CHAPTER II

REVIEW OF RELATED LITERATURE AND RESEARCH

Review of Related Literature

The use of solar energy in air conditioning systems had been being one of the most evident and insufficiently exploited applications of the source of renewable energy. The use of solar irradiation for cooling allows time synchronization between solar offer and cold demand since cold air was in general more necessary when the solar irradiation was high, and this reduces the need of storage systems, which was one of the drawbacks of the use of solar energy for heating. A paper by Best and Pilatowsky, (1998, pp. 150-159) reviewed many projects and technologies for solar cooling and refrigeration. An overview of solar assisted air conditioning systems could be found in Henning, (2007, pp. 1734-1749) while fundamental insights about controlled needs for solar thermal driven cooling in Europe were given in Balaras. et al., (2007, pp. 299-314).

1. Technologies for Solar-driven Cooling

In the principle, there were many different ways to convert solar energy into cooling or air conditioning process: an overview is given in Figure 1. A main distinction can be made between thermally and electrically operated systems. Solar energy can be converted directly into electricity using photovoltaic panels, to drive a vapor compression chiller with an electric motor. From the economic point of view, there is an even greater incentive if the price for electricity generated by solar energy is higher than that of electricity from conventional sources. Therefore, photovoltaically powered solar air conditioning systems are of minor interest from a system point of view. Among the thermally driven process, thermo-mechanical processes and processes based on heat transformation can be distinguished. The latter are all based on reversible thermo-chemical reactions with relatively low blinding energies. The solar cooling systems under consideration are generally divided into two main categories, the Closed Cycle System and the Opened Cycle System. The Opened Cycles are in contact with the atmosphere and always use water as the "refrigerant". The Closed Cycles can also use other refrigerants, e.g. ammonia. (Henning, 2004, unpaged).

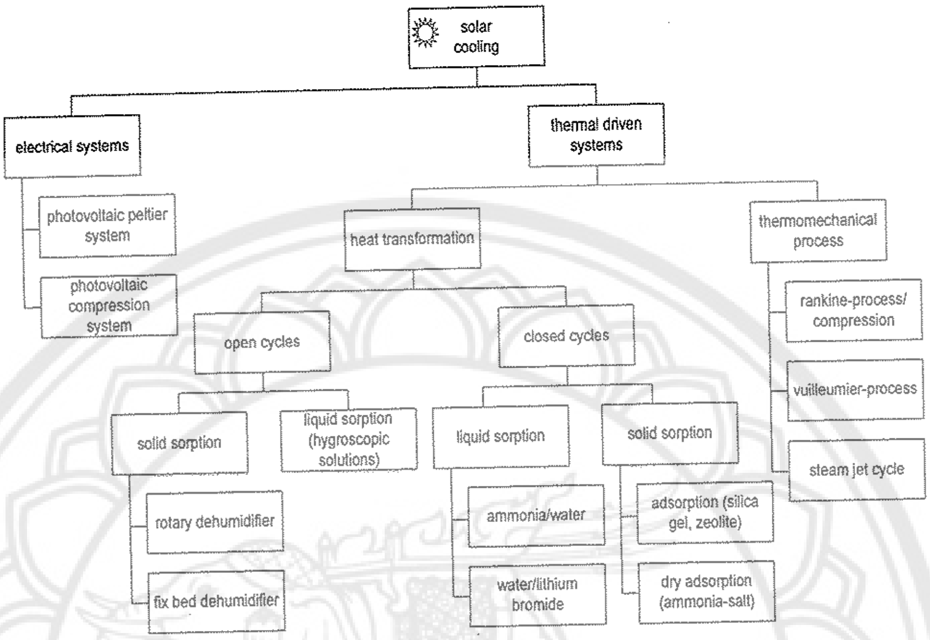


Figure 1 Overview on physical methods to use solar radiation for cooling

Source IEA-SHC TASK 25, 2004

1.1 The Closed-Cycle System

These types of systems are based mainly on the absorption cycle, every thermally driven chiller is characterized by three temperature level see Figure 2, The high temperature level at which the driven heat is absorbed, the low temperature level at which useful cooling is delivered (i.e. the heat from the air condition room is absorbed, and a mean temperature level at which the heat is rejected). In most cases, a wet cooling tower is used for heat transfer (International Energy Agency, 2004). A key figure to describe the performance of a thermally-driven chiller is the thermal Coefficient of Performance (COP), defined as the produced cold per unit of driving heat. Single-effect absorption systems are limited in COP to about 0.7 for LiBr water and to 0.6 for ammonia–water. These systems need; however, high temperature collectors, such as evacuated tube or concentrating collectors. The higher cost of the cooling machine and the solar collector should hence be considered.

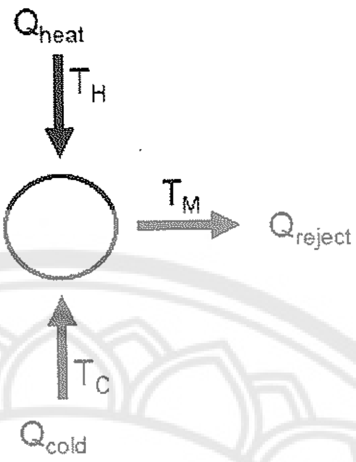


Figure 2 Basic process scheme of a thermally driven chiller

Source IEA-SHC TASK 25, 2004

The absorption system is hence a heat-driven heat pump; the heat may come from a variety of sources, including solar, waste heat and the like. The system operates between two pressure levels and interacts with heat sources/sinks at three temperature levels: The low temperature cooling in the evaporator, the intermediate temperature heat rejection in the absorber and condenser, and the high temperature (solar) heat supply in the generator. A variety of working fluids have been proposed. The two most common absorbent–refrigerant pairs are LiBr–water and water–ammonia. Most solar-powered absorption cooling projects to-date have utilized single-effect systems, with low-temperature solar collectors. Developments in gas-fired absorption systems in recent years for LiBr–water chillers have been made available in the market double-effect systems with COP in the range 1.0–1.2.Triple-effect systems are still under development but close to the market, with COP of about 1.7. These systems may be adapted to and employed in a solar-powered installation with high temperature solar collectors. Figure 3 compares the performance of several multi-effects chillers. Showing the COP as a function of the solar heat supply temperature for typical single-, double- and triple-effect chillers with the same component size and under the same operating conditions. The corresponding Carnot performance curve is also shown for comparison (Mahjouri, 2002). The single-effect system gives best results in the temperature range 80–100°C. For a higher supply temperature, it is worth switching to a double effect system, up to about 160°C, and then to a triple-effect. An advantage of LiBr–water system is in the high boiling point distance between the refrigerant and solvent, so when the refrigerant is expelled from the solution, pure refrigerant vapor develops.

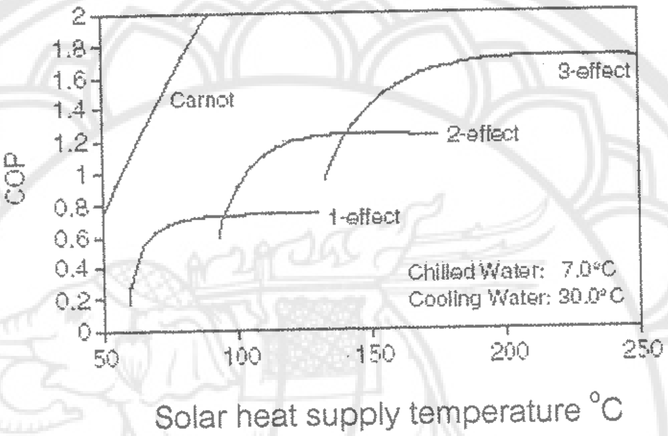


Figure 3 Coefficient of performance (COP) as a function of (solar) heat supply temperature for single -, double - and triple-effect LiBr–Water chillers.

Source IEA-SHC TASK 25, 2004

Adsorption chillers working with solid sorption materials are also available. The main difference compared to the absorption systems is that two or more absorbers are necessary in order to provide continuous operation. Adsorption systems allow for somewhat lower driving temperatures but have a somewhat lower COP compared to absorption systems under the same conditions. The simplicity of the process, the wide range of heating temperatures and other advantages such as noiseless operation could lead to a large number of small solar assisted air conditioning applications. Further research and development work on small-size adsorption machines is necessary in order to reduce their volume and increase the power density.

Table 1 Overview of the main methods of providing cold water with absorption and adsorption chiller for air conditioning

Method	**Absorption chiller**			**Adsorption chiller**
Stages	Single stage	Double stage	Single stage	Single stage
Sorbent	Lithium bromide		Water	Silica gel
Working fluid	Water		Ammonia	Water
Driving temperature	80 °C – 110 °C	140 °C – 160 °C	80 °C – 120 °C	60 °C – 95 °C
Driven by	Hot water, (steam possible, directly heated)	Hot water, steam, directly heated	Hot water, steam, directly heated	Hot water
COP	0.6 – 0.8	0.9 – 1.2	0.3 – 0.7	0.4 – 0.7
Capacity, market available	few manufacturers > 20 kW (hot water), many manufacturer > 100 kW	few manufacturers > 50 kW (hot water), several manufacturer > 100 kW	small capacity directly heated only, high capacity custom made	50 – 350 (Mayekawa) 250 – 500 (Nishyodo)
Manufacturer	York, Yazaki, EAW, Trane, Carrier, Broad, Ebara, LG Machinery, Sanyo-McQuay, Sulzer-Escher Wyss, Entropie, Century		**Directly heated:** Robur, Colibri, **Mattes;** hotwater, steam: Colibri, Matles	Mayekawa, Nishyodo

Source IEA-SHC TASK 25, 2004

The main methods of providing cold water are compared in Table 1 with regard to their key factors; a detailed description with example applications follows. In recent years, many new developments have been achieved to commercialize water chillers with small cooling capacities. Examples of theses are (International Energy Agency, 2005, unpaged):

1. Water-LiBr absorption chiller
2. Ammonia water systems with mechanical solution pump

3. Ammonia water-systems without mechanical solution pump
4. Solid sorption

1.2 The Open-Cycle Systems

Desiccant systems are essentially open sorption cycles, utilizing water as the refrigerant in direct contact with air. The desiccant (absorbent) can be either solid or liquid and is used to facilitate the exchange of sensible and latent heat of the conditioned air stream. The term 'open' is used to indicate that the refrigerant is discarded from the system after providing the cooling effect and new refrigerant is supplied in its place in an open-ended loop.

Using sorptive technology it is possible to condition air (i.e. to maintain its temperature and humidity in a comfortable range, with the aid of thermal driving energy). The function of such systems goes beyond delivering a cooling output, which makes it difficult to compare it directly with cold water supply systems. The definition of the COP for open air conditioning methods and the definition of refrigeration capacity and room cooling capacity are summarized in Figure 4.

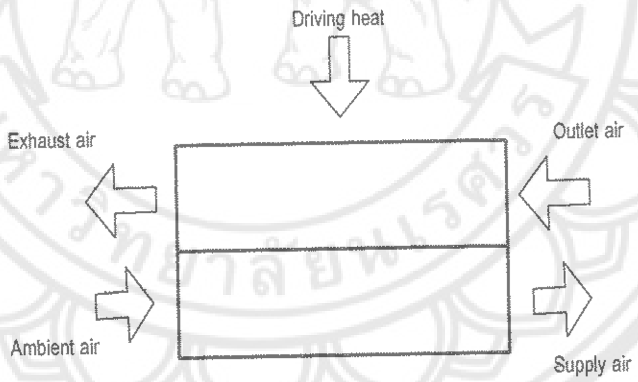


Figure 4 Definition of refrigeration capacity, cooling capacity, and COP in open methods

Source IEA-SHC TASK 25, 2004

For these systems, it equally true that a realistic comparison regarding energy efficiency requires a consideration of all energy consumptions. With open systems, the electrical energy powering the fan is particular important as a high number of a additional components which are usually install compared to conventional ventilation systems; thus entailing greater loss of pressure and therefore more electricity to move the air. The various methods are described below and examples of applications are including.

In the field of open cooling cycles, most new developments are focused on the application of liquid sorption due to its inherent advantages. First, it is more cold when using the sorption process. Second, the concentrated solution can be loss-free storage. Example of stored and provides high densities of new developments of open cooling cycles are (International Energy Agency, 2005, unpaged)

1. Menerga in Mulheim, Germany: new air handling unit using liquid sorption dehumidifier in combination with a standard indirect evaporative cooler
2. Technion Haifa in Israel: small system for treatment of fresh air using liquid sorption
3. ZAE Bayern in Munich, Germany: advanced open cooling system using liquid sorption; concentrated solution used as high energy density storage
4. Fraunhofer ISE in Freiburg, Germany: high efficient indirectly evaporative cooled sorption dehumidifier using a air-to-air plate heat exchanger coated with zeolite

2. Solar Absorption Cooling

Two approaches have been taken to solar operation of absorption coolers. The first is to use continuous coolers with energy supplied to the generator from the solar collector-storage –auxiliary system whenever conditions in the building dictate the need for cooling. The second is to use intermittent coolers similar in concept to that of commercially manufactured food coolers used many years ago in rural areas before electrification and mechanical refrigeration were widespread. Intermittent coolers have been considered for refrigeration but most of them work in solar air conditioning has been based on continuous cycles.

2.1 Continuous Absorption Cycles

These cycles can be adapted to operate from solar collector. The diagram of one possible arrangement is shown in Figure 5.

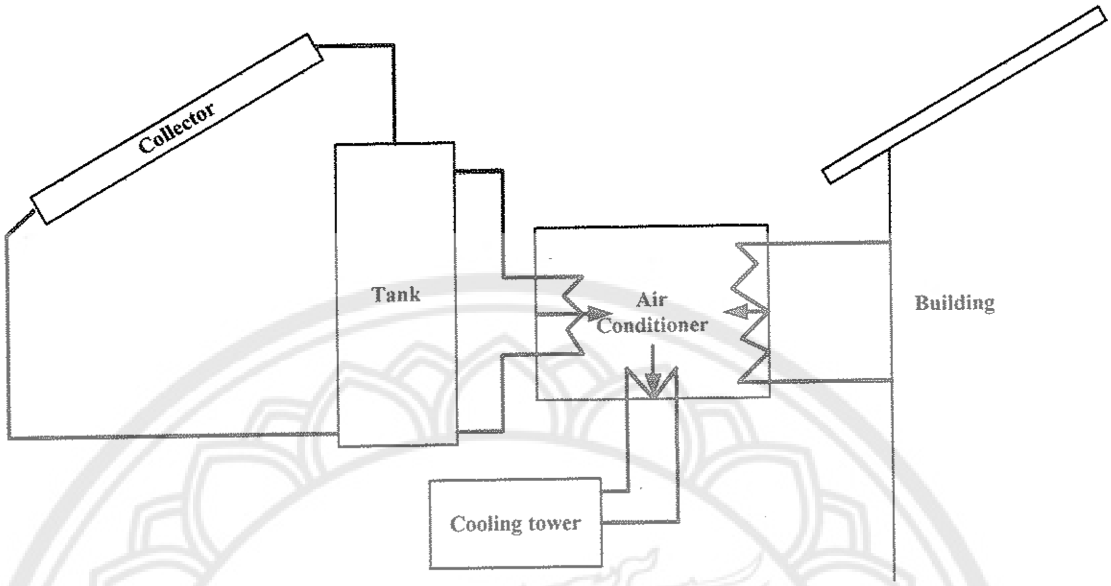


Figure 5 Simplified schematic of solar absorption air conditioning system

Source Solar Engineering of Thermal Processes: Third Edition, 2006

The present temperature limitations of solar collectors restrict consideration among commercial machines to lithium bromide-water systems. The LiBr-H_2O machines require cooling water for cooling the absorber and condenser. And in most applications, a cooling tower will be required. A commercial lithium bromide-water air conditioner, modified to allow supplying the generator with hot water rather than steam, was operation from a solar collector (Chung, Duffie, and Löf, 1963, pp. 132-146). An analytical study of solar operation of a LiBr-H_2O cooler and flat-plate collector combination by Duffie and Sheridan, 1965 identified critical design parameters and assessed the effects of operating conditions on integrated solar operation.

2.2 Intermittent Absorption Cycles

Most work to date on these cycles has been directed at food preservation rather than comfort cooling. These cycles may be of interest in air conditioning because they offer potential solutions to the energy storage problem. In this cycles, distillation of refrigerant from the absorbent occurs during the regeneration stage of operation and the refrigerant is condensed and stored. During the cooling portion of the cycles, the refrigerant is evaporated and reabsorbed.

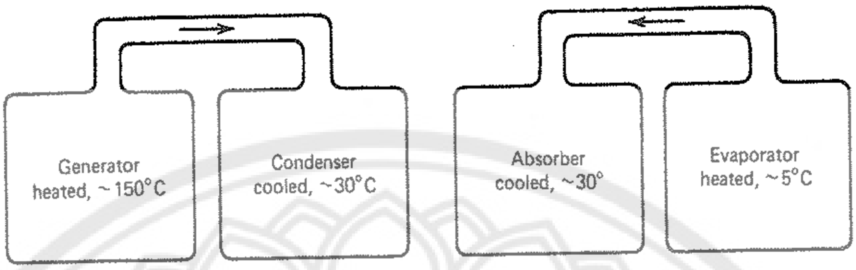


Figure 6 Schematic of an intermittent absorption cooling cycle

Source Solar Engineering of Thermal Processes: Third Edition, 2006

A schematic of the simple cycle in Figure 6 may result in an essentially continuous cooling capacity and improved performance. Refrigerant-absorbent systems used in intermittent cycles have been H_2SO_4-H_2O, NH_3-H_2O, and NH_3-NaSCH.

3. Theory of Absorption Cooling

Operation of absorption air conditioners with energy from solar collector and storage systems is the most common approach to solar cooling. A schematic of a solar absorption cooling system is shown in Figure 7.

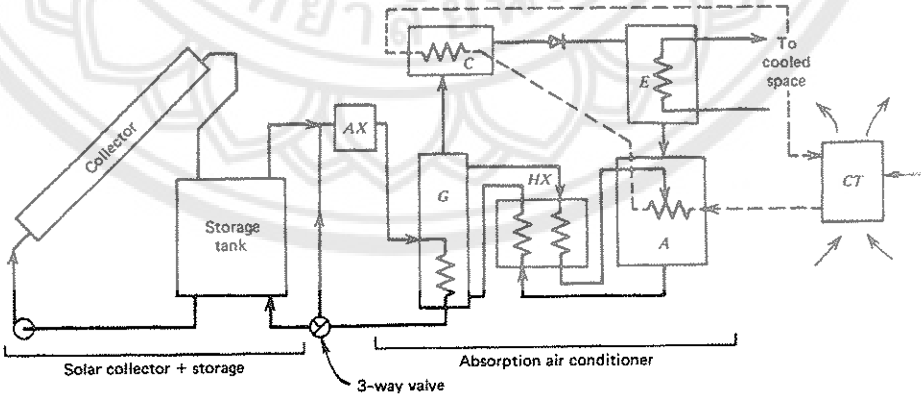


Figure 7 Schematic of a solar-operated absorption air conditioner

Source Solar Engineering of Thermal Processes: Third Edition, 2006

This system has been the basis of most of the experience to date with solar air conditioning. The essential components of the cooler are, A: absorber, B: generator, C: condenser, E: evaporator, HX: heat exchanger to recover sensible heat, CT: cooling tower, AX: auxiliary energy source. The coolers used in most experiments are LiBr-H_2O machines with water-cooled absorbers and condensers. A pressure-temperature-concentration equilibrium diagram for LiBr and H_2O is shown in Figure 8.

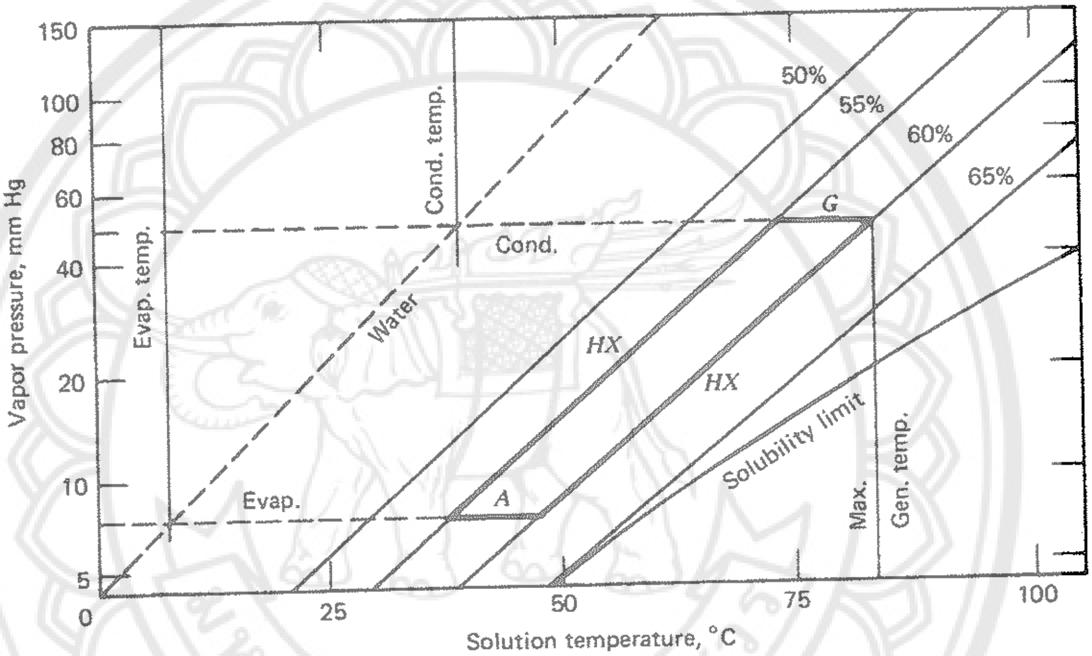


Figure 8 Pressure-temperature-concentration diagrams for LiBr-H_2O

Source Solar Engineering of Thermal Processes: Third Edition, 2006

The idealization of a cycle is indicated on the diagram. The pressure in the condenser and generator is fixed by the condenser fluid coolant temperature. The pressure in the evaporator and absorber is fixed by the temperature of the cooling fluid to the absorber. The letters on the lines representing the cycle correspond to the processes occurring in the components indicated in Figure 7. The generation process is one of increasing the concentration from 55 to 60 % while the equilibrium temperature of the solution rises from 72 to 82 °C at the pressure of the condenser. In the absorber, the solution concentration drops from 60 to 55% as the solution temperature drops from 48 to 38 °C, all at the evaporator pressure. In a real cycle, some sensible heat will have to be transferred in the generator and absorber (the amount dependent on the effectiveness of exchanger HX), There will be pressure changes through the generator due to hydrostatic head and there will be temperature differences across all heat exchangers. Exact pressures, temperatures and concentrations will vary with the machine and operating conditions. The number used here are for illustration of the nature of the process.

The maximum solution temperature in the generator is shown in Figure 8. The temperature of the heated fluid to the generator must be above the maximum generator temperature, which is determined by the condenser pressure and the concentration of the solution leaving the generator. The generator temperatures must be kept within limits imposed by the characteristics of the solar collectors. The critical design factors and operational parameters included solution concentrations, effectiveness of the heat exchangers and coolant temperature.

The pressure differences between the high- and low-pressure sides of LiBr-H_2O systems are small enough that these systems can use a vapor lift pump and gravity return from absorber to generator as an alternative to mechanical pumping to move the solution from the low-pressure to the high-pressure side. Early absorption machines used the vapor lift pump, but more recent designs use mechanical pumps because of improved performance (Duffie and Beckman, 2006, unpaged).

4. The Solar Cooling System: School of Renewable Energy and Technology (SERT), Thailand

4.1 Concept Design of Solar Cooling System

The Solar Cooling System, the first actual solar-thermal cooling system in Thailand, be installed in the School of Renewable Energy and Technology (SERT), Energy Park at Naresuan University in Phitsanulok, Thailand. This system was located at approximately 16 °N latitude and 100 °E longitude. The concept design of this system was achieve the target, not less than 70% energy supply from solar energy and not over than 30% supply from the backup, under metrological environment of Phitsanulok province and operation time was 07:00 – 17:00 of day.

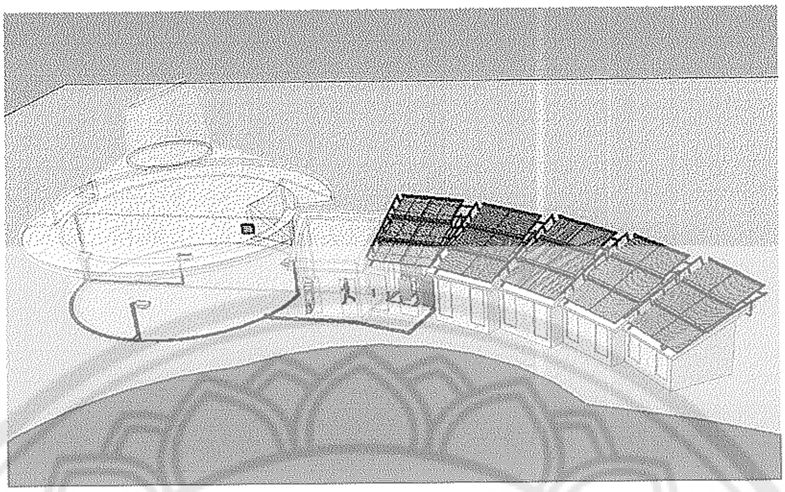


Figure 9 The 3-D graphic design for solar cooling system, SERT, Thailand

Source Ketjoy, Rakwichian, Laodee and Suponthana, 2005

The calculation of cooling load showed the maximums value was 24 kW in May so the 35 kW absorption system of Yazaki WFC SC-10 and 4x33,000 BTU fan coil unit of Carrier 42JB010-CW was chosen for this system.While the actual installed collector area was less than the calculation, so the water heater of Rainnai Infinity 32 as possible to supply 95 °C of water for this proposes (Ketjoy, Rakwichian, Laodee and Suponthana, 2005, unpaged). Figure 10 showed the schematic diagram of a solar absorption air-conditioning system comprised of four main flow circuits, taking into account the generator, condenser, absorber and evaporator. First of all, solar energy was absorbed by the collector and accumulated in the storage tank. Second, the auxiliary heat supplies to boost the temperature of water inlet became the reference range when solar energy was not sufficient to heat the water to the required temperature level needed by the generator. Then, the heat gained was supplied to the generator to boil off water vapor from a solution of LiBr-H_2O. Next, the water vapor was cooled down in the condenser and then passed to the evaporator wherein it again gets evaporated at low pressure, thereby providing cooling to the space to be cooled.

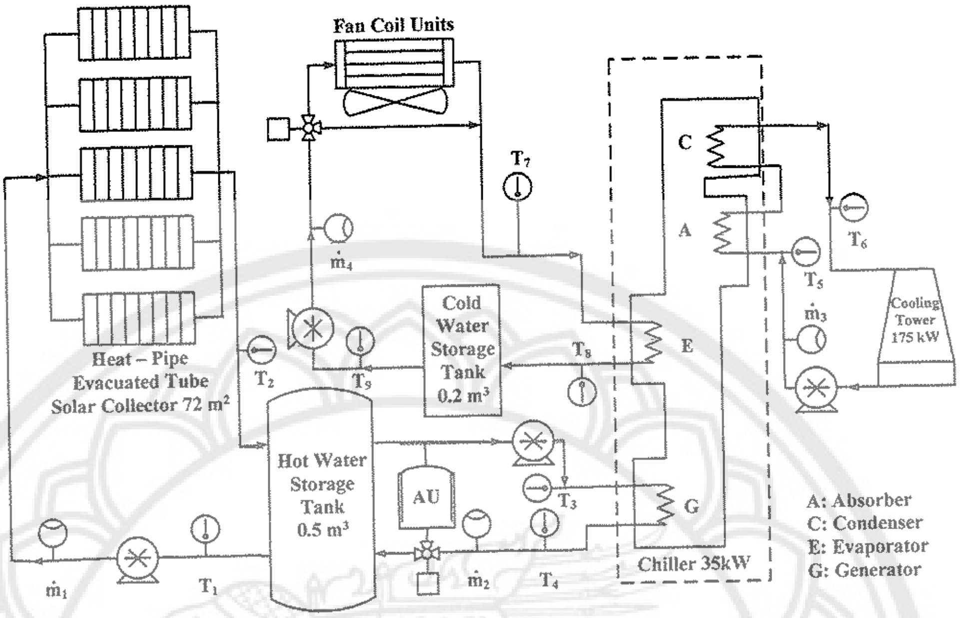


Figure 10 Schematic diagram of the solar absorption cooling system at SERT.

4.2 System Components

This system employed an evaluated heat pipe solar collector of area 72 m^2, a hot water and a cold water storage tank of 0.5 m^3 and 0.2 m^3 in volume, respectively, a 175 kW-cooling tower and a 35 kW nominal cooling capacity LiBr-H_2O absorption chiller that manufactured by Yazaki. The water flow rate in the system maintained 4 units of pumps that were generating the water flow rate via the components of the external circuits as follow; the water flow rate value via the solar collector and hot storage tank were set at 0.95 and 2.25 $kg.s^{-1}$ while the flow rate in the cooling tower and the fan coils were 5.25 and 1.92 $kg.s^{-1}$, respectively, Figure 10 required an auxiliary heat and a network of pumps and pipes, which had many components.

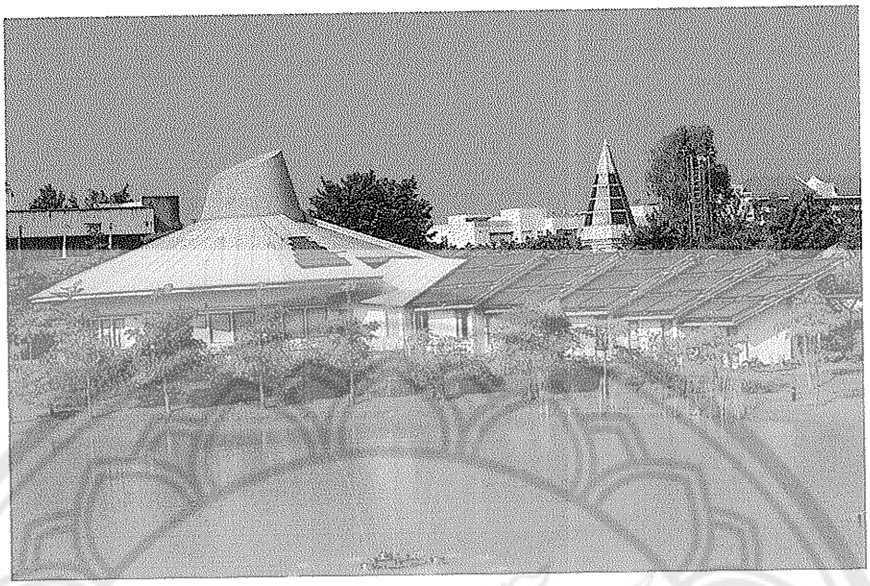
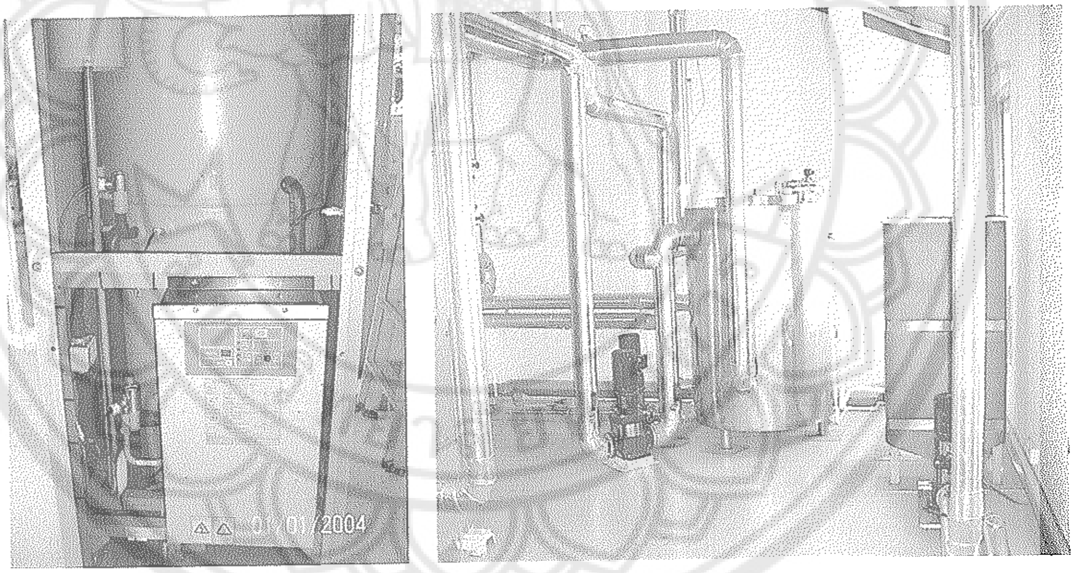


Figure 11 The 35 kW LiBr-H_2O solar cooling system at SERT, Thailand



(a) (b)

Figure 12 (a) The Yazaki WFC SC-10 chiller
(b) The 0.5 m^3 and 0.2 m^3 storage tanks

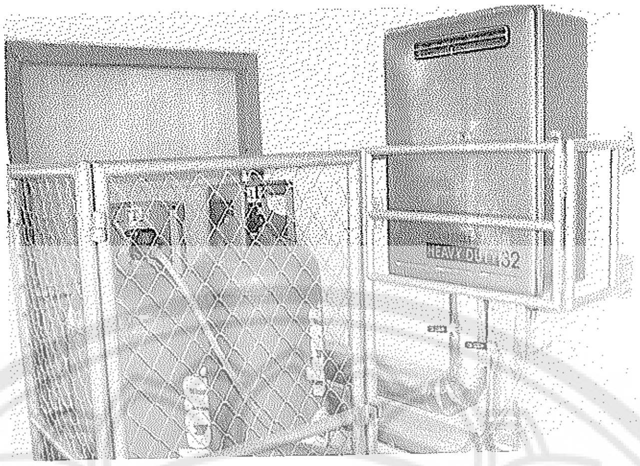


Figure 13 The Rainnai Infinity 32 and the LPG as the auxiliary heat sources

Figure 14 The control room of solar cooling system

Figure 15 The usage area, Testing Building of solar cooling system

Review of Related Researches

Computer modeling of thermal systems presented many advantages. The most important were the elimination of the expense of building prototypes, the optimization of the system components, estimation of the amount of energy delivered from the system and prediction of temperature variations of the system.

Various researchers recently presented modeling and simulation studies for design and optimization the components of solar cooling and air-conditioning systems.

Ghaddar, Shihad and Bdeir (1997, pp. 539-558) presented modeling and simulation of a solar absorption system for Beirut. The results showed that for each ton of refrigeration it was required to have a minimum collector area of 2.3 m^2 an optimum water storage capacity ranging from 1000 to 1500 l when the system was operated solely on solar energy for about 7 h per day. The monthly solar fraction of total energy used for cooling was determined as a function of collector area and storage tank capacity. The economic analysis performed showed that the solar cooling system was marginally competitive only when it was combined with domestic water heating.

Florids, Kalogirou, Tassou and Wrobel (2002, pp. 43-51) presented the modeling and simulation of an absorption solar cooling system with the TRNSYS simulation program in Nicosia, Cyprus. The optimized system consisted of a 15-m^2 parabolic collector tilted 30° and a 600-l hot water storage tank.

Assilzadeh, Kalogirou, Ali, and Sopian (2005, pp. 1143-1159) designed and optimized a solar cooling system for Malaysia that was using evacuated tube solar collector and LiBr absorption unit. The modeling and simulation was carried out with TRNSYS program by the weather parameters in Malaysia. The results presented the optimum system for a 3.5 kW (1 refrigeration) consists of 35 m^2 evacuated tubes solar collector sloped at 20 °.

Khattab (2006, pp. 823-833) developed a mathematic model for simulation and optimization the performance of a solar-powered adsorption refrigeration module with the solid adsorption pair was charcoal and methanol. The optimal ratio of the mass of steel pieces of the mass of the charcoal, $M_{st}/M_{ch}= 0.75$, increased the yearly average net COP from 0.146 to 0.1558 when comparison the model results.

One of the important objectives of investigating the performance of solar driven cooling and heating systems was to determine the optimum operating condition by some researchers.

Asdrubali and Grignaffini (2005, pp. 489-497) simulated and verified the performance of a single-stage H_2O-LiBr absorption machine with an electrical boiler supplied by varying the temperatures and the flow rate of water. The results showed that the machine could work with acceptable efficiency, with input temperature of about 65 – 70 °C.

Wu Chih., Chen Lingen. and Sun Fengrui (1997, pp. 203-208) optimized the collector temperature under the condition of either maximum COP or maximum cooling load for solar absorption refrigerator. The results provided a theoretical base for designing real solar refrigeration systems.

Göktun Selahattin (2000, pp. 625-631) optimized the performance of irreversible solar assisted ejector-vapor compression cascaded systems that focusing on the operating temperature of the solar collector and the maximum overall coefficient of performance of the cooling and heating modes. The results served as a good guide for the evaluation of existing real solar assisted hybrid systems.

Hou, Ye and Zhang (2005, pp. 143-149) presented a method of performance optimization of solar humidification-dehumidification desalination (HDD) process by adjusting mass flow rate ratio of water to dry air with Pinch technology. From Pinch technology analyzed an optimum mass flow rate ratio of water to dry air was 0.67 when the spraying water temperature was 80 °C and cooling water temperature was 30 °C.

Fortunately, the most related research for this work is presented by Zambrano, Bordons, Garcia-Gabin and Camacho (2007, pp. 213-228). It was described the dynamic model that included model development and validation base on first principles and a set of experiments, respectively, carried out on the real plant that has been operational since 2001. The plant used hot water coming from a field of solar flat collectors feed a single-effect absorption chiller of 35 kW nominal cooling capacity that used to cool the Laboratories of the System Engineering and Automation Department of the University of Seville, Spain. The models equations had been written as a Mixed Logic Dynamical (MLD) system which was a class of hybrid systems in which logic, dynamics and constrains were integrated. This equation generated the relative error obtained being less than 3% (in the worst case) on the out put temperature, flow and power of the main component. This simulation was a powerful tool for solar cooling systems both the design and also the development and testing of control strategies.

Several researchers were interested in optimization of economy or energy saving as Kilkis (2006, pp. 10-17) analysed the life cycles cost minimization algorithm used to optimize. The results indicated that with an optimize design; where low-enthalpy energy resources were available. This system may cost effectively reduced the fossil fuel dependency and harmful emission of building. And Colle Sergio and Vidal Humberto (2004, pp. 125-133) was use f–ϕ chart method for evaluation and optimization economy of lithium-bromide absorption chiller and R11 ejector cooling cycle. The optimization was carried out by using the life cycle cost saving function as the objective function that expressed in term of the capital cost, the operated cost, and the solar fraction. A numerical example was presented to compare the optimum bound regions for both cycles; the upper bounds for economical feasibility in terms of costs of the auxiliary energy and electric energy were also set down.

Due to the operational problem was come from the actual COP with the measured value being evaluated at 0.30 which was not as good as expected, because of the manufacturer showed a COP at nominal equal to 0.7. When the solar irradiation that struck on the solar collector was increased, the cooling capacity and the COP were noticeably increased and no variation of the solar fraction was detected when the water flow rate via the chiller was fixed.

Meanwhile this system operated with constant water flow rate condition through the major components of the chiller and system. It is presently not operating under optimized conditions as it is experiencing some energy losses. Therefore, this study would focus on the optimization of COP by adjusting the water flow rate for generating the energy balancing to attain the highest COP of the chiller. Reducing the heat losses also has the added advantage of reducing the consumption of auxiliary heat (i.e. LPG) and this can be both economically and environmentally beneficial.