]=0. x[8]=0. x[9]=0. x[10]=0. x[17]=0. x[18]=0. x[19]=0.

{Water Vapor Qualities}

q[1]=0 q[4]=0 q[11]=0 q[14]=0 q[18]=0 q[8]=0 q[10]=1.0

{Pressures}

Ph=pressure(WATER,T=T[18],x=q[18]) Pm=pressure(WATER,T=T[8],x=q[8]) PI=pressure(WATER,T=T[10],x=q[10]) P[1]=PI P[2]=Pm P[3]=Pm P[4]=Pm P[5]=Pm P[6]=PI P[7]=Pm P[8]=Pm P[9]=PI P[10]=PI P[11]=Pm P[12]=Ph P[13]=Ph P[14]=Ph P[15]=Ph P[16]=Pm P[17]=Ph P[18]=Ph P[19]=Pm

{Entropy}

S[1]=S_LiBrH2O(T[1],X[1]/100) S[2]=S LiBrH2O(T[2], X[2]/100) S[3]=S LiBrH2O(T[3], X[3]/100) S[4]=S_LiBrH2O(T[4], X[4]/100) S[5]=S LiBrH2O(T[5], X[5]/100)S[6]=S_LiBrH2O(T[6], XI6/100)*(1-q[6])+q[6]*Entropy(Water, P=P[6], X=1) S[7]=Entropy(Water, P=P[7], T=T[7]) S[8] =Entropy(Water, T=T[8], X=0) S[9] = Entropy(Water, T=T[9], X=0)*(1-Q[9])+Q[9]*Entropy(Water, T=T[9], X=1) S[10] = Entropy(Water, T=T[10], X=1)S[11]=S_LiBrH2O(T[11], X[11]/100) S[12]=S LiBrH2O(T[12], X[12]/100) S[13]=S LiBrH2O(T[13], X[13]/100) S[14]=S_LiBrH2O(T[14], X[14]/100) S[15]=S LiBrH2O(T[15], X[15]/100) S[16]=S LiBrH2O(T[16], X|16/100)*(1-q[16])+q[16]*Entropy(Water, P=P[16], X=1)+0.001 S[17]=Entropy(Water, P=P[17], T=T[17]) S[18] = Entropy(Water, P=P[18], X=0)S[19] = Entropy(Water, T=T[19], X=0)*(1-Q[19])+Q[19]*Entropy(Water, T=T[19], X=1) {Exergy Destructions} 0=M[27]*(H[27]-H[28]-T0*(S[27]-S[28]))+M[9]*(H[9]-H[10]-T0*(S[9]-S[10]))-ExDevp 0=M[23]*(H[23]-H[24]-T0*(S[23]-S[24]))+M[10]*(H[10]-T0*S[10])+M[6]*(H[6]-T0*S[6])-M[1]*(H[1]-T0*S[1])-ExDabs 0=M[25]*(H[25]-H[26]-T0*(S[25]-S[26]))+M[7]*(H[7]-T0*S[7])+M[19]*(H[19]-T0*S[19])-M[8]*(H[8]-T0*S[8])-ExDcon 0=M[2]*(H[2]-H[3]-T0*(S[2]-S[3]))+M[4]*(H[4]-H[5]-T0*(S[4]-S[5]))-ExDlthx 0=M[12]*(H[12]-H[13]-T0*(S[12]-S[13]))+M[14]*(H[14]-H[15]-T0*(S[14]-S[15]))-ExDhthx 0=M[21]*(H[21]-H[22]-T0*(S[21]-S[22]))+M[13]*(H[13]-T0*S[13])-M[14]*(H[14]-T0*S[14])-M[17]*(H[17]-T0*S[17])-ExDhtrg 0=M[17]*(H[17]-H[18]-T0*(S[17]-S[18]))+M[3]*(H[3]-T0*S[3])+M[16]*(H[16]-T0*S[16])-M[11]*(H[11]-T0*S[11])-M[4]*(H[4]-T0*S[4])-M[7]*(H[7]-T0*S[7])-ExDltrg 0=M[18]*(-T0*(S[18]-S[19]))-ExDval r h 0=M[8]*(-T0*(S[8]-S[9]))-ExDval_r_l 0=M[15]*(-T0*(S[15]-S[16]))-ExDval_s_h

0=M[5]*(-T0*(S[5]-S[6]))-ExDval_s_l

ExDTotal=ExDevp+ExDabs+ExDcon+ExDlthx+ExDhthx+ExDhtrg+ExDltrg+ExDval_r_h+ExDval_ r_l+ExDval_s_h+ExDval_s_l

ExLTotal=abs(M[25]*(H[25]-H[26]-T0*(S[25]-S[26])+M[23]*(H[23]-H[24]-T0*(S[23]-S[24]))))

ExSupply=M[21]*(H[21]-H[22]-T0*(S[21]-S[22]))

ExLoad=ExSupply-ExDTotal-ExLTotal

ExLoadcheck=abs(M[27]*(H[27]-H[28]-T0*(S[27]-S[28])))

ExEff=ExLoad/ExSupply

Appendix 2 Codes for the Unavoidable Exergy Destruction Analysis

{The state point assumptions for the cycles are based on Hu et al. (2011), Yin (2008), and Herold et al. (1996); only the differences from the codes in Appendix 1 are shown below.}

{Defined Delta Temperature Inputs}

DeltaTabs= 0.8 [K] DeltaTcon= 0.8 [K] DeltaTevp= 0.3 [K] DeltaThtrg= 0.8 [K] DeltaTltrg = 0.8 [K] Eff_HX=0.5

Appendix 3 Codes for the Endogenous and Unavoidable Endogenous Exergy Destruction Analysis in the Absorber

{The state point assumptions for the cycles are based on Hu et al. (2011), Yin (2008), and Herold et al. (1996); only the differences from the codes in Appendix 1 are shown below.}

{Defined Delta Temperature Inputs}

DeltaThtrg=0 [K] DeltaTcon=0.8 [K] DeltaTevp=0 [K] DeltaTabs=8 [K] {DeltaTabs=0.8 [K] {UN,EN}} DeltaTltrg=0 Eff HX=0.5

{Solution Expansion Valves} h[6]=h[5] - a {try and error, EN: a = -0.85; UN,EN,a = -1.1} CALL Q_LIBR(h[6],PI,x[5],2:q[6]*100,T[6],Xl6,hl6,hv6) S[6]=S_LiBrH2O(T[6], Xl6/100)*(1-q[6])+q[6]*Entropy(Water, P=P[6], X=1)

h[16]=h[15]- b{try and error, EN: b = -1.5; UN,EN, b = -2.52} CALL Q_LIBR(h[16],Pm,x[15],2:q[16]*100,T[16],XI16,hI16,hv16) S[16]=S_LiBrH2O(T[16], XI16/100)*(1-q[16])+q[16]*Entropy(Water, P=P[16], X=1)

{Refrigerant Expansion Valves}

$$\begin{split} S[9] &= S[8] \\ S[9] &= Entropy(Water, T=T[9], X=0)^{*}(1-Q[9])+Q[9]^{*}Entropy(Water, T=T[9], X=1) \\ H[9] &= Enthalpy(Water, T=T[9], X=0)^{*}(1-Q[9])+Q[9]^{*}Enthalpy(Water, T=T[9], X=1) \end{split}$$

$$\begin{split} S[19] &= S[18] \\ S[19] &= Entropy(Water, T=T[19], X=0)^*(1-Q[19])+Q[19]^*Entropy(Water, T=T[19], X=1) \\ H[19] &= Enthalpy(Water, T=T[19], X=0)^*(1-Q[19])+Q[19]^*Enthalpy(Water, T=T[19], X=1) \end{split}$$

Appendix 4 Codes for the Endogenous and Unavoidable Endogenous Exergy Destruction Analysis in the Condenser

{The state point assumptions for the cycles are based on Hu et al. (2011), Yin (2008), and Herold et al. (1996); only the differences from the codes in Appendix 1 are shown below.}

{Defined Delta Temperature Inputs} DeltaThtrg=0 [K] DeltaTcon=8 [K] {DeltaTcon=0.8 [K] {UN,EN}} DeltaTevp=0 [K] DeltaTabs=0 [K] DeltaTltrg=0 [K] Eff_HX=1

{Solution Expansion Valves} h[6]=h[5]- a{try and error, EN: a = -0.89; UN,EN,a = -1.10} CALL Q_LIBR(h[6],PI,x[5],2:q[6]*100,T[6],Xl6,hl6,hv6) S[6]=S_LiBrH2O(T[6], Xl6/100)*(1-q[6])+q[6]*Entropy(Water, P=P[6], X=1)

 $\label{eq:h16]=h16]-b $ try and error, EN: b =-2.85; UN,EN, b = -2.66 $ CALL Q_LIBR(h[16],Pm,x[15],2:q[16]*100,T[16],XI16,hI16,hv16) $ S[16]=S_LiBrH2O(T[16], XI16/100)*(1-q[16])+q[16]*Entropy(Water, P=P[16], X=1) $ The set of the$

{Refrigerant Expansion Valves}

$$\begin{split} S[9] &= S[8] \\ S[9] &= Entropy(Water, T=T[9], X=0)^{*}(1-Q[9])+Q[9]^{*}Entropy(Water, T=T[9], X=1) \\ H[9] &= Enthalpy(Water, T=T[9], X=0)^{*}(1-Q[9])+Q[9]^{*}Enthalpy(Water, T=T[9], X=1) \end{split}$$

$$\begin{split} S[19] &= S[18] \\ S[19] &= Entropy(Water, T=T[19], X=0)^*(1-Q[19])+Q[19]^*Entropy(Water, T=T[19], X=1) \\ H[19] &= Enthalpy(Water, T=T[19], X=0)^*(1-Q[19])+Q[19]^*Enthalpy(Water, T=T[19], X=1) \end{split}$$

Appendix 5 Codes for the Endogenous and Unavoidable Endogenous Exergy Destruction Analysis in the Evaporator

{The state point assumptions for the cycles are based on Hu et al. (2011), Yin (2008), and Herold et al. (1996); only the differences from the codes in Appendix 1 are shown below.}

{Defined Delta Temperature Inputs} DeltaThtrg=0 [K] DeltaTcon=0.8 [K] DeltaTevp=3 [K] {DeltaTevp=0.3 [K] {UN,EN}} DeltaTabs=0 [K] DeltaTltrg=0 [K] Eff_HX=1

{Solution Expansion Valves} h[6]=h[5]- a{ try and error, EN: a =-0.6 ; UN,EN: a =-1.04 } CALL Q_LIBR(h[6],PI,x[5],2:q[6]*100,T[6],Xl6,hl6,hv6) S[6]=S_LiBrH2O(T[6], Xl6/100)*(1-q[6])+q[6]*Entropy(Water, P=P[6], X=1)

 $\label{eq:h16} \begin{array}{l} h[16]=h[15]-b\{ try \ and \ error, \ EN: \ b=3 \ C, \ -2.15; \ UN, EN: \ b=-2.6 \ \} \\ CALL \ Q_LIBR(h[16], Pm, x[15], 2:q[16]*100, T[16], XI16, hI16, hv16) \\ S[16]=S_LiBrH2O(T[16], \ XI16/100)*(1-q[16])+q[16]*Entropy(Water, \ P=P[16], \ X=1) \end{array}$

{Refrigerant Expansion Valves}

S[9] = S[8] S[9] = Entropy(Water, T=T[9], X=0)*(1-Q[9])+Q[9]*Entropy(Water, T=T[9], X=1)H[9] = Enthalpy(Water, T=T[9], X=0)*(1-Q[9])+Q[9]*Enthalpy(Water, T=T[9], X=1)

Appendix 6 Codes for the Endogenous and Unavoidable Endogenous Exergy Destruction Analysis in the High Temperature Regenerator

{The state point assumptions for the cycles are based on Hu et al. (2011), Yin (2008), and Herold et al. (1996); only the differences from the codes in Appendix 1 are shown below.}

{Defined Delta Temperature Inputs} DeltaThtrg=8 [K]{DeltaThtrg=0.8 [K] {UN,EN}} DeltaTcon=0.8 [K] DeltaTevp=0 [K] DeltaTabs=0 [K] DeltaTltrg=0 [K] Eff_HX=1

{Solution Expansion Valves} h[6]=h[5]- a{ try and error, EN: a = 8C, -0.9; UN,EN, a= -1.03} CALL Q_LIBR(h[6],PI,x[5],2:q[6]*100,T[6],Xl6,hl6,hv6) S[6]=S_LiBrH2O(T[6], Xl6/100)*(1-q[6])+q[6]*Entropy(Water, P=P[6], X=1)

h[16]=h[15]- b { try and error, EN: b = -2.5; UN,EN, b = -2.66} CALL Q_LIBR(h[16],Pm,x[15],2:q[16]*100,T[16],XI16,hI16,hv16) S[16]=S_LiBrH2O(T[16], XI16/100)*(1-q[16])+q[16]*Entropy(Water, P=P[16], X=1)

{Refrigerant Expansion Valves}

$$\begin{split} S[9] &= S[8] \\ S[9] &= Entropy(Water, T=T[9], X=0)^{(1-Q[9])+Q[9]*Entropy(Water, T=T[9], X=1)} \\ H[9] &= Enthalpy(Water, T=T[9], X=0)^{(1-Q[9])+Q[9]*Enthalpy(Water, T=T[9], X=1)} \end{split}$$

Appendix 7 Codes for the Endogenous and Unavoidable Endogenous Exergy Destruction Analysis in the Low Temperature Regenerator

{The state point assumptions for the cycles are based on Hu et al. (2011), Yin (2008), and Herold et al. (1996); only the differences from the codes in Appendix 1 are shown below.}

{Defined Delta Temperature Inputs}

DeltaThtrg=0 [K] DeltaTcon=0.8 [K] DeltaTevp=0 [K] DeltaTabs=0 [K] DeltaTltrg=8 [K] {DeltaTltrg=0.8 [K] {UN,EN}} Eff_HX=1

{Solution Expansion Valves}

 $h[6]=h[5]-a{try and error, EN: a = -0.88; UN,EN: a = -1.05}$ CALL Q_LIBR(h[6],PI,x[5],2:q[6]*100,T[6],XI6,hI6,hv6) S[6]=S_LiBrH2O(T[6], XI6/100)*(1-q[6])+q[6]*Entropy(Water, P=P[6], X=1)

h[16]=h[15]-b{try and error, EN: b = -2.46; UN,EN: b = -2.65} CALL Q_LIBR(h[16],Pm,x[15],2:q[16]*100,T[16],XI16,hI16,hv16) S[16]=S_LiBrH2O(T[16], XI16/100)*(1-q[16])+q[16]*Entropy(Water, P=P[16], X=1)

{Refrigerant Expansion Valves}

$$\begin{split} S[9] &= S[8] \\ S[9] &= Entropy(Water, T=T[9], X=0)^{(1-Q[9])+Q[9]*Entropy(Water, T=T[9], X=1)} \\ H[9] &= Enthalpy(Water, T=T[9], X=0)^{(1-Q[9])+Q[9]*Enthalpy(Water, T=T[9], X=1)} \end{split}$$

Appendix 8 Codes for the Endogenous and Unavoidable Endogenous Exergy Destruction Analysis in the High Temperature Heat Exchanger

{The state point assumptions for the cycles are based on Hu et al. (2011), Yin (2008), and Herold et al. (1996); only the differences from the codes in Appendix 1 are shown below.}

{Defined Delta Temperature Inputs}

DeltaThtrg=0 [K] DeltaTcon=0.8 [K] DeltaTevp=0 [K] DeltaTabs=0 [K] DeltaTltrg=0 [K] Eff_HX_LTHX=1 Eff_HX_HTHX=0.5

{Solution Expansion Valves}

h[6]=h[5]-1.09 {try and error, 50%, -1.09} CALL Q_LIBR(h[6],PI,x[5],2:q[6]*100,T[6],XI6,hI6,hv6) S[6]=S_LiBrH2O(T[6], XI6/100)*(1-q[6])+q[6]*Entropy(Water, P=P[6], X=1)

h[16]=h[15]- 0.45 {nd error, 50%, -0.77} CALL Q_LIBR(h[16],Pm,x[15],2:q[16]*100,T[16],XI16,hI16,hv16) S[16]=S_LiBrH2O(T[16], XI16/100)*(1-q[16])+q[16]*Entropy(Water, P=P[16], X=1)

{Refrigerant Expansion Valves}

$$\begin{split} S[9] &= S[8] \\ S[9] &= Entropy(Water, T=T[9], X=0)^{*}(1-Q[9])+Q[9]^{*}Entropy(Water, T=T[9], X=1) \\ H[9] &= Enthalpy(Water, T=T[9], X=0)^{*}(1-Q[9])+Q[9]^{*}Enthalpy(Water, T=T[9], X=1) \end{split}$$

Appendix 9 Codes for the Endogenous and Unavoidable Endogenous Exergy Destruction Analysis in the Low Temperature Heat Exchanger

{The state point assumptions for the cycles are based on Hu et al. (2011), Yin (2008), and Herold et al. (1996); only the differences from the codes in Appendix 1 are shown below.}

{Defined Delta Temperature Inputs}

DeltaThtrg=0 [K] DeltaTcon=0.8 [K] DeltaTevp=0 [K] DeltaTabs=0 [K] DeltaTltrg=0 [K] Eff_HX_LTHX=0.5 Eff_HX_HTHX=1

{Solution Expansion Valves} h[6]=h[5]- 0.01 {try and error, 50%, -0.01} CALL Q_LIBR(h[6],PI,x[5],2:q[6]*100,T[6],XI6,hI6,hv6) S[6]=S_LiBrH2O(T[6], XI6/100)*(1-q[6])+q[6]*Entropy(Water, P=P[6], X=1)

h[16]=h[15]- 2.65 {nd error, 50%, -2.65} CALL Q_LIBR(h[16],Pm,x[15],2:q[16]*100,T[16],XI16,hI16,hv16) S[16]=S_LiBrH2O(T[16], XI16/100)*(1-q[16])+q[16]*Entropy(Water, P=P[16], X=1)

{Refrigerant Expansion Valves}

$$\begin{split} S[9] &= S[8] \\ S[9] &= Entropy(Water, T=T[9], X=0)^*(1-Q[9])+Q[9]^*Entropy(Water, T=T[9], X=1) \\ H[9] &= Enthalpy(Water, T=T[9], X=0)^*(1-Q[9])+Q[9]^*Enthalpy(Water, T=T[9], X=1) \end{split}$$

$$\begin{split} S[19] &= S[18] \\ S[19] &= Entropy(Water, T=T[19], X=0)^*(1-Q[19])+Q[19]^*Entropy(Water, T=T[19], X=1) \\ H[19] &= Enthalpy(Water, T=T[19], X=0)^*(1-Q[19])+Q[19]^*Enthalpy(Water, T=T[19], X=1) \end{split}$$