# การใช้ประโยชน์จากความร้อนปล่อยทิ้งในโรงไฟฟ้าพลังความร้อนร่วมโดยเครื่องทำความเย็นแบบดูดซึม

นายเชาวรินทร์ หาดเจียง

วิทยานิพนธ์นี้เป็นส่วนหนึ่งของการศึกษาตามหลักสูตรปริญญาวิศวกรรมศาสตรมหาบัณฑิต
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ปีการศึกษา 2554

# UTILIZING LOW GRADE WASTE HEAT IN COMBINED-CYCLE POWER PLANT BY ABSORPTION CHILLER

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A Thesis Submitted in Partial Fulfillment of the Requirements
for the Degree of Master of Engineering Program in Chemical Engineering
Department of Chemical Engineering
Faculty of Engineering
Chulalongkorn University
Academic Year 2011

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|---|---|
| Thesis Title | UTILIZING LOW GRADE WASTE HEAT IN COMBINED-CYCLE POWER PLANT BY ABSORPTION CHILLER |
| By | Mr. Chaowarin Hadchiang |
| Field of Study | Chemical Engineering |
| Thesis Advisor | Assistant Professor Amornchai Arpornwichanop, D.Eng. |

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Accepted by the Faculty of Engineering, Chulalongkorn University in Partial Fulfillment of the Requirements for the Master's Degree

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เชาวรินทร์ หาดเจียง : การใช้ประโยชน์จากความร้อนปล่อยทิ้งในโรงไฟฟ้าพลังความร้อนร่วมโดยเครื่องทำความเย็นแบบดูดซึม. (UTILIZING LOW GRADE WASTE HEAT IN COMBINED-CYCLE POWER PLANT BY ABSORPTION CHILLER) อ. ที่ปรึกษาวิทยานิพนธ์หลัก : ผศ.ดร. อมรชัย อาภรณ์วิชานพ, 57 หน้า.

การเพิ่มประสิทธิภาพของโรงไฟฟ้าแบบดั่งเดิมได้รับความสนใจเนื่องจากปัญหาวิฤตพลังงานและภาวะโลกร้อน การนำความร้อนปล่อยทิ้งอุณหภูมิต่ำมาใช้ประโยชน์เป็นแนวทางที่น่าสนใจเพื่อเพิ่มประสิทธิภาพของการผลิตพลังงานไฟฟ้า ปัจจุบันเครื่องทำความเย็นแบบดูดซึมได้ถูกใช้เพื่อนำความร้อนจากแหล่งพลังงานความร้อนอุณหภูมิต่ำ เช่น ไอน้ำปลดปล่อย และไอน้ำความดันต่ำ งานวิจัยนี้มีวัตถุประสงค์เพื่อศึกษาการใช้ก๊าซทิ้งจากหม้อต้มไอน้ำในโรงไฟฟ้า ซึ่งเป็นแหล่งความร้อนอุณหภูมิต่ำ สำหรับเครื่องทำความเย็นแบบดูดซึมในการผลิตน้ำเย็น การจำลองกระบวนการทำความเย็นแบบดูดซึมที่ประกอบด้วย เครื่องผลิตไอน้ำ เครื่องควบแน่น เครื่องระเหย และ เครื่องดูดซึม ถูกดำเนินการโดยใช้โปรแกรมจำลองกระบวนการโดยพิจารณากระบวนการถ่ายเทความร้อน ลิเทียมโบรมายด์ (LiBr) ถูกใช้เป็นสารดูดซึมในเครื่องทำความเย็นแบบดูดซึม สมรรถนะของกระบวนการทำความเย็นแบบดูดซึมถูกวิเคราะห์สภาวะที่เหมาะสมสำหรับการดำเนินงานเครื่องทำความเย็นแบบดูดซึมในการความร้อนเหลือทิ้งกลับมาใช้ประโยชน์ และการเปรียบเทียมเชิงเศรษฐศาสตร์ได้ถูกแสดงไว้ในงานวิจัยนี้ด้วย

ภาควิชา.......วิศวกรรมเคมี.......ลายมือชื่อนิสิต.......................................

สาขาวิชา.......วิศวกรรมเคมี.......ลายมือชื่อ อ.ที่ปรึกษาวิทยานิพนธ์หลัก...............

ปีการศึกษา.......2554.......

# # 5371475821 : MAJOR CHEMICAL ENGINEERING

KEYWORDS : ABSORPTION CHILLER / LOW GRADE WASTE HEAT / EXHAUST GAS/ POWER PLANT / COEFFICIENT OF PERFORMANCE

CHAOWARIN HADCHIANG : UTILIZING LOW GRADE WASTE HEAT IN COMBINED-CYCLE POWER PLANT BY ABSORPTION CHILLER. ADVISOR : ASST. PROF. AMORNCHAI ARPORNWICHANOP, D.ENG., 57 pp.

Performance improvement of a conventional power plant has been received much attention due to energy crisis and global warming problems. Utilization of low-temperature waste heat is considered a promising alternative to enhance the efficiency of power generation. Presently, an absorption chiller has been applied to recover heat from low-temperature energy sources such as bleeding steam and low pressure steam. The aim of this work is to study the use of an exhaust gas from a steam boiler in a power plant, as the low temperature heat source, for an absorption chiller to produce chilled water. Simulations of the absorption chiller process consisting of generator, absorber, condenser and evaporator, were performed using a flowsheet simulator taking into account detailed heat transfer processes. LiBr was represented an absorbent in the absorption chiller. The performance of the absorption chiller process was analyzed. The suitable condition for the absorption chiller operation to recover the waste heat and economic comparison are also identified.

Department : Chemical Engineering Student's Signature ..............................

Field of Study : Chemical Engineering Advisor's Signature ..............................

Academic Year : 2011

# ACKNOWLEDGEMENTS

I would like to express the greatest gratitude to my advisor, Assistant Professor Amornchai Apornwichanop, for invaluable suggestions and guidance throughout this study. Without his support, this research can not be completed. I would be also grateful to Assistant Professor Anongnat Somwangthanaroj as the chairman of the thesis committee and Assistant Professor Soorathep Kheawhom and Yaneeporn Patcharavorachot as the member of the thesis committee, for their kind suggestions.

Support from the Computational Process Engineering Research Group, Department of Chemical Engineering, Chulalongkorn University is gratefully acknowledged. In addition, I wish to thank all my friends at Department of Chemical Engineering for their assistance and friendship.

Finally, I would like to express my highest gratitude to my parents who always take care of me all the times, for their support and encouragement. The success of this graduate study is devoted to my parents.

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# CHAPTER 1

# INTRODUCTION

Due to industrial development and population growth, there has been an enormous increase in energy demand. In addition, the emission of pollutants such as $CO_2$, CO, $SO_2$ and $NO_x$ from a fuel combustion process causes a major environmental problem. As a result, the development of new and renewable sources of energy and the efficient usage of energy have become important issues.

When considering industrial plants, a heat recovery within a production process should be considered so as to improve the process effeicient. In general, there are two kinds of waste heat produced in the process, i.e., high-temperature heat and low temperature heat (temperature below 200 ºC). The high temperature heat is found in various sources and can be used in many applications; for example, a high-pressure steam is used to drive steam turbines and a combustion gas is employed to drive gas turbine generators or to generate steam. Compared with the high-temperature heat, the low-temperature heat seems to be less useful because it has low enthalpy. Therefore, it is commonly released to atmosphere or decrease temperature by cooling water. This type of heat is also known as "a low-grade waste heat." The low-grade waste heat can be widely found in production proessess. In petrochemical and refinery processes, air from air compressors is an example of the low-grade heat, which is normally cooled down by an inter cooloer using cooling water. Another low-temperature heat sourece is an overhead gas from distillations or absorbers which is condensed by overhead condensers. For a combined cycle power plant, steam from steam turbines and exhaust gas from steam boiler are also found to be low-grade heat sources.

There are several ways to utilize a low-grade waste heat (Burer et al., 2003; Xiling et al., 2003; Jaruwongwittaya and Chen, 2010; Yin et al., 2010; Somer et al., 2011). Xiaojun et al. (2009) investigated the use of the low-grade waste heat in a combine cycle power plant for the Rankine system to produce additional power. In this system, the ammonia-water mixture was used as a working fluid. Heat and mass balances were performed to analyze the system. The result showed that the proposed combined cycle has good performance with the net electrical efficiency of 33%. The power output is equal to 1.25 MWh per kg of the ammonia–water mixture. About 0.2 MW of electrical power for operating sea water pumps can be saved. Parametric analyses of the proposed combined cycle were conducted to evaluate the effect of key factors on its performance. The simulation result revealed that the maximum net electrical efficiency can be obtained when the inlet pressure of ammonia turbine increases and the peak value increases with an increase in the ammonia mass fraction. Hisazumi et al. (1998) presented the utilization of low-grade waste heat, that includes the steam-turbine's condensate and the exhaust gas from the waste-heat recovery boiler, as heat sources. The system consisted of the Rankine cycle using a freon mixture and liquidfied natural gas (LPG) as working fluid. In order to find out the optimal condition of the system, several factors such as gas turbine combustion pressure, steam pressure, condensing temperature in the combined cycle, composition of freon mixture and natural gas vaporizing pressure, were evaluated by process simulation. The results of these studies showed that in the total system, about 400 kWh can be generated by vaporizing one ton of LNG, including about 60 kWh/LNG ton recovered from the LNG cold energy when supplying NG in 3.6 MPa. About 8.2 MWh can be produced by using one ton of LNG as fuel, compared with about 7 MWh by the conventional combined system. A net efficiency of over 53 %HHV could be achieved by the proposed system. In the case of the LNG terminal receiving five million tons of LNG per year, this system can generate 240 MW and reduce the power of the sea water pump by more than 2MW.

In addition, the low-grade waste heat can be used in an absorption refrigeration system, which is known as an absorption chiller, and an absorption heat pump system. The absorption chiller system composes of four main heat exchangers (i.e., generator, condenser, evaporator and absorber) in which water is a working fluid and LiBr is an absorbent. The system requires the low-grade waste heat as a driving force and the cooling water to remove heat from the absorber and the condenser.

Commonly, the absorption chiller uses for air conditioning (Bahador et al., 2011) because chilled water is produced for use in a Heat Ventilating and Air Conditioning system (HVAC). For the absorption heat pump, it is often used in the winter season to heat air for heaters. The use of absorption chiller and absorption heat pump systems presents many advantages. Firstly, there is no need to use electricity. Secondly, the systems do not require frequently maintenance because they have less moving parts. Finally, they can use water as working fluid and are thus the friendlily environmental systems (Boonnasa et al., 2006). As a result, many researchers have focused on the study and development of absorption chiller and absorption heat pump systems. Kevin et al. (2000) investigated the theoretical efficiency of single and half effect of the absorption chiller based on mass and energy balances. Heat transfer coefficients for the inside and outside tubes of each component such as generator, absorber, condenser and evaporator, were considered. Their study showed the advantage and disadvantage of the single and half effects of absorption chiller. For the single effect, it provides a higher coefficient of performance (COP) but a narrower temperature operating range. The advantage of the half effect is the capability of using a lower temperature hot water; however, the presence of lower COP increases operating cost and requires a larger cooling tower. Boonnasa et al. (2006) presented the utilization of absorption chiller to improve the capacity of the combined cycle power plant by cooling down the temperature of intake air to around 15 ºC (100% RH) before entering the air compressor of a gas turbine. Exhaust steam from the generator is a low-grade waste heat which is used for driving the absorption. The result showed that the power output of the gas turbine and and the combined cycle power plant increase about 10% and 6.24%, respectively. Bahador et al. (2011) presented the experiment and simulation of an absorption heat pump with a cooling capacity of 14 kW. The results were obtained from the variation of the most five influential parameters by considering the machine performance, as described by the coefficient of performance (COP) and cooling capacity. Moreover, the model of the $H_2O$–LiBr absorption heat pump was developed. It was shown that the simulation results is in good agreement with experimental results. The cooling water flow rate and temperature significantly affected on the performance of the machine. It also reported that the absorption chiller has wide applications for utilizing a low-grade waste heat with good efficiency.

An exhaust gas from steam boilers in a combined heat and power system is considered a low-grade waste heat that can be applied to an absorption chiller. This study presents an analysis of the use of the exhaust gas as the heat source for the absorption chiller to produce chilled water. Simulations of the absorption chiller process consisting of generator, absorber, condenser and evaporator, are performed by using a process simulator. The feasibility study and performance of the absorption chiller process is investigated with respect to key operating parameters. Furthermore, the suitable condition for the absorption chiller operation to recover the waste heat is identified. At last, the cost of absorption chiller is compared with a convention chiller to ensure that the absorption chiller is suitable for implementing to industries.

**1.1 Objective of this work**

The aim of this work is to investigate the use of an exhaust gas from a steam boiler in a combined-cycle power plant as the low grade waste heat for driving an absorption chiller to produce chilled water. The performance of the absorption chiller process is analyzed with respect to key operating parameters. The suitable condition for the absorption chiller operation to recover the waste heat is also identified.

**1.2 Scope of Research**

1. Simulate an absorption chiller process consisting of a generator, absorber, condenser and evaporator, by using an exhaust gas of a steam boiler as a heat source. LiBr is used as an absorbent in the absorption chiller.

2. Simulate the absorption chiller process using a mixed gas and steam as a heat source.

3. Analysis the performance of the absorption chiller process in terms of a coefficient of performance (COP)

4. Compare the cost of the absorption chiller with a conventional chiller.

# CHAPTER 2

# LITERATURE REVIEWS

This chapter presents a literature review on the study related to the recovery of a low-grade waste heat. An absorption chiller process used to recover the low grade waste heat is also presented.

## 2.1 Recovery of low-grade waste heat

The consumption of fossil fuels has been being increased according to the demand for energy and electricity. This factor results in serious energy shortage and environmental pollution. In order to save energy and protect the environment, greater attention has been paid to the utilization of low grade waste heat to generate power in recent years, leading to an efficient energy usage (Xiaojun and Defu, 2009). Since the temperature of low-grade waste heat sources is less than 200 ºC, it is especially important that suitable technologies for heat recovery at such a temperature range are implemented. Heat recovery options can be broadly classified into three strategies (Carolyn, 2009):

- Recycling energy back into the process
- Recovering energy for other on-site uses
- Using energy to generate electricity in combined heat and power systems

Heat recovery technologies may be also classified as either passive or active methods. The passive heat recovery makes use of various heat exchangers types to transfer heat from a higher temperature source to a lower temperature stream. Passive heat recovery technologies do not require significant mechanical or electrical input for their operation, except for auxiliary equipment such as pumps or fans. On the other hand, the active heat recovery technologies require the input of energy to "upgrade" the waste heat to provide a higher temperature or produce electricity. These technologies include industrial heat pumps and combined heat and power systems.

- The most appropriate technology depends largely on the temperature of heat sources as: Passive heat recovery: Temperatures greater than about 93 ºC
- Industrial closed-cycle mechanical heat pumps: Temperatures less than about 93 ºC
- Absorption chillers and heat pumps: Temperatures between 93 ºC and 200 ºC
- Rankine cycle for a combined heat and power system (CHP): Temperatures between 150 ºC to 400 ºC
- Kalina cycle for CHP: Temperatures between 121 ºC to 540 ºC

Xiaojun et al. (2009) proposed the system consisting of the Rankine cycle, which uses a ammonia–water mixture as the working fluid, and the LNG power generation cycle, to convert the remaining energy in the low-temperature waste heat into power. In this system, the low-temperature waste energy can be recovered from the exhaust of industrial processes and power plants. The steady-state component models were used to determine the performance of the proposed combined system. Each component in the system was modeled by taking into account mass, energy and species balances. The component models were composed of feed pump, heat exchanger, separator, ammonia vapor turbine, condenser, condensate pump, mixer, LNG pump and LNG turbine. The schematic diagram of the proposed system is shown in Fig. 2-1.

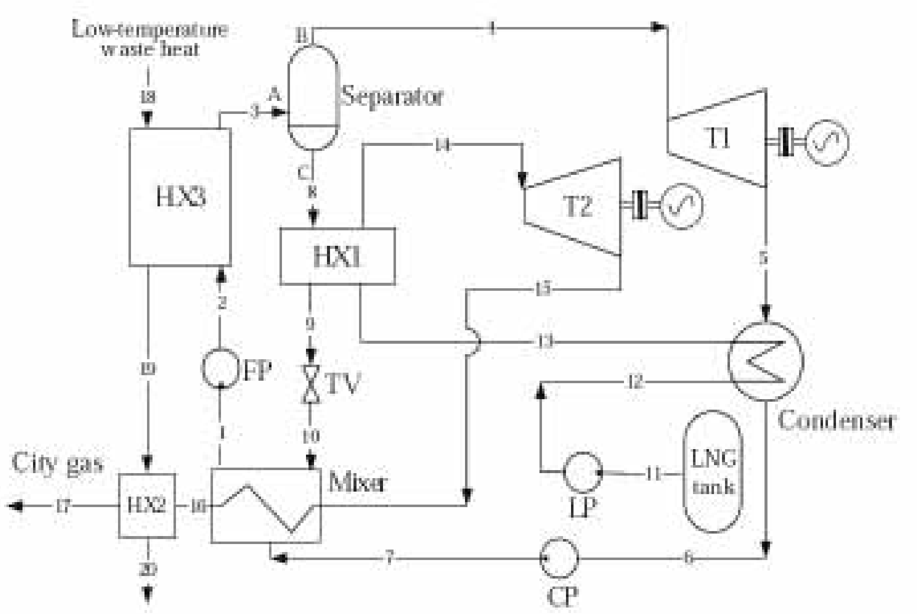


**Fig. 2-1** Schematic diagram of the proposed combined power cycle (Xiaojun et al., 2009).

It is noted that the symbols, demonstrating in Fig.2-1, can be explained as follows: A is the basic solution of ammonia–water, B is the ammonia vapor, C is the weak solution of ammonia–water, CP is the condensate pump, FP is the feed pump, HX is the heat exchanger, LP is the LNG pump, T1 is the ammonia turbine, T2 is the LNG turbine, TV is the throttle valve. The results showed that the net electrical efficiency is 33.28%. The waste heat recovery efficiency of the proposed combined cycle can be reached to 86.57%. About 16.36 kg $s^{-1}$ of natural gas at temperature of 35.8 °C and pressure of 0.3 MPa can be delivered to the natural gas supplying system and no less than 0.2 MW of electrical power for operating sea water pumps can be saved.

Due to the large LNG consumption in Japan, Hisazumi et al. (1998) proposed the high efficiency LNG power generation system based on a new concept in which the cold energy is fully utilized without discarding into the sea. The system was composed of the combined cycle and the LNG power-generation plant. A Rankine cycle using a freon mixture, natural-gas Rankine cycle and a combined cycle with gas and steam turbines were considered. The heat sources for this system are the latent heat from the steam-turbine condenser and the sensible heat of exhaust gas from the waste heat recovery boiler. The performances of the studied system were evaluated by using process simulator. To improve the total efficiency of the system, a simulation was conducted to evaluate several factors, such as the composition of the freon mixture, natural gas send-out pressure, combustion pressure, steam inlet pressure and steam-condensing temperature of the combined cycle. The simulation results obtained in this study revealed that this system can generate a power of about 240 MW when 600 tons of LNG is used in an hour. With the elimination of about 24,000 tons per hour of sea water, which is used for vaporizing 600 tons/h of LNG in the conventional system, no less than 2 MW of electric power for operating sea water pumps can be saved.

## 2.2 Absorption chiller technology

Absorption technology uses thermal energy rather than mechanical shaft energy for its operation. Absorption units are referred to as absorption chillers when they are used to provide chilled water for air conditioning or refrigeration using waste heat as an energy source. The same technology is referred to as Type-I absorption heat pumps when they are used for heating. Absorption chillers/heaters are designed to switch between heating and cooling modes.

Absorption chillers and Type I heat pumps are most economical when using a waste heat source with the temperature ranging from 93 ºC and 200 ºC. Its efficiency would be enhanced when there is a need for heating and cooling loads at the same time. The low grade waste heat source can be a low pressure steam and hot gaseous or liquid streams. The operating temperature of absorption units varies depending on the working fluid. Lithium bromide-based chillers can cool down to about 4 ºC, while ammonia-based chillers can cool down to about -3 ºC. The Type I heat pump can have the output temperature up to about 94 ºC and achieve the maximum temperature lift up to about 50 ºC. Another less common type of absorption technology is the Type-II absorption heat pump, also known as a heat transformer. Heat transformers take waste heat at an intermediate temperature that is too low to be useful and upgrade to a higher useable temperature. A portion of the heat must also be rejected to a lower temperature heat sink. Up to 50% of the waste heat can be upgraded. Heat transformers can have the output temperatures up to about 150 ºC. Lithium bromide based units can achieve temperature lifts of 7.2 ºC to 32 ºC from waste heat sources at temperatures of 80 ºC to 93 ºC. Absorption technology has been received a growing interest in waste heat recovery. The capacity of absorption chillers and chiller/heaters is extended from 3 tons to more than 1,500 tons (Caloryn, 2009).

Boonnasa et al. (2006) proposed the improvement of a power plant. Since the ambient temperature all the year in Bangkok is very high, leading to a lower efficiency of the system substantially, a steam absorption chiller was introduced to cool a compressor intake air to the desired condition. The absorption chiller has a certain advantage because it does not need additional power for compressors as the mechanical chiller does. Furthermore, it has a very few moving parts and thus, a standby unit is not necessary. Although the absorption chiller may use a fraction of low-pressure steam, this energy requirement is less than that used by a mechanical chiller. They suggested a way to improve the performance of one gas turbine unit using one lithium-bromide absorption chiller. The chiller was used to produce chilled water by using the heat input from a low pressure steam (about 0.6–0.8 MPa) in a heat recovery steam generator (HRSG). The storage tank that stores chilled water was installed between the absorption chiller and the compact heat exchanger (HE). Chilled water pumps was separated into two groups. The first group was a primary pump which circulates chilled water between the storage tank and the AC. The second group was a secondary pump which circulates chilled water between the storage tank and the HE. The results showed that the fuel consumption of the improved system was increased by about 0.4 kg/s, while the power output of the gas turbine increases by around 10.8 MW (10.16%) and the power output of the steam turbine is decreased about 1.25 MW (2.85%). The economic analysis was based on the retail prices of electricity and demand charges for the time of use tariff (TOU): on-peak = 0.06534 US$/kWh; off-peak = 0.029315 US$/kWh

Kevin et al. (2000) examined the performance of a current single-effect absorption chiller and determined what design changes are needed so that the chiller can operate using low temperature hot water. The second phase of the study involves designing a half-effect cycle to be operated on low-temperature hot water. The performance characteristics of the half-effect cycle are examined by changing the hot water flow rate, the cooling water temperature and the flow arrangements of the cooling and heat source flows. The half-effect and single-effect cycles are compared to see which one is more competitive for a specified fuel source temperature. A detailed computer model was written for the single and half effect cycles based on heat transfer coefficients for the inside and outside tubes of each component (generator, absorber, condenser and evaporator), energy, mass and salt balances, and rate equations. The cooling tower was modeled using the analogy approach, calibrated and validated with performance data from a cooling tower manufacturer. The advantage of the single effect over the half effect cycle is the higher COP but the single effect has a narrower temperature operating range. The half effect has the capability of using lower hot water temperature but the lower COP increases operating cost and requires investing a larger cooling tower. The results showed that based on the hot water temperature of 107 ºC and flow rate of 7,570 L/min, the cost to maintain capacity starts to change for the single effect cycle. At the temperature of 99 ºC the cost to maintain capacity increases rapidly. The half effect cycle can maintain capacity at hot water temperature and flow rate as low as 82 ºC and 7,570 L/min, respectively, without a large increase in capital investment. The capital cost for the half effect chiller system is 57 $/kW, which is higher that the single effect, using hot water temperatures above 93 ºC. The single effect cycle can only be competitive with an electric centrifugal chiller if the heat source is free or a combination of high electrical cost with a low cost of heat.

Bahador et al. (2011) studied an absorption heat pump because it can recover useful energy from waste energy. This unit can be widely used for various applications and are environmentally friendly because working fluids used are not toxic and do not cause ozone depletion. In general, absorption chiller and heat pump can be run by low driving temperatures and generally used in air conditioning. They can be effectively integrated to load shifting during summer peak electricity demand. However, they are rarely commercially available with a small cooling capacity. The aim of this study was to design the absorption cycle. Experimental results were used to validate the developed model, which requires minimal information about the working fluid and characteristics of the heat exchangers. The proposed model is suitable for rapid and reliable simulation of an existing absorption cycle, evaluation and comparison of different working fluids and design and dimensioning of new absorption heat pump.

**2.3 Use of exhaust gas as a heat source.**

Exhaust gas from steam boilers in a combined heat and power system is considered a low grade waste heat, which can be applied to an absorption chiller. There are many studies focused on the use of the exhaust gas to improve the efficiency of steam production.

Vhutshilo et al. (2011) studied the integrated gasification combined cycle (IGCC) process as an innovative electrical power generation system that provides a large share of the future world's energy needs. The combination of these two technologies can divide the IGCC into two subsystems: the gas-side subsystem and the steam-side subsystem. These two subsystems are integrated by a gas cooler (often referred to as a boiler) and a heat recovery steam generator (HRSG). The purpose of their system is to recover a low potential heat from the gas turbine exhaust stream of the IGCC plant by using a contact economizer system (CES) to further improve the overall efficiency of the IGCC. Most previous studies were concentrated on improving or optimizing the performance of components of the gas-side subsystem, i.e., the coal gasification unit, the gas turbine and the air separation unit. Only a few researchers have focused on improving the steam-side subsystem and optimizing the use of energy available within the system. The proposed method in their work is divided into two parts: (1) determining the maximum boiler feedwater temperature and (2) applying Mickley's graphical technique for dehumidification to demonstrate how the aforementioned temperature is achieved. The result showed that when applied to a case study on the heat-integrated design of the Elcogas plant, the developed method is capable of increasing the gross efficiency.

In conclusion, it can be seen that there are several methods to utilize worthy low grade waste heat. Among the various technologies, the absorption chiller offers several advantages that can be applied to low grade waste heat. Considering the source of low temperature waste heat, it is found that the use of exhaust gas has been received much attention because it can increase heat transfer between the exhaust gas and the boiler feed water. Interestingly, there are a few previous publications focused on the utilization of exhaust gas as the low temperature heat source for absorption chiller. As a consequence, this study will be concentrated on the investigation of utilizing an exhaust gas as the low grade waste heat for driving an absorption chiller. Simulation of absorption chiller is performed by using a process simulator in order to examine the system performance with respect to key operating conditions of the absorption chiller.

# CHAPTER 3

# THEORY

This chapter presents a basic knowledge of a refrigeration system, absorption chiller and heat exchanger sizing.

## 3.1 Refrigeration system

### 3.1.1 General

A refrigeration is a process in which heat is transferred from a lower- to a higher-temperature level by doing work on a system. In some systems, heat transfer is used to provide the energy to drive the refrigeration cycle. All refrigeration systems are heat pumps ("pumps energy from a lower to a higher potential"). The term heat pump is mostly used to describe refrigeration system applications where heat rejected to the condenser is of primary interest. (Robert and Don, 1999).

A basic principle of the refrigeration is derived from thermodynamics. Since the aim is to understand a refrigeration process, refrigeration cycles are of the crucial interest. The Carnot refrigeration cycle is reversible and consists of adiabatic (isentropic due to reversible character) compression (1-2), isothermal rejection of heat (2-3), adiabatic expansion (3-4) and isothermal addition of heat (4-1). The temperature-entropy diagram is shown in Fig.3-1. The Carnot cycle is an unattainable ideal which serves as a standard for comparison and it provides a convenient guide to the temperatures that should be maintained to achieve the maximum effectiveness.

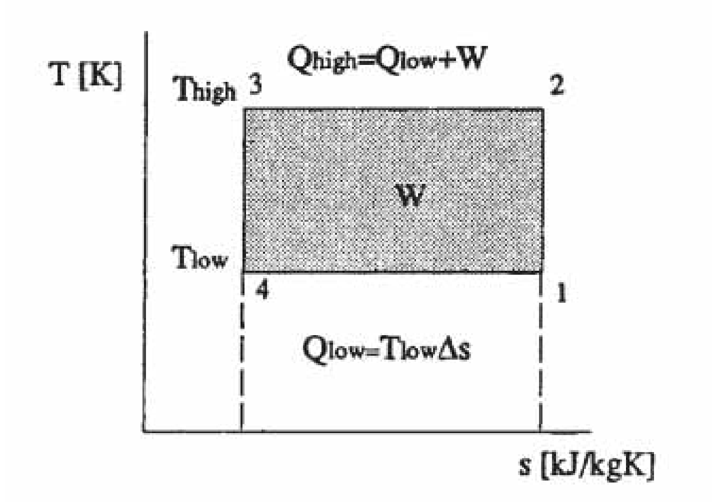


**Fig. 3-1** Temperature-entropy diagram of the Carnot cycle (Robert and Don, 1999).

A key parameter used to measure the system performance is a coefficient of performance (COP). For refrigeration applications, COP is the ratio of heat removed from the low-temperature level ($Q_{low}$) to the energy input (W):

$$\mathrm{COP_R} = \frac{Q_{low}}{W} \tag{1}$$

For the heat pump (HP) operation, heat rejected at the high temperature ($Q_{high}$) is considered as follows:

$$COP_{HP} = \frac{Q_{high}}{W} = \frac{Q+W}{W} = COP_R + 1 \tag{2}$$

For the Carnot cycle (where $\Delta Q = T\Delta s$), the COP for the refrigeration application becomes (note that $T$ is the absolute temperature (K)):

$$COP_R = \frac{T_{low}}{T_{high} - T_{low}} \tag{3}$$

and for the heat pump application is:

$$COP_{HP} = \frac{T_{high}}{T_{high} - T_{low}} \tag{4}$$

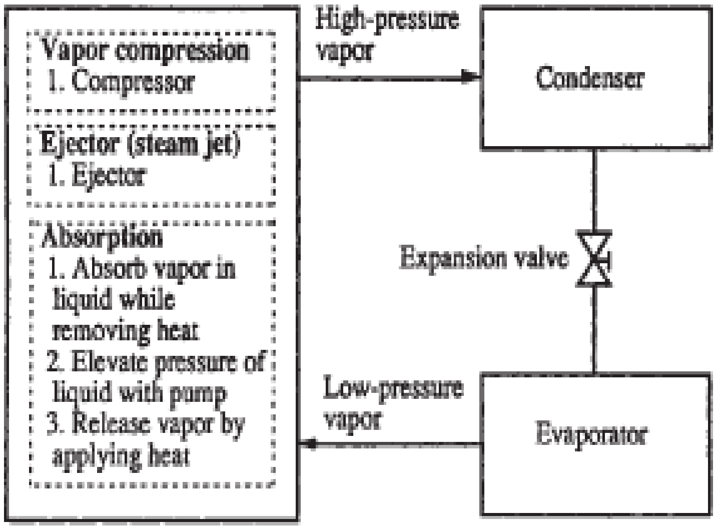


**Fig. 3-2** Methods of transforming low-pressure vapor into high-pressure vapor in refrigeration systems (stoecker, refrigeration and air conditioning) (Robert and Don, 1999)**.**

The COP in real refrigeration cycles is always less than for the ideal (Carnot) cycle and there is constant effort to improve the actual refrigeration cycles to achieve this ideal value.

### 3.1.2 Type of refrigerations

The main components of a refrigeration cycle are a condenser and an evaporator. Each refrigeration type has a difference in the way compression is being done. Fig.3-2: for vapor compression using mechanical work (in compressor) for ejector use stream jet to increase the pressure and for absorption and desorption use pump to increase solution pressure.

### 3.1.3 Mechanical refrigeration (vapor-compression system)

The most widely used refrigeration principle is based on a vapor-compression cycle. Isothermal processes are realized through isobaric evaporation and condensation in the tubes. Standard vapor compression refrigeration cycle (counterclockwise Rankine cycle) is marked in Fig. 3-3 by 1, 2, 3 and 4. Work that could be obtained from a turbine is small and thus the turbine is substituted by an expansion valve. For the reason of proper compressor function, a wet compression is substituted for an compression of dry vapor. Although the T-s diagram is very useful for thermodynamic analysis, the pressure enthalpy diagram is used much more in refrigeration practice due to the fact that both evaporation and condensation are isobaric processes so that heat exchanged is equal to enthalpy difference, $\Delta Q = \Delta h$. For the ideal case, isentropic compression, the work could be also presented as enthalpy difference $\Delta W = \Delta h$. The vapor compression cycle (Rankine) is presented in Fig. 3-3 in the P-h coordinates.

Fig. 3-4 presents the actual versus standard vapor-compression cycles. In reality, flow through the condenser and evaporator must be accompanied by pressure drop. There is always some subcooling in the condenser and superheating of the vapor entering the compressor suction line, both due to continuing process in the heat exchangers and the influence of the environment. Subcooling and superheating are usually desirable to ensure only liquid enters the expansion device. Superheating is recommended as a precaution against droplets of liquid being carried over into the compressor.

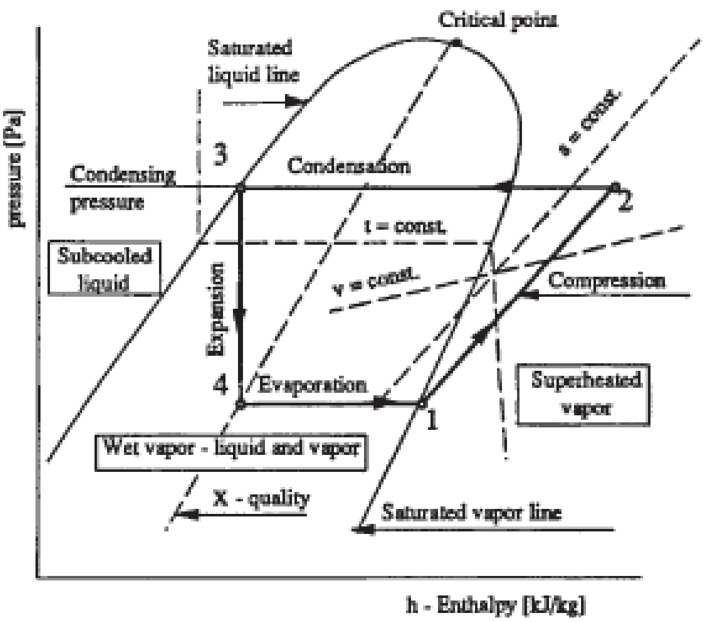


**Fig. 3-3** P-h diagram for vapor-compression cycle (Robert and Don, 1999).

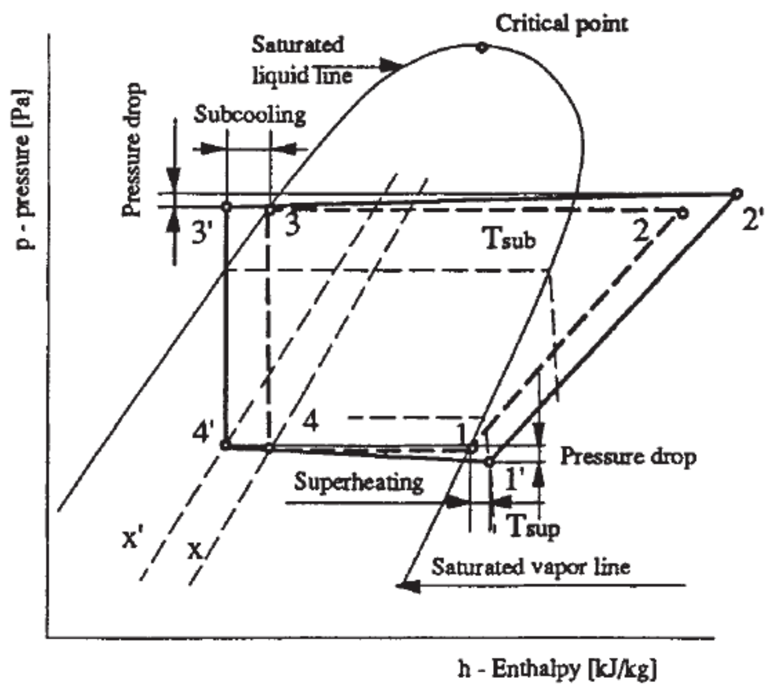


**Fig. 3-4** Actual vapor-compression cycle compared with standard cycle (Robert and Don, 1999),

There are many ways to increase the cycle efficiency (COP). Some of them are better suited to one, but not for the other refrigerant. Sometimes, for the same refrigerant, the impact on COP could be different for various temperatures. One typical example is the use of a liquid-to-suction heat exchanger (Fig.3-5).

The suction vapor coming from the evaporator could be used to subcool the liquid from the condenser. Graphic interpretation in T-s diagram for such a process is shown in Fig.3-6. The result of the use of suction line heat exchanger is to increase the refrigeration effect ($\Delta Q$) and to increase the work ($\Delta W$). The change in COP is then:

$$\Delta COP = COP' - COP = \frac{Q + \Delta Q}{(P + \Delta P) - Q / P} \tag{5}$$

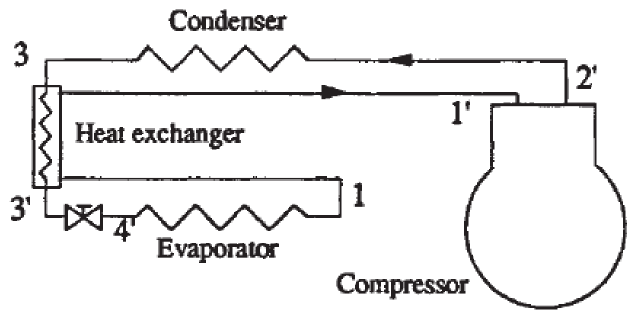


**Fig. 3-5** Refrigeration system with a heat exchanger to subcool the liquid from the condenser (Robert and Don, 1999).

When dry, or superheated, vapor is used to subcool the liquid, the COP in the R12 system will increase, and decrease the COP in NH3 systems. For R22 system, it could have increase or decrease, depending on the operating regime. Generally, this measure is advantageous (COP is improved) for fluids with high, specific heat of liquid (less-inclined saturated-liquid line on the p-h diagram), small heat of evaporation $h_{fg}$, when vapor-specific heat is low (isobars in superheated regions are steep), and when the difference between evaporation and condensation temperature is high. Measures to increase COP should be studied for every refrigerant. Sometimes the purpose of the suction-line heat exchanger is not only to improve the COP, but to ensure that only the vapor reaches the compressor, particularly in the case of a malfunctioning expansion valve.

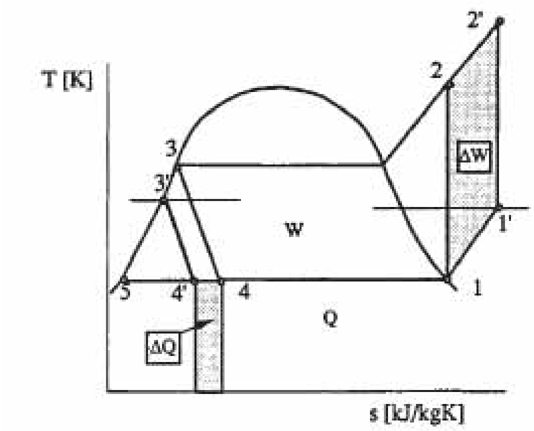


**Fig. 3-6** Refrigeration system with a heat exchanger to subcool the liquid from the condenser (Robert and Don, 1999).

The system shown in Fig. 3-5 is direct expansion where dry or slightly superheated vapor leaves the evaporator. Such systems are predominantly used in small applications because of their simplicity and light weight. For the systems where efficiency is crucial (large industrial systems), recirculating systems (Fig.3-7) are more appropriate.

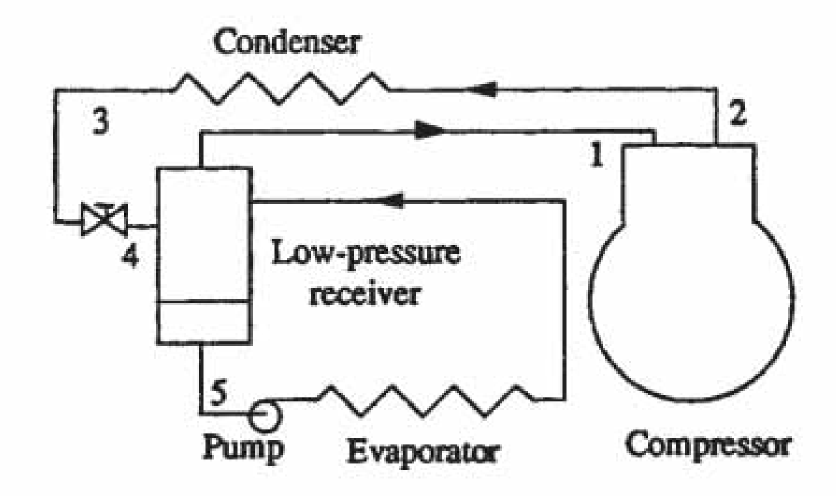


**Fig. 3-7** Recirculation system (Robert and Don, 1999).

Ammonia refrigeration plants are almost exclusively built as recirculating systems. The main advantage of recirculating versus direct expansion systems is better utilization of evaporator surface area. It is clear that heat-transfer characteristics will be better if the outlet quality is lower than 1. Circulation could be achieved either by pumping (mechanical or gas) or using gravity (thermosyphon effect: density of pure liquid at the evaporator entrance is higher than density of the vapor-liquid mixture leaving the evaporator). The circulation ratio (ratio of actual mass flow rate to the evaporated mass flow rate) is higher than 1 and up to 5. Higher values are not recommended due to a small increase in heat-transfer rate for a significant increase in pumping costs.

### 3.1.4 Other refrigeration systems

Absorption refrigeration systems two main absorption systems are used in industrial application: lithium bromide-water and ammonia-water. Lithium bromide-water systems are limited to evaporation temperatures above freezing because water is used as the refrigerant, while the refrigerant in an ammonia-water system is ammonia and consequently it can be applied for the lower-temperature requirements. Single-effect indirect-fired lithium bromide cycle is shown in Fig.3-8. The machine consists of four major components: *Evaporator* is the heat exchanger where refrigerant (water) evaporates (being sprayed over the tubes) due to low pressure in the vessel.

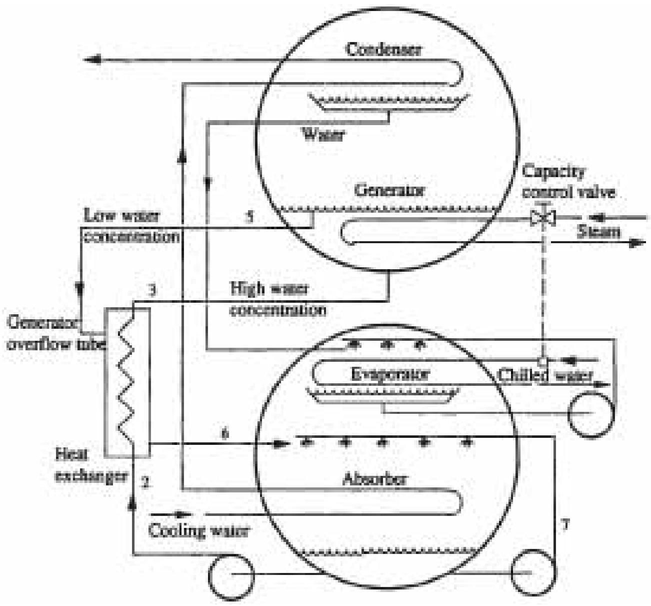


**Fig. 3-8** Two-shell LiBr-water cycle chiller (Robert and Don, 1999).

Evaporation chills water flow inside the tubes that bring heat from the external system to becooled.

*Absorber* is a component where strong absorber solution is used to absorb the water vapor flashed in the evaporator. A solution pump sprays the lithium bromide over the absorber tube section. Cool water is passing through the tubes taking refrigeration load, heat of dilution, heat to cool condensed water, and sensible heat for solution cooling. Heat exchanger is used to improve efficiency of the cycle, reducing consumption of steam and condenser water.

*Generator* is a component where heat brought to a system in a tube section is used to restore the solution concentration by boiling off the water vapor absorbed in the absorber.

*Condenser* is an element where water vapor, boiled in the generator, is condensed, preparing pure water (refrigerant) for discharge to an evaporator.

Heat supplied to the generator is boiling weak (dilute) absorbent solution on the outside of the tubes. Evaporated water is condensed on the outside of the condenser tubes. Water utilized to cool the condenser is usually cooled in the cooling tower. Both condenser and generator are located in the same vessel, being at the absolute pressure of about 6 kPa. The water condensate passes through a liquid trap and enters the evaporator. Refrigerant (water) boils on the evaporator tubes and cools the water flow that brings the refrigeration load. Refrigerant that is not evaporated flows to the recirculation pump to be sprayed over the evaporator tubes. Solution with high water concentration that enters the generator increases in concentration as water evaporates.

The resulting strong, absorbent solution (solution with low water concentration) leaves the generator on its way to the heat exchanger. There the stream of high water concentration that flows to the generator cools the stream of solution with low water concentration that flows to the second vessel. The solution with low water concentration is distributed over the absorber tubes. Absorber and evaporator are located in the same vessel, so the refrigerant evaporated on the evaporator tubes is readily absorbed into the absorbent solution. The pressure in the second vessel during the operation is 7 kPa (absolute). Heat of absorption and dilution are removed by cooling water (usually from the cooling tower). The resulting solution with high water concentration is pumped through the heat exchanger to the generator, completing the cycle.

Heat exchanger increases the efficiency of the system by preheating, that is, reducing the amount of heat that must be added to the high water solution before it begins to evaporate in the generator.The absorption machine operation is analyzed by the use of a lithium bromide-water equilibrium diagram, as shown in Fig.3-9. Vapor pressure is plotted against the mass concentration of lithium bromide in the solution. The corresponding saturation temperature for a given vapor pressure is shown on the left-hand side of the diagram. The line in the lower right corner of the diagram is the crystallization line. It indicates the point at which the solution will begin to change from liquid to solid, and this is the limit of the cycle. If the solution becomes over concentrated, the absorption cycle will be interrupted owing to solidification, and capacity will not be restored until the unit is desolidified. This normally requires the addition of heat to the outside of the solution heat exchanger and the solution pump.

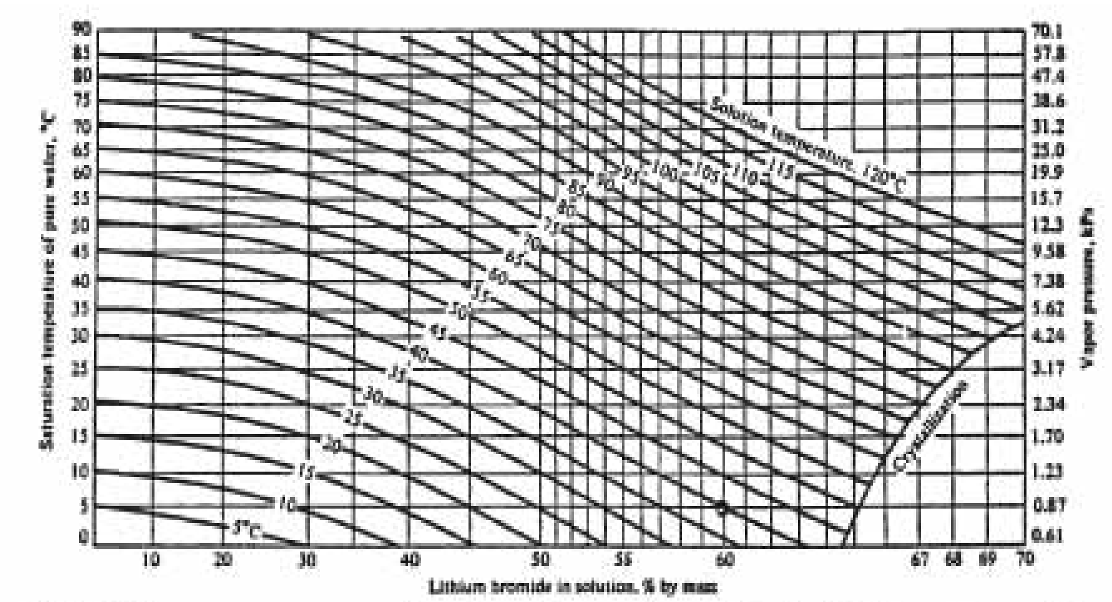


**Fig. 3-9** Temperature-pressure-concentration diagram of the saturated LiBr-water solution (Robert and Don, 1999).

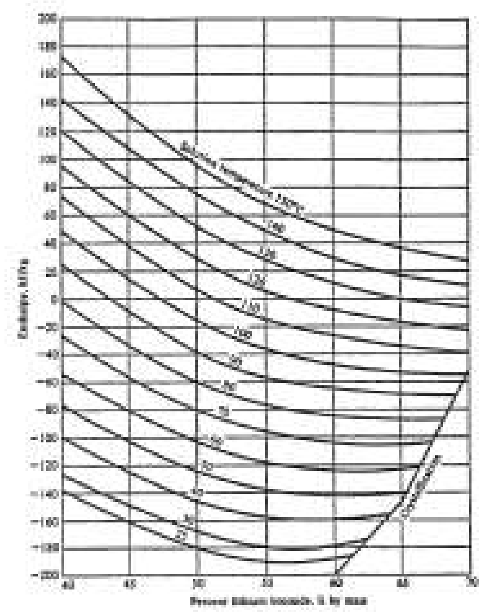

**Fig. 3-10** Enthalphy of LiBr-water solution (Robert and Don, 1999).

The diagram in Fig.3-10 presents enthalpy data for LiBr-water solutions. It is needed for the thermal calculation of the cycle. Enthalpies for water and water vapor can be determined from the table of properties of water. The data in Fig.3-10 are applicable to saturated or subcooled solutions and are based on a zero enthalpy of liquid water at 0°C and a zero enthalpy of solid LiBr at 25°C. Since the zero enthalpy for the water in the solution is the same as that in conventional tables of properties of water, the water property tables can be used in conjunction with diagram in Fig.3-9

Coefficient of performance of the absorption cycle is defined on the same principle as for the mechanical refrigeration:

$$COP_{abs} = \frac{\text{useful effect}}{\text{heat input}} = \frac{\text{refrigeration rate}}{\text{heat input at generator}} \tag{6}$$

but it should be noted that here denominator for the COPabs is heat while for the mechanical refrigeration cycle it is work. Since these two forms of energy are not equal, COPabs is not as low (0.6–0.8) as it appears compared to COP for mechanical system (2.5–3.5). The double-effect absorption unit is shown in Fig. 3-11. All major components and operation of the double-effect absorption machine is similar to that for the single-effect machine. The primary generator, located in the vessel 1, is using an external heat source to evaporate water from dilute-absorbent (high water concentration) solution.

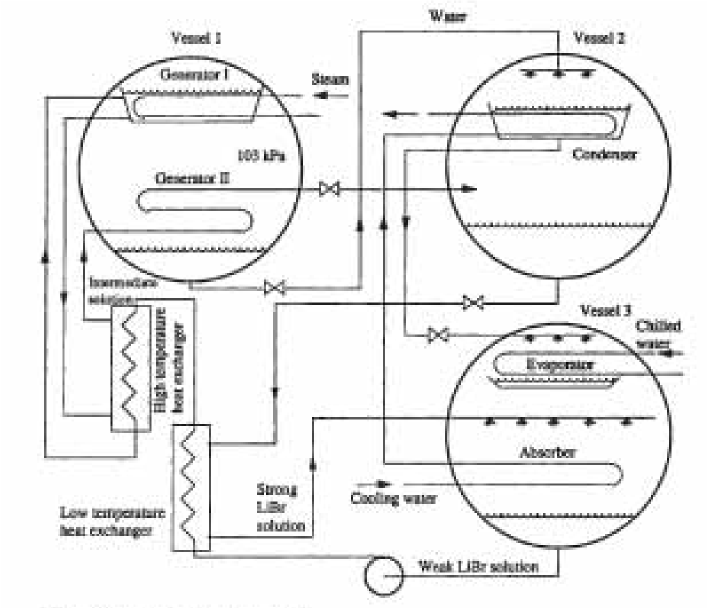


**Fig. 3-11** Double-effect absorption unit (Robert and Don, 1999).

Water vapor readily flows to the generator II where it is condensed on the tubes. The absorbent (LiBr) intermediate solution from generator I will pass through the heat exchanger on the way to the generator II where it is heated by the condensing water vapor. The throttling valve reduces pressure from vessel 1 (about 103 kPa absolute) to that of vessel 2. Following the reduction of pressure some water in the solution flashes to vapor, which is liquefied at the condenser. In the high temperature heat exchanger intermediate solution heats the weak (high water concentration) solution stream coming from the low temperature heat exchanger. In the low temperature heat exchanger strong solution is being cooled before entering the absorber. The absorber is on the same pressure as the evaporator. The double-effect absorption units achieve higher COPs than the single stage.

The ammonia-water absorption system was extensively used until the fifties when the LiBr-water combination became popular. Fig.3-12 shows a simplified ammonia-water absorption cycle. The refrigerant is ammonia, and the absorbent is dilute aqueous solution of ammonia. Ammonia-water systems differ from water-lithium bromide equipment to accommodate major differences: Water (here absorbent) is also volatile, so the regeneration of weak water solution to strong water solution is a fractional distillation. Different refrigerant (ammonia) causes different, much higher pressures: about 1100–2100 kPa absolute in condenser.

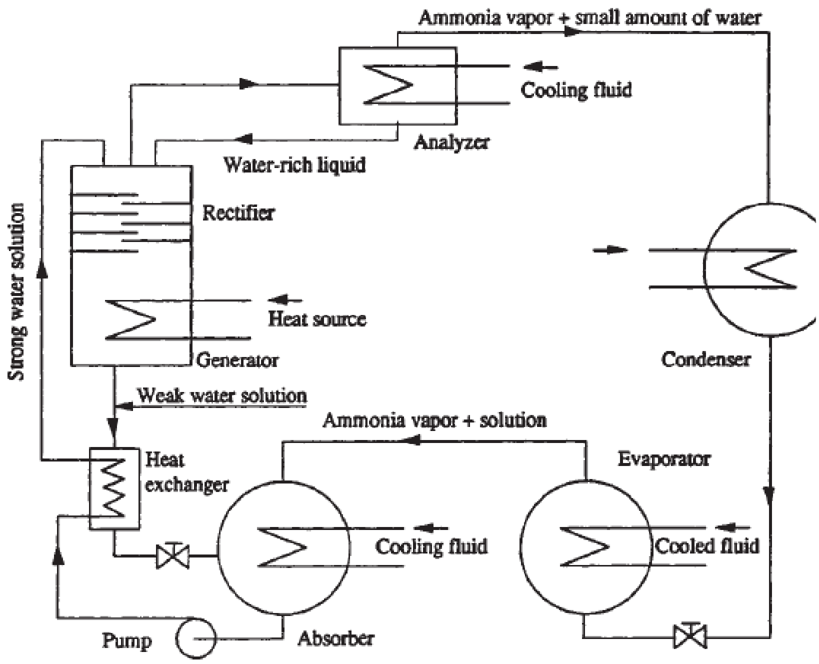


**Fig. 3-12** A simplified ammonia-water absorption cycle (Robert and Don, 1999).

## 3.2 Heat exchanger design

### 3.2.1 General heat exchanger design

Design methods for several important classes of process heat-transfer equipment are presented. Mechanical descriptions and specifications of equipment are given in this section and should be read in conjunction with the use of this material. It is impossible to present here a comprehensive treatment of heat-exchanger selection, design, and application.

Approach to Heat-Exchanger Design The proper use of basic heat-transfer knowledge in the design of practical heat-transfer equipment is an art. Designers must be constantly aware of the differences between the idealized conditions for and under which the basic knowledge was obtained and the real conditions of the mechanical expression of their design and its environment. The result must satisfy process and operational requirements (such as availability, flexibility, and maintainability) and do so economically. An important part of any design process is to consider and offset the consequences of error in the basic knowledge, in its subsequent incorporation into a design method, in the translation of design into equipment, or in the operation of the equipment and the process. Heat-exchanger design is not a highly accurate art under the best of conditions. The design of a process heat exchanger usually proceeds through the following steps:

1. Process conditions (stream compositions, flow rates, temperatures, pressures) must be specified.

2. Required physical properties over the temperature and pressure ranges of interest must be obtained.

3. The type of heat exchanger to be employed is chosen.

4. A preliminary estimate of the size of the exchanger is made, using a heat-transfer coefficient appropriate to the fluids, the process, and the equipment.

5. A first design is chosen, complete in all details necessary to carry out the design calculations.

6. The design chosen in step 5 is evaluated, or rated, as to its ability to meet the process specifications with respect to both heat transfer and pressure drop.

7. On the basis of the result of step 6, a new configuration is chosen if necessary and step 6 is repeated. If the first design was inadequate to meet the

required heat load, it is usually necessary to increase the size of the exchanger while still remaining within specified or feasible limits of pressure drop, tube length, shell diameter, etc. This will sometimes mean going to multiple-exchanger configurations. If the first design more than meets heat-load requirements or does not use all the allowable pressure drop, a less expensive exchanger can usually be designed to fulfill process requirements.

8. The final design should meet process requirements (within reasonable expectations of error) at lowest cost. The lowest cost should include operation and maintenance costs and credit for ability to meet long-term process changes, as well as installed (capital) cost. Exchangers should not be selected entirely on a lowest-first-cost basis, which frequently results in future penalties.

Overall Heat-Transfer Coefficient The basic design equation for a heat exchanger is

$$dA = \frac{dQ}{U\Delta T} \tag{7}$$

where dA is the element of surface area required to transfer an amount of heat dQ at a point in the exchanger where the overall heat transfer coefficient is U and where the overall bulk temperature difference between the two streams is $\Delta T$ . The overall heat-transfer coefficient is related to the individual film heat-transfer coefficients and fouling and wall resistances by Eq. (8) Basing $U_o$ on the outside surface area $A_o$ results in

$$U_o = \frac{1}{1/h_o + R_{do} + xA_o / k_w A_{wm} + (1/h_i + R_{di})A_o / A} \tag{8}$$

Equation (7) can be formally integrated to give the outside area required to transfer the total heat load $Q_T$:

$$A_o = \int_0^{Q_T} \frac{dQ}{U_o \Delta T} \tag{9}$$

To integrate Eq. (9), $U_o$ and $\Delta T$ must be known as functions of $Q$. For some problems, $U_o$ varies strongly and nonlinearly throughout the exchanger. In these cases, it is necessary to evaluate $U_o$ and $\Delta T$ at several intermediate values and numerically or graphically integrate. For many practical cases, it is possible to calculate a constant mean overall coefficient $U_{om}$ from Eq. (8) and define a corresponding mean value of $\Delta T_m$, such that

$$A_o = \frac{Q_T}{U_{om} \Delta T_m} \tag{10}$$

Care must be taken that $U_o$ does not vary too strongly, that the proper equations and conditions are chosen for calculating the individual coefficients, and that the mean temperature difference is the correct one for the specified exchanger configuration.

### 3.2.2 Mean Temperature Difference

The temperature difference between the two fluids in the heat exchanger will, in general, vary from point to point. The mean temperature difference ($\Delta T_m$ or MTD) can be calculated from the terminal temperatures of the two streams if the following assumptions are valid:

1. All elements of a given fluid stream have the same thermal history in passing through the exchanger.

2. The exchanger operates at steady state.

3. The specific heat is constant for each stream (or if either stream undergoes an isothermal phase transition).

4. The overall heat-transfer coefficient is constant.

5. Heat losses are negligible.

Countercurrent or cocurrent Flow If the flow of the streams is either completely countercurrent or completely cocurrent or if one or both streams are isothermal (condensing or vaporizing a pure component with negligible pressure change), the correct MTD is the logarithmic- mean temperature difference (LMTD), defined as

$$LMTD = \Delta T_{lm} = \frac{(t_1' - t_2'') - (t_2' - t_1'')}{\ln\left(\dfrac{t_1' - t_2''}{t_2' - t_1''}\right)} \tag{11}$$

for countercurrent flow (Fig. 3-13a) and

$$LMTD = \Delta T_{lm} = \frac{(t_1' - t_1'') - (t_2' - t_2'')}{\ln\left(\dfrac{t_1' - t_1''}{t_2' - t_2''}\right)} \tag{12}$$

for cocurrent flow (Fig. 3-13b)

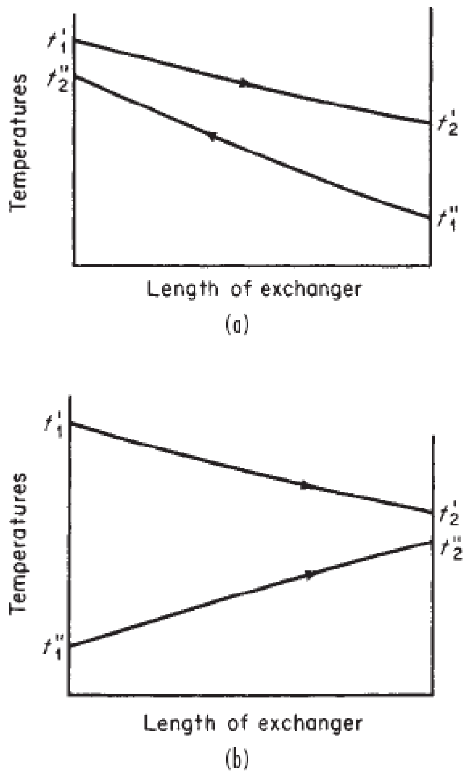


**Fig. 3-13** Temperature profiles in heat exchangers: (a) countercurrent and (b) cocurrent flows (Robert and Don, 1999).

If U is not constant but a linear function of DT, the correct value of $U_{om}\,\Delta\,Tm$ to use in Eq. (10) is

$$U_{om}\Delta T_m = \frac{U''_o(t'_1 - t''_2) - U'_o(t'_2 - t''_1)}{\ln\left(\dfrac{U''_o(t'_1 - t''_2)}{U'_o(t'_2 - t''_1)}\right)} \tag{13}$$

for countercurrent flow, where $U''_o$ is the overall coefficient evaluated when the stream temperatures are $t'_1$ and $t''_2$ and $U'_o$ is evaluated at $t'_2$ and $t''_1$. The corresponding equation for cocurrent flow is

$$U_{om}\Delta T_m = \frac{U''_o(t'_1 - t''_1) - U'_o(t'_2 - t''_2)}{\ln\left(\dfrac{U''_o(t'_1 - t''_1)}{U'_o(t'_2 - t''_2)}\right)} \tag{14}$$

where $U'_o$ is evaluated at $t'_2$ and $t''_2$ and $U''_o$ is evaluated at $t'_1$ and $t''_1$. To use these equations, it is necessary to calculate two values of $U_o$.

The use of Eq. (13-14) will frequently give satisfactory results even if $U_o$ is not strictly linear with temperature difference.

3.2.3 Reversed, Mixed, or Cross-Flow

If the flow pattern in the exchanger is not completely countercurrent or cocurrent, it is necessary to apply a correction factor FT by which the LMTD is multiplied to obtain the appropriate MTD. These corrections have been mathematically derived for flow patterns of .

For a common flow pattern, the 1-2 exchanger (Fig. 3-14), the correction factor $F_T$ is given in Fig. 3-15a, which is also valid for finding $F_T$ for a 1-2 exchanger in which the shell-side flow direction is reversed from that shown in Fig. 3-14.

Fig. 3-15a is also applicable with negligible error to exchangers with one shell pass and any number of tube passes. Values of $F_T$ less than 0.8 (0.75 at the very lowest) are generally unacceptable because the exchanger configuration chosen is inefficient; the chart is difficult to read accurately; and even a small violation of the first assumption underlying the MTD will invalidate the mathematical derivation and lead to a thermodynamically inoperable exchanger.

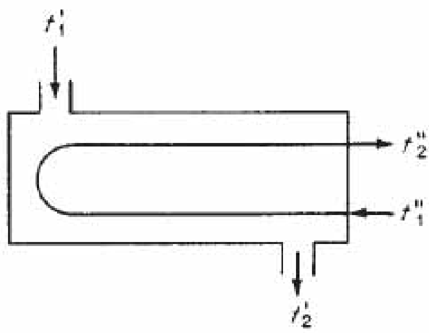


**Fig. 3-14** Diagram of a 1-2 exchanger (one well-baffled shell and two tube passes with an equal number of tubes in each pass) (Robert and Don, 1999).

Correction-factor charts are also available for exchangers with more than one shell pass provided by a longitudinal shell-side baffle. However, these exchangers are seldom used in practice because of mechanical complications in their construction. Also thermal and physical leakages across the longitudinal baffle further reduce the mean temperature difference and are not properly incorporated into the correction-factor charts. Such charts are useful, however when it is necessary to construct a multiple-shell exchanger train are included here for two, three, four, and six separate identical shells and two or more tube passes per shell in Fig. 3-15b, c, d, and e. If only one tube pass per shell is required, the piping can and should be arranged to provide pure countercurrent flow, in which case the LMTD is used with no correction. Cross-flow exchangers of various kinds are also important and require correction to be applied to the LMTD calculated by assuming countercurrent flow. Several cases are given in Fig. 3-15 f, g, h, i, and j. Many other MTD correction-factor charts have been prepared for various configurations. The $F_T$ charts are often employed to make approximate corrections for configurations even in cases for which they are not completely valid.

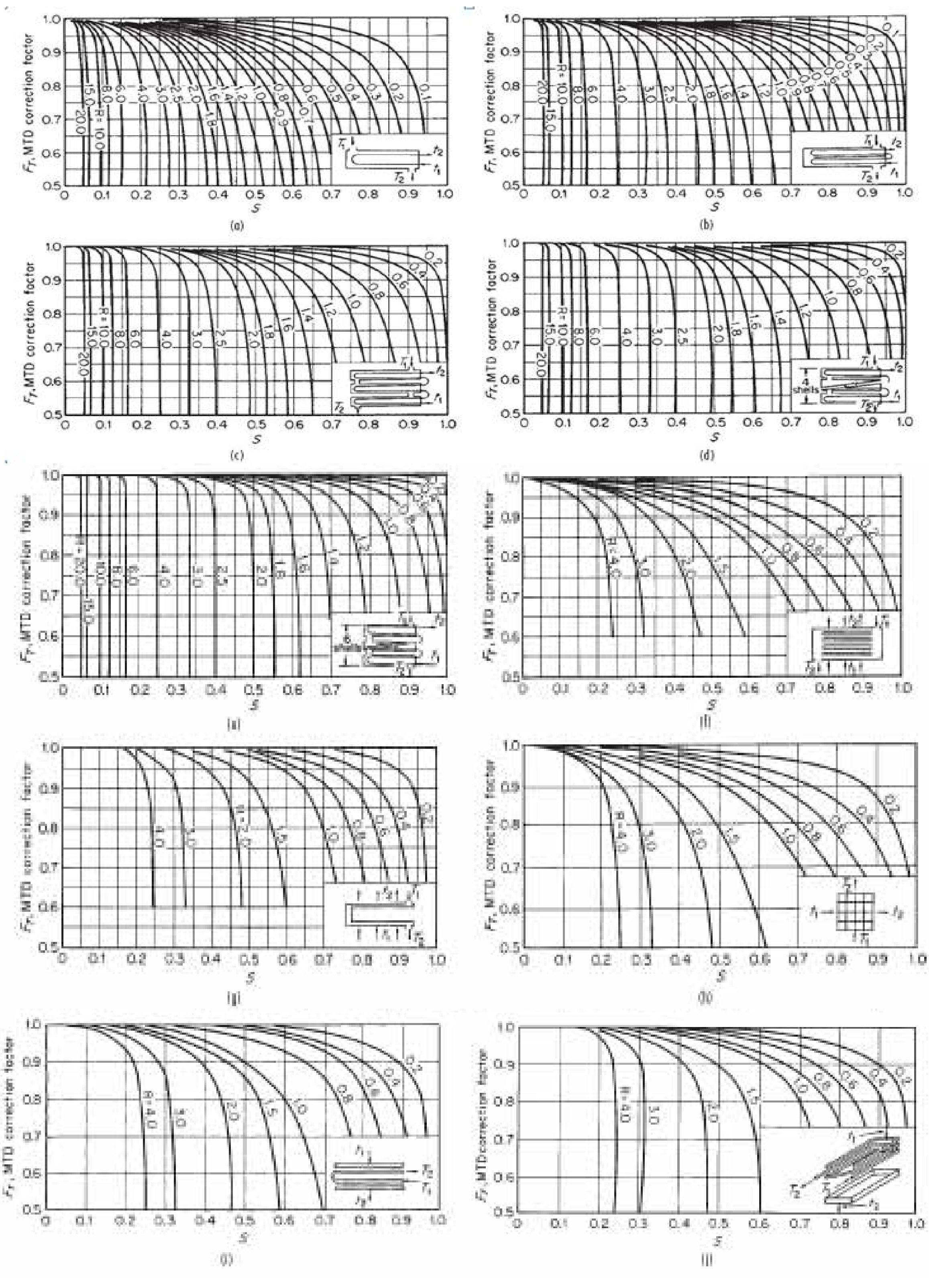


**Fig. 3-15** LMTD correction factors for heat exchangers (Robert and Don, 1999).

# CHAPTER 4

# SIMULATION OF AN ABSORPTION CHILLER

## 4.1 Description of absorption chiller

Absorption chiller has five major steps (Carolyn, 2009): condensing (condenser), expansion (control valve), evaporation (evaporator), absorption (absorber) and generator/concentrator (Fig. 4-1). Similar to vapor compression chillers, absorption chillers have a high-pressure side (i.e., generator and condenser) and a low-pressure side (i.e., expansion pipe, evaporator and absorber). Description of the unit operation in the absorption chiller will be given as follows:

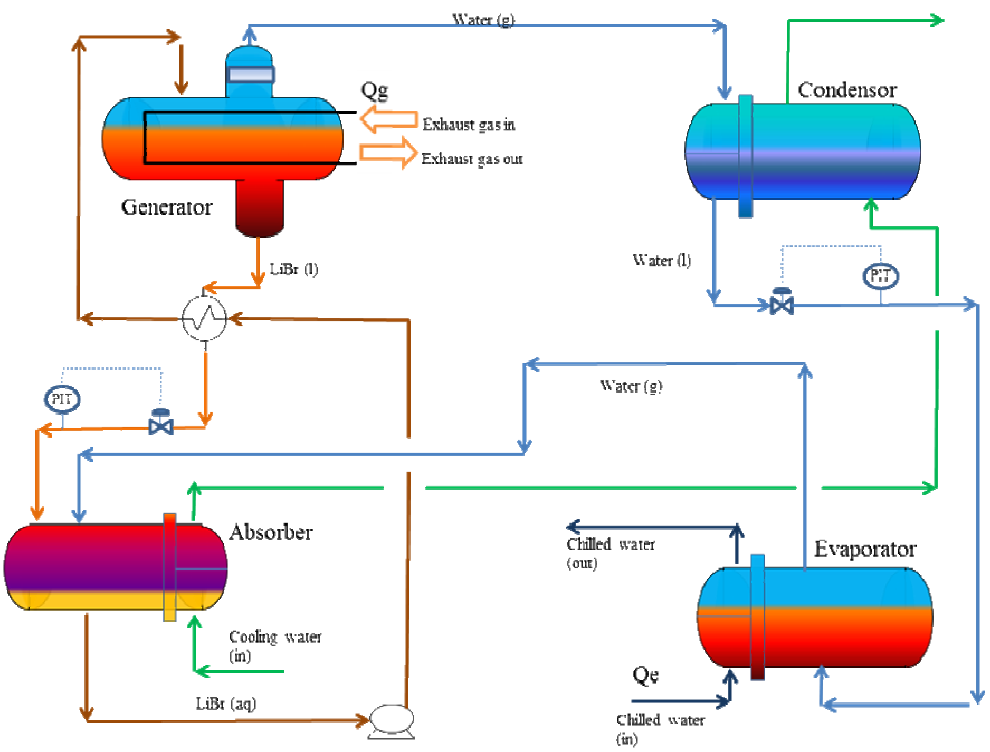


**Fig. 4-1** Flow diagram of absorption chiller process.

• Condenser

In the condenser, the cooling water absorbs the heat of condensation from the vaporized water, changing the water in the gas phase to the liquid phase (Vincent, 2009). After that, water flows via a control valve where its pressure decreases from 6.8 kPa to 0.68 kPa before sending to the evaporator. The cooling water used at the condenser is obtained from the absorber. The final temperature of the cooling water is based on the absorbed heat duty and pressure of cooling water (normal operating pressure of 600 kPa).

• Evaporator

Water from the condenser is sent to the evaporator in which a chilled water is used as the heating medium. In the evaporator, water is sprayed on the tube bundle and changed to the vapor. Under the low evaporator pressure (less than 0.68 kPa), the water vaporizes at the temperature approximately 2 °C. At this stage, heat is removed from the chilled water of which the temperature decreases by 10°C. The chilled water at the outlet is a product of the absorption chiller system.

• Absorber

In the absorber, the water vapor from the evaporator is absorbed by a lithium-bromide liquid solution. The heat of absorption is removed by cooling water from a cooling tower. The inlet temperature of cooling water is based on the wet bulb temperature of air at surrounding. The lithium-bromide solution is removed from the absorber via pump with the outlet pressure of 6.8 kPa and then preheated before being fed to the generator/concentrator.

• Generator/concentrator

The lithium-bromide solution enters the generator/concentrator where it is heated by a low grade waste heat to the desired temperature at which water in the lithium-bromide solution can vaporize. The water vapor obtained goes to the condenser, completing the refrigerant cycle. The concentrated LiBr solution flows down to the absorber.

**4.2 Simulation of absorption chiller process**

Simulation of an absorption chiller process starts by assigning the initial conditions and the operating parameters (Pratihar et al., 2011). In this study, the exhaust gas with the flow rate of 33,588 kg/h is considered a low grade waste heat. The inlet and outlet temperatures of the exhaust gas are 130 and 100 °C, respectively. The exhaust gas contains water, oxygen, nitrogen and carbon dioxide with the mass fraction of 0.23, 0.02, 0.69 and 0.049. The flow rate of LiBr as an absorbent is 2,000 kg/h. Water is a working fluid or refrigerant and its flow rate is 2,000 kg/h. The temperature of cooling water inlet is 29 °C and chilled water inlet temperature is 20 °C.

Fig. 4-2 shows the flowsheet of the absorption chiller system. It is noted that the pressure drop across heat exchangers and pipe line is neglected (Somers et al., 2011).

**Property method**

To simulate the absorption chiller process, LiBr solution is added and the electrolyte property model as “ELECNRTL” is selected as a thermodynamic method to calculate the fluid properties in the process (Somers et al., 2011).

**Unit operations**

Generator: in this study, a heat exchanger is employed to simulate the generator. The exhaust gas is used as a heat source and its temperature decreases from 130 °C to 100 °C, which equals tothe heat flow of 346 kW. The 50% LiBr solution with the inlet temperature of 32 °C is fed to the generator and its outlet temperature is

72 °C. A flash separator is used to separate the concentrate LiBr and water. It is operated under the temperature of 72 °C and pressure of 6.8 kPa. Water is removed from the top of the separator (stream no.003) and concentrate LiBr at the bottom (stream no.007).

Condenser: water vapor from the generator is condensed to the liquid phase in the condenser operated at pressure of 6.8 kPa. At the standard condition, the heat flow of 250 kW is necessary to remove from the condenser by cooling water. This causes an increase in the cooling water temperature from 36 °C to 41 °C.

Control valve: the outlet pressure of valve is specified at 0.68 kPa.

Evaporator: the simulation of the evaporator is the same as the condenser. The evaporator is operated at pressure of 0.68 kPa. Chilled water with the flow rate of 20.5 $m^3$/h is required to vaporize water from the condenser.

Absorber: it is assumed that the LiBr solution exiting from the absorber is the saturated liquid (Somers et al., 2011). The water vapor from the evaporator is absorbed by LiBr solution and flow out the absorber at the temperature of 22 °C. Heat of absorption (333 kW) is removed by cooling water with flow rate 40 $m^3$/h. The inlet cooling water temperature is 29 °C.

Pump: it is used to build up the pressure of LiBr solution to 6.8 kPa.

Under the standard conditions, the COP of the absorption chiller system is 0.68 and the simulation data are shown in Table 4-1.

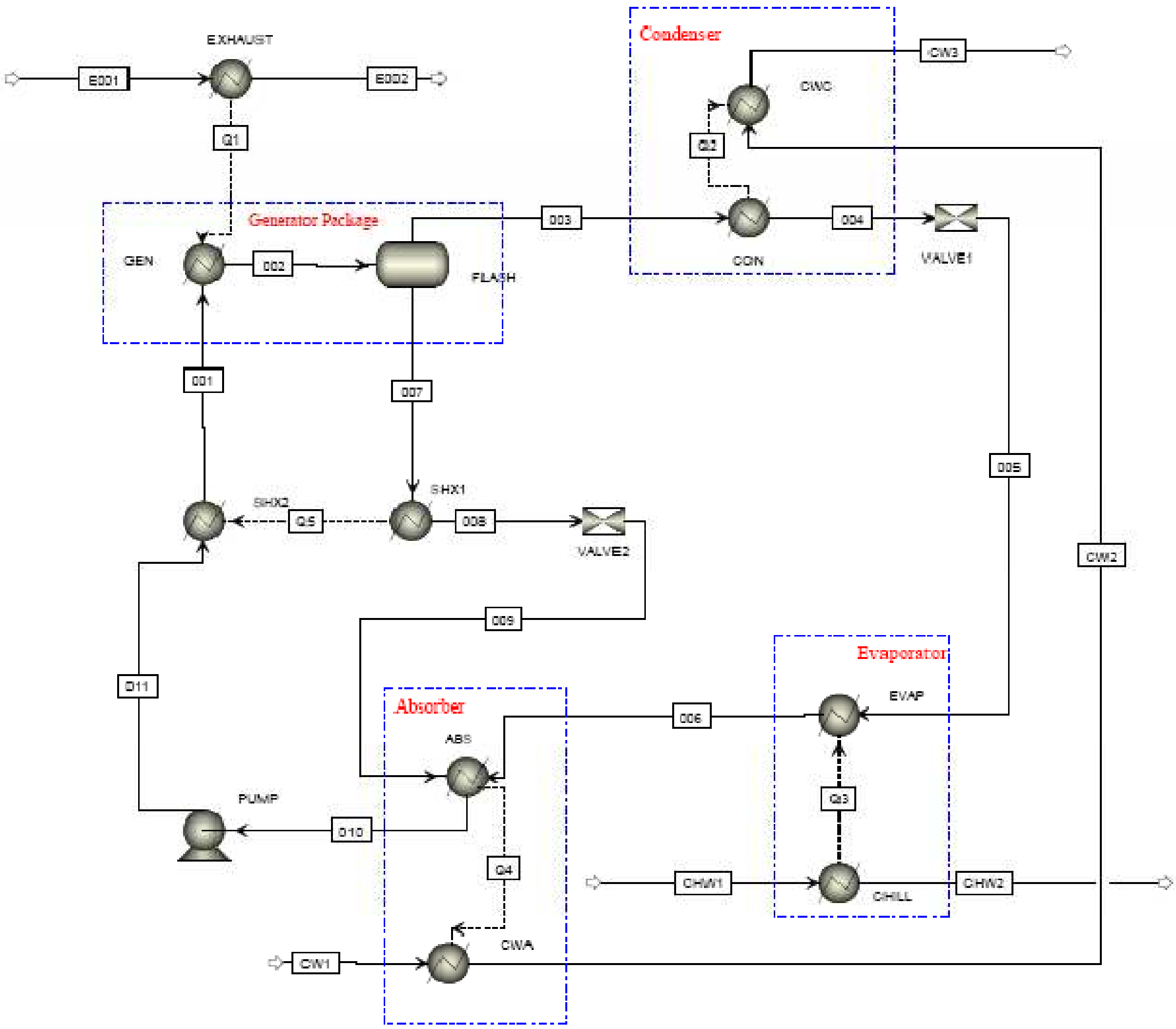


**Fig. 4-2** Flowsheet of the absorption chiller.

Table 4-1 Stream data for absorption chiller

| | | 001 | 002 | 003 | 004 | 005 | 006 | 007 | 008 | 009 |
|---|---|---|---|---|---|---|---|---|---|---|
| From | | SHX2 | GEN | FLASH | CON | VALVE1 | EVAP | FLASH | SHX1 | VALVE2 |
| To | | GEN | FLASH | CON | VALVE1 | EVAP | ABS | SHX1 | VALVE2 | ABS |
| Substream: MIXED | | | | | | | | | | |
| Phase: | | Liquid | Mixed | Vapor | Liquid | Mixed | Vapor | Liquid | Liquid | Mixed |
| Mass Flow | KG/HR | 4000 | 4000 | 363.26 | 363.26 | 363.26 | 363.26 | 3636.74 | 3636.74 | 3636.74 |
| Volume Flow | CUM/HR | 2.7 | 8389.15 | 8386.77 | 0.37 | 4149.8 | 66798.91 | 2.39 | 2.37 | 15691.82 |
| Temperature | C | 32.18 | 71.97 | 71.97 | 38.65 | 1.59 | 1.6 | 71.97 | 60 | 31.13 |
| Pressure | kPa | 7 | 7 | 7 | 7 | 1 | 1 | 7 | 7 | 1 |
| Vapor Fraction | | 0 | 0.13 | 1 | 0 | 0.06 | 1 | 0 | 0 | 0.03 |
| Average Molecular Weight | | 25.47 | 25.47 | 18.02 | 18.02 | 18.02 | 18.02 | 26.56 | 26.56 | 26.56 |

| | | 010 | 011 | CHW1 | CHW2 | CW1 | CW2 | CW3 | E001 | E002 |
|---|---|---|---|---|---|---|---|---|---|---|
| From | | ABS | PUMP | | CHILL | | CWA | CWC | | EXHAUST |
| To | | PUMP | SHX2 | CHILL | | CWA | CWC | | EXHAUST | |
| Substream: MIXED | | | | | | | | | | |
| Phase: | | Liquid | Liquid | Liquid | Liquid | Liquid | Liquid | Liquid | Vapor | Vapor |
| Mass Flow | KG/HR | 4000 | 4000 | 20500 | 20500 | 40000 | 40000 | 40000 | 33588 | 33588 |
| Volume Flow | CUM/HR | 2.69 | 2.69 | 20.53 | 20.5 | 40.15 | 40.24 | 40.33 | 43898.07 | 40616.51 |
| Temperature | C | 22.3 | 22.31 | 20 | 10 | 29 | 36.18 | 41.56 | 130 | 100 |
| Pressure | kPa | 1 | 7 | 600 | 600 | 600 | 600 | 600 | 101 | 101 |
| Vapor Fraction | | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 1 | 1 |
| Average Molecular Weight | | 25.47 | 25.47 | 18.02 | 18.02 | 18.02 | 18.02 | 18.02 | 25.28 | 25.28 |

## 4.3 Model validation

In order to verify the reliability of the ELECNRTL property method used in simulations of the absorption chiller process, the vapor pressure of the LiBr solution predicted by the model is compared with data in literature (Robert and Don, 1999). Fig. 4-3 shows a comparison of model prediction and data of the vapor pressure of the 50% LiBr solution at different temperatures (30-85 °C). The average error is 6.25% and this ensures that the ELECNRTL property method can be applied for simulation of the absorption chiller process.

The proposed model of the absorption chiller is then validated. The simulation result is compared with experimental data reported by Bahador et al. (2011). They investigated the performance of the absorption chiller by varying the inlet temperature of a driving heat source. The simulations are performed based on the inlet condition of the heat source data from Bahador et al. (2011). Fig. 4-4 compares the heat duty at each unit (i.e., absorber, evaporator and condenser) and the COP predicted by the model and experimental data. It is found that the model prediction agrees with the experimental data.

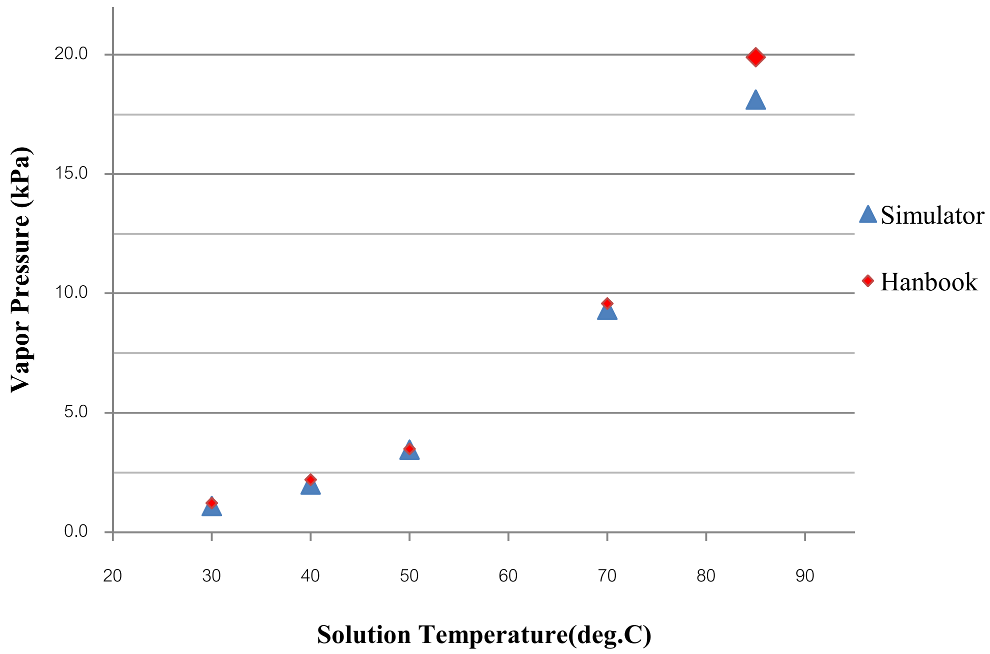


**Fig. 4-3** Vapor pressure of LiBr solution at different temperatures: model prediction and data reported in literature.

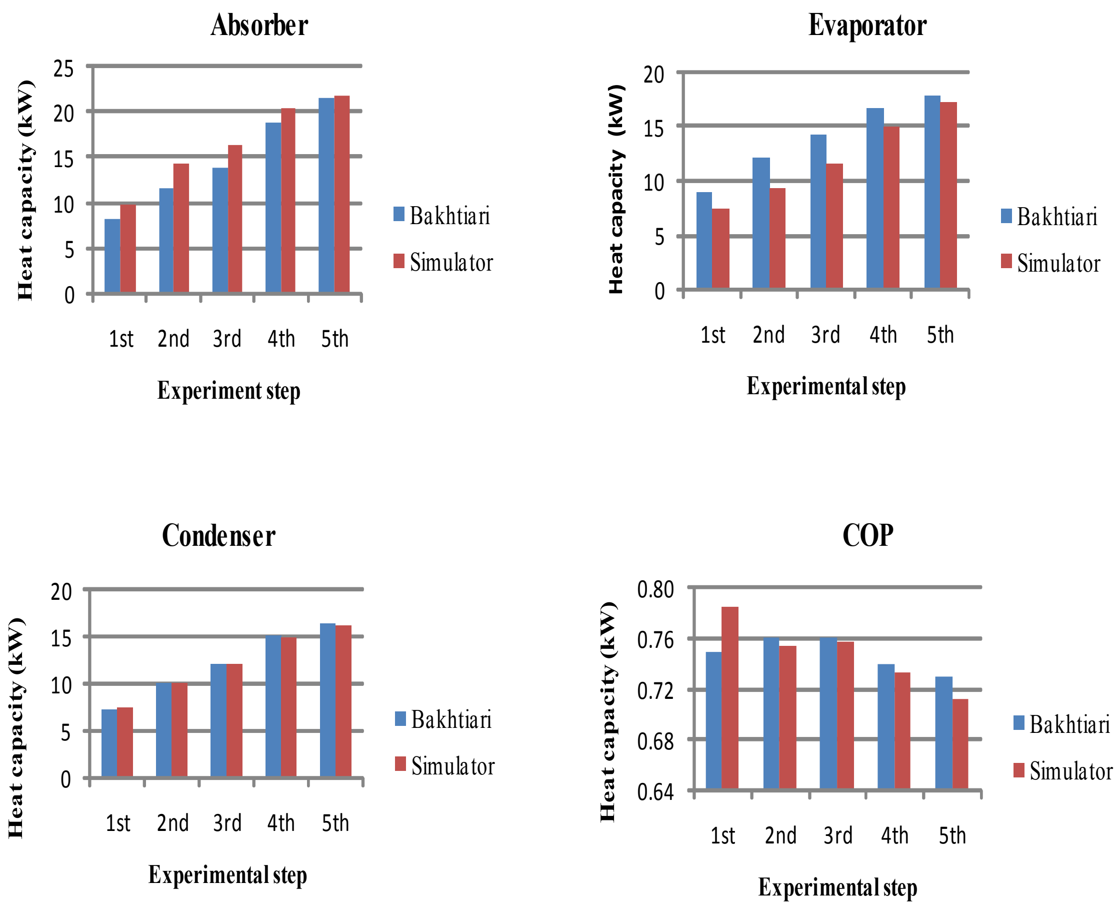


**Fig. 4-4** Comparison between predicted results and experimental data of the heat capacity at each unit and COP.

# CHAPTER 5

# RESULTS AND DISCUSSION

Simulations of the absorption chiller system are performed to investigate the effect of key operating parameter on its performance in terms of COP. The approach to keep the performance of the absorption chiller is presented when the exhaust gas from a steam boiler changes. In addition, the cost comparison between the absorption chiller and a electrical chiller is also given.

## 5.1 Sensitivity analysis

Key operating variables considered consist of the flow rate and temperature of exhaust gas, chilled water and cooling water. These variables affect the performance of the absorption chiller.

### 5.1.1 Effect of exhaust gas inlet temperature

In this study, an exhaust gas from the steam boiler is employed as a low grade waste heat to drive the absorption chiller. Analysis of effect of the exhaust gas conditions on the absorption chiller is interesting. Fig. 5-1 shows effect of the exhaust gas inlet temperature on the COP of the absorption chiller. The inlet temperature of exhaust gas is varied from 110 °C to 135 °C. It is noted that the maximum of inlet temperature is basically not over than 135 °C because in the combined cycle power plant, portion of the exhaust gas is employed for a steam generation section and thus its temperature decreases. It is found that the COP increases with increasing the exhaust gas inlet temperature. This because temperature is the driving force for heat transfer of exhaust gas to generator in absorption chiller system.

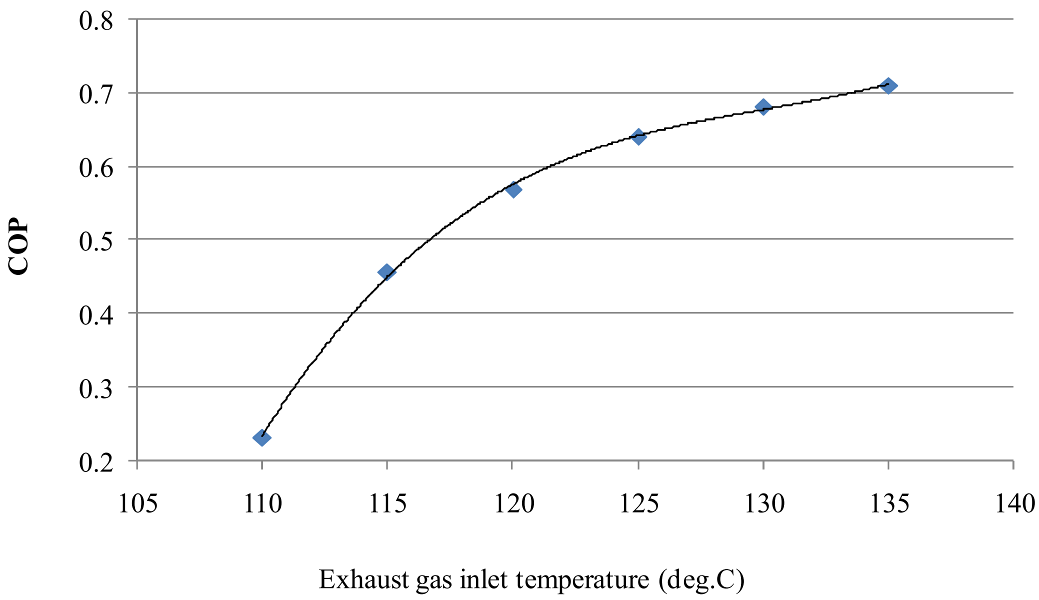


**Fig. 5-1** Effect of exhaust gas inlet temperature.

### 5.1.2 Effect of chilled water inlet temperature

Chilled water is the product of an absorption chiller and used for an air conditioning system. The temperature of the chilled water at the outlet of the air conditioning system increases to 20°C. Then, the chilled water is returned to the absorption chiller system to decrease its temperature. Thus, the inlet temperature of the chilled water is varied from 12-25 °C. Fig. 5-2 shows that the COP increases from 0.2 to 1 when the chilled water inlet temperature increases. At the chilled water inlet temperature of 25°C, the COP is higher than 1 but in reality it difficult to find the chilled water inlet temperature more than 1 because the limitation of HVAC system.

### 5.1.3 Effect of chilled water flow rate

Chilled water is used to supply heat to the evaporator to vaporize a condensate water at low pressure. Chilled water flow rate is the factor that affects the COP as shown in Fig. 5-3. In the figure, the chilled water flow rate is varied from 5 to 11 kg/s and this causes the COP of the absorption chiller system to increase from 0.5 to 0.7. An increase in the flow rate of chilled water rise the COP because a higher heat can be removed from the evaporator. The chilled water flow rate should be selected to compromise both the condensate flow rate and heat exchanger size.

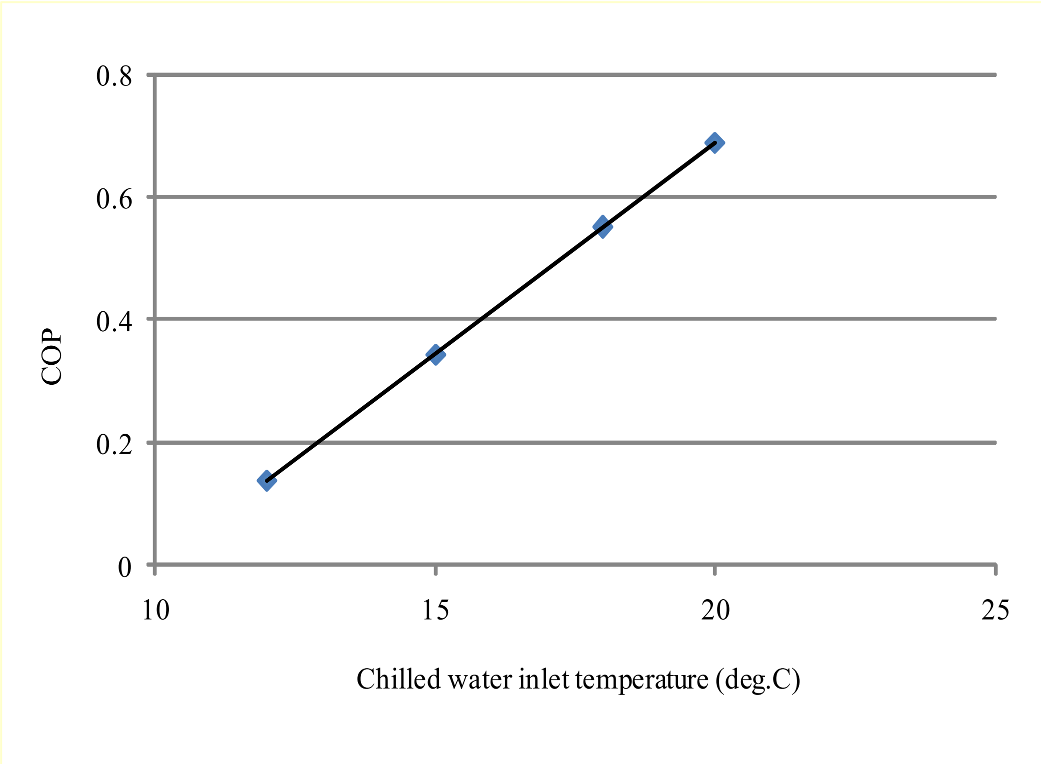


**Fig. 5-2** Effect of chilled water inlet temperature.

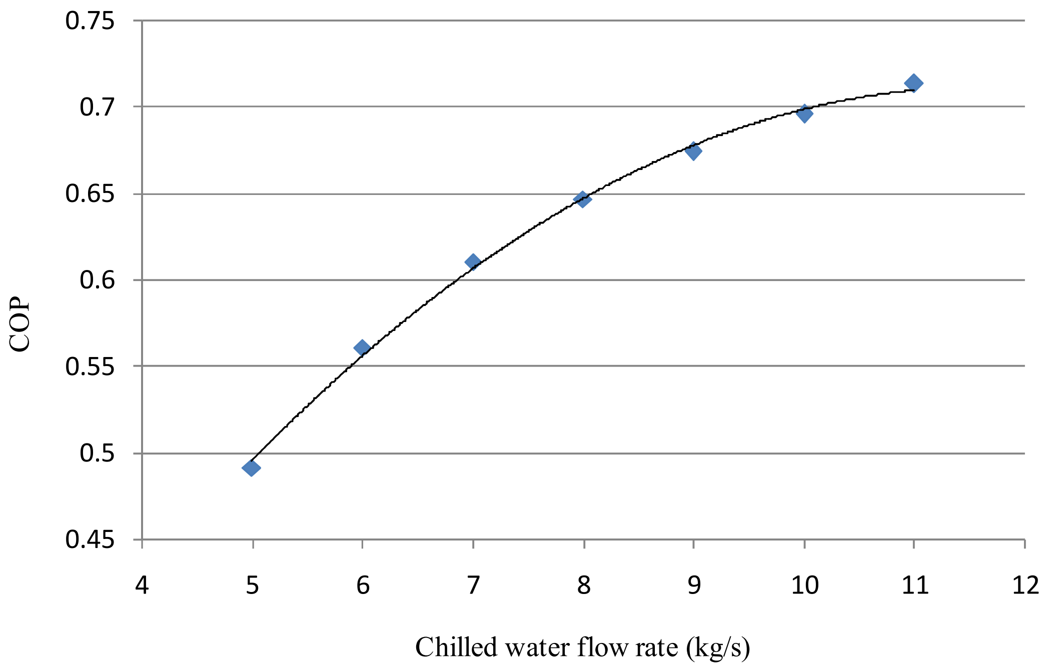


**Fig. 5-3** Effect of chilled water flow rate.

### 5.1.4 Effect of cooling water inlet temperature

Cooling water is an important utility in the absorption chiller and used to remove heat from the absorber and the condenser. In general, cooling water is first sent to the absorber and then go to the condenser. After that, it will be returned to a cooling tower to cool down its temperature. The temperature of cooling water depends on the wet bulk air temperature. Fig. 5-4 shows the effect of cooling water inlet temperature on COP. The increased cooling water temperature has a negative effect on the COP. COP of this study more than 1 when temperature of cooling water inlet less than 24 °C but it difficult to find the cooling water inlet temperature less than 29 °C because the limitation of weather in Thailand. ,

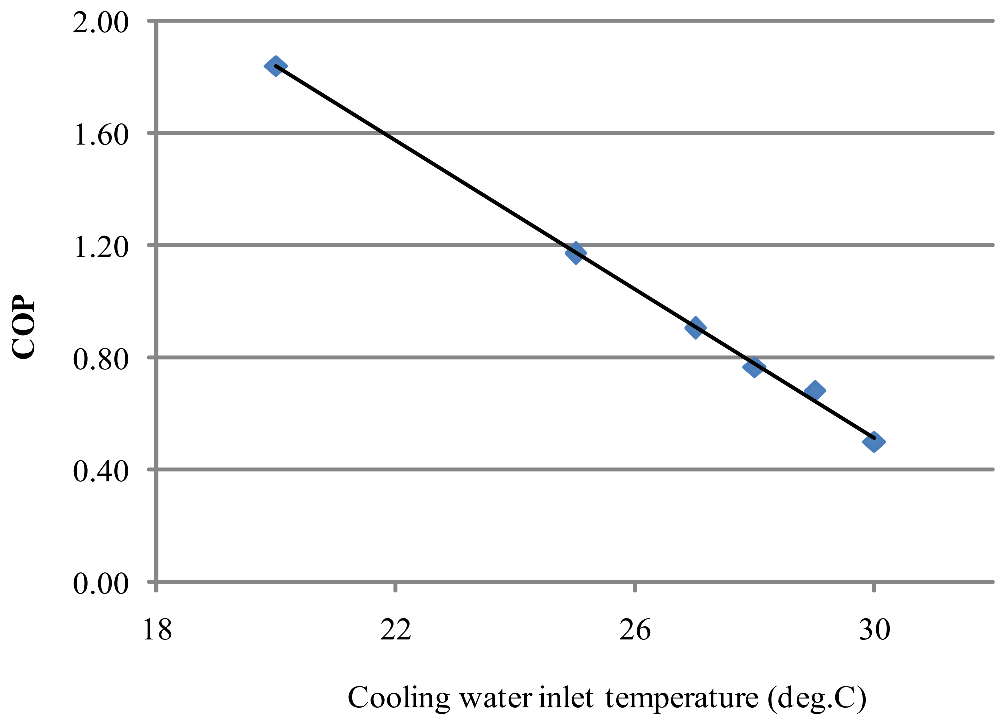


**Fig. 5-4** Effect of cooling water inlet temperature.

### 5.1.5 Effect of cooling water flow rate

Heat duties from the absorber and condenser are removed by cooling water. Fig. 5.5 shows the effect of cooling water flow rate on the COP. It is found that the performance of the absorption chiller improved at high cooling water flow rate. However, in normal operation, the flow rate of cooling water should be minimized for operating cost saving.

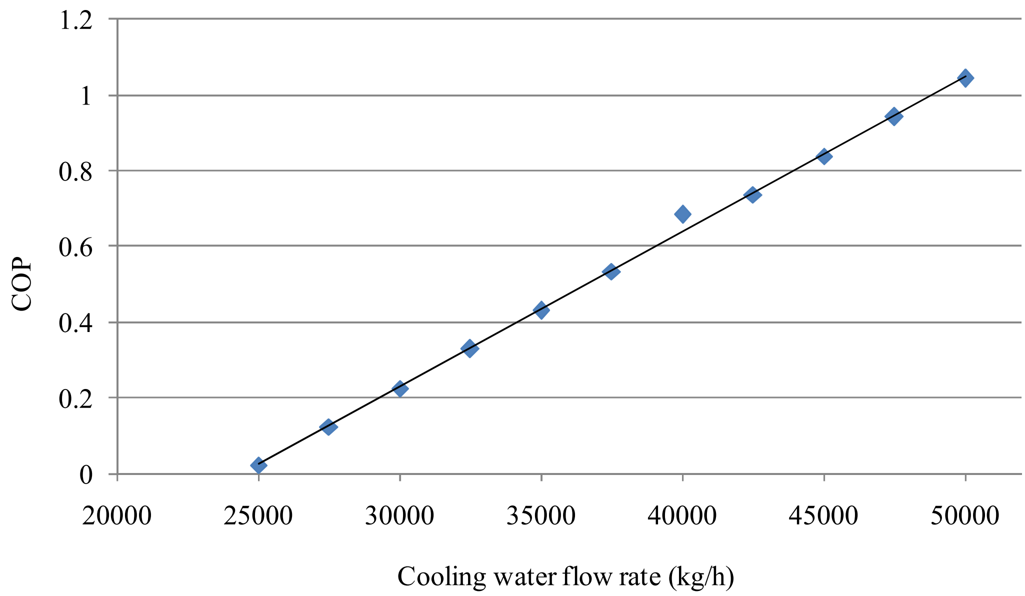


**Fig. 5-5** Effect of cooling water flow rate.

### 5.1.6 Effect of water content in exhaust gas

In general, the exhaust gas from a steam boiler contains water and its content depends on source of fuel used (Myers, 2009). Fig. 5-6 shows the fraction of water in the exhaust gas obtained from the steam boiler using different fuels. It is found that the water content in the exhaust gas slightly affects the COP of the absorption chiller system. The exhaust gas obtained from the steam boiler using coal has a higher water fraction. When it is used as the heat source for the absorption chiller, the COP of the system shows the highest value. This because water have highest specific heat capacity when compare with another composition in exhaust gas. In Fig. 5-7 ensure that water content in exhaust gas is strongly effect with COP.

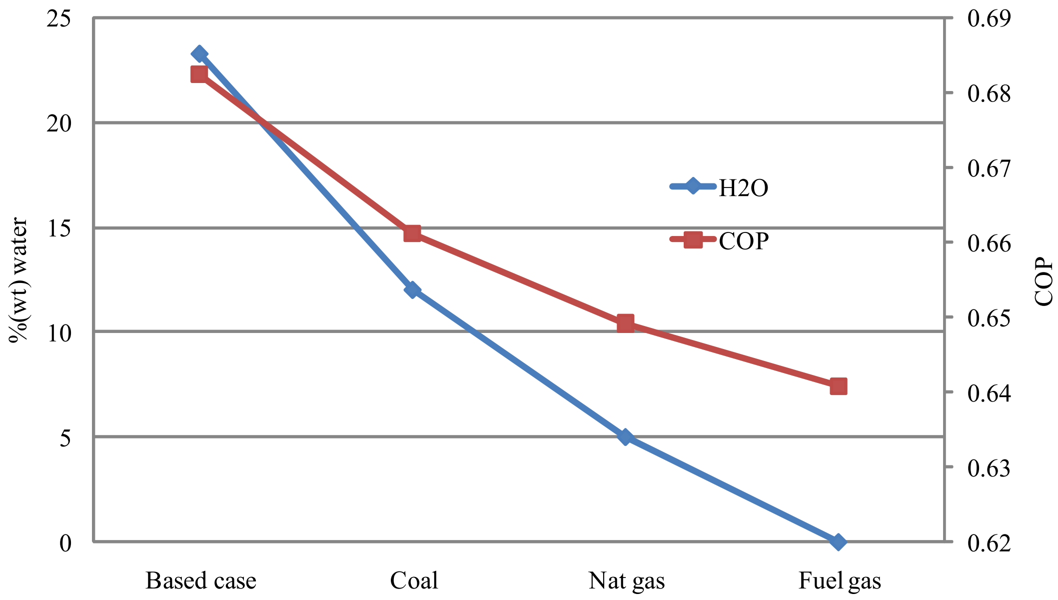


**Fig. 5-6** Comparison of water content and system COP using exhaust gas derived from the steam boiler using different fuels.

From the sensitivity analysis of the absorption chiller as mentioned above, it is found that the cooling water inlet temperature has the strongest effect on the performance of the absorption chiller.

### 5.2 Strategy for remain COP of the system

In normal operation, the fluctuation of the exhaust gas can be occurred and as a result, the exhaust gas may not be sufficient to supply for the absorption chiller. In such a situation, the addition of a low pressure steam (800 kPa) in the exhaust gas is proposed to maintain the production of chilled water (Fig. 5-7). When the heat from the exhaust gas is insufficient, the low pressure steam is required as shown in Fig. 5-8. When the amount of the exhaust gas decreases from 10 to 50%, the flow rate of the low pressure steam required to keep the constant heat input increases from 1200 to

6200 kg/h. However, the addition of steam causes an increase in the operation cost of the absorption chiller. Fig. 5.9 shows the cost of the required low pressure steam when the exhaust gas decreases. The calculation is based on the steam price of 867 bath/$m^3$.

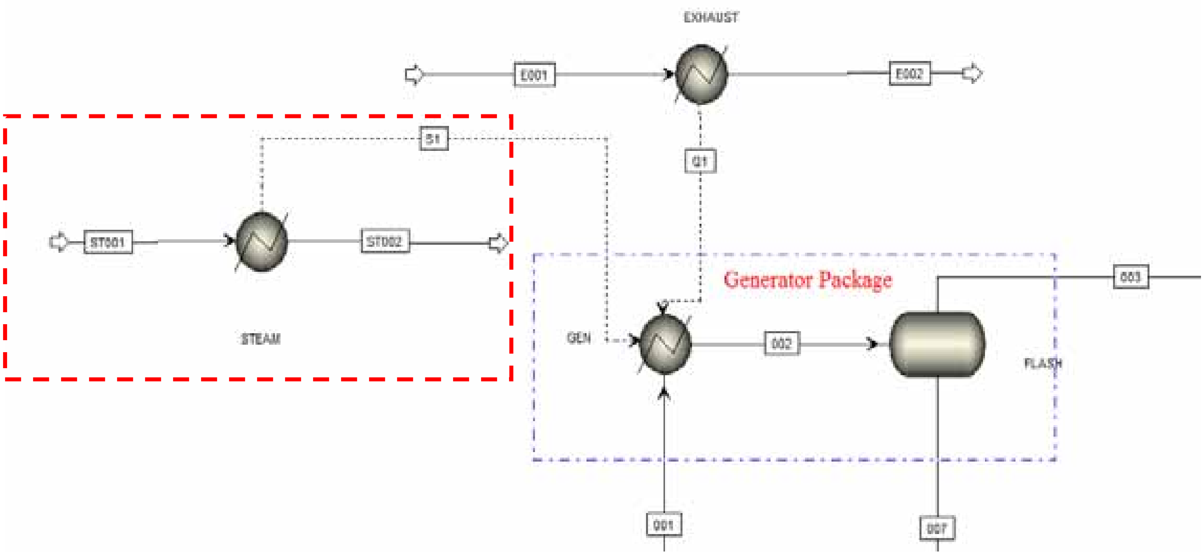


**Fig. 5-7** Flow diagram of the generator when a low pressure steam is added to the exhaust gas.

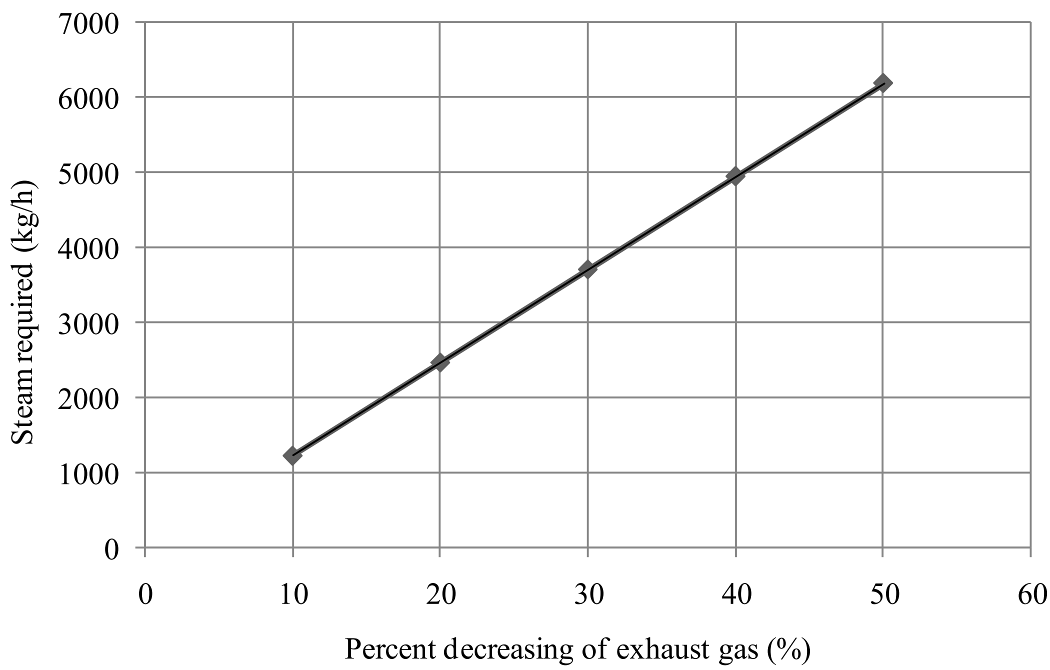


**Fig. 5-8** Relation of decreased exhaust gas and the required steam.

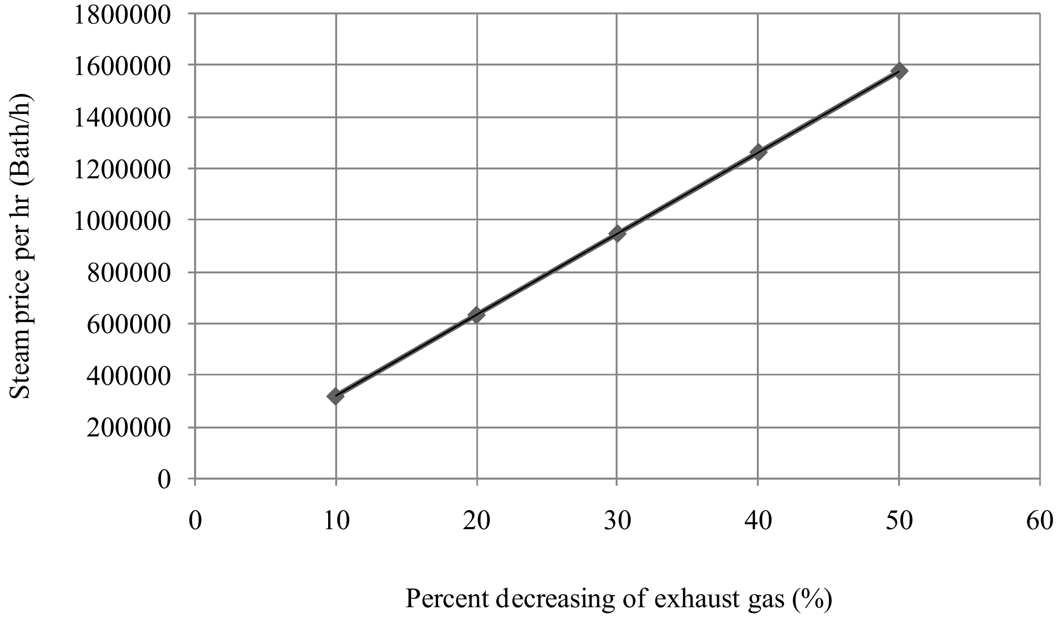


**Fig. 5-9** Cost of the required low pressure steam when the exhaust gas decreases.

### 5.3 Economic comparison

In this section, the capital cost of the absorption chiller is determined and compared with a conventional electric chiller. Data used for cost evaluation is obtained from the study by Boonnasa et al. (2006). Costs of major components such as absorption chiller, heat exchanger, chilled water pump, condensate pump, control system and storage system are considered. The results show that based on the same evaporator heat duty (ton refrigerant, RT), the cost of the absorption chiller (0.18 MUS$) is lower than the electric chiller (0.38 MUS$). Appendix A shows the cost calculation in detail.

# CHAPTER 6

# CONCLUSIONS

In this study, simulation of the absorption chiller system which composes of a generator, condenser, evaporator and absorber was performed. Exhaust gas from a steam boiler was used as a driving heat source and lithium-bromide solution (LiBr) and water are the absorbent and the refrigerant, respectively. Effect of operating condition on performance of the absorption chiller was analyzed. Cost comparion between the absorption chiller and convention chiller was also given.

## 6.1 Simulation of absorption chiller

Simulation of the absorption chiller was performed to investigate the effect of key operating parameters on its performance. Under standard conditions, the exhaust gas with the flow rate of 33,588 kg/h was used as a low grade waste heat. The inlet and outlet temperatures of the exhaust gas is 130 °C and 100 °C, respectively. The exhaust gas composes of water, oxygen, nitrogen and carbon dioxide at the mass fraction of 0.23, 0.02, 0.69 and 0.049, respectively. The flow rates of LiBr and water are equal to 2,000 kg/h. The inlet temperatures of cooling water and chilled water are 29 °C and 20 °C. The simulation result showed that the absorption chill process can produce the chilled water at the production rate of 20.5 $m^3$/h and the COP of the process is 0.68.

Effects of key parameters such as exhaust gas inlet temperature, chilled water inlet temperature, chilled water flow rate, cooling water inlet temperature, cooling water flow rate and water contain in exhaust gas were investigated and it was found that the cooling water inlet temperature has a strong effect on the performance of the absorption chiller in terms of the COP.

Since a variation of the exhaust gas flow rate may cause non-stable performance of the absorption chiller and affect the productivity of chilled water. Low pressure steam (800 kPa) was added into the absorption chiller process to maintain the heat input of the generator. The results showed that the addition of steam can keep COP of the process constant but leads to a high operation cost.

## 6.2 Economic comparison

The cost analysis of the absorption chiller based on data reported in literature (Ref.) showed that at the same capacity of absorption chiller (refrigerant ton), the cost of the absorption chiller is lower than a electrical chiller.

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# APPENDICES

# APPENDIX A

# COST ESTIMATION OF CHILLERS

This appendix explains an example of calculating the cost of absorption chiller and conventional chiller (electric chiller) for economic comparison.

Table A-1 shows the capital cost of the absorption chiller based on data from the study by Boonnasa et al. 2006. The cost estimation of the chiller can be performed step-by-step as follows:

From Table A-1, the capital cost of the absorption chiller per unit refrigerant ton (RT) (excluding the compact heat exchanger) = 0.0025 MUS$/RT

| | | | |
|---|---|---|---|
| In this study, a heat load at the evaporator | = | 236.43 | kW |
| | = | 806,425.2 | Btu/h |
| (12,000 Btu/h = 1 ton RT) | = | 67.2 | RT |

Thus, the cost of the absorption chiller (excluding preheater)

| | | | |
|---|---|---|---|
| | = | 0.1684 | MUS$ |
| Heat exchanger cost per unit area | = | 0.31 | MUS$/m2 |

**Table A-1** Capital costs for absorption chiller (Boonnasa et al. 2006)

| Component | Unit | Price(MUS$) |
|---|---|---|
| Absorption chiller (3232 RT) | 1 | 1.79 |
| Compact heat exchanger (7544 m$^2$) | 1 | 0.313 |
| Chilled water pump | 4 | 1.3 |
| Condensate pump | 2 | 0.097 |
| Control system | 1 | 0.81 |
| Storage tank (24432 m$^2$) | 1 | 3.2 |
| Miscellaneous | 1 | 0.9 |
| Total capital cost | | 8.41 |

| | | |
|---|---|---|
| Cost of preheater (area = 168 $m^2$) | = 0.007 | MUS$ |
| Total capital cost for this research | = 0.1753 | MUS$ |
| | = 5.3 | MTHB |

Reference prices for the electric chiller:

| | | |
|---|---|---|
| Electric chiller capacity | = 165 | RT |
| Electric chiller price | = 27.8 | MTHB |
| Prices for 67.2 RT Electric chiller | = 11.3 | MTHB |

From the calculation, the result shows that the electrical chiller is more expensive than the absorption chiller about 6 MTHB (0.2 MUS$).

# VITAE

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