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Solar Air-Conditioning Systems

Focus on components and their working principles

Nathan Rona

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Abstract

Air-conditioning is one of the major consumers of electrical energy in many parts of the world today and already today air-conditioning causes energy shortage in for example China. The demand can be expected to increase because of changing working times, increased comfort expectations and global warming. Air-conditioning systems in use are most often built around a vapor compression systems driven by grid-electricity. However, most ways of generating the electricity today, as well as the refrigerants being used in traditional vapor compression systems, have negative impact on the environment.

Solar air-conditioning might be a way to reduce the demand for electricity. In addition many solar air-conditioning systems are constructed in ways that eliminate the need for CFC, HCFC or HFC refrigerants.

Alternatives to using solar energy are to use waste heat from different industrial processes such as refineries, garbage treatment facilities etc. Even driving the air-conditioning systems directly with fossil fuels might in some cases be a more environmental friendly alternative than using electricity

This report deals with a wide range of components, from room air-conditioners to solar collectors, which can be used as subcomponents in a solar air-conditioning system. However, most of the components and subsystems covered are not only suitable for solar air-conditioning.

Some components are used for electrically, mechanically or heat driven air-conditioning. And of course other sources of mechanical energy or heat could be used for powering these components.

Other components are used for solar energy collection and storage, which can be used in solar energy system with other purposes than just driving a solar air-conditioning system.

An aim of the report is to describe and explain the working principles of the components and subsystem in such general terms that the report is usable not only to those specifically interested in solar air conditioning, but to anyone interested in air conditioning, heat driven air-conditioning and solar energy.

The last section of the report briefly deals with how the components can be combined to form a complete solar air-conditioning system.

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Keywords: Air conditioning, Absorption chillers systems, Cogeneration, Heat driven air conditioning, Solar air conditioning, Solar energy, Solar equipment, Solar heat engines,Thermal Heat Pumps, Waste heat utilization.

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Preface

This report is the result of of two individual adapted advanced courses in solar air-conditioning, given by the department of Building Technology at Chalmers University of Technology. These courses will be part of my master of science degree in engineering.

The report could not have been written without all the assistance I have received from people, both at the department and outside the department. My thanks are therefore due to all those who have assisted, provided information or given feedback to the report, whether named or unintentionally forgotten.

Particular thanks are due to Professor Per Fahlén, Doc. Jan-Olof Dalenbäck and especially Dr. Torbjörn Lindholm (my primary supervisor).

Special thanks also due to Professor Gershon Grossman, head of Energy Engineering Research Center, Faculty of Mechanical Engineering, Technion - Israel Institute of Technology, Wimolsiri Pridasawas Ph.D. student at the Division of Applied Thermodynamics and Refrigeration, Department of Energy Technology, Royal Institute of Technology, Sweden, and Dr. Warick for their valuable assistance and feedback.

Thanks also to Alain Dahan at SOLEL Solar Systems Ltd. and to Erwin Kose at microtherm Energietechnik GmbH for providing me with information, as well as to all those who those who have given me permission to use there illustrations in the report, especially to those who gave it free of charge (all except Taylor & Francis).

If you come to appreciate this report, give a thought of thanks also to the ones mentioned above who have helped me writing it, but above all, to the One above who gave me the abilities, the understanding and the wisdom needed to write it.

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Any feedback on the report will be highly appreciated.

Göteborg, Kislev 5765/Nov. 2004

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1. Introduction

1.1 General Background

Air-conditioning is one of the major consumers of electrical energy in many parts of the world today. For example about 40 % of the summer electricity used by commercial buildings in the United States is used for air-conditioning (Syed 2002a), and in Egypt at least 32 % of the electrical energy used by the domestic sector is for air-conditioning. (Elsafty 2002). Further more, the demand is growing, and this trend is expected to continue as traditional working times in many locations with midday siesta is changed to the rhythm typical to Central Europe (Lamp 1998). The demand is also growing due to increasing comfort expectations and cooling loads (Biezanowski 2003).

The average temperature worldwide is also expected to rise because of the global warming and this might already have begun. WMO (World Meteorological Organization) writes in a prerelease 2 July 2003 that "...the increase in temperature in the 20th century is likely to have been the largest in any century during the past 1000 years...1990s were the warmest decade and 1998 the warmest year... Global average land and sea surface temperatures in May 2003 were the second highest since records began in 1880. Considering land temperatures only, last May was the warmest on record "(WMO 2003).

Nearly all air-conditioning systems in use are built around vapor compression systems driven by grid-electricity. However, most ways of generating electricity used today has some kind of negative impact on the environment, whether it is emissions of carbon-, sulphur-, and nitrogen dioxide (fossil fuel plants), radioactive waste (nuclear power), destroyed rivers and water falls (hydropower) or noise pollution (wind power). Therefore it is desirable to reduce or at least to prevent the increase of electrical demand.

Refrigerants being used in traditional vapor compression systems also have negative impacts on the environment. The CFC (Chloro-Flouro-Carbon) and HCFC (Hydro-Chloro-Flouro-Carbon) refrigerants also known under the trademark Freon®s are known to be harmful to the ozone layer, and therefore they are currently being phased out. An other problem is that the CFC and HCFC refrigerants as well as the HFC (Hydro- Flouro-Carbon) refrigerants (which are often used to replace the CFCs and HCFCs,) are all greenhouse gases (Pita 1998).

Solar air-conditioning might be a way to reduce the demand for electricity. The constructions of many solar air-conditioning systems today are also such that CFC, HCFC or HFC refrigerants are not used.

Other options are to use waste heat from different processes such as industry, refineries, garbage treatment facilities etc. Even driving the air-conditioning systems directly with fossil fuels might in some cases be a more environmental friendly alternative than using electricity

1.2 Report background, aims and coverage

A lot of material has been published that briefly describes some different solar air-conditioning and refrigeration system. There are also a larger number of publication available that in depth cover either a specific solar air-conditioning system or different components that can be used for a solar air-conditioning system.

However, a publication that collects, covers and describes, a little more in depth the working principles of a wider range of different components and subsystem, from cold air distribution to solar collector of a solar air-conditioning has not been found.

The primary aim of this report is to fill this gap, to collect and explain the principles of these building blocks that together make up a solar air conditioning system.

The report also briefly deals with how the building blocks can be combined to form a complete system.

Most of the components and subsystems covered are not only useable for solar air-conditioning.

Some component are for electrically, mechanically or heat driven air conditioning systems. And of course other sources of mechanical energy or heat could be used for powering these components.

Others components are for solar energy collection and storage, which can be used also in solar energy system with other purposes than driving a solar air-conditioning system.

An aim of the report is to describe and explain the working principles the components and subsystem in such general terms that the report is usable not only to those specifically interested in solar air conditioning, but to anyone interested in air conditioning, heat driven air conditioning and solar energy.

The in depth analysis of the different components are limited to exploitation of their function. Detailed mathematical models are not presented. One major reason for this is that also readers who do not necessary have a scientific or engineering background should be able to read and understand this report.

1.3 Report structure

The report consists of three parts, introduction included.

Part 2, is the main part of the report. It handles the different components and subsystems that make up a solar air conditioning system. It consists of 5 sections each handling different parts of the system. Each chapter is split into different sub sections covering different components. Only one of the 5 sections, section 2.3, is directly linked to solar energy. The other covers components that can be used in solar driven as well as other air conditioning system.

Part 3 briefly handles how the components and subsystems in part 2 can be combined to form a complete system.

1.4 Definitions and concepts

Whenever the word *air-conditioning* is used in this report it refers to *dehumidification* and *cooling*. Likewise *solar air-conditioning* refers to *active solar cooling*. *Passive solar cooling systems* (*s*hading of solar insolation, solar assisted ventilation etc.) are not included in this report.

Whenever the word *vapor pressure* is used it refers to *saturation vapor pressure*.

The following concepts are explained in appendix A: *Mechanical work, heat, vapor, partial pressure, vapor pressure (, saturation vapor pressure, saturation pressure), evaporation, boiling, boiling point, heat of evaporation, latent heat, sensible heat, condensing, dew point, relative vapor pressure, relative humidity, superheated vapor* and *subcooled liquid.*

2. Components

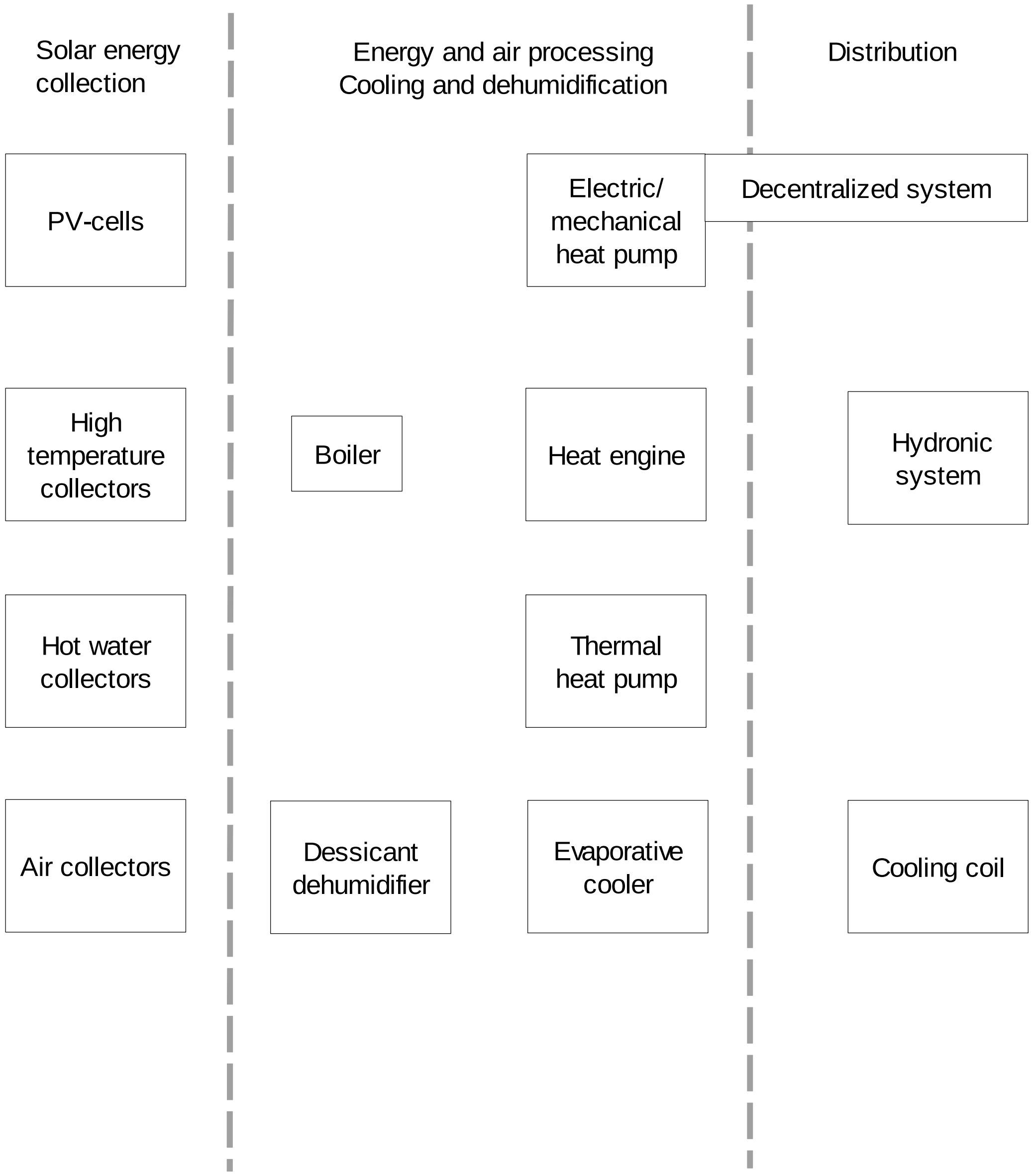

Fig 2.1 Components of solar air-conditioning systems

Figure 2.1 shows the components of different solar air-conditioning systems available, heat rejection, energy storage and auxiliary heat supply or auxiliary cooler excluded. This schematic is divided into three layers: Solar energy collection, processing (cooling and dehumidification) and distribution (of chilled brine and air).

From a user perspective the two things that are needed, removal of sensible and latent heat, are accomplished either with a supply of treated air or by air-conditioning devices within the occupied space. For treating the supply air-cooling coils can be used. Both the air-cooling coils and local air-conditioning devices need either, chilled brine or refrigerant to operate. All these components, together with ductworks for air and pipes for brine, are grouped together in the distribution layer, which is handled in section 2.1.

The object of components in the processing layer, which are handled in section 2.2, is to supply treated air and chilled brine or refrigerant to the distribution layer. Some of the components do not service the distribution layers directly; instead they provide necessary resources to other components within the layer. The processes done by the components of the processing layer require energy. In the case of a solar air-conditioning system this energy is solar energy that is collected by the components in the solar energy collection layer. The solar collection components are treated in section 2.3. In many of the processes heat needs to be rejected, methods and components for this is handled in section 2.4.

In section 2.5 components and principles for energy storage is handled.

2.1 Distribution

The two aspects of air-conditioning that are of interest to this report, cooling and dehumidification, can be obtain in two ways. Either the air presently in the room is replaced with supply air that is cooler and dryer than the air being replace (all-air-systems), or cold surfaces inside the room are used (hydronic systems or decentralized systems with small room air-conditioners).

2.1.1 All-air-systems

In an all-air-system (fig. 2.2) the outdoor and, in some systems, part of the return air normally passes a cooling coil before being supplied throughout the building by a duct-work.

Since air has low specific heat capacity and the temperature of the supply air cannot be too low due to comfort problems, the air-flow has to be large (often several times that needed for ventilation) (Abel 2003). This has the effect that a lot of space is needed for ducts and shafts and that additional fan power is needed.

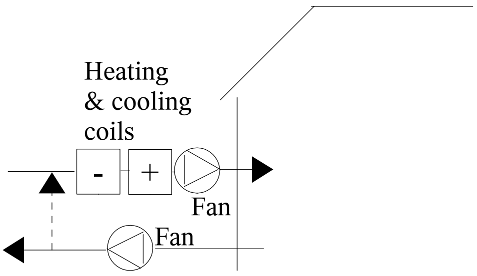

Fig. 2.2 Schematic all-air-system

To accomplish dehumidification the temperature of the cooling coil is often held slightly above 5 °C. When latent (humidity) loads are high, the air has to be cooled to very low temperatures to achieve the desired dehumidification. In most of these cases the air needs to be reheated to avoid comfort problems (Lindholm 2003a). Reheating can be accomplished by letting the cooled air pass through a heating coil or air-air heat exchanger. Alternatively the treated air can be mixed with untreated air to accomplish a mixed air stream with desired temperature and humidity.

Alternative techniques for cooling and dehumidification are evaporative cooling and desiccant dehumidification. These techniques are described in sections 2.2.4 and 2.2.5.

An advantage of the all-air-system is that when outdoor temperature and humidity is lower than required supply air condition, air-conditioning of the interior space can be accomplished without using refrigeration machinery (Abel 2003).

2.1.2 Hydronic systems

In a hydronic system (fig. 2.3) heat is transferred from the room by transferring the heat from the interior air through cold surfaces to a brine. The interior air comes to contact with the cold surfaces either through forced convection, or through natural convection.

Devices using force convection are Fan-coils and induction units. Air is forced through cooling coils and returns to the room cooled.

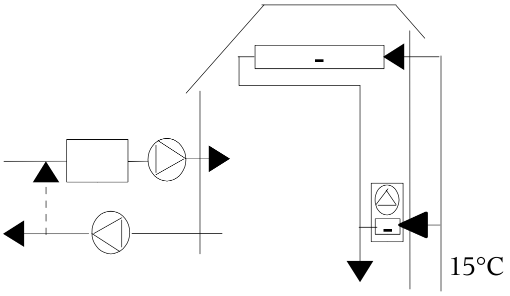

Fig. 2.3 Schematic hydronic system

Ceiling cooling panels work by natural convection. When the warm interior air comes in contact with the (cool) ceiling panels the air is chilled. As the air is chilled its density increases and therefore it falls down towards the floor. Thus air circulation is created in the room and the chilled air is distributed. Ceiling panels also take up heat radiation emitted from surfaces and people in the room.

The brine, in most cases water, is constantly being replaced with new cold brine. The supply temperature in a hydronic system is normally set above the dew point (the point at which condensation occurs), typically around 15 °C (Lindholm 2003a).

Since the temperature of the surfaces is above the dew point, no dehumidification is achieved in the room. Therefore latent (humidity) loads, internal (from people etc.) as well as from leakage of humid outdoor air, must still be handled through supply of dry air (Lindholm 2003a).

An alternative is to work with a brine supply temperature below the dew point but this requires a condensate drain system, which can be difficult and costly, and requires additional maintenance (ASHRAE 2000). However, not all equipment can be equipped with a condensate drain system, this is true for most ceiling panels for example.

2.1.3 Decentralized systems

An alternative to the all-air and the hydronic system is to use a decentralized system where each room (or office, level etc.) is conditioned by one (or several) small local air-conditioning unit(s). In this system either window/wall units or split units are used.

A window/wall unit is a small air-conditioner (1.2-10 kW) where the compressor, the evaporator and the condenser coils together with needed fans and air filters is packed into one unit. Thus it includes everything needed to cool, dehumidify and filter the indoor air. It may also include means for ventilating and/or exhausting and heating (ASHRAE 2000). The unit can be mounted though a window, or through a hole in an exterior wall. There are also units where the whole unit is located indoors and the heat is rejected, together with exhaust air, through a short duct to the outside.

Figure 2.4 shows a typical window/wall air-conditioning unit in cooling mode. Warm room air passes over the cooling coil (evaporator) and gives up sensible and latent heat. The conditioned air is then recirculated in the room by a fan or blower (ASHRAE 2000).

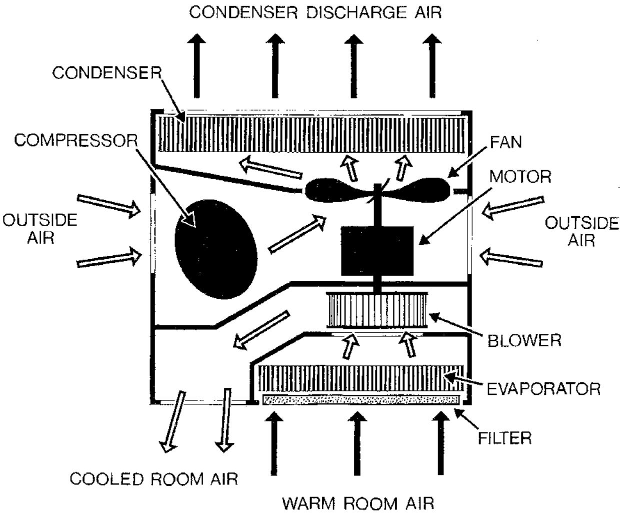

Fig. 2.4 Schematic drawing of typical window/wall air conditioning unit (ASHRAE 2004$^{©1}$*)*

In a split unit, the air-conditioner is split in to two units,one outdoor unit containing the condenser and the compressor and one indoor unit containing the evaporator. The two units are connected by a refrigerant loop transporting the refrigerant between the outdoor and the indoor unit. There are also systems where a single outdoor unit supplies refrigerant to several indoor units. These systems are called multisplit systems.

A decentralized system has several advantages; the initial cost is lower, no ductwork or pipes are needed and the installation is simple (ASHRAE 2000). The later makes it especially attractive for retrofit application. Disadvantages are short equipment lifetime (10 yrs.), noise and require routine unit maintenance (ASHRAE 2000). An other aspect that might be a disadvantage when it comes to solar air-conditioning is that it is an all-electric system.

There are also systems where small decentralized air-conditioning units reject heat to a central cooling water loop. The heat rejection is thus centralized. These systems are called water loop systems and are somewhat in between the hydronic and the decentralized systems.

2.2 Processing (Refrigeration and dehumidification)

2.2.1 Mechanic and electric heat pumps

Mechanic and electric heat pumps are both working between two temperature levels. The basic principle of a heat pump working between two temperature levels are shown in figure 2.5.

Heat $\dot{\mathbf{Q}}_0$ collected at a lower temperature level $\mathbf{T_0}$ is lifted and ejected at a higher temperature level $\mathbf{T_1}$. To accomplish this lift power $\dot{\mathbf{W}}_t$ is needed, during the process this power is degraded to heat and ejected together with collected heat $\dot{\mathbf{Q}}_0$ at temperature level $\mathbf{T_1}$. $(\dot{\mathbf{Q}}_1 = \dot{\mathbf{Q}}_0 + \dot{\mathbf{W}}\mathbf{t})$

When used for air-conditioning heat is collected at a temperature below room temperature and ejected at a temperature above that of some available heat sink.

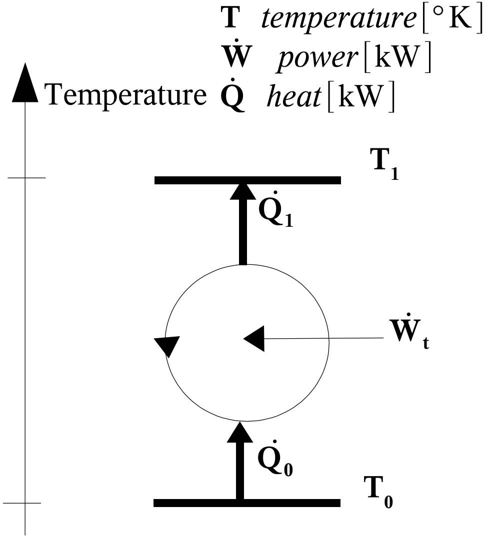

Fig. 2.5 Schematic heat pump working between two temperature levels

For example in an air-conditioner heat can be collected with a cooling coil at 5 °C and ejected to the outdoor air with a heating coil at 50 °C.

A useful measure to compare performance of air-conditioning and refrigeration equipment is the Coefficient of Performance (COP), which is defined as:

$$\mathbf{COP} = \dot{\mathbf{Q}}_0 / \dot{\mathbf{W}}_t \quad \text{(e1) (COP = useful cooling capacity/power input)}$$

The ideal refrigeration process is given by the so-called Carnot refrigeration cycle. An extensive explanation of this cycle is outside the scope of the report, but since this it is an ideal refrigerator it is used to define the upper COP-limit. The Carnot COP for a refrigeration process working between two temperatures is:

$$\mathbf{COP_{Carnot}} = \mathbf{T_0/(T_1 - T_0)} \quad \text{(e2)}$$

A refrigeration process' efficiency can thereby also be declared as its efficiency compared to an ideal refrigerator by the so-called Carnot efficiency:

$$\boldsymbol{\eta}_{\mathbf{Carnot}} = \mathbf{COP/COP_{Carnot}} \quad \text{(e3)}$$

2.2.1.1 Vapor compression systems

The vapor compression system is the dominant system today for cooling and refrigeration and is being used in almost all kind of application. It is available in a large range of sizes from 50 W up to 50 MW (Lindholm 2003a).

Because of its dominant position and since it simplifies the understanding (and explanation) of other techniques it is explained in more detail.

A schematic flow diagram of the vapor compressor system and its components are shown in figure 2.6.

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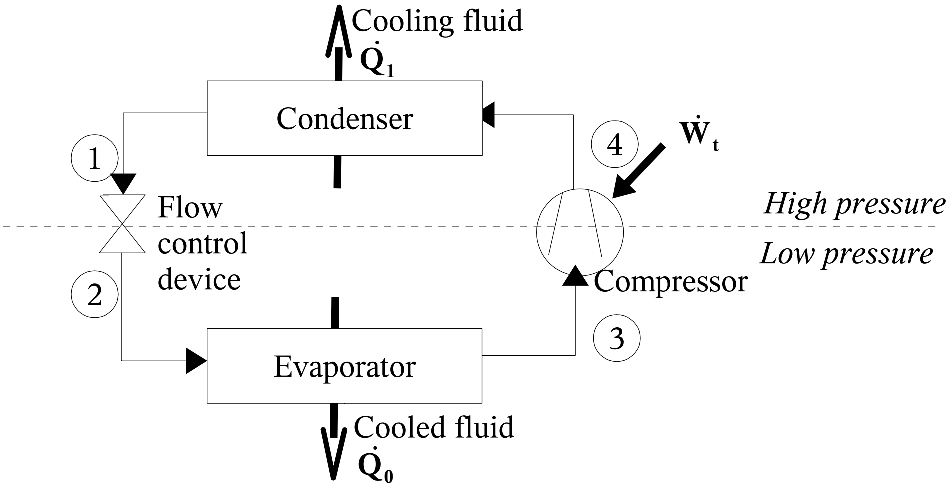


Fig. 2.6 Schematic vapor compression cycle

Pita (1998) explains the vapor compression cycle as follows:

PROCESS 1-2. At point (1), the refrigerant is in the liquid state at a relatively high pressure and high temperature. It flows to (2) though a restriction, called the *flow control device* or *expansion device.* The refrigerant loses pressure going through the restriction. The pressure at (2) is so low that a small portion of the refrigerant *flashes* (vaporizes) into a gas. But in order to vaporize, it must gain heat (which it takes from that portion of the refrigerant that did not vaporize), thus cooling the mixture and resulting in a low temperature at (2).

PROCESS 2-3. The refrigerant flows through a heat exchanger called the *evaporator.* This heat exchanger has two circuits. The refrigerant circulates one, and in the other the fluid to be cooled (usually air or water) flows. The fluid to be cooled is at a slightly higher temperature than the refrigerant , therefore heat is transferred from it to the refrigerant, producing the cooling effect desired. The refrigerant boils because of the heat it receives in the evaporator. By the time it leaves the evaporator (4), it is completely vaporized.

PROCESS 3-4. Leaving the evaporator the refrigerant is a gas at a low temperature and low pressure. In order to be able to use it again to achieve refrigerating effect continuously, it must be brought back to the conditions at (1)-a liquid at a high pressure. The first step in this process is to increase the pressure of the refrigerant gas by using a *compressor*. Compressing the gas also results in increasing its temperature.

PROCESS 4-1. The refrigerant leaves the compressor as a gas at high temperature and pressure. In order to change it to a liquid, heat must be removed from it. This is accomplished in a heat exchanger called the *condenser*. The refrigerant flows through one circuit in the condenser. In the other Circuit, a cooling fluid flows (air or water) at a temperature lower than the refrigerant. Heat therefore transfers from the refrigerant to the cooling fluid, and as a result, the refrigerant condenses to a liquid (1).

A compressor can be driven mechanically from an external source by an exposed shaft (with a seal to prevent leakage), or be built as a single unit with an electric driving motor and the actual compressor sealed in a casing. The former is called an *open* compressor and the later a *hermetic* compressor.

Open compressors are available in all sizes while hermitic are available mostly as smaller units primary for household refrigeration and freezing applications (Lindholm 2003a). Smaller air conditioning units like window/wall and split units as well as resident heat pumps also use hermetic compressors. Also medium size chillers seem to be obtainable with hermetic compressors. An advantage of the hermitic compressor is that there is no risk for leakage at the shaft seal (Lindholm 2003a). A disadvantage of hermetic compressors is that if the available energy is mechanical work it must first be transformed into electricity and then back into mechanical work inside the sealed unit. This double transformation results in poorer efficiency.

The COP value of a vapor compression systems for air conditioning seem to be from around 3 for smaller to medium size units and up to 4-5 for larger systems (Gordon 2000, Filipe Mendes1998, ASHRAE 2000). A rule of thumb estimate of the COP of vapor compression system is given by:

$$\mathbf{COP_{tot}=0.45*COP_{Carnot}} \quad \text{(e4)}$$

This estimate however seems to be a bit conservative especially for larger systems.

Important notice: For a vapor compression system three COPs can be defined .

- $\mathbf{COP_d}$ is the ideal COP for a vapor compression system ($\mathbf{COP_d}$ is not to be confused with $\mathbf{COP_c}$ which is the ideal COP for any refrigeration process)
- $\mathbf{COP_{compressor}}$ is the COP of the system when internal losses in compressor are taken in to consideration. In other word it gives the shaft power input per unit of refrigeration $\mathbf{COP_{compressor}=\dot{Q}_0/\dot{W}_{t,compressor}}$ (e5). This is the COP value that according to ASHRAE (2000) is used for open compressor, therefore this book defines it as **COP (open).** When literature and articles refers to COP:s around 5 it is most likely this COP that is meant.
- $\mathbf{COP_{tot}}$ is the COP when also losses in driving engine and transmission is included. It gives the input power to the driving engine per unit of refrigerating capacity $\mathbf{COP_{tot}=\dot{Q}_2/\dot{W}_{tot}}$ (e6). The efficiency of an electric motor is about 0.75 for a 1 kW motor and 0.95 (Lindholm 2003a) and 0.95 for 75 kW and above (ASHRAE 2000). The efficiency of a transmission is usually around 0.95 (Lindholm 2003a). Since this is the main COP of interest to a hermetic compressor it is also called $\mathbf{COP_{hermetic}}$ (ASHRAE 2000).

2.2.1.2 Stirling cooling

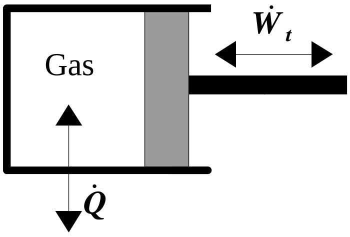


Fig 2.7 Closed cylinder with a contained gas and a moveable piston.

Figure 2.7 shows a closed cylinder with a contained gas and a moveable piston. When the piston is moved the gas either heats up or cools down depending on the direction of the movement. If the piston is forced inwards the gas is compressed and the temperature rises. If the piston moves outward the gas expands and is cooled down. If heat is allowed to pass in and out of the cylinder, heat will pass into the cylinder during decompression and out of the cylinder during compression. These are all very basic principles and are used as examples to understand fundamental thermodynamics in textbooks.

If the piston is pushed in and pulled out while the cylinder is in a closed room little will be gained. However if the piston is pulled out (causing the gas to expand) in one closed room, and the cylinder is then moved over to another closed room and there pushed in (causing the gas to compress), heat will be taken from the first room to the second. By repeating this the first room will be cooled and the second heated. This is the basic principle of a Stirling heat pump.

Moving the whole cylinder from one room to another is not very convenient; therefore the cylinder (or gas container) is divided into two parts. In one part the gas expands and heat is absorbed, in the other the gas is compressed and heat is rejected. The gas is moved between these two parts. This can be done in two ways. Either two pistons are used moving the gas between them (α-configuration) or a *displacer* is used to force the gas to the hot or the cold part (β,γ-configuration) . To prevent heat from flowing with the gas from the hot to the cold side, and thereby increase efficiency, the gas passes a *regenerator* (heat barrier and store) where it leaves some of its heat. Figure 2.8 shows a complete cycle with a *α and β-configuration* Stirling heat pump.

The process can be driven either with mechanical work (for example through a crank shaft) or with electricity through a linear engine and sealed together with the Stirling heat pump in a casing. The later, called free piston heat pump, have the same advantages and disadvantages as a hermitic compressor compared with an open compressor (section 2.2.1.1). Other methods than piston to compress the gas, such a rotation and even acoustic waves are also possible but out of the scope of this report.

Stirling refrigerators are currently being used for cryogenic cooling applications, where they are superior to vapor compression systems. Helium is used as the working gas when used for cryo-cooling. When used for air conditioning however helium has the disadvantage that it is difficult to contain. If there is a seal to allow mechanical power input it is likely that there will be some leakage and need for periodic refill. For near ambient temperature refrigeration, air can be used at a little less efficiency. Air is much easier to contain (Walker 1994).

The COP for a Stirling heat pump, at temperatures normal to air conditioning, seems to be about the same as for a vapor compressor. The COP of the Stirling heat pump improves compared to that of the vapor compressor heat pump as the temperature lift between T_0 and T_1 increases.

The main advantage of the Stirling heat pump for air conditioning application (compared to the vapor compressor system) seems to be that it does not use any potentially harmful refrigerants.

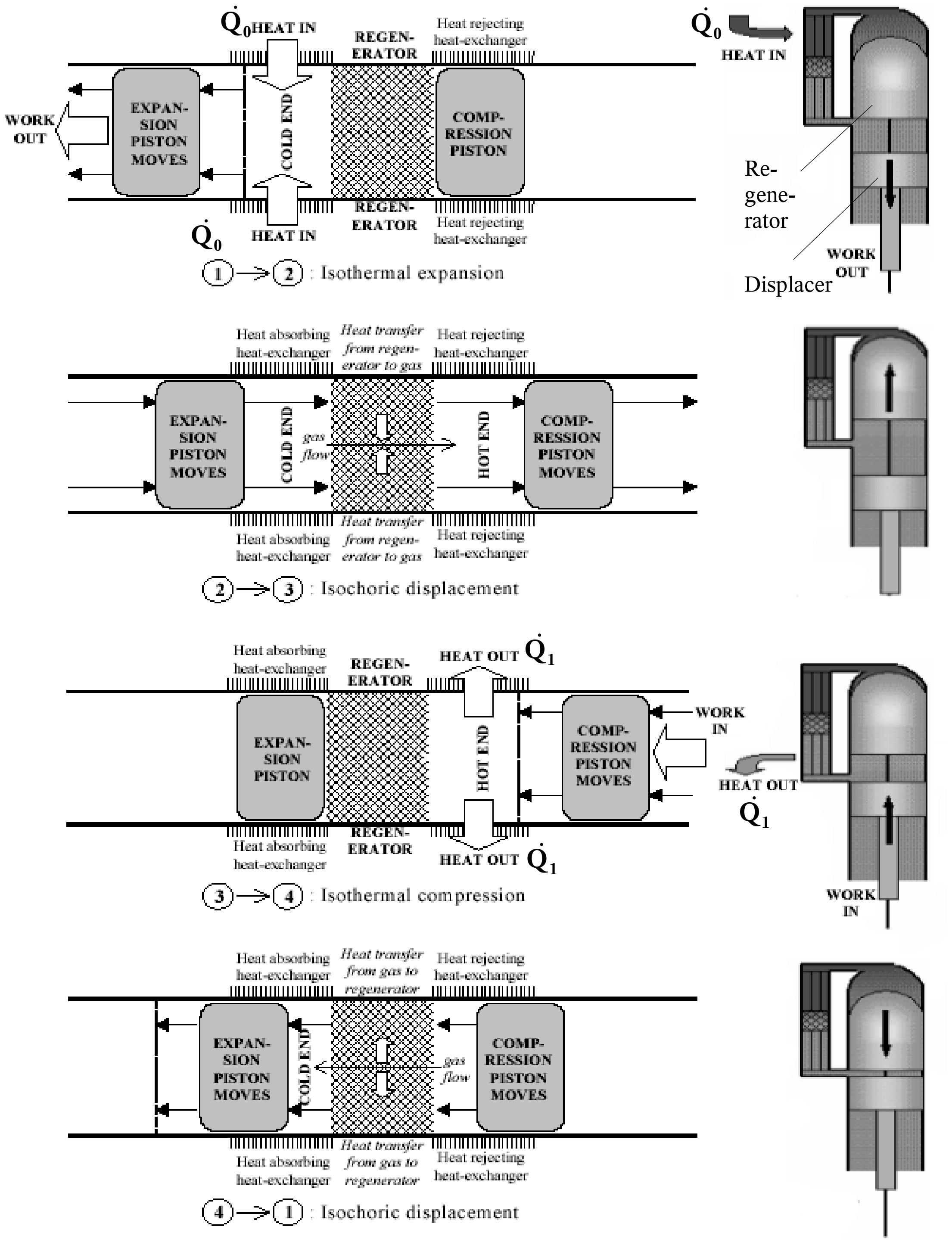


Fig. 2.8 Thermodynamic processes in the Stirling-cycle refrigerator or heat pump shown on simplified a and b-configuration machines (Haywood 2002 [©2]*, NASA 2003)*

2.2.1.3 Brayton refrigeration cycle

The Brayton refrigeration cycle, or the gas refrigeration cycle as it is also called have similarities with both the vapor compression system and the Stirling refrigeration system. The closed Brayton refrigeration cycle is illustrated in figure 2.9.

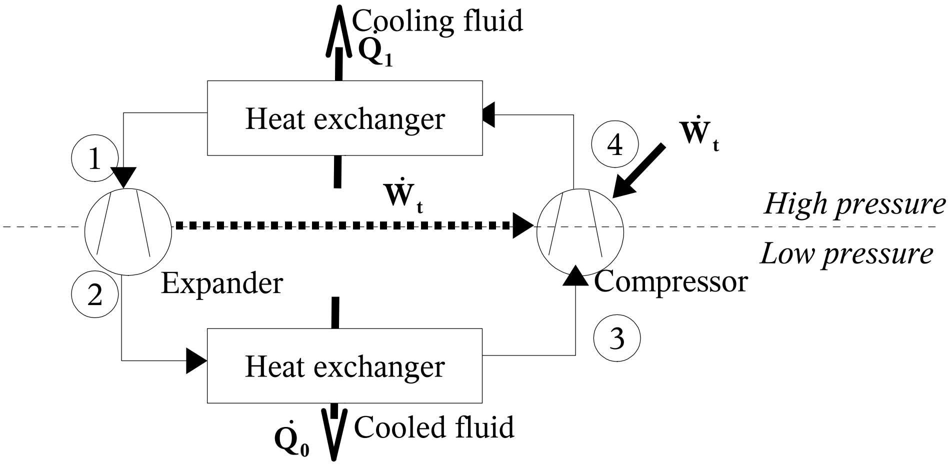


Fig. 2.9 Schematic Brayton refrigeration cycle

The main difference from the vapor compression cycle is that the working fluid is constantly in gas phase. Therefore there is no condenser and no evaporator. Instead there are two heat exchangers, one warm on the high-pressure side and one cold on the low-pressure side. The process works in the following way:

Process 1-2 At point (1) the working fluid is at a relatively high pressure and high temperature. It flows from point (1) to point (2) through an *expander*, which is basically the reverse of a compressor. Gas is expanded and work is taken out. This work is used to help power the compressor. When the gas expand its temperature drops.

Process 2-3 As in the vapor compression system the working fluid flows through a heat exchanger where it takes up heat from the fluid that is to be chilled. However, since the working fluid is already in the gas phase no evaporation takes place. The process is a heat exchange process instead of a boiling process and the temperature varies along the working fluids flow path.

Process 3-4 After the working fluid has been heated in the heat exchanger it is compressed and transferred from the low-pressure side to the high-pressure side. As the gas is compressed its temperature increases.

Process 4-1 The high temperature gas is cooled by a cooling fluid in a heat exchanger and the cycle is completed.

The reason why an expander is used instead of the much simpler flow control device used in the vapor compression system, is that since no phase transformation takes place a much larger volume flow of working fluid is needed. If a simple flow control device were to be used the energy losses would be far more than acceptable. Still, even with the expander this cycle is much less efficient for air-conditioning than the vapor compression and the Stirling cycles. It is however used for cryogenic cooling.

In the open Brayton cycle the low-pressure heat exchanger is removed. Instead ambient air is sucked in at point (3), compressed, cooled, expanded and ejected into the conditioned space. This system is used in commercial jet aircraft. No compressor is needed in this case since compressed air is available from the planes engines.

2.2.1.4 Thermoelectric modules

Thermoelectric (TE) coolers, sometimes-called thermoelectric modules or Peltier coolers are small all-electric heat pumps. They work by the so-called Peltier effect; that when a current passes through the circuit of two dissimilar conductors, depending on the current direction, the junction of the two conductors will either absorb or release heat (Kryotherm 2003). If these two conductors are coupled in a closed circuit with a voltage source as in figure 2.10 heat $\dot{Q}$ will be pumped from the cold junction **A** to the hot junction **B**. (Ferrotec America 2003a) (, in accordance with the principles of two temperature heat pumps as stated above and shown in figure 2.5). The carriers of heat in TE cooler are electrons. In the cold junction **A** the electrons absorbs heat by and increases their (kinetic) energy. These high-energy electrons flow against the electric current **I** over to the hot junction **B** where they by the Peltier Effect loose energy to a heat sink (Ferrotec America 2003a). The Peltier effect is utilized to its maximum when thermocouples are made of material of different conductivity (Kryotherm 2003). Materials used today are different kinds of semiconductors (Ferrotec America 2003a).

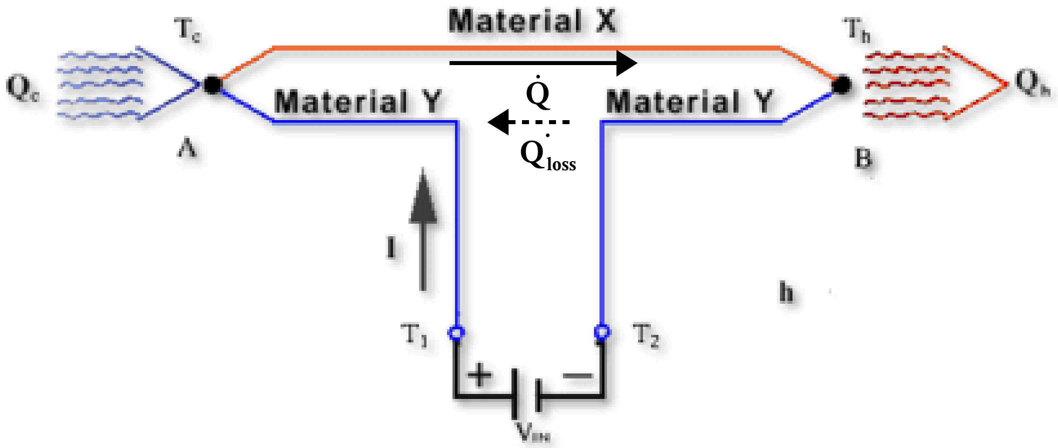
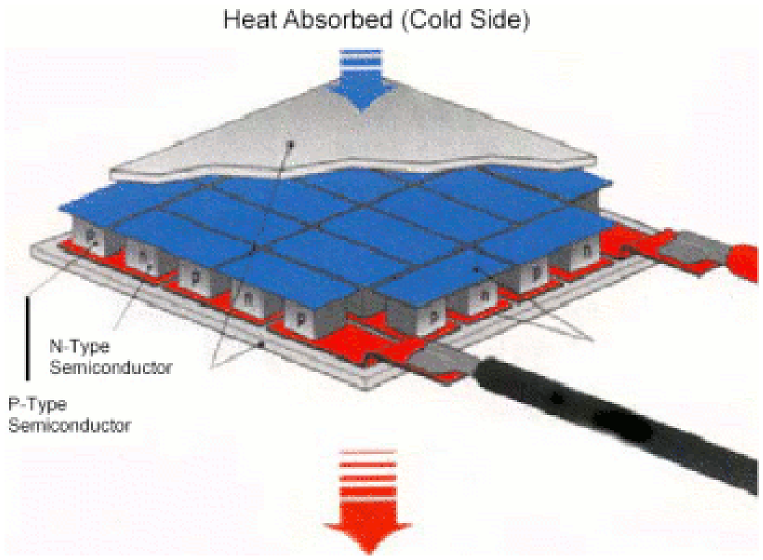


Fig. 2.10 Schematic TE module (Ferrotec America 2003a$^{\copyright 3}$)



Heat rejected(Hot side)

Fig. 2.11 TE module (Solid State Cooling systems 2003$^{\copyright 4}$)

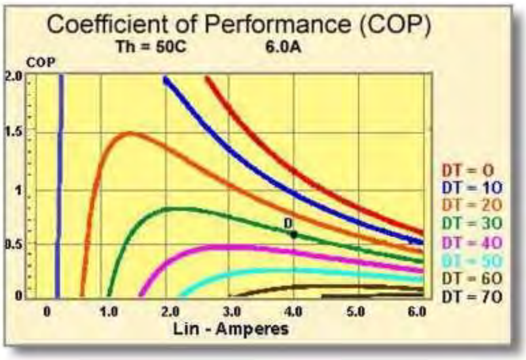


Fig. 2.12 COP TE module (Ferrotec America 2003b©3)

A disadvantage with TE coolers are their poor COP. This can be seen in figure 2.12, which illustrates the COP of a 6-ampere (9.8 volt⇒ $\dot{W}_{tmax}=50\ W$) TE module as a function of input current (I) at some different temperature differentials (DT) . $\mathbf{T_1}$ is set to 50 °C. At point **D** with an input of 6.7 volt and 4 ampere (26.8 W, 46 % of maximum input) and a heat lift of 30 °C (T_c=10 °C) a COP of only 0.58 is achieved.

This is because, in addition to conducting electricity (and thereby the high energy elections), the semiconductor material also conducts heat (Cool chips 2003). While the high-energy electrons are controlled by the voltage source to flow from the cold to the hot side, heat flow in accordance with the laws of thermodynamics, from the hot to the cold side. This heat flow, $\dot{\mathbf{Q}}_{\mathbf{loss}}$ in fig. 2.10, then causes a thermal short-circuit which results in that a lot of the heat pumped from the cold to hot side returns to the cold site instead of going to the intended heat sink.

A company called Borealis Exploration Limited claims to have solved this problem by being able to manufacture chips with a small (vacuum) gap in-between the hot and the cold side. Heat cannot pass this gap, electrons however, through an effect called tunneling can, if the gap is small enough (Cool Chips 2003). Projected Carnot efficiency for their chip is said to be η_c=0.55, which is the same or slightly better than that of a vapor compression system. With this efficiency and conditions as in the example above (T_0=10°C, T_1=50°C) the resulting COP_{tot} is 3.9. This technique is currently in a prototype stage (Cool Chips 2003).

Advantages of TE coolers are (primary) no moving part (resulting in silent operation, low maintenance and long life), small size and weight and no (potentially harmful) refrigerants.

Another aspect is that TE modules are DC-driven. This can be a disadvantage as well as an advantage. For grid-powered applications AC-current first has to be converted to DC-current, and this results in less overall efficiency. On the other hand together with a DC power source, for example PV-cells this is an advantage as no DC-AC modulation is needed.

TE coolers are generally available with cooling capacities of less than 1 W up to around 4 kW.

If the problem with low efficiency can be solved TE coolers would indeed be an interesting alternative to vapor compression systems in a number of applications, air-conditioning among them.

2.2.2 Heat engines

A heat engine, working between two temperatures (fig 2.13), is the opposite of a two-temperature heat pump. Heat $\dot{\mathbf{Q}}_2$ flows from a higher temperature $\mathbf{T_2}$ towards a lower temperature $\mathbf{T_1}$. Some of the heat is transformed into mechanical work (or electricity) $\dot{\mathbf{W}}_t$ and the rest $\dot{\mathbf{Q}}_1$ is ejected at the lower temperature $\mathbf{T_1}$.

Just as a heat engine in theory is the inverse of a heat pumps, so too the methods of constructing heat engine cycles is the inverse of the methods above used for heat pump cycles.

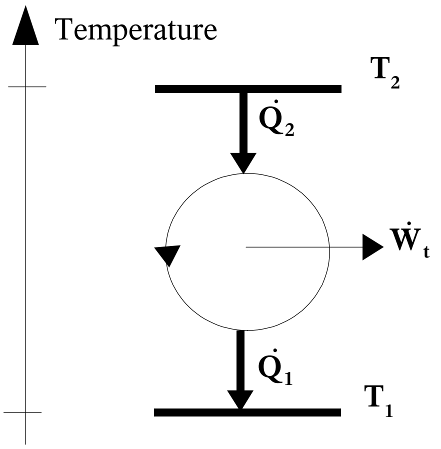


Fig. 2.13 Schematic heat engine working between two temperature levels

The efficiency of a heat engine is given by:

$$\eta = \dot{\mathbf{W}}_t / \dot{\mathbf{Q}}_2 \quad \text{(e4)}$$

The ideal heat engine is given by the Carnot heat engine whose efficiency is:

$$\eta_{\mathbf{carnot}} = \mathbf{1} - \mathbf{T}_1 / \mathbf{T}_2 \quad \text{(e5)}$$

(In addition to the information below, information about heat engines can be obtained from the reed rock heat engine projects homepage by Johnson (2003).)

2.2.2.1 Rankine power cycle

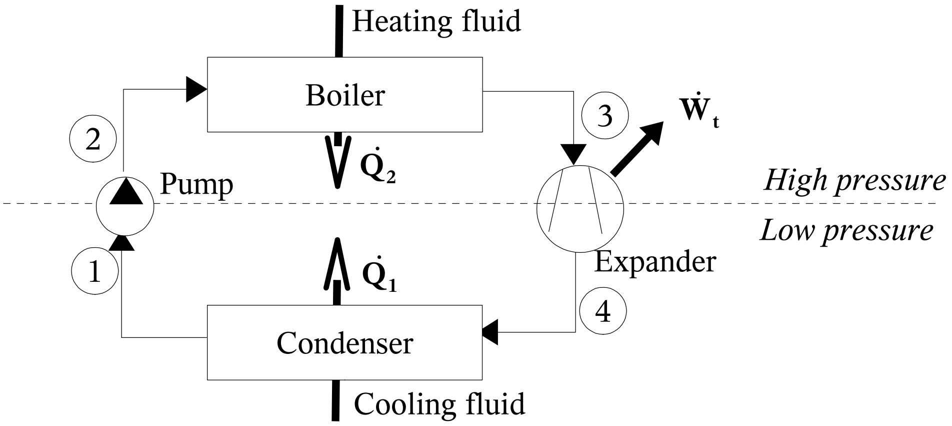


Fig. 2.14 Schematic Rankine power cycle

The Rankine power cycle is almost the inverse of the vapor compression cycle. The only difference is the need for a pump to transport the working fluid from the condenser the evaporator, which in a Rankine power cycle is called boiler. The basic Rankine cycle is shown if figure 2.14. The cycle consists of (Goswami 1999):

Process 1-2 Saturated liquid from condenser is pumped to the boiler (isentropically).

Process 2-3 The liquid is heated in the boiler. The temperature and vapor pressure of the liquid rises until it becomes saturated liquid. Further additional heat vaporizes the liquid at constant temperature until all of the liquid turns into saturated vapor. Any additional heat superheats the working fluid (vapor).

Process 3-4 The vapor expands (isentropically) through an expander (see 2.2.1.4) and work is taken out.

Process 4-1 The vapor exiting the turbine is condensed at a constant pressure until it returns to state (1) as saturated liquid.

The Rankine cycle is used in most existing thermal power plants (fired by nuclear, fossil fuels, bio mass etc.) , and thus a larger part of the worlds electric energy is generated by Rankine cycles. In these large system water is used as working medium and turbines as expanders. For smaller systems piston steam engines can be used. Another technique is to use an inverted scroll compressor (Johnson 2003).

By far the majority of the Rankine cycles used are based on water as working fluid. A disadvantage of using water is that it needs to be superheated. Without superheating, the final moisture content after expansion would be too high, which would cause the turbine blades to erode too much (Verschoor 1995).

For high temperature Rankine cycles, water still is the preferred working fluid. For lower temperature cycles (below 400 °C (Verschoor 1995), 220-500 °C (Bronicki 1996) or even below 200 °C (Badr 1990)), organic substance might be a better choice. This is because they do not need superheating before expansion to avoid a too high moisture content. Instead it will by itself become superheated and dryer as it is expanded in the turbine (Verschoor 1995, Bronicki 1996). The (maximum) boiler temperature may therefore be equal to the evaporation temperature.

Since the temperature after dry expansion is rather high, internal heat recovery between the fluid entering and exiting the condenser is needed to utilize this heat in a useful way within the cycle (Verschoor 1995). The heat exchanger used for this is (sometimes) called regenerator.

For an organic Rankine cycle the turbine and pipe sizes are smaller and thus less costly due to the organic fluids higher density. The condensing pressure in an organic cycle is generally above atmospheric pressure, thus eliminating the need for vacuum and gas purging equipment that is utilized in steam condensing cycles (Bronicki 1996). On the other hand the heat exchanging surfaces in the organic Rankine cycle need to be considerably larger than in a steam Rankine cycle (Bronicki 1996).

One should be cautiously aware that not all organic substances are environmentally friendly. Many of the organic substances used, are the same as used as refrigerants, including CFCs, HCFCs and HFCs.

The efficiency (η) for a steam Rankine cycle with a heat source temperature of 510 °C seems to be around 38-40 %, and with heat source temperatures of 330-400 °C around 30-38 % (Goswami 1999). A Rankine steam cycle with a heat source temperature of 166 °C would have an efficiency of 14.5 % (Bronicki 1996). This seems to correspond to a Carnot efficiency (η_{Carnot}) of about 0.6 for high temperatures down to about 0.4 for low temperatures.

The efficiency for some organic Rankine cycles can be read from figures 2.15 and 2.16.

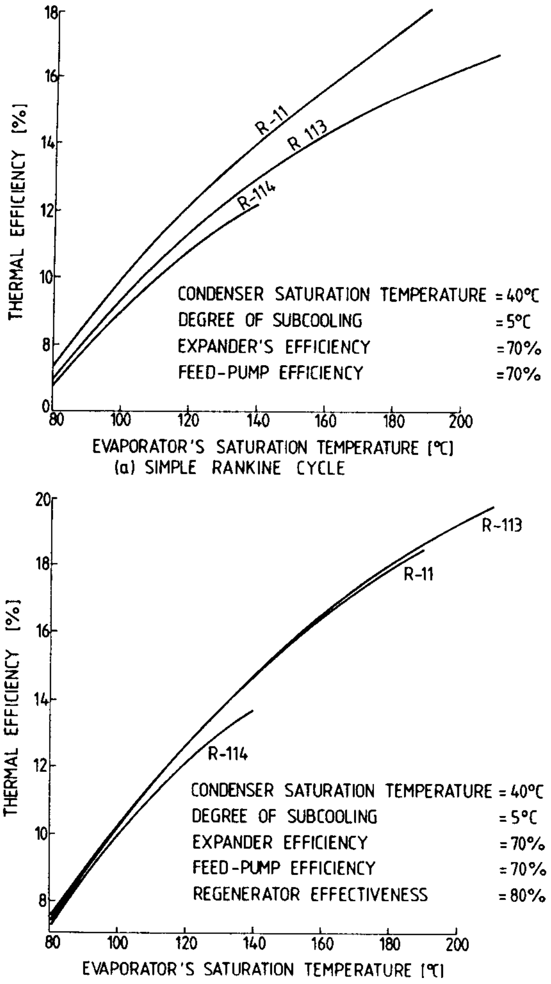


Fig. 2.15 Efficiency of organic Rankine cycles with three different working fluids, with (below) and without (above) a regenerator. (Badr 1990©5)

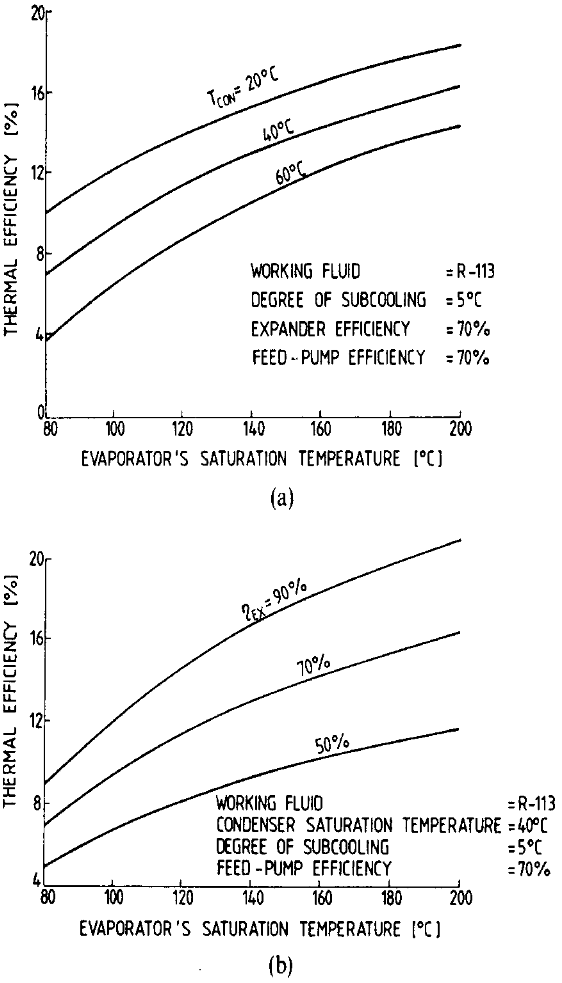


Fig. 2.16 Influence of condenser temperature (a) and expander efficiency (b) on the efficiency of a ORC with R113 as working fluid (Badr 1990©5)

An organic Rankine cycle has been patented in which the pump is eliminated. Instead a sealed chamber with an entry and a exit valve is used. This chamber seems to work as a pressure lock (rather than a pump), allowing liquid working fluid to pass from the low pressure to the high-pressure side (Sterling 1997). A 1-hp (746 W) prototype of this cycle has been built. The cycle seems to be primary intended for low temperature (down to 65 °C) , 1-4 kW systems (Sterlingsolar 2003). Commercial release is said to be planned in 2004 .

2.2.2.2 Stirling engine

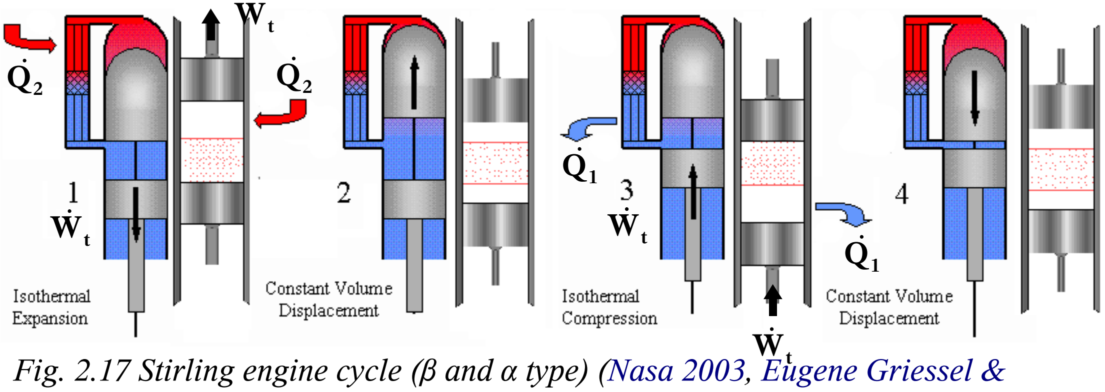


Fig. 2.17 Stirling engine cycle (β and α type) (Nasa 2003, Eugene Griessel & Dynagen Systems 1997$^{©6}$)

The Stirling engine is identical in construction with the Stirling heat pump. The heat flows are the same with heat flowing into the cylinder during expansion and out of the cylinder during compression. The difference lay in that in heat pump mode expansion and compression was achieved by mechanical force on a piston while in engine mode expansion and compression is achieved by heating and cooling the cylinder. The cold side in the heat pump has thus becomes the hot side, and the hot side becomes the cool side. During operation the hot side is constantly heated and the cool side constantly cooled. The cycle works as follows: (illustrated in figure 2.17):

Process 1 The working gas is at the hot side. As the gas is heated its pressure increases and it strive to expand. This will forces the piston to move outwards from the cylinder. Work is taken out.

Process 2 The gas is forced to the cold side through the regenerator (see section 2.2.1.2)

Process 3 As the gas is cooled down its pressure will decrease and it will strive to compress. The piston is thus drawn in to the cylinder. Work is taken out.

Process 4 The working gas is forced to the hot side and the cycle is closed.

Helium or hydrogen is preferable to air as the working gas due to a higher heat transfer rate (Goswami 1999, Walker 1994). At lower temperatures however, air or nitrogen, which are cheaper and easier to contain, can be used (BSR 2002).

The Strirling engine can work on as low temperature differences as the difference between the palm of a hand and ambient air. However low temperature engines would be very large and heavy. A general guideline is that if the hot side of the engine is less than 260 °C the engine will be too bulky (American Stirling Company 2002). Stirling engines can reach a Carnot efficiency of 50%. Most Stirling engines in use seem to be designed for heating temperatures in the range of 700-1000 °C with an efficiency (η) of around 20-40%.

BSR Solar claims that their free piston Stirling engine should be able achieve a Carnot efficiency of 75 %. Accordingly they claim an efficiency of 36 % at a heating source temperature of 300 °C (cool side temperature 30 °C) (BSR 2002). Figure 2.18 shows the (by BSR solar specified) efficiency of a typical Stirling engine, of the BSR solar's Stirling engine SUNPULSE and of the ideal Carnot cycle.

Stirling engines are used to power submarines as it can be fired directly with fuel plus oxygen and do not need air as internal combustion engines. It is also quiet which is essential for this application.

There are some Stirling engines commercially available for those who have a military or hefty commercial budget. There is also one 750 W diesel fired engine available which is designed to heat and power yachts and remote residences (American Stirling Company 2002).

A lot of small model, demonstration and laboratory Stirling engines are commercially available, and there seem to be a few manufactures that plan to start manufacturing Stirling engines for a broader market.

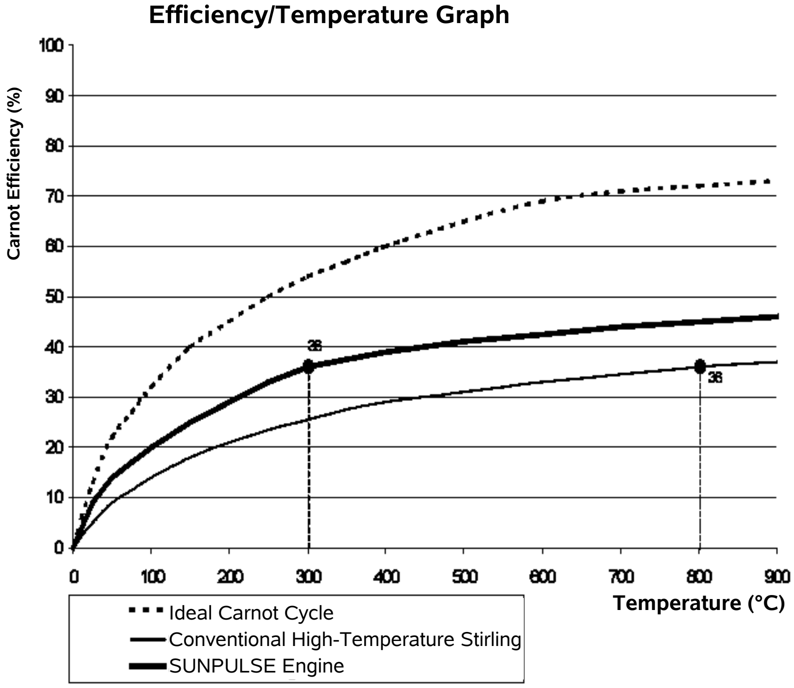


Fig. 2.18 Efficiency of a typical Stirling engine, SOLARPULSE Stirling engine and Carnot cycle. (BSR 2002©[7])

2.2.2.3 Brayton power cycle

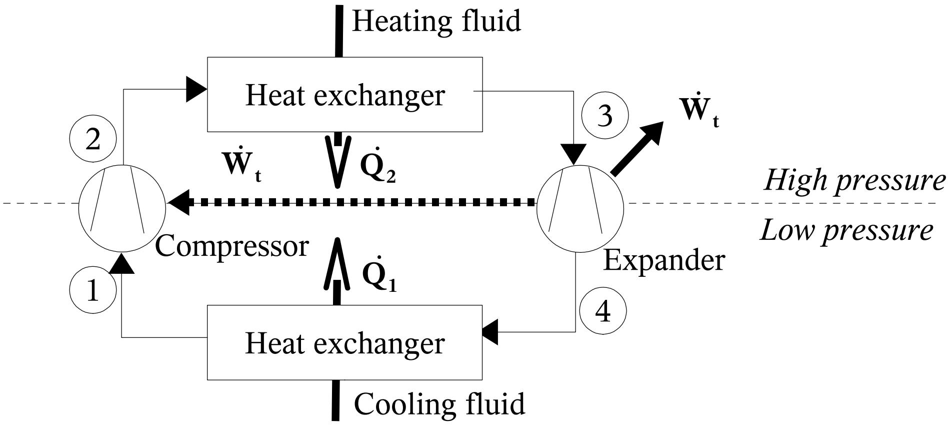


Fig. 2.19 Schematic Brayton power cycle

The Brayton power cycle is the inverse of a the Brayton refrigeration cycle. Heat is added at the high-pressure side and removed on the low-pressure side. The closed Brayton power cycle is shown in figure 2.19. It consists of:

Process 1-2 Gas is compressed to high pressure. Mechanical work is used.

Process 2-3 The compressed gas is heated.

Process 3-4 The hot compressed gas passes through an expander in which some of the heat added to the gas is converted into mechanical work. Some of the mechanical work is used to power the compressor, the rest is taken out.

Process 4-1 The gas is cooled further in a heat exchanger and the cycle is closed.

As with the Brayton refrigeration cycle there is also an open variant where the low-pressure heat exchanger is removed and ambient air is used as working fluid. The hot air that leaves the expander is thus rejected to the atmosphere.

In a gas turbine, which is a variant of the open Brayton power cycle, the heat is added through internal combustion instead of through a heat exchanger. Gas is compressed into a burning chamber between the compressor and the expander. In the burning chamber the gas is combusted with oxygen from the compressed air. The combustion generates hot exhaust gasses. The hot exhaust gases leaves the burning chamber through the expander and is ejected to the atmosphere.

The Brayton power cycle seems to be suitable for higher heat source temperatures than the Rankine power cycle. It also seems like Brayton power cycle is less efficient than the Stirling heat engine. Since the heat source temperature is high and the efficiency is rather low, the gas that exits the expander has a high temperature and contains a lot of heat.

The Brayton power cycle can be used (as a top cycle) together with a Rankine power cycle. Heat rejected from the Brayton cycle is in this case used to fire the Rankine cycle. Together they will have a rather good efficiency. This combination is often refereed to as "combined cycle" and is used in many modern power plants.

2.2.2.4 Thermoelectric generator

The thermoelectric generator is not really a heat engine in as much as no mechanical work is produced. In a thermoelectric generator heat is transformed directly into electric energy. It works as an inverse of the thermoelectric cooler described in section 2.2.1.4. While the thermoelectric cooler uses the Peltier effect, the thermoelectric generator uses the Seebeck effect (which was discovered first and then led to the discovery of the Peltier effect). The Seebeck effect is: If a circuit is made of two dissimilar conductors and the junctions are held at different temperatures a current will flow through the circuit (Ferrotec America 2003a). Thus if the same kind of chip that is used for thermoelectric coolers are heated on one side and cooled on the other, a low voltage DC source is obtained.

As with the thermoelectric coolers thermoelectric generators are currently very ineffective due to large leakage of heat through conduction from the hot to the cold side. This could as in the case with the thermoelectric coolers be prevented by a small vacuum gap between hot and cool side. Projected Carnot efficiency for a thermoelectric generator with a vacuum gap is according to Borealis exploration limited 70-80%, which is a very high efficiency (Power chips 2003).

2.2.3 Thermal Heat Pumps

The basic principle for a thermal heat pump working between three temperature levels is shown in figure 2.20.

Heat $\dot{Q}_2$ from a high temperature level T_2 is used to pump heat $\dot{Q}_0$ from a low temperature level T_0. $\dot{Q}_2$ and $\dot{Q}_0$ are ejected at a temperature level T_1 in between T_2 and T_0 as $\dot{Q}_1$. $(\dot{Q}_1=\dot{Q}_2+\dot{Q}_0)$

The ideal refrigeration machine working between three temperatures is a Carnot heat engine cycle driving a Carnot refrigeration cycle. The COP of such a combination is:

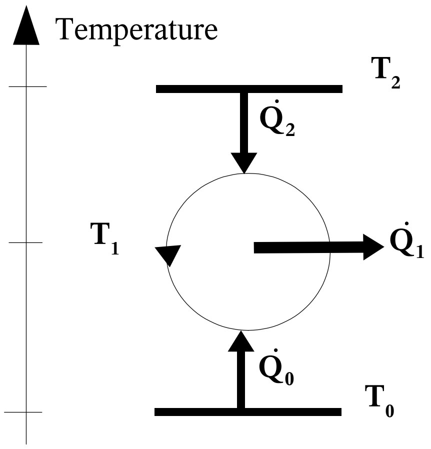
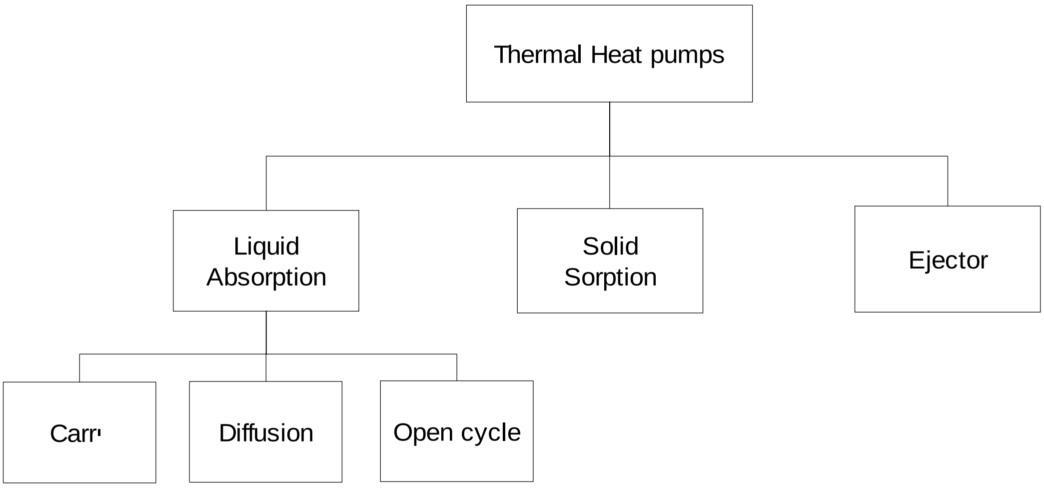


Fig. 2.20 Schematic heat pump working between three temperature levels

$$\mathbf{COP}_{\mathbf{carnot,3T}}=\eta_{\mathbf{carnot}}*\mathbf{COP}_{\mathbf{carnot,2T}}=\left(1-\frac{T_1}{T2}\right)*\left(\frac{T_0}{(T_1-T_0)}\right) \quad (e6)$$

It could be argued that this combination is in fact not working between three temperature levels but four since the heat sink temperature T_1 for the Carnot heat engine and T_1 for the Carnot refrigerator does not need to be the same. This is true as well for many of the practically used three temperature refrigeration machines. Hellman (2002) has handled this issue in an article called “Carnot-COP of a heat pump working between four temperatures”. Most practical realizations of thermal heat pumps however use a single heat sink (for cooling both the “engine” and “heat pump”) giving a three-temperature situation (at least as seen from an external perspective (from outside the machine)).

The chart below shows some techniques used to achieve the thermally driven heat pumps.



2.2.3.1 *Carré cycle absorption heat pump*

Working principles.

Absorption refrigeration machines based on the Carré vapor absorption cycle, seems to have as dominant position among thermal refrigeration machines as the vapor compression system has among mechanic/electric refrigeration machines, and most solar cooling systems in use includes an absorption chiller (IEA 2003).

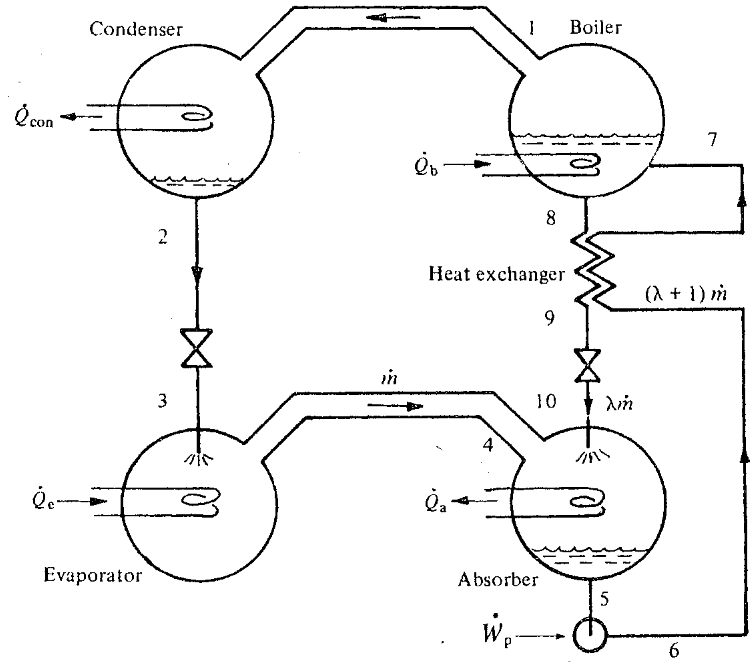


Fig. 2.21 Simple lithium-bromide/water absorption system (Gosney 1982©8)

Figure 2.21 shows the principles of a simple H_2O /LiBr (water/lithium-bromide) single effect vapor absorption cycle. It can be seen (in one way) as a vapor compression system where the compressor has been replaced by the *absorber/generator* couple. The generator can also be called boiler as in figure 2.21 or desorber.

The condenser and evaporator works basically in the same way as described for the vapor compression system described in section 2.2.1.1, with water used as refrigerant. (However, the evaporator is often a flooded evaporator where a pool of liquid refrigerant is maintained (Pita 1998) in equilibrium with refrigerant vapor, unlike the DX (Direct Expansion) evaporator explained in section 2.2.1.1 where in the end of the evaporator only refrigerant vapor exist)

Instead of being compressed and lifted by a compressor to the high-pressure side the refrigerant vapor in an absorption-cycle is absorbed by a absorbent-refrigerant solution. In the case of a water/lithium-bromide system lithium-bromide is the absorbent. Many other combinations can be used. Another common combination is NH_3/ H_2O (ammonia/water) where ammonia works as refrigerant and water is the absorber. Others are H_2O/LiCl, NH_3/ $SrCl_2$ etc.

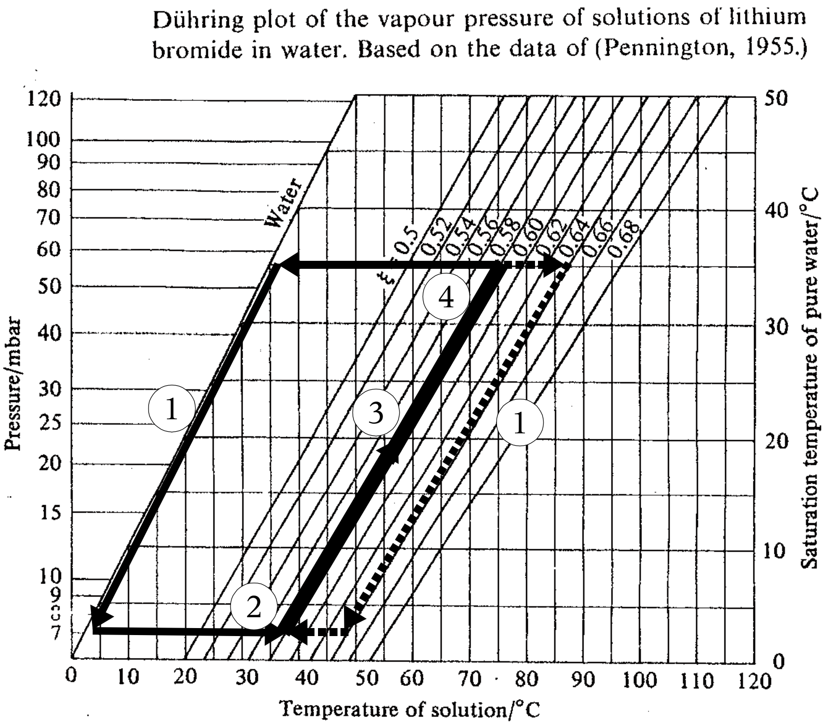


Fig. 2.22 The Carré cycle in a Dühring plot(Dühring plot: Gosney 1982[©8]*)*

(1) The refrigerant (left, solid line) and the strong solution (right, dotted line) pass through restrictors from the higher pressure level to the lower. The refrigerant is evaporated at 7°C. The strong solution is let into the absorber usually at the top, and trickles down over the tubes with the coolant, while being exposed to refrigerant vapor in order to absorb it. It ends up as weak solution in the bottom of the absorber (Grossman 2003). (For improved efficiency the strong solution normally passes a heat exchanger where it exchanges heat with the cooler weak solution from the absorber, see fig 2.21 and below.)

(2) The refrigerant vapor is absorbed in the weak solution in the absorber and heat is rejected at 35°C. The mass fraction LiBr in the absorber is 0.58.

(3) The weak solution is pumped from the lower pressure level to the higher (and through the heat exchanger).

(4) The refrigerant is distilled from the solution in the generator at 85°C. The mass fraction LiBr in generator is 0.63. The refrigerant vapor is cooled and condensed at 35°C.

The absorbent has to have a (substantially) lower vapor pressure than the refrigerant. The solution of the absorbent mixed with refrigerant will thereby have a lower vapor pressure than the refrigerant in its pure state (Gosney 1982). This is seen in figure 2.22 showing a Dühring plot for H_2O/LiBr. In this Dühring plot the saturation temperature for the pure refrigerant is plotted against the saturation temperature for the solution at different solution concentrations (ξ denotes the mass fraction of absorbent (here LiBr) in the solution). On the left axis the vapor pressure of the solution as well as pure refrigerant (water) can be read. (The vapor pressure of pure LiBr is almost zero and need not to be taken into consideration.) A cycle using a heat-sink with a temperature of 30°C, a heat-source with a temperature of 90 °C (allowing 5°C difference for heat transfer through absorber and generator heat exchangers) and an evaporation temperature of 7 °C is inserted into the plot.

Strong solution (containing a low concentration of absorbent) from the generator enters and tickles down over tubes inside the absorber.

The vapor pressure of the solution trickling down the absorber needs to be kept slightly below the vapor pressure of the pure refrigerant in the evaporator. This vapor pressure difference will cause the refrigerant to evaporate in the evaporator and be absorbed by the solution.

Mixing an absorbent with a refrigerant results in heat, called heat of solution, being released within the solution. When the refrigerant vapor is absorbed by the solution this heat of solution as well as the latent heat of the refrigerant (which has been taken up by the refrigerant in the evaporator when it was evaporated,) is released. This heat must be removed to prevent the vapor pressure of the solution from rising (because of increased temperature). The heat removed from the absorber is in figure 2.21 denoted as $\dot{\mathbf{Q}}_{\mathbf{a}}$.

As the solution trickles down the absorber and absorbs refrigerant it gets weaker. (Eventually, if given enough time, it becomes to weak and its vapor pressure to high to absorb any more refrigerant.) It ends up as a weak solution (containing a low concentration of absorbent) in the bottom of the absorber. From the bottom of the absorber the weak solution is pumped to the generator for regeneration.

The weak solution thus carries the absorbed refrigerant from the absorber to the generator. A low concentration of absorbent in the weak solution leaving the absorber and a high concentration of absorbent in the strong solution entering the absorber, will result in a more efficient transport of refrigerant from absorber to boiler, since less strong solution is needed per kilo refrigerant.

In the generator the refrigerant is desorbed from the solution through input of heat $\dot{\mathbf{Q}}_{\mathbf{b}}$. Similarly to the evaporation in the evaporator, the desorption is driven by a vapor pressure difference between the generator and the condenser. Thus the vapor pressure of the solution in the generator must be slightly higher than the vapor pressure of the refrigerant in the condenser. To accomplish both a high vapor pressure and a high concentration high temperature is needed (as seen in figure 2.22). When the refrigerant vapor is desorbed it becomes superheated since the vapor pressure of the desorbed pure refrigerant is much higher than the solution's (at the same temperature). The heat input $\dot{\mathbf{Q}}_{\mathbf{b}}$ equals the latent and sensible heat of the superheated refrigerant leaving the generator and the heat leaving the generator with the strong solution (due to the difference in heat of solution and sensible heat between the weak solution entering and the solution leaving generator).

The superheated vapor then flows to the condenser and thereby the cycle is closed.

In a NH_3/ H_2O system a rectifier is needed between the generator and condenser to purify the refrigerant vapor from remains of absorbent. This is because water, in opposite of salts has a significant vapor pressure and thus there will be a significant amount of water in the vapor leaving the solution. This rectifier requires additional heat.

The relation between the Carré cycle illustrate in figure 2.22 and the theoretical three temperature heat pump illustrated in figure 2.21 is:

$$\dot{\mathbf{Q}}_2=\dot{\mathbf{Q}}_b \quad \dot{\mathbf{Q}}_1=\dot{\mathbf{Q}}_{con}+\dot{\mathbf{Q}}_a \quad \dot{\mathbf{Q}}_0=\dot{\mathbf{Q}}_E$$

To reduce the heat transfer from the generator to the relatively cooler absorber a heat exchanger is inserted between the flow entering and the flow leaving the generator. In this heat exchanger the hot strong solution leaving the generator heats up the relatively cold weak solution from the absorber before entering the generator. A heat exchanger, called *subcooler,* can also be place between cold refrigerant vapor flow leaving the evaporator and refrigerant flow entering the evaporator, in order to prevent heat transfer from the condenser to the evaporator. This can be seen in schematics of some NH_3/H_2O systems.

Just as in the vapor compression system there are two pressure levels to enable evaporation at a low temperate and condensation at a higher temperature. The condenser and generator are at the high-pressure side. The evaporator and absorber are at the low-pressure side. The pressure difference is usually smaller than in a vapor compression system. When water is used as refrigerant both pressure levels are sub-atmospheric.

The pump in between the absorber and the generator and a restrictor between the condenser and the evaporator maintains this difference. The pump demands an input of mechanical work $\dot{\mathbf{W}}_p$, but this work is only a small fraction of what is needed in a compressor system (Gosney 1982), therefore this work is often disregarded when COP values is presented. (It would also be problematic to include it as mechanical work is a higher form of energy (see appendix A) therefore it would have to be weighted somehow).

There can also be a small pressure differences between the evaporator and the absorber, and between the generator and the condenser because of pressure losses due to vapor flow. For an H_2O/LiBr system this is below 1 mbar (Gosney 1982).

To reduce pressure losses between the evaporator and the absorber, and between the generator and the condenser those parts are often built together into two shells. One shell contains the evaporator and the absorber and the other contains the generator and the condenser as shown in figure 2.23. One manufacture has taken this a step further and included all four components in to one single shell divided into a higher and lower pressure side with a diaphragm (Gosney 1982).

The pressure in all parts is the same as the vapor pressure of the liquid in that part. The vapor pressure in each liquid is as explained before (above and in appendix A) and as seen from figure 2.22, a function of temperature and concentrations. In an H_2O/LiBr system the absorbent has a very low vapor pressure, and therefore there is only refrigerant in the condenser and the absorber. The vapor pressure in the condenser and the evaporator is thus a function of the temperature (only). In an NH_3/H_2O system the amount of absorbent in the condenser and the evaporator depends on the efficiency of the rectifier. Remains of absorbent in the evaporator will result in a higher evaporator temperature at a given evaporator pressure.

Note should be taken that the Carré absorption cycle is a steady state system in opposite to a cyclic system. Concentrations in all parts are held just above or below the saturation point.

From the Dühring plot in figure 2.22 it can be seen that a higher condensation temperature requires a higher generator temperature, and that a higher absorber temperature requires a higher evaporation temperature, in order to maintain the same mass fraction lithium-bromide. Else the mass fraction of LiBr in the generator must decrease and the mass fraction in the absorber increase. Otherwise the temperature in the generator would become lower than the saturation temperature of the strong solution and if this happens no refrigerant can be distilled.

The mass fraction in the generator and in the absorber can be decreased and increased only until they are equal (at which state the system stops working). This means that for every combination of condenser/absorber-evaporator temperatures there is a minimum value for the generator temperature (and for every combination of generator-evaporator temperatures there is a maximum condenser/absorber temperature). Figure 2.24 illustrates this for an H_2O/LiBr system at various evaporation temperatures. Due to crystallization of the strong solution at the outlet of the heat exchanger there is also a maximum generator temperature. This depends on the efficiency of the heat exchanger. The maximum allowed temperature for two different values of heat exchanger efficiency (1, 0.75) is also plotted in figure 2.24.

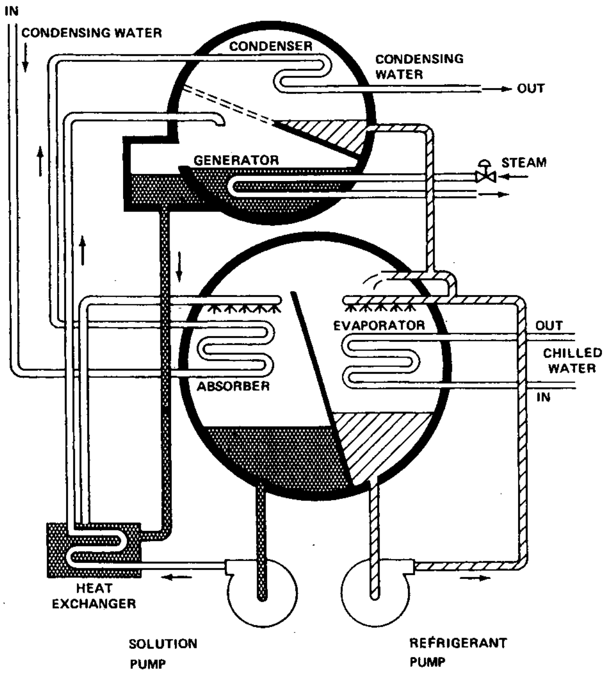
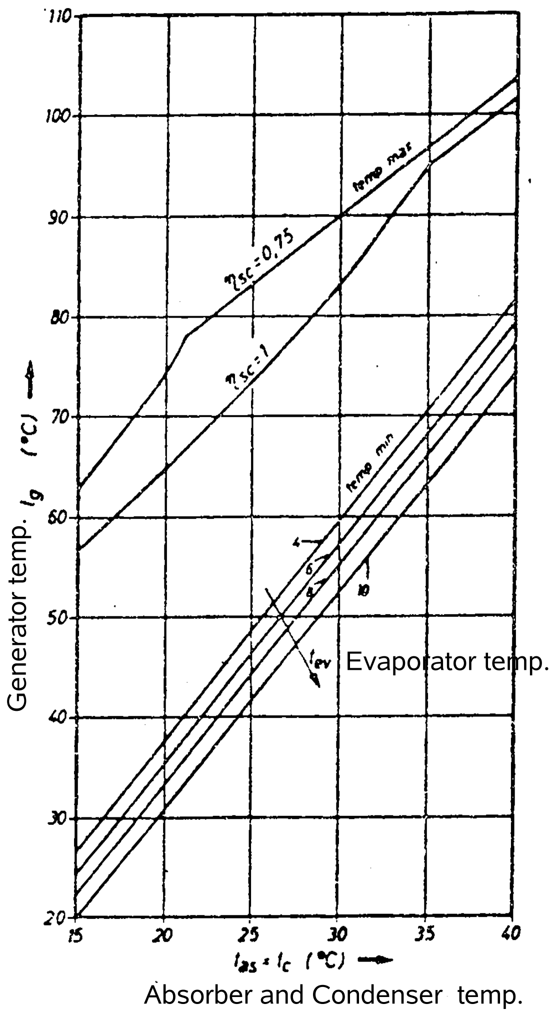


Fig. 2.23 Two shell absorption system (ASHRAE Refrigeration 2002$^{©9}$*)*



Fig

Fig. 2.24 Maximum and minimum generator temperature (Lazzarin1979).

Efficiency (COP) and capacity

Four external temperatures affect the COP and capacity of an absorption chiller; these are generator heating-, condenser cooling-, absorber cooling- and chilled fluid temperature. If the same cooling fluid is used to cool both the absorber and the condenser, either in series (as common in water cooled chillers) or in parallel, this is reduced to three different temperatures; heating-, cooling- and chilled fluid temperature.

In the most common plot of COP for this kind of systems the COP value is plotted against the heating fluid supply temperature while cooling fluid supply temp for the absorber and the condenser, as well as the chilled fluid temperature are held fixed. (Note that the generator and evaporator temperatures are always lower than heating and chilled fluid temperatures and that condenser and absorber temperatures are always higher than cooling fluid temperature, this is because of the necessary temperature difference for heat exchange.) To also illustrate effects of different cooling and chilled water temperatures several configurations can be drawn in a single plot. Figure 2.25 shows the COP (solid lines) and normalized capacity $\dot{\mathbf{Q}}_E$ (capacity/design point capacity) of a single effect H_2O/LiBr absorption chiller at different cooling water temperatures $\mathbf{T}_C$. Design point conditions are indicated in the diagram with a black dot.

It is evident from figure 2.25 that the COP increases with the heating fluid temperature and decreases with the cooling water temperature, as may be expected from the increase in the Carnot COP (e1) under the same temperature variation. However the absorption system does not exactly follow the Carnot trend. As the heating fluid temperature $\mathbf{T}_h$ increases, the absorption COP tends to level, rather than continue to increase, and with a further increase in $\mathbf{T}_h$, even drops somewhat (Gommed 1990). Also the minimum value previously described is clearly visible.

Gommed and Grossman have explained the reason for this in an article 1990.

The two major kind of energy losses that exist in an absorption chiller and result in this characteristic behavior are: *Circulation losses* and *mixing losses*.

Circulation losses are associated with the imperfect heat transfer in the heat exchanger between the absorber and the generator. This heat exchanger, even if ideal, can only prevent transfer of sensible heat from the generator to the absorber. The heat of solution, caused by the difference in concentration between strong and weak solution, goes straight through. Circulation losses increase with the solutions circulation factor λ.

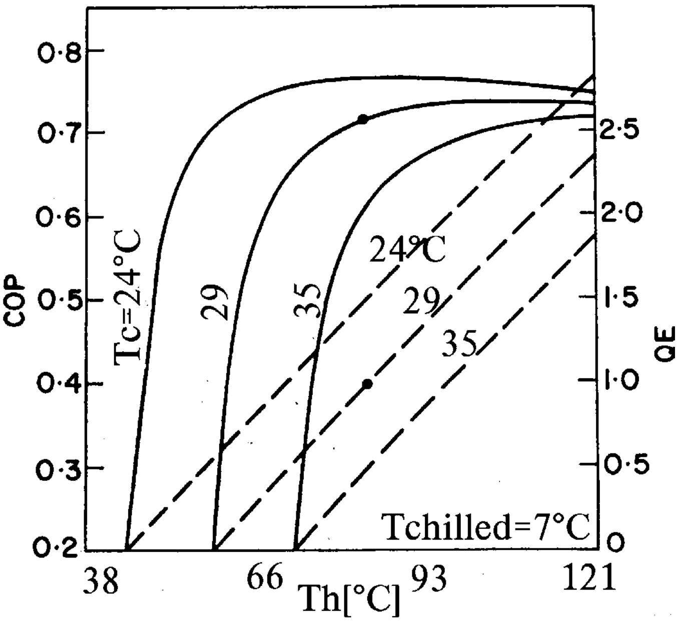


Fig. 2.25 COP and capacity of a single effect H_2O/LiBr absorption chiller. (based on Gommed 1990©10)

$$\lambda = \frac{\text{mass flow strong solution}}{\text{mass flow pure refrigerant}} = \frac{\xi_{\text{weak}}}{\xi_{\text{strong}} - \xi_{\text{weak}}} \qquad (e7)$$

Mixing losses are associated with the desorption of the refrigerant in the generator. The refrigerant vapor leaves the generator superheated, since the temperature required for desorption from the solution is higher than the evaporation temperature of the pure refrigerant under the same pressure. This superheat energy is rejected in the condenser and constitutes a thermodynamic loss. Mixing losses increases with increased concentration of absorbents in the strong solution (ξ_{strong}).

When the system operates with low heating fluid temperatures, circulation losses dominate. As stated above a low generator temperature in results in a small differences between absorbent concentration in the weak solution leaving the absorber ξ_{weak} and the strong solution leaving the generator ξ_{strong}. This results in a high circulation factor λ. As the heating fluid temperature increases, so does the difference in concentration and thereby the circulation factor. As the circulation factor decreases and the relative importance of the circulation losses to the COP value diminishes.

When operating at full design capacity most of the heat supplied to the generator is used to vaporize the refrigerant. Since the amount of heat needed to distill the refrigerant from the solution is fairly insensitive to the generator temperature and is proportional to the amount of refrigerant distilled, the ratio of evaporator to generator heat tends to be constant. This explains the "leveling off" of the COP at higher temperatures.

Further increase in heating fluid temperature will lead to further superheating of the refrigerant vapor leaving the generator and thereby higher mixing losses and a slight decline in the COP.

Figure 2.26 shows predicted COP for an (rebuilt previously direct fired) absorption chiller, with the effect of both increased chilled water and decrease cooling fluid temperature are included. Decreasing chilled water and increasing cooling fluid temperature seems mostly to effect the circulation losses. This effectively lowers the minimum generator temperature and enables higher COP values at lower temperatures. (From e7 it seems like lowering the absorber temperature and thereby ξ_{weak} could have a greater effect on λ than increasing ξ_{strong}).

Points 1, 2,3 in figure 2.26 are real experimental values. They well illustrate the fact that real values often are less than the theoretically predicted.

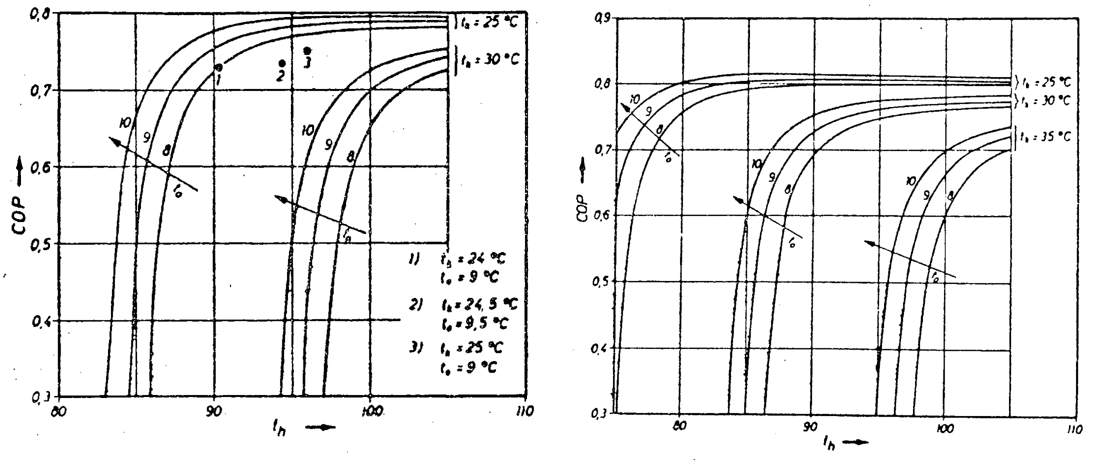


Fig. 2.26 Predicted and experimental COP for a absorption chiller. Left: Original heat exchangers. Right improved heat exchangers. (Lazzarin 1979)

In addition to the energy losses, there are also (as mentioned above) temperature losses. Temperature differences exist between the heating fluid and the generator, the cooling fluid and the absorber/condense and between and the chilled fluid and the evaporator. The temperature difference depends on efficiency of the heat exchangers in those parts. The effect off heat exchanger efficiency is seen is figure 2.26. The left half shows predicted COP for a chiller with original and right part with improved heat exchanging surfaces.

The capacity of an absorption chiller is given by:

$$\dot{m} * (h3 - h4) \approx \dot{m} * (\text{heat of evaporation})_{\text{refrigerant}} \qquad (e8)$$

$$\dot{m} = \textit{massflow refrigerant}$$

It can be seen from e8 that the capacity depends on the mass flow of refrigerant through the system. As seen in figure 2.25 the capacity does not have the same characteristic “level off” and the slight dip at higher temperatures as the COP value has. Instead it seems to be proportional to the temperature once the temperature has reached above the value where the system starts working.

Double and multiple effect absorption chillers

One way to improve COP at high temperatures is to use a double effect absorption chiller. In a double effect chiller two generators at different pressures and temperature levels are used. The high-pressure generator is heated by a high temperature heat source. The vapor that leaves the-high pressure generator is heavily superheated (because of its high temperature and pressure) and thus contains a lot of heat. In a single effect absorption chiller high temperatures, as explained before, leads to a “level off” and decreases in the COP, because a lot of heat is then rejected to the cooling fluid through the condenser In the double effect chiller this heat is instead used to heat the low-pressure generator. In this way an internal heat recovery is introduced reducing the mixing losses.

The two generators can be connected in three different ways. The most efficient way is to connect them in parallel as shown in figure 2.27. The weak solution from the absorber is divided between the high pressure and the low pressure generator (s13 fig 2.27. Strong solution from both generators are mixed (M12 2.27) and supplied to the single absorber. Alternative configurations are to connect the generators in series,

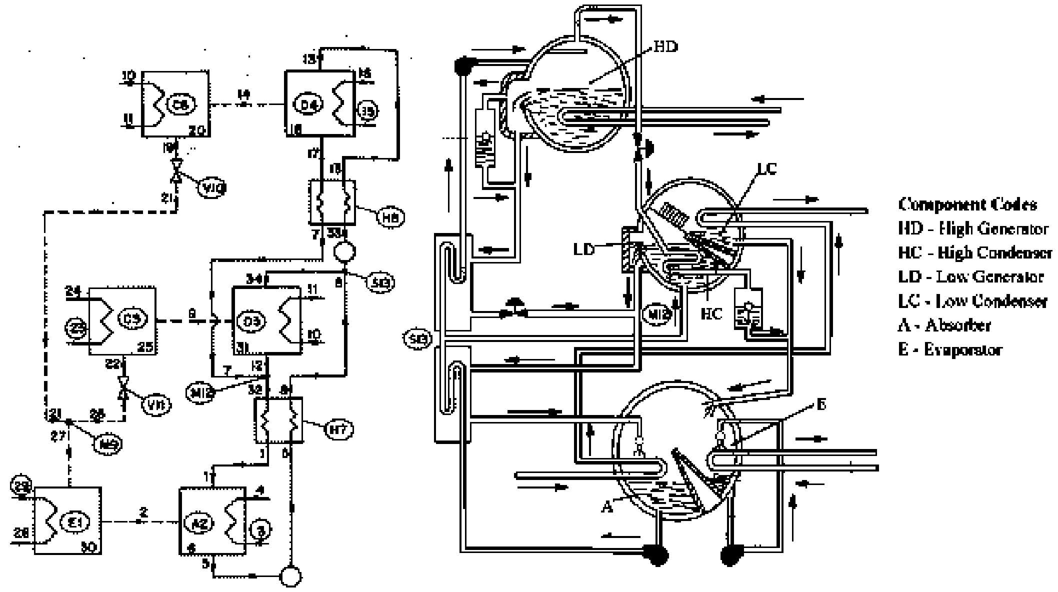


Fig 2.27 Double effect $H_2 0$/LiBr chiller (Gommed 1990[©10], Biezanowski 2003)

either by having the weak solution pass the low-pressure generator first and then the high-pressure generator (Serial Flow type I), or by having it pass the high-pressure generator first and then the low-pressure generator. The efficiency of these two configurations is pretty much the same (Gommed 1990).

An aspect that could be of importance is that in the Serial Flow type I configuration the concentration of the solution in the low-pressure generator is lower than in the high-pressure generator. Thus the temperature of the low-pressure generator in a Serial Flow type I configuration is lower than it is in a Serial Flow type II configuration. In the parallel configuration the concentration and temperature of the low-pressure generator can be controlled rather freely between a minimum and a maximum (same as for a single effect chiller, see above).

In similar way as a single effect chiller can be expanded to a double effect chiller a double effect chiller can be expander further to a triple effect chiller by adding a third generator at a higher pressure. The COP of typical single double and triple effect absorption chillers is shown in figure 2.28.

It can be seen from figure 2.28 that by multi-staging the chiller the COP curve is moved diagonally upwards compared to with a single effect chiller. As a result the minimum heating fluid temperature at which the chiller can operate is also shifted upwards. However, if the chiller is built with parallel flows to the generators, and all generators can be supplied with external heat, a double effect chiller should be downgradable to a single effect chiller, and a triple effect to a double and a single effect chiller. In this way a single chiller should be able to work efficiently over a wide range of temperatures.

A triple effect cycle can also be achieved through cascade coupling of two single effect absorption chillers. Heat from the condenser and the absorber of the chiller working at high temperature (, which due to it high temperature can not use H_2O/LiBr,) is used to provide heat to the other working at normal temperature. The evaporators of both chillers are used to remove heat from the chilled fluid (air or water). A theoretical COP of 1.8 is obtainable from such a system with high temperature generator temperature 200-203 °C, mid temperature (in between the two chillers) 93 °C and chilled water temperature 7 °C. An advantage for this system over the single unit triple effect chiller is that in a cascade coupled system all parts can be kept at sub-atmospheric pressure, while in the single unit system the highest temperature generator has to be at high pressure requiring costly pressure vessel design and a high pressure solution pump. In both systems high temperature creates problem with solution stability and material corrosion (ASHRAE Refrigeration 2002). However, in the cascade coupled system different absorbents/refrigerant-pairs can be used in the high and the low temperature part, therefore a solution optimized to the temperature of the specific part can be used.

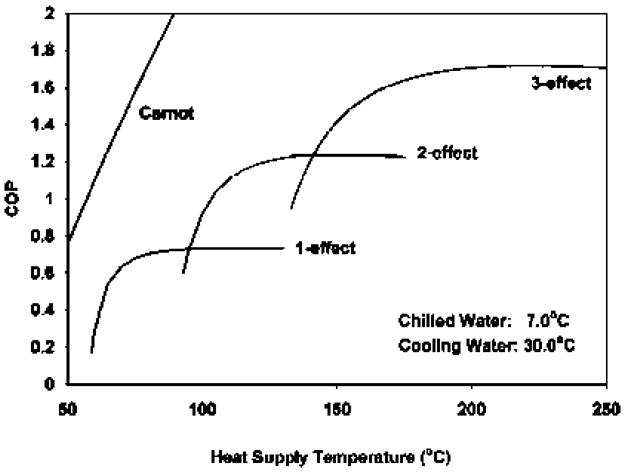


Fig. 2.28 COP as a function of heat supply temperature for single-, double- and triple-effect H_2O/LiBr absorption chillers (Grossman 2002 $^{\copyright 11}$)

Other approaches to obtain higher

COP through multiple effect cycles are the AHE and the GAX (or GAHX) cycles.

In the AHE (Absorber Heat Exchange) cycle heat recovery is achieved with the use of double generators and absorbers. The double generators seem to work similarly to a Serial type I double effect absorption chiller described above. (The weak solution first passes the low temperature generator (with a lower concentration ξ) in which part of the refrigerant is distilled. It then flows to the externally heated high temperature generator (with a higher ξ) where additional refrigerant is distilled. Superheated refrigerant vapor from the high temperature generator provides heat $\dot{\mathbf{Q}}_{\mathbf{AHEG}}$ to the low temperature generator.) However, in the AHE cycle both generators are at the same pressure level. The double absorbers work in a similar way. The strong solution flows from the generator to the higher temperature absorber with a high concentration ξ where it absorbs some refrigerant vapor. It then flows to the externally cooled low temperature absorber with lower concentration where it absorbs additional refrigerant. From the low temperature absorber it flows back to the generator. Before reaching the generator it is used to cool, and thereby recovers heat $\dot{\mathbf{Q}}_{\mathbf{AHEA}}$ from the high temperature absorber. In addition a conventional heat exchanger between strong and weak solution can be added (Engler 1997).

In the full GAX (Generator Absorber heat eXchanger) cycle an additional generator and an additional absorber is added to the AHE cycle. The additional absorber is placed before the solution and the externally cooled absorbers (in the absorbent loop). Thus this absorber has the highest solution concentration and the highest temperature. Heat $\dot{\mathbf{Q}}_{\mathbf{GAX}}$ from this absorber is used to heat a low temperature generator, which is placed before the solution and the externally heated generators. It thus has the lowest temperature and the lowest concentration (of the three generators). When GAX is used there is generally no heat to be recovered by conventional solution heat exchange and this is therefore not needed (Engler 1997). The cycles are illustrated schematically in figure 2.30.

Figure 2.29 shows a NH_3/H_2O GAX cycle. Both the triple generators and absorbers are integrated into single units. The temperature and concentration differences seem to be kept by flow direction in absorber and density differences in generator. By stopping the GAX-loop pump the cycle is downgraded to an AHE cycle.

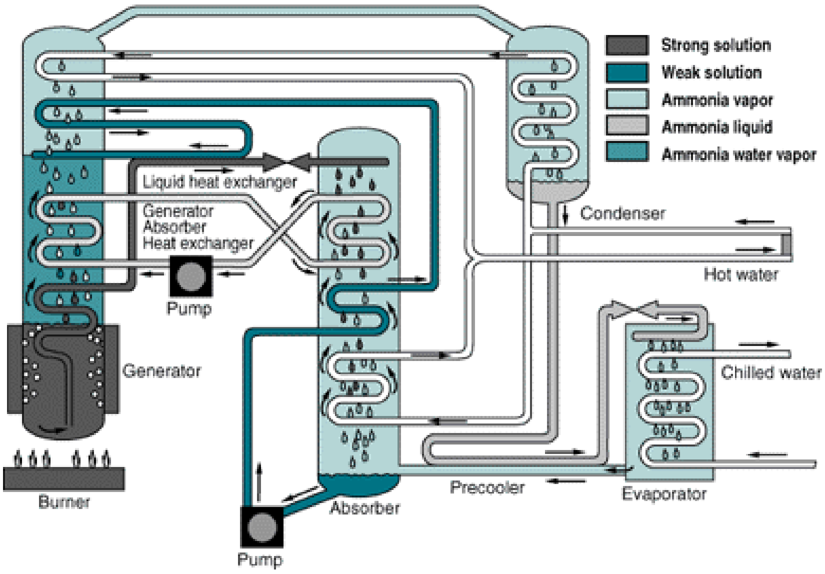


Fig. 2.29 GAX-cycle (MEDS 2004[©12])

GAX cycles often use NH_3/H_2O (ammonia/water) as refrigerant-absorber couple. H_2O/ LiBr cannot be used because the concentration of the strong solution is limited upwards by crystallization (see below). This imposes a limit on the maximum absorber temperature, which is too low to allow for the necessary temperature gap ($\mathbf{T_{diff}}$ in fig 2.30) between the absorber and the generator.

Current gas fired water/ammonia (double effect) GAX-cycle absorption heat pumps has COP of 0.7 in cooling mode. There are programs directed towards cycle COP of 0.9 (ASHRAE Refrigeration 2002).

Figure 2.31 shows simulated COP for an AHE cycle (with and without heat exchanger), a GAX cycle, a single effect absorption cycle with subcooler (SECP) and a single effect absorption cycle without subcooler (BSEC). All cycles use NH_3/H_20.

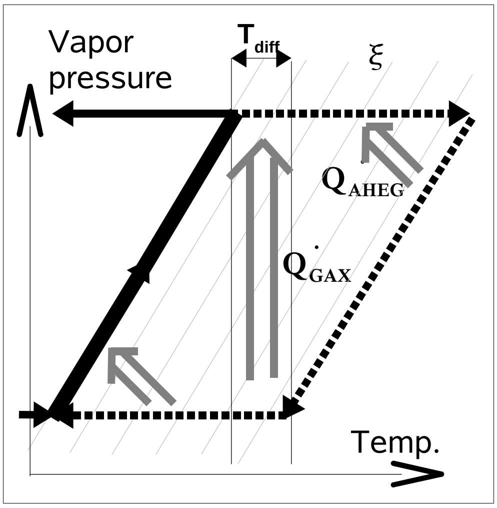


Fig. 2.30 Schematic GAX-cycle (see also fig. 2.22)

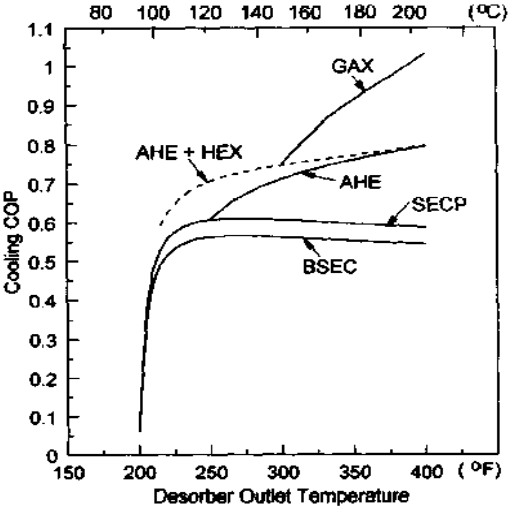


Fig. 2.31 Cooling COP as a function of the strong solution outlet temperature from the desorber, for various simulated NH_3/H_20 cycles (Engler 1997[©13a])

Multiple heat sources and heat sinks

If two heat sources are available, one with low temperature and one with high temperature, then the low temperature generator in a multi effect chiller could be heated with heat from the high temperature generator or absorber as well as heat from the low temperature heat source. To effectively heat a generator with two different heat sources they should be of similar temperatures (Grossman 2002). The temperature of the low temperature generator (and thus its supply temperature) therefore needs to be matched with the temperature of the low heat source. As seen above it is in some multiple effect chillers easier to regulate low temperature generator temperature than in others. A chiller using this principle has been developed within a European joint project. It is based on a direct gas-fired double-effect chiller, which has been modified so that lower temperature heat can be added directly to the low temperature generator through a hot water loop (Lamp 1998).

In a multiple effect system consisting of two cascade coupled chillers, additional heat to the low temperature chiller, (from a lower temperature heat source,) could be added to the heat transfer loop between the two chillers.

As already mentioned, the above methods require that temperatures are matched. This could result in that the temperature, and thereby the concentration ξ_{strong}, in the low temperature generator has to be lower than what had otherwise been optimal (if heated only by heat from the high temperature generator).

An alternative to match two heat sources temperature is to use an additional generator in series at the same pressure that will work in a similar way as the solution and the GAX heated generators in the AHE and GAX cycles (Grossman 2002). The difference is that it will be fired from an external heat source instead of an internal. In a similar way a double absorber of different temperatures could be used if for example heat rejection to a low temperature heat sink is limited. This could be the case if a wet cooling tower is needed (as often is the case) and the water supply is limited, or if a heat storage is used as a low temperature heat sink. Since both generators/absorbers are at the same pressure it should be possible to put them in a single shell in the same way as the generator/condense and evaporator/absorber couples are put in a singe shells (se fig. 2.23). A thermally insulated partition might be needed to keep them apart if this cannot be done by flow directions and density differences as in the GAX-cycle (see fig. 2.29).

This double generator system can be used in both single effect and in all stages of a multiple effect chiller.

Note that when more than one heat source is used it is no longer a heat pump working between three temperatures, but between four (high temperature heat source, low temperature heat source, cooling water and chilled water temperature). Therefore $\mathbf{COP}_{\mathbf{Carnot,\,3T}}$ is not applicable.

Refrigerant/Absorbent couples

NH_3/H_2O (Ammonia/Water)

Comparing figure 2.28 and 2.31 highlights the main problem with this pair, that at the same operating temperature it has a COP consistently lower than H_2O/LiBr. This is in part due to rectification losses (Gommed 1990).

An other disadvantage of ammonia is that it is harmful to the human body, and it is flammable when mixed with air (in a certain mixture even explosive). However, it has a strong smell that can be easily detected when released to the environment. It is not a greenhouse gas and leakage does not harm the ozone layer. Further more, even if leaking out it will not be long lived in the environment since it is absorbed by water and thereby diluted to harmless concentrations.

Advantage for the ammonia/water couple are: High-pressure refrigerant with a low specific volume (compared to water) makes the unit more compact.

No problem with crystallization since both ammonia and water are fluids. This makes high absorbent concentration unproblematic and thereby direct air-cooling of the absorber easy (since the it can operate at relatively high concentrations and temperatures).

All parts have a pressure higher than the atmosphere; therefore no air purge system is needed. Subzero temperatures achievable and the unit can be located outdoors (Ryan 2002, Biezanowski 2003, Pridasawas 2002).

H_2O/LiBr (Water /Lithium-bromide)

In a chiller using LiBr (or any other salt) as absorbent crystallization is an issue. Crystallization occurs when the concentration of the absorbent ξ in the solution becomes too high at a given temperature. Crystals of lithium bromide then form and plug the machine (usually the heat exchanger section). The temperatures and concentrations at which crystallization occurs in LiBr-water solution is seen from the crystallization limit in figure 2.32. Crystallization can, as previously mentioned, occur in the heat exchanger when the strong solution leaving the generator is cooled by weak solution. It could also occur in other parts (primary in the generator where concentration is high) for example when the chiller cools down after shutdown. Because of this the solution has to be diluted before or at shutdown of a liquid absorption chiller. There are technical solutions that prevent crystallization during normal operation and to guarantee dilution and safe shutdown even in case of a total power failure (Tri-state 2004).

As seen from the temperature-pressure-concentration diagram in figure 2.32 crystallization sets a limit for the maximum concentration of the strong solution and thereby the maximum absorber temperature at a given evaporator temperature (left axis). For example, if an evaporation temperature of 5 °C is desired the absorber temperature could maximum be 50 °C. Higher absorber temperature would require the concentration of the absorbent solution to be past the crystallization limit. If the evaporator temperature is raised to 15 °C then the maximum absorber temperature is increased to 65 °C. Note that these are the maximum values requiring in principle infinite flow of strong and weak solution. For efficient operation the temperature should be less so that the strong solution entering the absorber can be diluted somewhat before returning to the absorber.

Because of the limited absorber temperature air-cooling of the absorber is complicated as it can be hard to reached low enough temperatures in the absorber a hot summer day. The chiller could have to operate very close to the crystallization limit.

Another disadvantage of LiBr is that it is highly corrosive in contact with oxygen, therefore disposal is closely controlled, (Biezanowski 2003) and care must be taken to prevent air in the chiller. Air in the chiller is prevented through proper sealing and periodic evacuation with a purge system.

Air in the chiller could also result in an increased generator temperature and thereby crystallization (Tri-state 2004).

Inhibitors are also used to prevent corrosion and crystallization. Crystallization inhibitors seem to be especially useable in air-cooled chillers or when heat from the absorber is recovered for some use and a high absorber temperature is therefore desired.

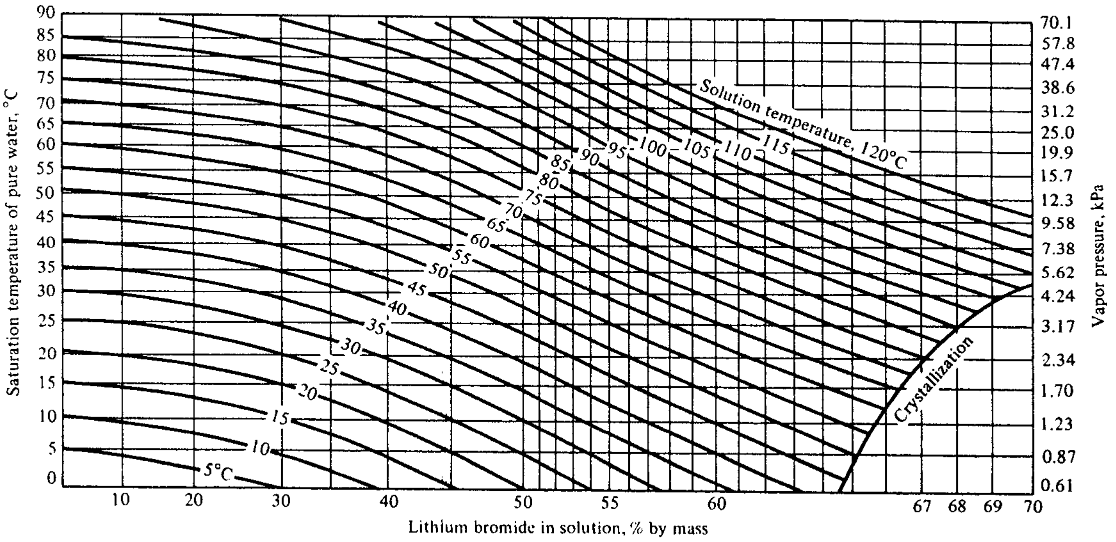


Fig. 2.32 Temperature -pressure-concentration diagram of saturated LiBr-water solutions (Stoecker 1982$^{©13b}$)

Other

Other refrigerant/couples than ammonia/water and water/lithium bromide have been suggested, but no other than NH_3/H20 and H_20/LiBr seems to be commonly used.

Chillers, available and under development

Heat can be provided to an absorption chiller directly from a burning flame and its exhaust gasses (direct fired), by hot water or by steam. (Direct-fired absorption chillers can often be modified to accept hot air or exhaust from a gas turbine or engine. (Tri-state 2004)). Single effect chillers can be direct, water, or steam fired. Commercially available double effect chillers seem to be either direct or steam fired. Triple effect chillers seem not yet to be commercially available, but they are under development and should be close to the market (Grossman 2002).

The smallest water fired absorption heat pump now commercially available seems to be a 10-ton (35 kW) (nominal cooling capacity) H_2O/LiBr heat pump from Yazaki's (Yazaki 2003). Smaller units with cooling capacities of 3 ton (10 kW) have been commercially available especially for solar cooling, but production of these has been discontinued (Grossman 2002). There are today only a few in the capacity range below 100 kW while many chillers are available in the high capacity range typically above 200 kW (IEA posters 2002).

Direct fired air cooled ammonia/water heat pumps are available also in smaller sizes with cooling capacities of 3 and 5 ton.

Research is in progress on developing water-fired chillers with capacities around 10 kW. NH_3/H20 and H_20/LiBr as well as other couples are used in those units (IEA 2002).

After many years of commercialization, absorption chillers have shown, that properly maintained, they tend to have a long life, lasting 20 to 30 years (Biezanowski 2003). There seems to be different opinions on whether they require more or less maintenance than a vapor compression system.

2.2.3.2 Diffusion (Platen Munters) absorption cycle.

In the diffusion absorption cycle, patented by Platen and Munters 1928 the need of a solution pump, and to decrease the pressure in restrictor valves are eliminated by arranging the whole circuit to a single pressure level. It was commonly used in household refrigerators in the 1920's (Gosney 1982).

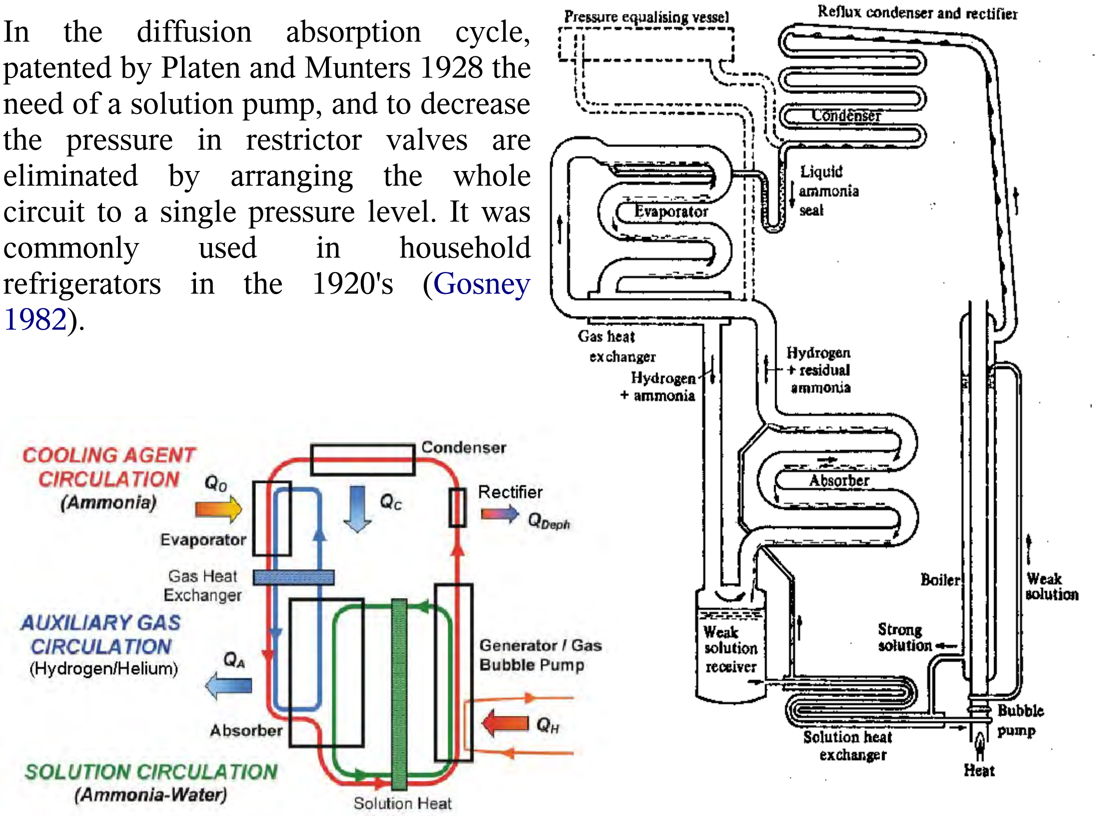


Fig. 2.33 Platen Munters absorption system (Jakob 2002b©14, Goesney 1982©8)

In figure 2.33 a water-fired (left) and a direct-fired (right) heat pump based on the Platen Munters cycle is illustrated. In the evaporator refrigerant evaporates into a gas-mixture consisting of the refrigerant and an inert auxiliary gas. The evaporation is achieved by diffusion (instead of boiling as in the Carré cycle (section 2.2.3.1)). The driving force for evaporation thorough diffusion is that the partial pressure of the refrigerant in the gas-mixture is lower than the refrigerant's vapor pressure (see appendix A). Because of the presence of the auxiliary gas, evaporation can thus take place at much higher absolute pressure than in a chiller based on the Carré cycle. The evaporator can therefore be at the same absolute pressure level as the generator/condenser and there is no need for two pressure levels as in the Carré cycle. A feature of evaporation into a gas-mixture compared to evaporation through boiling is that the temperature of the evaporating liquid does no remain constant. The coldest part of the evaporator is the top where the refrigerant enters, because the partial pressure of refrigerant in the gas-mixture is lowest at this point (Gosney 1982).

As the refrigerant is evaporated into the gas-mixture the amount of refrigerant in it is increased, it becomes rich (with refrigerant). The increased amount of refrigerant in the gas-mixture results in an increased density. It then flows, forced by gravity, down towards the absorber. The rich gas mixture leaving the evaporator is replaced with poor gas mixture (containing a low amount of refrigerant) with lower density from the absorber. On the way towards the absorber the rich gas-mixture passes through a gas-to-gas heat exchanger where it extracts heat from the poor gas mixture, preventing this heat from entering the evaporator.

In the absorber the refrigerant is absorbed from the gas mixture by a strong solution (with high absorbent concentration) counter-flowing the gas mixture. Thus the gas mixture is as richest and the solution as strongest when they enter the absorber. As they pass through the absorber they gradually get poorer and weaker.

For absorption of refrigerant into the strong solution to occur the strong solutions vapor pressure must be lower than the partial pressure of the refrigerant in the gas mixture. Thus the minimum partial pressure of refrigerant in the gas mixture leaving the absorber is set by the vapor pressure of the strong solution entering the absorber (which is a function of concentration and temperature). In the same way the minimum concentration of the solution leaving the absorber is given by the partial pressure of refrigerant in the rich gas mixture entering the absorber.

As in systems based on the Carré cycle there is a solution heat exchanger between the weak and strong solution flowing between the absorber and the generator.

The solution is lifted to the top of the generator by a bubble pump. When the solution in the bubble pump is heated and brought to boiling the solution with bubbles will strive upwards, drawing weak solution from the absorber to replace it. At the top of the generator, where the weak solution enters, the bubbles are released. Weak vapor rises from the generator to a reflux condenser and rectifier, where it is cleaned from remains of absorbent, which flow back into the generator. Strong solution flows from the bottom of the generator, through the solution heat exchanger, to the top of the absorber.

The simple bubble pump can be used because no pressure lift is necessary. The pump only needs to circulate the solution, to compensate for pressure losses due to friction in pipes and to compensate for the small altitude difference between the solution inlet and outlet in the absorber.

Cleaned refrigerant vapor is condensed in the condenser and flows through a liquid seal into the evaporator. The seal prevents uncondensed refrigerant vapor from entering the evaporator.

As refrigerant-absorbent couple in this cycle NH_3/H_2O is commonly used. The original Platen Munters heat pump uses hydrogen as auxiliary gas (Gosney 1982). An alternative is to use helium (IEA 2002).

The Platen Munters heat pump was originally designed for cooling household refrigerators. Electrically fired they were used until the 1960th when compressor refrigerators, which are much more efficient reached the market. It is still used for refrigerators in hotel rooms (mini bars) where silent operation is desired and for gas-fired refrigerators. For theses applications the cooling-capacity of the heat pumps is rather small, typically below 100 W.

Two projects with the aim of scaling up the diffusion absorption heat pump are described in the IEA Task-25 final report (2002). One is the JOULE-CRAFT project working on a system called DACM (Design of a Solar Driven Cooling Unit based on the diffusion absorption principle). It uses the same cycle as above described with an aimed cooling capacity of 2.5 kW. The other, the EC-project LSSC uses a slightly modified cycle. They call their system Advanced Ammonia Absorption Cooling (AAAC). Different sizes have been proposed. One idea is to combine parallel-coupled 500 W units to an arbitrary size unit. SolarFrost have made prototype with a capacity of 2 kW, and plans to build one with 4 kW cooling capacity (SolarFrost 2003).

The diffusion ammonia-water absorption systems require rather high generator temperatures. As an example the design temperatures of the DACM-system is given in table 2.1.

Table 2.1 Design temperatures DACM (Jakob 2002)

Generator inlet	101 °C
Generator outlet	112 °C
Evaporator	5 °C
Condenser	45 °C
Absorber	45 °C

If the generator temperature is low, then the solution leaving the generator will be weaker and thus the solutions vapor pressure when entering the absorber will be higher. A higher solution vapor pressure will lead to a higher partial pressure of ammonia in the poor gas mixture entering the evaporator, and thus a higher evaporation temperature. Insufficient cooling of the condenser and absorber will give similar result. As these systems are generally air-cooled high ambient temperatures can cause insufficient cooling.

At a certain point the system will stops operating and at very low heating temperatures the bubble pump collapses and the system needs to be repaired to restart (Solarfrost 2003).

The modification done by Solarfrost is to add a “bypass” into the system. In this bypass, strong hot solution from the generator (with a high vapor pressure) is brought into contact with rich gas mixture from the evaporator after it passed the gas heat exchanger. Then additional refrigerant will be desorbed from the solution into the gas mixture. The solution and the gas mixture then passes heat exchangers and enter the absorber. This will result in a stronger solution and a richer gas mixture entering the absorber. The partial pressure of the poor gas mixture leaving the absorber can thus be poorer and the solution leaving the absorber weaker. Required heating temperature is thus reduced to about 75-80 °C. This principle is illustrated in figure 2.34.

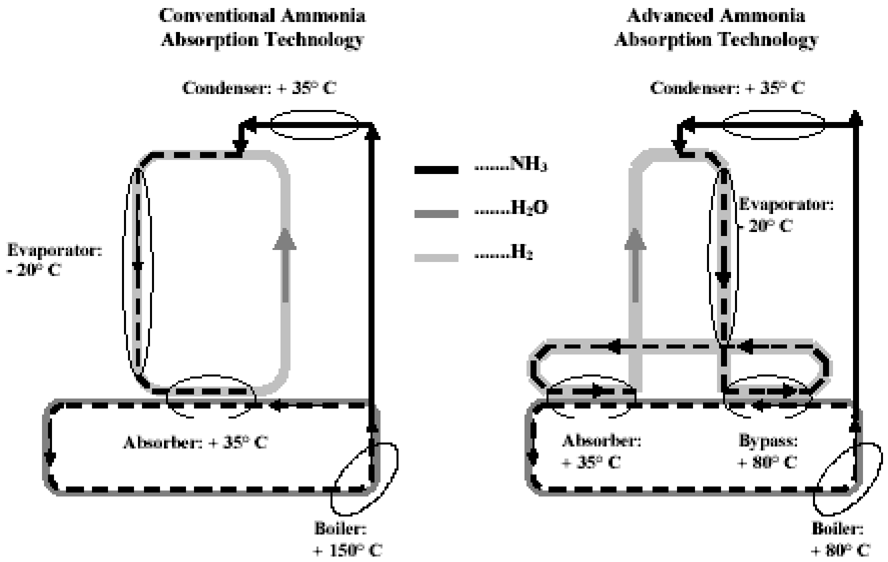


Fig. 2.34 Principle of AAAC (IEA 2002©15)

The Carnot efficiency of available high capacity diffusion absorption chillers prototypes seems to be much less than for systems using the Carré cycle. Maximum COP obtained from the DACM prototype is 0.25. Projected COP for the DACM is 0.53 without and 0.72 with heat recovery of rectifier losses, at generator temperature 117 °C, absorber/condenser temperature 45 °C and evaporator temperature 5 °C (Jakob 2002). If assumed that this means heating temperature ~130 °C, cooling fluid temperature (ambient air) ~32 °C and chilled water temperature ~7 °C, this results in a Carnot COP (e6) and efficiency (e3) of:

$$COP_{Carnot}=(1-\frac{32+274}{130+274})*(\frac{7+274}{32-7})=2.72 \qquad \eta_{Carnot}=\frac{0.53}{2.72}=0.19 \text{ or } \frac{0.72}{2.72}=0.26$$

As comparison the COP for a single effect water chilled lithium-bromide chiller with heating fluid temp 93 °C, cooling water temp 35 °C and chilled water temperature 7 °C is ~0.68 (fig. 2.25), this results in a Carnot COP and efficiency of:

$$COP_{Carnot}=(1-\frac{35+274}{93+274})*(\frac{7+274}{35-7})=1.59 \qquad \eta_{Carnot}=\frac{0.68}{1.59}=0.43$$

One should however keep in mind that requiring cooling water of 35 °C from ambient air of 32 °C would most likely require wet cooling towers.

Detailed COP values for the AAAC have not been obtained. Projected COP is said to be 0.3-0.7 (Solar Frost 2003), but it is not clearly specified at which temperatures this applies.

A disadvantage of the diffusion absorption system is that it seems to grow rather much in height when scaled up, and thus scaled up units becomes rather tall.

The main advantage seems to be the use of a bubble pump. Since a bubble pump is used it has no moving parts and can therefore easily be hermetically sealed. It should be noted however that Yazaki's absorption chillers, which works in accordance with the Carré cycle, according to their own description also seems to be working with a thermally driven solution pump, without any moving parts (Yakzaki 2003).

A general advantage of the Munters-Platen concept, and other types of absorption refrigerators and air-conditioners using ammonia, is that the manufacturing of such an equipment can be made with rather simple tools and components. This aspect is especially interesting for applications on developing countries where a low first cost is more important than a high COP (Lindholm 2003b).

Delano (1997) describes a different diffusion absorption cycles using ammonia-butane-water, which was patented by Albert Einstein 1930.

2.2.3.3 Open absorption cycle

In an open absorption cycle the refrigerant vapor from the generator is discarded to the atmosphere, instead of flowing to a condenser to be reused. Since refrigerant is constantly lost, new refrigerant must constantly be added. The refrigerant in question here must be safe to release to the environment, cheap and easily obtained, therefore water is used as the refrigerant.

In principle the absorption heat pump in figure 2.21 can be transformed to an open absorption heat pump by simply removing the condenser and let the refrigerant vapor from the generator flow out to the environment. The evaporator would then be feed with a supply of make-up water (for example tap water). However done in this way the refrigerant would still have to be desorbed by boiling. Boiling the solution at atmospheric pressure would require very high temperatures.

If instead the solution is allowed to be in direct contact with ambient air, it is enough that the vapor pressure of the solution is raised to above the partial pressure of water vapor in the ambient air. The partial pressure of water vapor in air can be read from a psychometric or Mollier chart such as included in appendix B. As can be seen from the Mollier and psychometric charts, the partial pressure of water vapor in air varies from as low as a few mbar while the vapor pressure in a closed system usually is around 40-50 mbar (Collier 1979). (For example the partial pressure a humid summer day with 25 °C and 70 % rh would be 22 mbar.) Generally an open system seems to work with lower generator temperatures than a closed system. However required temperatures for desorbtion cannot be fully compared simply by comparing pressures, since other absorbents and different solution concentrations are generally used. Direct contact between solution and air with large contact surfaces also results in efficient mass and heat transfer.

Generally two kinds of generators seem to be useable in these kind of systems, either a combined generator and solar collector or an open flow-through desorption chamber of the type commonly used in liquid desiccant systems.

Since no condenser is used there is one component less in this system compared to a closed system. If a solar generator is used there is no need for additional solar collectors. Since the desorption is done at atmospheric pressure there is no need for a pressure vessel for the generator and the condenser. An open system can thus be cheaper than a closed.

Note however, that since the generator is at atmospheric pressure the lift necessary solution pump lift is considerable higher than in a closed system. A vacuum pump is also needed to constantly purge gases that follow the solution into the low-pressure part after the solution has been exposed to the atmosphere. The energy consumption of the air purge system is thus higher than in a closed system (Collier 1979), where it only needs to run about one hour a week (Tri-state 2003). There could also be problems with accumulation of hardness salts and scaling in the evaporator (Gosney 1982).

A complete system using a combined generator and solar collector is shown in figure 2.35.

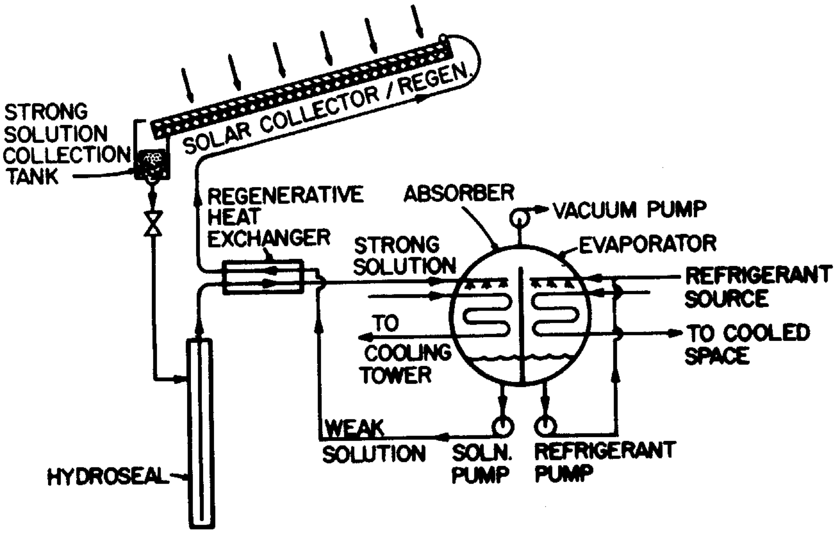


Fig. 2.35 Solar generator open absorption system (Collier 1979$^{©16}$)

The open generator

The combined generator and solar collector consist of a tilted surface, preferable of a material that is good at absorbing solar radiation and have a roughness to provide a large surface area. Kakabaev and Rakhmanov (1971) used a surface covered with Ruberoid. However any blackened tilted surface would work (Collier 1979). The back of the surface is thermally insulated. Weak solution is released from near the top of the tiled surface and flows along the surface from the top towards the bottom. As the solution flows downwards the surface and the sun heats it and this results in water being desorbed to the ambient air. Thereby the solution is getting stronger and at the end of the surface the strong solution is collected and returned to absorber.

The other kind of open generator consists of a chamber in which an airflow of ambient air is forced through a counterflowing solution from the bottom towards the top. The solution is heated, either in a fluid heat exchanger before entering the chamber or by heating coils inside the chamber (ASHRAE 2000). As the hot solution meets the relatively dry air, water is desorbed from the solution to the air.

Several ways of bringing the solution in contact with the air can be used. A rather common method is to spray the solution over a packing (fig. 2.36). Alternatively it can be sprayed onto wet surfaces, or let onto wet surfaces using overflowing channels above the surfaces (Forkosh 2002, Kessling 1998) or by wicks (Lowenstein 1998). The solution then flows down the wet surfaces as a falling film

In the generator shown in fig. 2.36 solution from a sump in the bottom of the chamber is heated and recirculated into the chamber. The circulated solution is sprayed over the packing and flows from the top of the chamber towards the bottom. The circulated solution flows into the sump where it mixes with strong solution. An air-to-air heat exchanger is used between incoming and outgoing air to reduce the amount of heat lost from the desorption chamber and to preheat the air entering the chamber. Preheating the air entering results in lowering the air's relative humidity so that the air can take up water more efficiently, and reduces the incoming air's cooling effect on the solution.

Disadvantages of open generators are: Contamination of solution by dust and dirt from ambient air and polluted rain and potential carry over of sorbent from the solution to the air. This may require replacement of the solutions after some period of op-

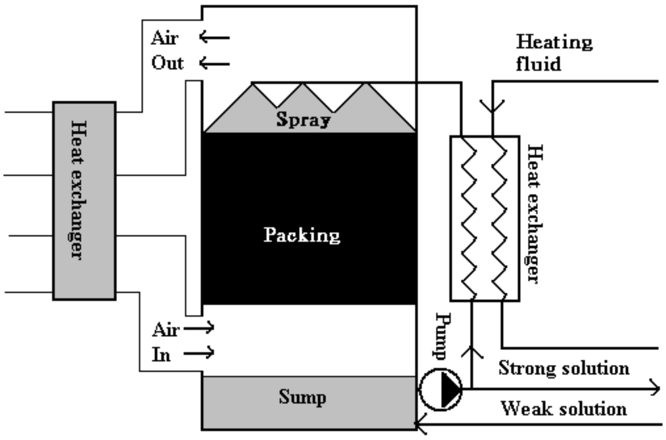


Fig. 2.36 Open generator with spray chamber

eration. Carry over of solution will result in a loss of absorbent (Grossman 2002). If a salt is used as absorbent (which is most often the case) it could also result in a highly corrosive substance being let out into the ambient air, where it could come in contact with materials that are subject to corrosion. Since the generator is in contact with both air rich in oxygen and the potentially corrosive solution the demand on the materials used are high.

To prevent carryover in generators in which the solution is spray-distributed, a filter, called mist eliminator, is often used to catch drops of solution from the exiting airflow.

In a solar generator, rainwater must be prevented from diluting the solution (Collier 1979). In a solar generator described by A. Johannsen (Johannsen 1983) (fig 2.37), rain is prevented from coming in contact with the solution by a glass surface covering the collector area. The covering is open at the top and bottom to allow ambient air to flow through.

Because of cost and corrosion considerations, Collier (1979) chose to use LiCl instead of LiBr as absorbent. Also Kakabaev and Rakhmanov (1971) used LiCl while the system simulated by Johannsen and Grossman (1983) worked with triethylene glycol as absorbent. Glycol is less corrosive but has a much higher vapor pressure than LiCl and carry over could thus be a problem with this absorbent (Lowenstein 1998).

Calculating a COP-value comparable with that of a closed system is complicated by factors that influence an open system but not a closed. Those are the energy cost of the make-up water, the partial pressure of water in the ambient air, and the flow rate of ambient air through or past the generator. This also means that $COP_{Carnot,\ 3T}$ is not really appropriate as the ideal COP for these systems.

Make-up water is often not included in the COP. In areas where sweet water is available in abundance, make-up water could be seen as a resource free from energy input (except for pump work needed) and needs not to be included. However this is not the case in many parts of the world. And if the make-up water has been desalinated, or treated in any other way, the energy used for that should be taken into consideration when calculating the total COP of the system.

The cost for make-up water will also be an operational cost for an open system. Water

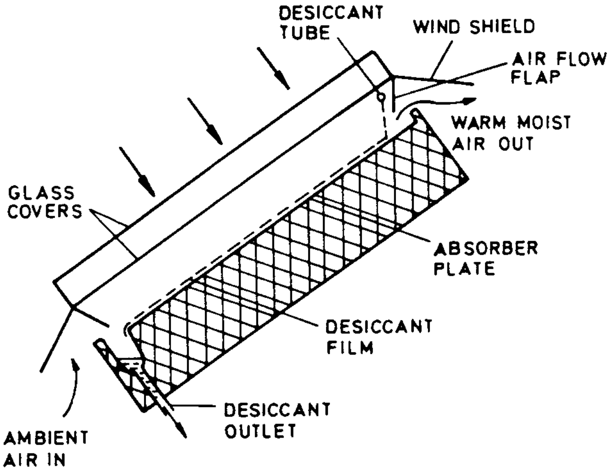


Fig. 2.37. Solar generator with glass covering (desiccant=absorbent-water solution) (Johannsen 1983©17)

is, however, generally cheap. It can be relatively cheap even in areas with a shortage of water, because, since water is essential for industry and agriculture it is often subsidized. However one should consider if it is wise to use such an essential substance as water for cooling in areas where there is shortage even if it is economical.

With the motive to save water as well as removing the other disadvantages of an open generator. Kakabev and Golaev 1971 suggested covering the solar generator with a glass surface and to seal it making the system closed. The glass surface will then work as a condenser. Condensed refrigerant flows along the glass surface down to the bottom where it is collected and returned to the evaporator. Some of the simplicity of an open system would remain, such as no need for a heat rejection device or solar collectors and there is only three parts instead of four. However, it would remove the advantage of desorbtion to the ambient air, and result in the need to boil the solution at atmospheric pressure.

2.2.3.4 Solid sorption heat pumps

A solid sorption heat pump uses either a dry adsorbent or a dry absorbent to ad- or absorbs vapor from an evaporator. This is similar to how vapor is absorbed in a solution in the liquid absorption heat pump. The important difference is that since the sorbent is solid it cannot be cycled by pumps between tanks at different pressure. This is why the cycle and the construction of a dry sorption heat pump is rather different from the construction of absorption heat pumps base on the Carré-cycle.

The difference between absorption and adsorption is that when a substance (the refrigrant) is absorbed it is bound *into* the absorbent solution. When a substance is adsorbed it is bound only to *the surface* of the adsorbent.

Or as defined by American Heritage Book of English (1996): "*Absorption* indicates an active ongoing process in which something is taken up by something else by various physical actions: *The absorption of spilled juice into a paper towel occurs by capillary action. Adsorption,* in contrast, describes the holding or accumulation of something, such as a gas, a liquid, or a solute (a substance that has been dissolved in another substance), on the surface of a solid or liquid.

Figure 2.38 illustrates schematically the difference between absorption (left) and adsorption (right). The left part of figure 2.39 (the enlarged part inside the circle) illustrates how sorbent molecules are adsorbed on the surfaces of sorbent molecules.

Fig. 2.38 Schematic description of the difference between absorption (left) and adsorption (right) (Servitec 2004©18)

The most commonly used refrigerant is water and the most commonly used dry sorbents are adsorbents like silica gel or zelolite. Because of this dry sorption systems is often referred to and explained as adsorption systems. The focus then often tends to be on the difference between absorption and adsorption, which is explained above, rather than the difference between liquid and solid sorption. However the difference between absorption and adsorption has little effect on the design of the machine and even less to the use of it.

Since the dry sorbent can not be easily moved, sorption as well as desorption has to be done in a single container, called reactor or sorption chamber, which is shifted between sorption and desorption mode.

A simple dry sorption heat pump can consist of a single chamber containing the sorbent and a combined evaporator and condenser. However this machine works discontinuously. Therefore it cannot pump any heat while the refrigerant is desorbed and condensed and not use any available driving heat while the refrigerant is being evaporated and ad/absorbed. To enable continuous cold (or heat) supply and to take care of available driving heat, heat and cold storage is needed.

To improve the efficiency both commercially available adsorption chillers uses two chambers as shown in figure 2.39. This gives the following cycle.

1. Sorbent surfaces in the right chamber are rich with refrigerant and adsorbent surfaces in the left chamber poor ("rich" with adsorbent). The upper vapor valve of the right chamber leading to the condenser and the lower valve of the left chamber leading to the evaporator are open. Sorbent surfaces of the right chamber are heated and refrigerant are thereby desorbed from the surfaces as vapor. The refrigerant vapor flows to the condenser where it is condensed to liquid. The liquid flows through a restrictor to the evaporator where it evaporates at low pressure. The evaporated refrigerant vapor flows to the left chamber where it is adsorbed. Heat of adsorption is released and removed by cooling the left chamber with a cooling fluid. The process would continue until saturation, when no more refrigerant can be desorbed from the right chamber or no more refrigerant can be adsorbed by the left chamber at current heating, cooling and chilled fluid temperatures. However, since the capacity of the chiller decreases as the sorbent in the left chamber is getting rich with refrigerant and the right chamber getting poor, the process is aborted before that.
2. Valves are closed and the cycle is reversed. To achieve this the right chamber has to be cooled and the left heated and the pressure situation reversed. To conserve heat, heat from the right chamber can be used to preheat the left chamber. One way is to open a valve between the two chambers. This will even out pressure difference and transfer some heat from the left chamber with higher temperature and pressure to the right chamber with lower temperature and pressure. Another way is to circulate the cooling fluid through right and left chamber in series. Both these methods are used in a commercial adsorption chillers marketed by HIJC USA INC (HIJC 2003).
3. The upper valve of the left chamber and the lower valve of the right chamber are opened. The left chamber is heated and the right chamber is cooled. Opposite of 1. occur.
4. Opposite of 2. and thereby the cycle is closed.

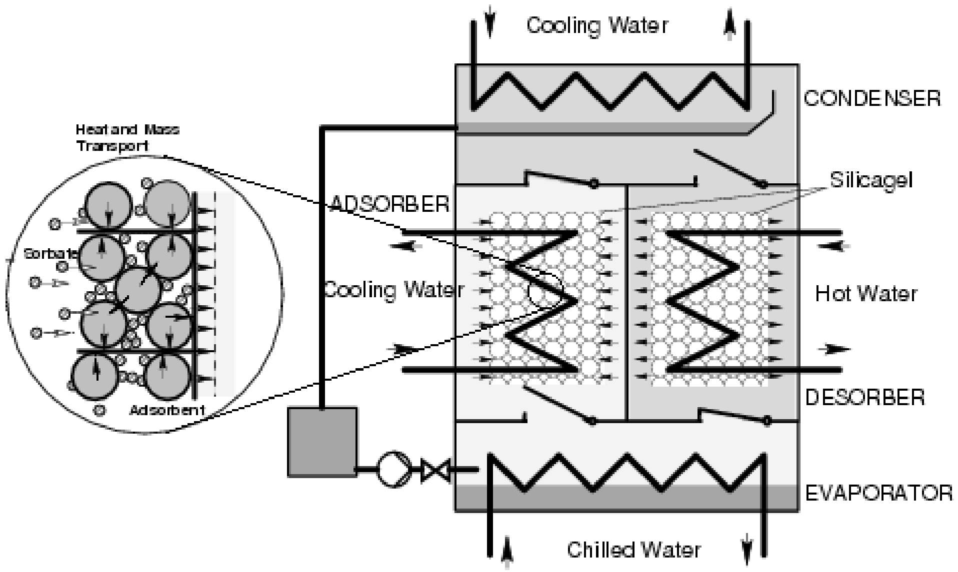


Fig. 2.39 Two chamber adsorption chiller (Oertel 1997b©19 ,1998©19)

As seen in the above description of the cycle, it is not fully continuous, since interruptions in chilled fluid supply and heating fluid demand occur during shift of mode, until the chambers have reached working temperatures and pressures. The capacity of the chiller also decreases towards the end of the cycle. To make it more continues it has been proposed to use four chambers. A four chamber chiller prototype has been built (IEA 2002).

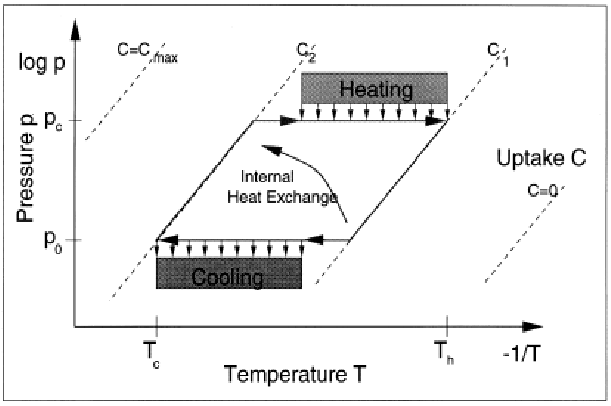
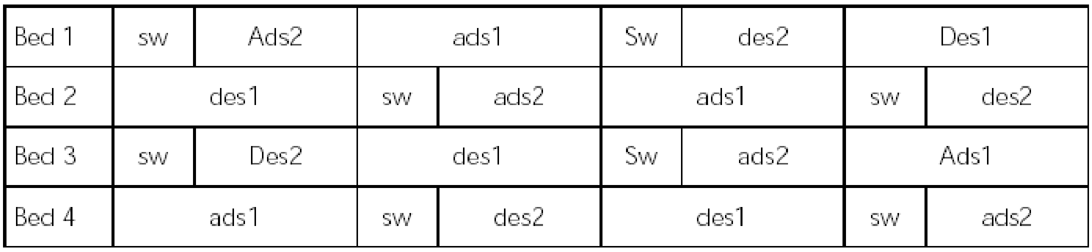


Fig. 2.40 Claperyron diagram for solid sorption cycle (Oertel 1998©20)

An aspect of the solid sorption chiller's intermedient cycle is that the demanded temperature actually changes throughout the cycle. Heating fluid temperature could be lower in the beginning of the cycle than in the end. In the same way cooling temperature could be higher in the beginning than in the end. This can be seen in the Claperyron diagram of the cycle (similar to a Dühring plot) in figure 2.40.

An application for this could be to use two heat sources of different temperatures (similar to the double generator system in section 2.2.3.1). First the lower temperature heat source could be used to desorb as much refrigerant as possible. Then the higher temperature heat source could be used to desorb additional refrigerant. Similarly heat sinks of different temperatures can be used. (For reasons to do so see “Multiple heat sources and heat sinks” section 2.2.3.1). Using multiple heat sources and heat sinks would increases the amount of discontinuity in the system, such a system is therefore probably best realized as a four-chamber system.

In the four-chamber adsorption chiller prototype the above mentioned aspect is used. Absorbers and desorbers are cooled and heated in serial with the one demanding lowest/highest temperature first. Thus the temperature difference between heat inlet and outlet may be up to 20 °C, compared to 5 °C for a two-chamber adsorption chiller. Figure 2.41 shows how this is done. Two heat sources or heat sinks (with different temperatures) could then provide heat/cold by heating or cooling the supply fluid in serial (IEA 2002).



Bed 1	sw	Ads2	ads1		Sw	des2	Des1	
Bed 2	des1		sw	ads2	ads1		sw	des2
Bed 3	sw	Des2	des1		Sw	ads2	Ads1	
Bed 4	ads1		sw	des2	des1		sw	ads2

Fig. 2.41 Switching schedule for a 4-bed adsorption chiller over two cycles. “ads” is adsorption, “des” is desorption, and “sw” is switching from adsorption to desorption and vice versa. Number 1 – the bed receives the cooling stream directly from the condenser or the heating stream directly from the heat source. Number 2 – the bed receives cooling from “ads1” or heating from “des2” (IEA 2002).

An adsorption chiller can be operated at lower heating fluid temperatures than absorption chillers. However at higher temperatures it has a lower COP (than an absorption chiller). Figure 2.41a show the COP of a two-chamber water/silica-gel adsorption chiller compared to that of a water/lithium-bromide absorption chiller. The COP and capacity of an adsorption chiller is, in addition to temperature, affected by the cycle length as seen above (1. in description of cycle). Longer cycles give a better COP but less capacity (Saha 1995, IEA 2002). Too short cycles result in low capacity as well as low COP (Saha 1995). Determining cycle time is thus an optimization between capacity and efficiency. Figure 2.41b shows the COP's dependency on cycle time.

To acquire low evaporation temperatures other refrigerants like methanol can be used. A system using methanol will have worse performance than a system using water due to the lower specific phase change energy of methanol (Oertel 1997a).

Adsorbents are not corrosive like the absorber-salts and ammonia used in absorption chillers. Therefore inhibitors are not needed, and disposal should be easier. According to supplier data there is no need for replacement of adsorbent during the chillers lifetime (HIJC 2003).

The adsorption chillers are more expensive per kW cooling capacity than absorption chillers. They also requires more effort in design and control due to their cyclic nature and they are big and heavy (IEA 2002).

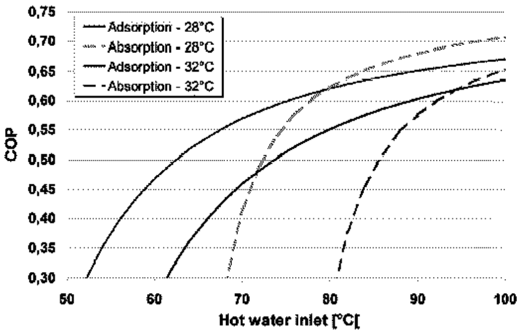
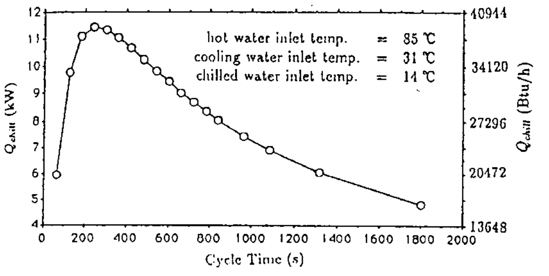
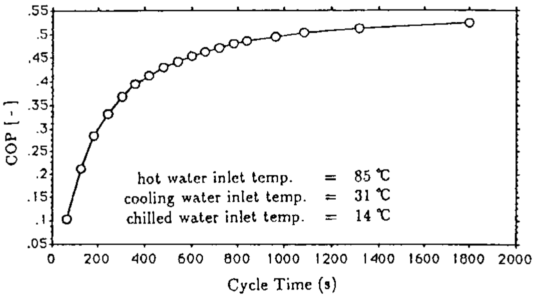


Fig. 2.42a COP of adsorption and absorption chiller as a function of heating water temperature (chilled water 9° C,cooling water 28 and 32 °C) (IEA posters 2003©44)



Effect of cycle time on cooling output.



Effect of cycle time on COP.

Fig. 2.42b COP and capacity of silica gel adsorption chiller as function of cycle time (Saha 1995©21)

2.2.3.5 Ejector (steam/vapor jet) cycle

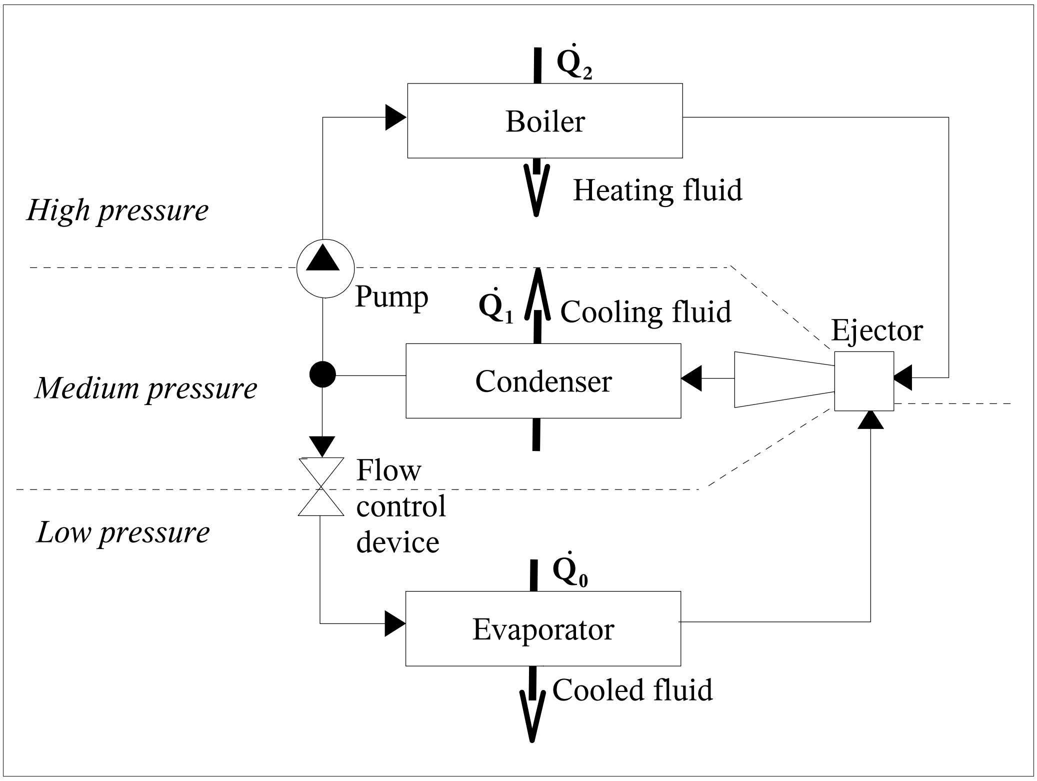


Fig. 2.43 Schematic ejector refrigeration cycle

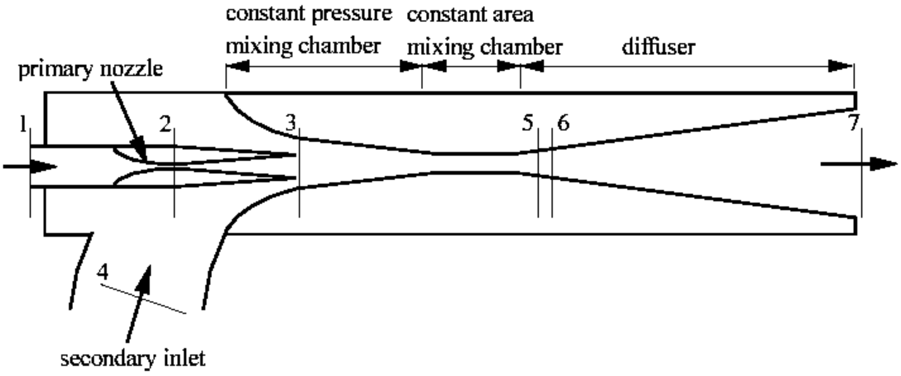


Fig. 2.44 Cross section of an ejector (Wolpert 2000©22)

The ejector or steam/vapor-jet cycle, illustrated by figure 2.43 can be seen as a combination of the vapor compression cycle explained in section 2.2.1.1 and the Rankine power cycle explained in section 2.2.2.1. The heart of the cycle is the ejector that replaces both the compressor in the vapor compression system and the expander in the Rankine power cycle.

A typical ejector cross section is shown in figure 2.44. Inside the nozzle the vapor from the boiler is accelerated to a high velocity (typically mach no. >2) (1-3) (Wolpert 2000). After the vapor nozzle, the supersonic vapor blows into the mixing chamber. This causes a low pressure in the mixing chamber. Because of the low pressure, refrigerant is sucked into the mixing chamber through the secondary inlet (4); thereby a low pressure in the evaporator is achieved. In the mixing chamber the vapor from the boiler and vapor from the evaporator are mixed (3-5). After mixing, the combined stream will become a transient supersonic stream. The velocity of the combined fluid must be high enough to increase the pressure after deceleration in the diffuser (5-7) to a suitable condensing pressure (Pridasawas 2002).

The rest of the two sub-cycles function in the same way as in the vapor compression and the Rankine power cycles (section 2.2.1.1 and 2.2.2.1)

Different refrigerants, both natural and organic (CFC, HCFC and HFC) can be used in an ejector refrigeration cycle.

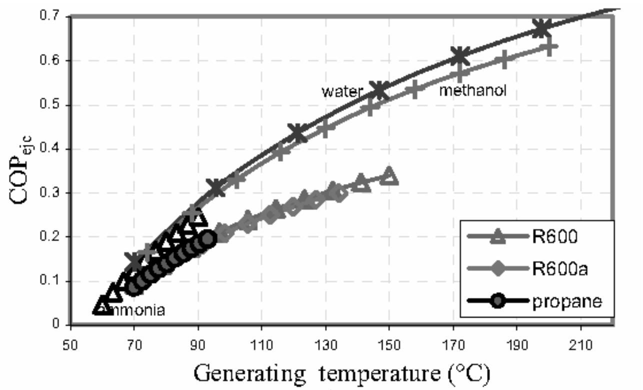


Fig. 2.45 COP ejector refrigeration system. Evaporating temp. 10 °C Condensing temp. 30 °C (Pridasawas 2003$^{\copyright 23}$)

Fig 2.45 shows the COP as a function of the boiler temperature for the natural refrigerants water methanol, butane (R600), isobutane (R600a) and propane.

Data for more refrigerants (natural, and organic) is available in appendix C. In appendix C the COP-values dependence on condensation and evaporation temperature is also included.

Water shows the highest efficiency of the refrigerants. Carnot efficiency around 10 % can be obtained. The disadvantage with water as refrigerant (as stated before in section 2.2.3.1) is its high specific volume resulting in a high volumetric flow and a large size of ejector is required. The evaporation temperature cannot be below 0 °C (Pridasawas 2003). Water is the most environmentally friendly option.

Methanol shows to be a little bit less efficient than water. However, it has a specific vapor volume that is much lower. Thus the ejector size required is small (Pridasawas 2003). It is environmentally friendly but flammable.

Ammonia can also be used; it has been used as a refrigerant for a long time and gives a good performance. The Carnot efficiency is quite high. The pressure-enthalpy characteristic curve of ammonia shows that it requires more superheating than other refrigerants due to a negative slope of the saturated vapor line. Thus condensation may occur inside the ejector, which causes failure in operation (Pridasawas 2002). Ammonia has a low specific volume. For other aspects of ammonia see section 2.2.3.1.

Other refrigerants in figure 2.45 and appendix C seem to be less efficient and results in lower COP values. However other considerations like system-size could make them interesting. For example using the HFC-refrigerant R134a would result in a quite small ejector compared to other refrigerants (Pridasawas 2002). As mentioned before, the use of some HFC-refrigerants can be questioned because of their global warming potential (there ability to contribute to global warming if released to the atmosphere).

Changing the evaporating or the condensing temperature affects the COP of the system more than shifting the boiler temperature. One degree Celsius shift of the evaporating or the condensing temperature changes the COP approximately 2% while one degree Celsius shift of the boiler temperature changes the COP only about 0.3% (Pridasawas 2002).

Electric power input to the pump is not included in the COP for the same reasons as for not including electricity consumption for the solution pump in a absorption machine (see section 2.2.3.1). However, more energy seems to be needed from the pump in an ejector cycle than in an absorption cycle. In a system described by Wolpert 2000 the electric demand for the pump is about 1/4th of the heat demand. Low COP, as seen from figure 2.45 and appendix C, is a drawback of this system; however, when the temperature lift is small, the ejector cycle yields a COP higher than the corresponding COP of an absorption system (Pridasawas 2002). The COP-plots further seem to suggest that it does not have as drastic collapse of COP as the absorption system at low boiler temperatures.

To improve the efficiency, multi-stage ejectors have been proposed by Chen (1997) and Grazzini (1998). In Chen's setup, shown in figure 2.46a, the exit stream from the first stage ejector is sucked through the secondary inlet of the second stage ejector. In Grazzini's setup, shown in figure 2.46b, the exit vapor from the first stage ejector instead enters the primary inlet of the second stage ejector.

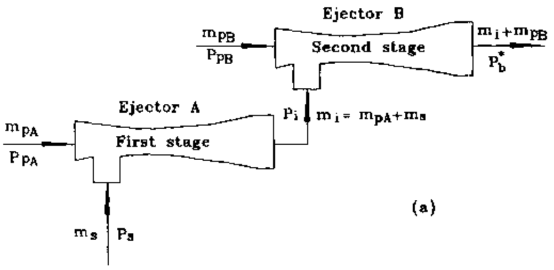


Fig. 2.46a Multistaged ejectors as suggested by Cheng (1997[©24])

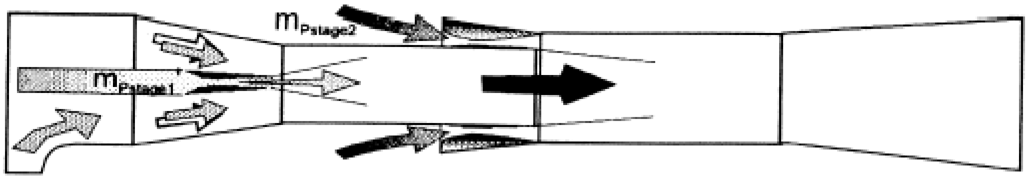


Fig. 2.46b. Multistaged ejectors as suggested by Grazzini (1998[©25])

The ejector cooling system is quite cheap compared to other heat operated cooling systems i.e. at a low generating temperature, this system is cheaper than a absorption cooling system. Its simplicity in installation, design and operation are other advantages (Pridasawas 2002).

Its simplicity and low COP makes it somewhat similar to the diffusion absorption system (section 2.2.3.2). Like for the diffusion absorption system this can make it an interesting option for developing counties.

If the pump can be eliminated by using the device suggested (for replacing the pump in a Rankine cycle) by Sterling (1997) (see section 2.2.2.1) the cycle will be without moving parts (valves excepted).

An ejector can also be used as a vacuum pump to purge non-condensable gases from an absorption or adsorption system.

2.2.4 Evaporative cooling

Evaporative cooling means using evaporation of water to cool a passing airflow. The airflow passes through either a spray of water droplets (size 1-15 µm (Lindholm 2000)) or a wetted material. As the (unsaturated) airflow comes in contact with the water, water evaporates into the air (see appendix A). If no heat is added, the heat required for evaporation will be taken from the sensible heat of the air. The sensible heat of the air and thus its (dry-bulb) temperature will then be reduced. The latent heat of the air will at the same time be increased by the same amount as sensible heat of the air is reduced. The total energy of the air thus remains constant. The change of state of the air towards saturation described on a psycrometric or Mollier chart (fig. 2.47), essentially follows the state line for wet bulb temperature of the air, which is approximately equal to the state line for enthalpy.

The above holds when the water temperature is equal to the wet bulb temperature of the air. If the water is colder than the air wet bulb temperature, then additional sensible cooling will take place. If the water is warmer, then sensible heating, which to some degree counteract the evaporative cooling,will occur.

Figure 2.48 illustrates a device in which water is sprayed into a passing airflow. Eliminators remove remaining water droplets from being carried away with the cooled air. A pool in the bottom of the device collects water that has not been evaporated. This water can then be recirculated and reused. When the water is continuously recirculated its temperature will become equal to the passing airflow's wet bulb temperature. To prevent the build up of the concentration of (for example) mineral salts an amount of water from the pool needs to be discharged. To replace the water evaporated and discharged an amount of makeup water needs to be supplied. This device is called a spray humidifier or an air scrubber/washer. The later because it also has the effect that it washes the passing air from particles and dust.

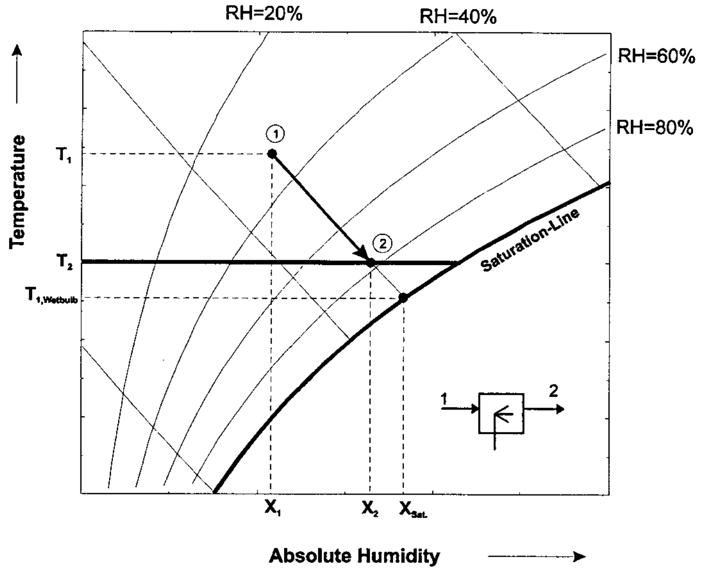


Fig. 2.47 Evaporative cooling in Mollier chart (Dittmar 1997)

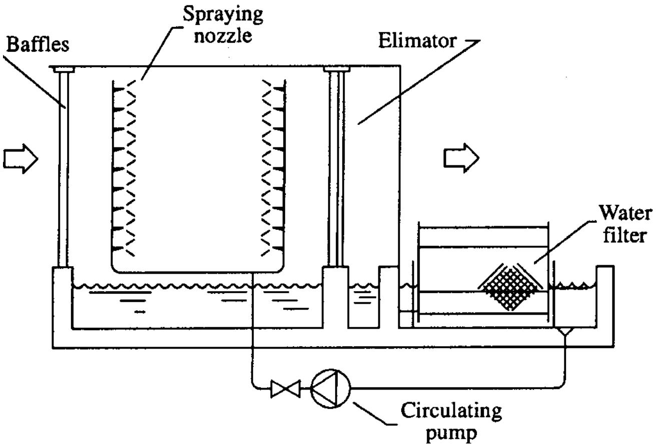


Fig. 2.48 Spray humidifier (Wang 1999$^{©26}$)

Other devices use wetted materials such as pads of treated wooden fibers, rigid or corrugated plastic, impregnated cellulose, or glass fiber (Wang 1999). Water is let onto the material from above wetting it. As air flows through the material water evaporates from the surfaces of the material into the air stream. In this way of humidification no water droplet can be carried away with the airflow (provided that the speed of the airflow is not too high) (and thus no eliminators are needed). Excess water is, as in the spray humidifier, collected from the bottom of the humidifier. This kind of device is called an evaporative humidifier.

Studies have shown that evaporative humidifiers are more hygienic than spray humidifiers (Lindholm 2002). This is because of the possibility of carryover of contaminated water in a spray humidifier. The contaminants of concern are types of bacteria and microorganisms that can cause what is known as humidifier fever, a sickness with similar symptoms as mild influenza.

In some cases (for example because of hygienic concerns,) non of the excess water is circulated. Instead all excess water is drained away and all water used for humidification is supplied externally. The water consumption in a direct water humidifier is then of cause larger than when circulated water is used.

The effectiveness of a humidifier used for evaporative cooling can be expressed as (Dittmar 1997):

$$\eta_{\text{Hum}} = \frac{T_1 - T_2}{T_1 - T_{1,\text{Wetbulb}}} \qquad (e9)$$

Where T_1 is the incoming air temperature, T_2 is the exiting air temperature and $T_{1,\text{Wetbulb}}$ is the entering air wet bulb temperature.

2.2.4.1 Direct evaporative cooling

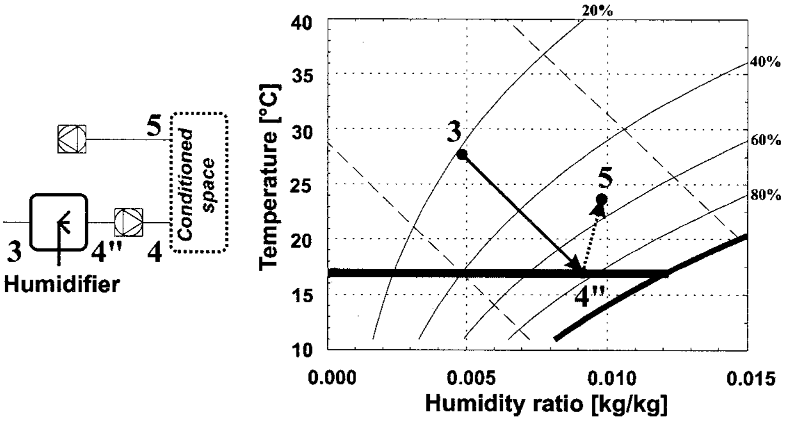


Fig. 2.49 Direct evaporative cooling (Lindholm 2000[©27])

Direct evaporative cooling is when the evaporatively cooled airflow is used as supply air as illustrated in figure 2.49. How much the air can be cooled is limited by how humid the air entering the conditioned space can be accepted to be. This depends on the size if internal latent loads (internally generated humidity) and accepted internal humidity level. According to Dittmar (1997) up to 80% relative humidity is acceptable (within the comfort zone). This however, seem to be a very generous limit. Others recommend for personal and building health reasons an upper limit at 60% Rh.

2.2.4.2 Indirect evaporative cooling

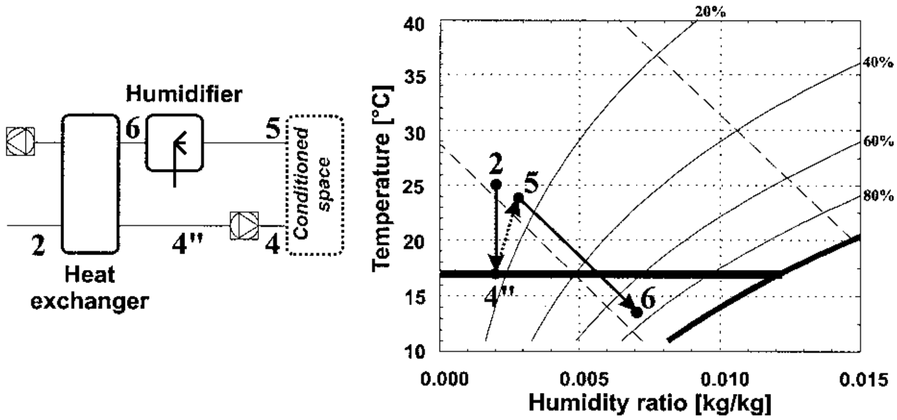


Fig. 2.50 Indirect evaporative cooling (Lindholm 2000[©27]*)*

Indirect evaporative cooling is when an airflow other than the airflow used as supply air is evaporatively cooled. Heat is, as illustrate in figure 2.50, transferred from the supply airflow to the evaporatively cooled airflow using a heat exchanger. The humidifier and heat exchanger can also be integrated into a single unit.

An advantage of indirect evaporative is that no humidity is added to the supply airflow. The supply air is therefore, as seen in figure 2.50, much drier than when direct evaporative cooling is used. An other advantage is that that contaminants from the water cannot (easily) contaminate the supply airflow, therefore the demand on the water quality is less. A disadvantage is that thermal losses in the heat exchanger are introduced.

2.2.4.3 Indirect-direct evaporative cooling

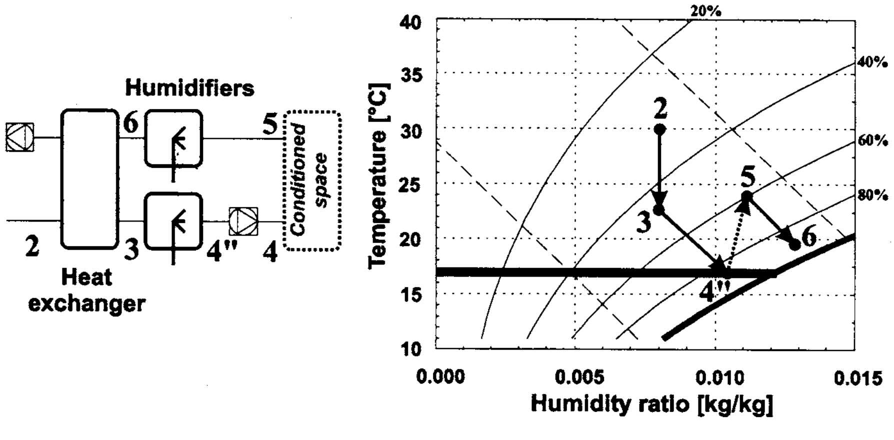


Fig. 2.51 Indirect-direct evaporative cooling (Lindholm 2000[©27]*)*

Indirect-direct evaporative cooling is, as illustrated in figure 2.51, a two stage evaporative cooling where the supply airflow is cooled through indirect evaporative cooling (2-3) and then further cooled through direct evaporative cooling (3-4). In this way desired supply air conditions can be met at tougher ambient air conditions (compare state 2 fig. 2.49 with state 3 fig. 2.50 and state 2 fig. 2.51).

2.2.5 Desiccant dehumidification

Desiccant dehumidification works on the same basic principles as absorption and adsorption heat pumps explained in sections 2.2.3.1-4. Water vapor is ab/ad-sorbed from the air to the sorbent or desiccant because of a difference between the partial pressure of water vapor in the air and the vapor pressure of the desiccant. (Similar to how ammoniac is absorbed in the absorber of the diffusion absorption cycle section 2.2.3.2, and opposite to the situation in the open generator section 2.2.3.3)

As in the heat pumps both solid and liquid desiccants can be used.

2.2.5.1 Solid desiccant dehumidification

The material most often used as solid desiccants are silica gel, lithium chloride, lithium bromide, zeolites (aluminum silicate, molecular sieves), various metal oxides or combination thereof (Lindholm 2000). Of those silica gel and zeolites seem to be most commonly used. Salts like lithium chloride, and lithium bromide have the disadvantage when they get saturated with water they turn from solid phase to liquid, thus there is a risk of carryover of these highly corrosive substances.

A common way of utilizing solid desiccants is in a desiccant wheel illustrate in figure 2.52. The wheel consists of a honeycomb structure coated with desiccant. The air to be dehumidified flows through the cells on the process side of the wheel. As the moist air comes into contact with the desiccant, water vapor is ab/ad-sorbed by the desiccant. When the water vapor is ab/ad-sorbed it releases its heat of evaporation and its heat of absorption and since there is no internal cooling of the system the air temperature increases. This process is illustrated in figure 2.53 and is approximately the opposite of the evaporative cooling process (see section 2.2.4). Simultaneously a heated airflow, the regeneration flow, flows through the regeneration side of the wheel. This airflow heats up the desiccant and as a result the vapor pressure of the desiccant becomes higher than the partial pressure of water in the regeneration airflow. Thus water vapor evaporates from the desiccant and is taken up by the regeneration airflow. By slowly rotating the wheel a constant shift of saturated cells from the process side to the regeneration side and of regenerated cells from the regeneration side to the process side is achieved. The function of the purge sector is to cool down

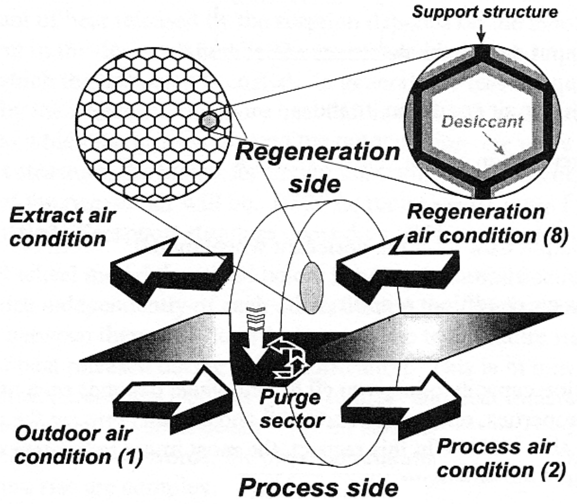


Fig. 2.52 Schematic diagram of a desiccant wheel (Lindholm 2000[©27])

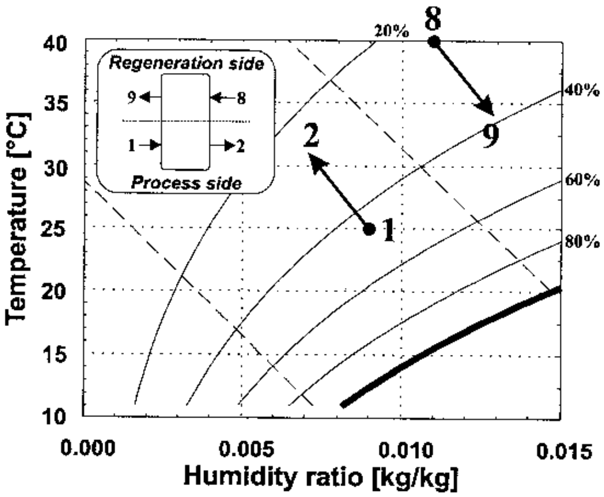


Fig. 2.53 Desiccant wheel dehumidification process in a Mollier chart (Lindholm 2000[©27])

the cells before entering the process flow. This is needed because dehumidification cannot take place if the desiccant is to hot (because the desiccant will the have to high vapor pressure).

Regeneration temperatures vary depending on use, equipment design and the local climate. Required regeneration temperature according to Kessinger (1998) is 80-100 ° C and according to Edward (1999) 74-121 °C. However, in a study for British climate done by Halliday (1999) the regeneration temperatures were as low as 55-65 °C.

An alternative to using the desiccant wheel is to use static desiccant beds through which the air passes. At least two beds, which are switched between process and regeneration mode in the same way as in the two-chamber solid sorption heat pump is section 2.2.3.4, are needed. A setup like this is illustrated in figure 2.54. An advantage of using desiccant beds could be that they could be internally cooled in the same way as done in the solid sorption heat pump (not done in fig 2.54).

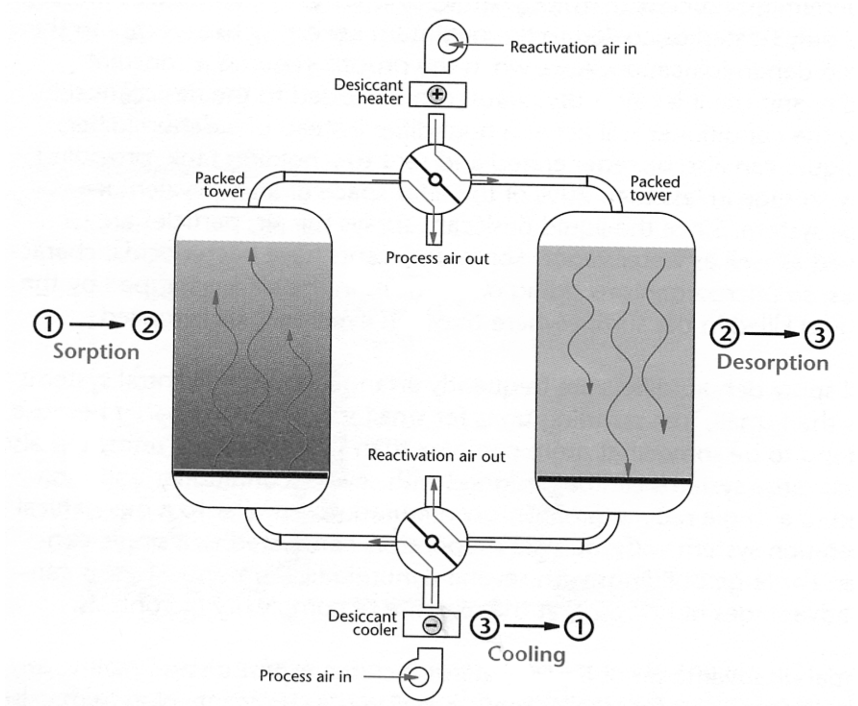


Fig. 2.54 Schematic diagram of a desiccant bed dehumidification system (Harriman 1990[©28]*)*

2.2.5.2 Liquid desiccant dehumidification

Liquid desiccants commonly used are lithium chloride, lithium bromide, calcium chloride, mixtures of these solutions and triethylene glycol (Goswami 1999). Lithium chloride is cheaper and less corrosive than lithium bromide. Glycol has the advantage of not being corrosive but it has a rather high vapor pressure making carryover of desiccant to the process and the regeneration air flows a serious problem, both from pollution and economic viewpoint.

The dehumidification takes place in a chamber similar to the open generator described in section 2.2.3.3. An airflow of ambient air is forced through counterflowing solution from the bottom towards the top. The solution is cooled to reduce its vapor pressure, either in a fluid heat exchanger before entering the chamber, by cooling coils inside the chamber (ASHRAE 2000), or evaporatively with wetted air channels passing through the chamber (similar to a indirect evaporative cooler) (Lowenstein 1998). As the solution meets the air, water vapor is absorbed by the solution.

As with the open generator, several ways of bringing the solution in contact with the air can be used. A rather common method is to spray the solution over a packing (as in fig 2.55). Alternatively it can be sprayed onto wet surfaces, or let onto wet surfaces using overflowing channels above the surfaces (Forkosh 2002, Kessling 1998) or by wicks (Lowenstein 1998). The solution then flows down the wet surfaces as a falling film. Using a falling film reduces the risk of carryover of sorbent to the process air. Carryover is an even more serious problem here than in the open generator as the process air will enter the air-conditioned space. If spray injection is used a mist eliminator (filter) is needed.

As in the solid desiccant system heat is released when the water vapor is absorbed. However, in a liquid system this heat can by carried away both by the air and by the liquid desiccant. Because the liquid has a much higher heat capacity than air it is often the primary source of cooling in a liquid desiccant system. The temperature of the process airflow leaving the dehumidifier can be lower compared to a desiccant wheel since the heat is removed primary by the liquid desiccant and not by the process airflow. The airflow leaving the dehumidifier can even be colder than the

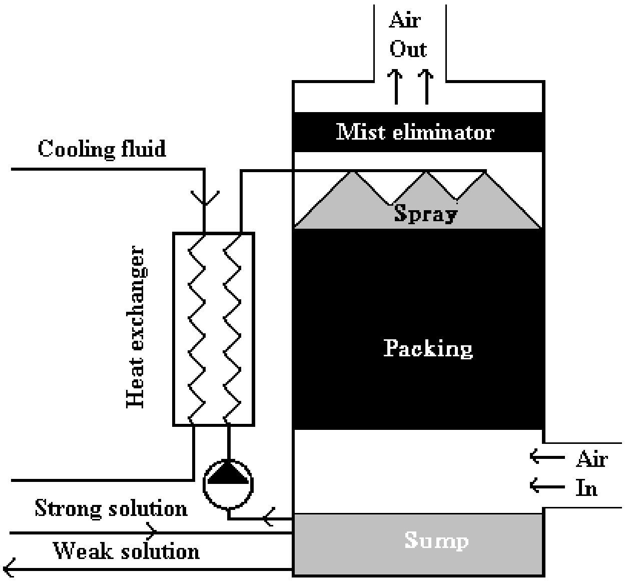


Fig. 2.55 Schematic diagram of a liquid desiccant dehumidifier

airflow entering, because the (chilled) liquid desiccant, in addition to removing the heat of evaporation and heat of absorption, also can remove some of the process air's sensible heat.

In the dehumidifier shown in fig. 2.55 solution from a sump in the bottom of the chamber is cooled externally and recirculated into the chamber. The circulated solution is sprayed over the packing and flows from the top of the chamber towards the bottom. The desiccant is strongest and coldest in the top of the chamber. As it flows downwards and absorbs water vapor it gets weaker and, if no internal cooling is done, warmer (because it has taken up heat from the dehumidification process). When the concentration decreases and its temperature increases the vapor pressure increases. Since the dehumidification is dependent on a pressure difference between the desiccant's vapor pressure and the partial pressure of water in the air, there is a limit of how high the vapor pressure can be. This sets the limit for how low the flow rate of desiccant in the dehumidification chamber can be. Too low flow rate will result in that the desiccant will be too weak and too warm towards the bottom of the chamber.

If the chamber is internally cooled the increase of the vapor pressure is less since the temperature of the solution does not increase, therefore the solution can be weaker in the lower part of the chamber.

A greater desiccant concentration difference between the inlet and the bottom of the chamber can be held and the flow rate of desiccant can be less. A low flow rate is according to Lowenstein (1998) desirable since it reduces the size and cost of the equipment, allows for increased air velocity and reduces the pump sizes.

An other reason for desiring a large concentration difference is to reduce the circulation factor (Lowenstein 1998), which affects the efficiency of the system in a similar way as in an absorption heated pump (see section 2.2.3.1). A large concentration difference and a low flow rate also enable an efficient storage system where concentrated and diluted desiccant is stored in different tanks. Thereby, dehumidification can be performed even when regeneration of the desiccant cannot take place (Kessling 1998). This ability for storage could be a major advantage for the liquid desiccant technology over the solid desiccant technology.

A problem encountered in internally cooled system is leakage between the absorption and the cooling side.

Figure 2.56 illustrates a low flow rate internally cooled falling film dehumidifier.

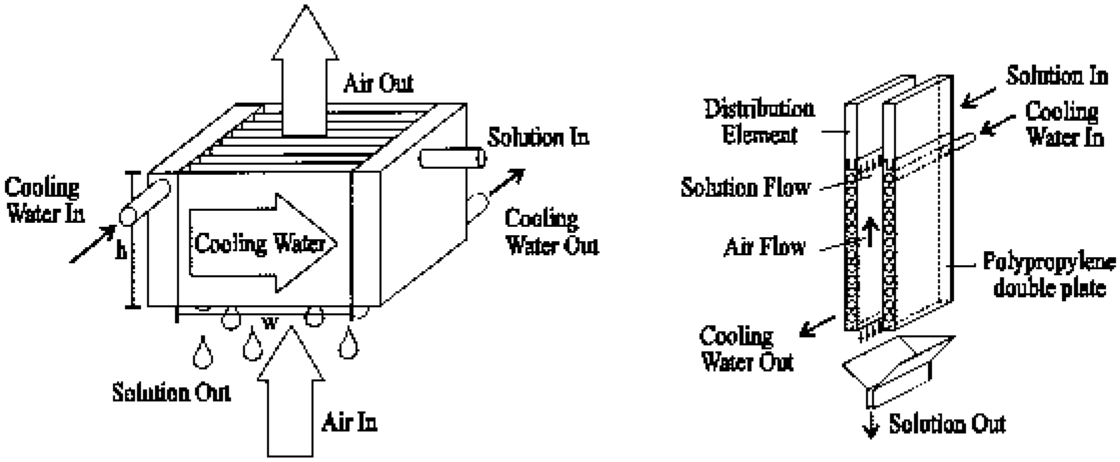


Fig. 2.56 Schematic diagram of a liquid desiccant, low flow rate, internally cooled, falling film dehumidifier (Kessling 1998©29)

The desiccant can be regenerated either in a closed generator (single or multi stage) such as those used in absorption heat pumps (section 2.2.3.1) or in an open generator (section 2.2.3.3). Liquid desiccant systems with open generators generally require lower regeneration temperatures than solid desiccant systems. Required regeneration temperature is according to Kessinger (1998) 50-80 °C, but it can be considerably lower. Lowenstein (1998) shows in an example that if ground water with a temperature of 14 °C is used to cool a liquid desiccant system using lithium chloride (16 %) then the heat from ambient air is enough to regenerate the desiccant when the outdoor conditions are 26 °C 50 % Rh.

Additional advantages for the liquid desiccant technology using salts is that it can kill bacteria and microorganism in the supply air, and that the dehumidifier easily can be transformed in to a humidifier by simply injecting sweet water instead of desiccant solution when humidification is needed instead of dehumidification (Harriman 1990).

The main disadvantage for the liquid desiccant technology is the corrosiveness of the desiccants.

2.3 Solar energy collection

Solar energy collection, as the name suggest, is to collect energy provided from the sun and to transform this energy into something useable, electricity, heat or both.

The energy emitted from the sun, mainly in the form of visible light and heat, can be modeled either as a electromagnetic wave or as a flow of tiny particles called photons. This is called the double nature of light. In the wave model the radiation is said to have an amplitude and a frequency. In the particle model it would be more appropriate to speak about the amount of photons (corresponding to the amplitude) and the energy contents of each photon (corresponding to the frequency). However, for simplicity, photons will in this section be said to have a frequency.

The solar radiation that falls from the sun towards the atmosphere of the earth is called *extraterrestrial* solar radiation. As the extraterrestrial solar radiation passes through the atmosphere, part of it is reflected back into space, a part is absorbed by air, water vapor, aerosols and dust particles (fig. 2.57). The part of the solar radiation that reaches the surface of the earth with essentially no change in direction is called *direct* or *beam radiation.* The scattered diffuse radiation reaching the surface from the sky is called the *sky diffuse radiation.* Although extraterrestrial radiation can be predicted with certainty, radiation levels on the earth are subject to considerable uncertainty resulting from local climatic interaction (clouds, rain, snow, dust storms etc. blocking the direct radiation). The best way to predict radiation levels on the earth is through statistical methods using data collected during many years (30 or more) (Goswami 1999).

The efficiency of a solar collector is defined as:

$$\eta_c = \frac{\mathbf{E}}{\mathbf{I_c A_c}} \quad \text{or} \quad \eta_c = \frac{\int \mathbf{E\,dt}}{\int \mathbf{I_c A_c\,dt}} \qquad (\mathbf{e9})$$

$\mathbf{E}$ *collected energy* [W]
$\mathbf{I_c}$ *solar radiation* [W/m^2]
$\mathbf{A_c}$ *solar collector aperture area* [m^2]

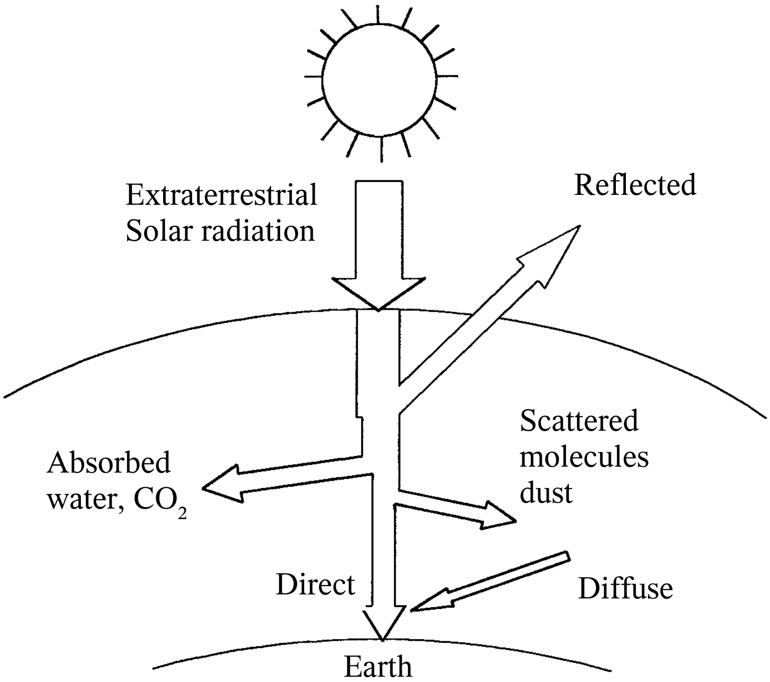


Fig. 2.57Attenuation of solar radiation as it passes through the atmosphere (Goswami 1999$^{©30}$*)*

2.3.1 PV. (Photovoltaic) cells

In a Photovoltaic cell or a solar cell, solar radiation is transformed directly into electricity.

The solar cell consists of a disc or surface with two thin layer of differently doped semiconductor material, often silicon, forming a junction in between (fig. 2.58). Metal stripes runs along the front of the surface and along the back is a metal plate. When solar radiation hits the top of the upper layer, the disc is polarized. The upper layer becomes negatively charged and the lower layer becomes positively charged (Andrén 2001). If the metal stripes and plate are connected in a closed circuit, an electrical current will flow through the circuit. Thus electrical power is accessible.

The voltage obtained from a single disc is rather low, in the range of 0.5 V. To obtain higher voltage, several discs are connected in series. To increase the current rows of serially connected solar cells can be connected in parallel. Thus solar cell panels, also called modules, are constructed. The cells are encapsulated in a transparent material (often plastic and low-iron glass) to protect them from the environment (but not to heat insulate them). Several solar cell panels can be combined into a solar cell array. This is illustrated in figure 2.59. Commonly the output voltage from solar cell panels seems to be in the range of 12-24 V.

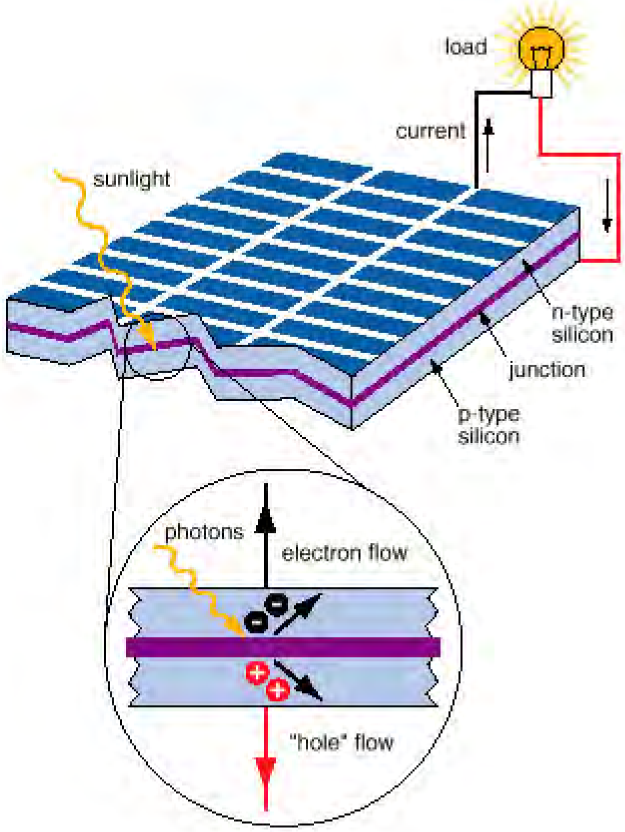


Fig. 2.58 Principles of a silicon photovoltaic cell (Carr 1999©31)

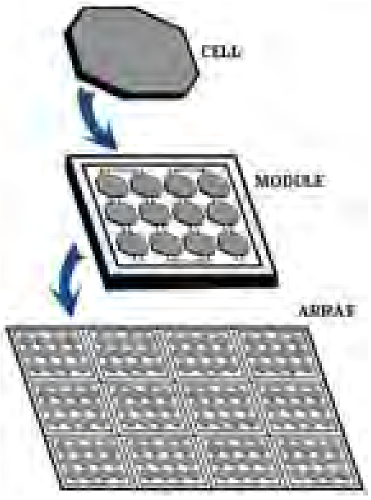


Fig. 2.59 Solar cell, module and array (DayStar 2002)

Trackers (fig 2.60) are used to keep PV.-panels directly facing the sun, thereby increasing the output from the panels. Trackers can nearly double the output of an array (see figure 2.61). Careful analysis is required to determine whether the increased cost and mechanical complexity of using a tracker is cost effective in particular circumstances (Carr 1999).

Fig. 2.60 Tracked PV. Array containing 16 panels (Carr 1999©31).

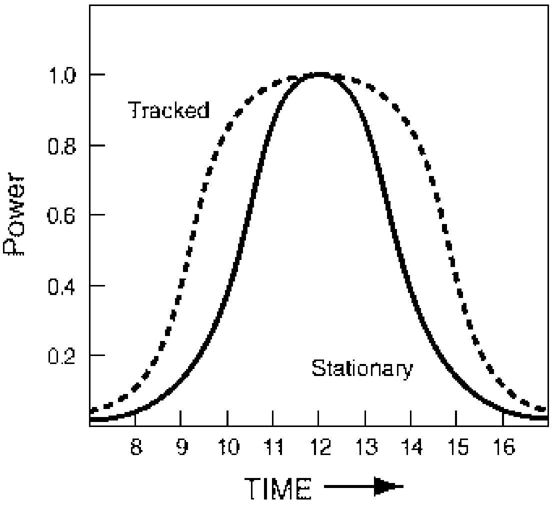


Fig. 2.61 Graph showing power output for a tracked and a non tracked array (Carr 1999©31).

To increase the amount of radiation that falls on each cell, concentrators in form of reflectors or focusing lenses can be used. Depending on their construction they can be more or less concentrating. A commonly used division is to divide between imaging and non-imaging concentrators. Imaging concentrators have a sharp focus point and thus create an "image" of the solar radiation that hits the larger reflector or lens area onto the smaller solar cell area (fig 2.62a). Non-imaging concentrators create a more diffuse "image" of the solar radiation (fig 2.62b). The imaging concentrators are more effective in concentrating the direct solar radiation than the non-imaging. However, because of their strong focus they need to be constantly directed towards the sun, thus tracking mechanisms are required. A system with non-imaging concentrators can also collect diffuse radiation something a system with imaging concentrators cannot do (Carr 1999). The reason for using concentrators is that the material used is much cheaper per m^2 than solar cells. This price gain is however reduced a bit by the fact that more expensive solar cells might be needed to handle the more intense radiation, and in addition to that might come the cost for a tracking and a cooling system.

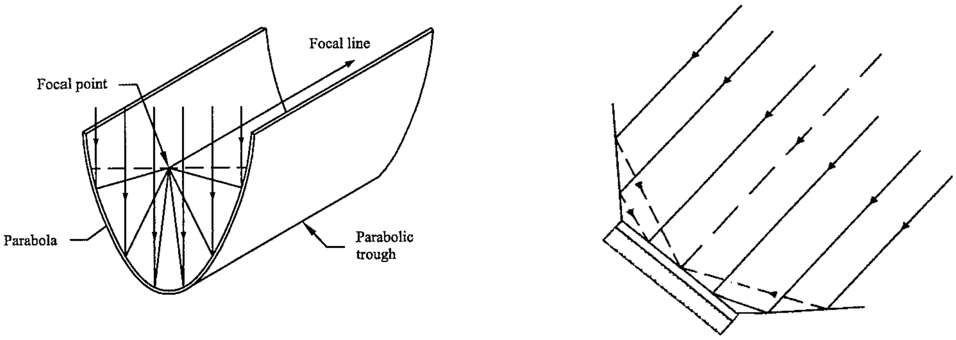


Fig. 2.62 Principles of concentrating collectors(Goswami 1999©30) .
(a) imaging concentration *(b) non imaging concentration*

Understanding the mechanisms for the electrical polarization requires an extensive explanation of the basic principles of semiconducting materials. Goswami (1999) and Tiwari (2003) as well as different Internet sites give thorough explanations of the fundamentals of the conversion principles. Even though these detailed explanations are outside the scope of this report; two things that are connected to these basic principles are of special importance and should be mentioned.

The first is that only the part of light that has a certain frequency (color) can be transformed effectively into electricity. This is because every photon of light can only affect one electron in the solar cell. Photons that have too low "frequency" will not move any electrons, but only cause heat generation. Photons that have too high "frequency" will cause electrons to move, but they will also cause excess heat generation. Because of these two effects the majority of the energy emitted from the solar cell is heat and not electricity.

The second is that the efficiency of the solar cell is connected to its temperature. The warmer the cell becomes the less efficient it becomes. Therefore it is essential that the generated heat is removed from the solar cell.

Most often the heat is rejected directly from the cell to the ambient air, primary through the backplane of the solar cell, but in some application water-cooling of the backside of the cell is used. The reasons to use water-cooling is either that more efficient cooling is needed (for example if the ambient temperature is high or concentrators are used) or that heat from the solar cell is desired for some application. In the first case the solar cell will have a lower temperature than if air-cooled. In the second case it might have to have a higher temperature. The lower electrical efficiency of the cell can then be motivated by the fact that the cell also generates useable heat. In those cases the solar cells might even be capsulated in an insulated casing with an insulating glass cover similar to flat hot-water collectors (section 2.3.2.2). There are also systems where air heated by heat rejected from solar cells is used either for space heating or processing.

Electrical efficiency of a cell using only silicon (which is most common today) is up to 15 % (Andrén 2001). To increase the efficiency there is much research focused on the possibilities to use several materials in a single cell, which are sensitive to different frequencies, thus overlapping each other. The hope is to reach up to 50% electrical efficiency (Claesson 2003).

An option in large-scale applications using heliostat fields and central receivers (see below) is to filter out the frequencies that fit the solar cells and send only this part to the solar cells. The rest is sent to high temperature (thermal) collectors (Segal 2003).

2.3.2 Thermal collectors

Converting the sun's radiant energy into heat is the most common and well-developed solar conversion technology today.

The basic principle of solar thermal collection is that when solar radiation strikes a surface, part of it is absorbed, thereby increasing the temperature of the surface. The efficiency of that surface as a solar collector depends not only on the absorption efficiency (how much of the incoming solar radiation that is absorbed and transformed to heat) but also on how the thermal and reradiation losses to the surrounding are minimized and how the heat from the collector is removed for useful purposes. Various solar thermal collectors range from unglazed flat plate-type collectors operating at about 5-10°C above the ambient temperature, to central receiver concentrating collectors operating at above 1000°C. Table 2.2 lists various types of solar thermal collectors and their typical temperature and concentration ranges (Goswami 1999).

Table 2.2 Various types of solar thermal collectors (Goswami 1999[©30])

Type of Collector	Concentration Ration	Typical Working Temperature Range (°C)
Flat plate collectors	1	≤70
High efficient flat plate collectors	1	60-120
Fixed concentrators (non imaging)	3-5	100-150
Parabolic trough collectors	10-50	150-350
Parabolic dish collectors	200-500	250-700
Central receivers	500->3000	500-<1000

A division can be made between high temperature collectors with a temperature range above 150 °C, medium temperature collectors with a temperature range of 30-150 °C and low temperature collectors with a temperature range below 30°C.

Two types of special surfaces of great importance in solar thermal collection systems are *selective* and *reflecting surfaces.*

Selective surfaces combine a high absorptance for solar radiation with a low emittance for the required temperature range (Goswami 1999, Tiwari 2003). This combination is possible because 98 percent of the energy in incoming solar radiation is contained within wavelengths below 3 μm (radiation with frequencies above 333 kHz), whereas 99 percent of the radiation emitted by a black or gray surface at 400 K (127 °C) is at wavelengths longer than 3 μm (Goswami 1999). Examples of coatings with good selective properties used for solar thermal collectors are Black nickel on nickel, Black chrome on nickel, Black chrome, Black copper and Iron oxide. However, any black or dark surface works to some degree as a selective surface.

Highly reflective surfaces are needed for concentrating solar collectors. Reflecting surfaces are usually highly polished metals or metal coatings on suitable subtracts.

Heat losses from a solar collector primary occur because of *convection* and *radiation*.

Convection is when the heat is carried away with a moving fluid (gas or liquid). When the fluid motion is created by the temperature difference between the fluid and the hot surface the convection is called *free convection.* Another source of convection is wind.

Heat losses through radiation (emittance of electromagnetic waves) occur when a hot surface (for example the absorber of a solar collector) faces a colder surface or body. For practical purposes the sky can be approximated as a surface with an equivalent surface temperature $\mathbf{T_{sky}}$. Different suggestion for the formula of $\mathbf{T_{sky}}$ exists (Tiwari 2003).

To prevent losses from the collector, insulation and glazing is used. Insulation materials such as mineral wool can naturally only be used where solar radiation does not have to pass through (back, sides and pipes). Where the radiation needs to pass through to reach the absorber glazing is used. The role of the glazing is to admit the maximum possible radiation and to minimize the loss of heat. The most commonly used glazing material is glass as it can transmit 90% of the incident short wave radiation while it transmittance to long wave heat radiation (5.0 to 50 µm), emitted by the absorber, is negligible. Plastic films and sheets may also be used for this as they possesses high transmittance to short wave solar radiation, but transmittance in the middle of the thermal radiation spectrum and dimensional changes in the temperature range restricts their use as a good glazing surface (Tiwar 2003). Plastic can also deteriorate over years because of ultraviolet solar radiation.

The collected heat from a solar thermal collector can be modeled using a simple energy balance as (Goswami 1999):

$$\mathbf{q_{cT}} = \eta_0 \, \mathbf{I_c} \mathbf{A_c} - \mathbf{U_L}(\mathbf{t_c} - \mathbf{t_a}) \mathbf{A_r} \quad (\mathbf{e10})$$

η_0 *Optical efficency*

$\mathbf{I_c}$ *Solar radiation* $[\mathrm{W/m^2}]$

$\mathbf{A_c}$ *Solar collector aperture area* $[\mathrm{m^2}]$

$\mathbf{U_L}$ *Overall heat loss cofficient* $[\mathrm{W/(m^2 \cdot {}^oC)}]$

$\mathbf{t_c}$ *Collector (mean) temperature*

$\mathbf{t_a}$ *Atmospheric/ambient temperature* $[{}^o\mathrm{C}]$

$\mathbf{A_r}$ *Reciver area* $[\mathrm{m^2}]$

For a collector without concentrators A_r=A_c. Thus, using e10, the efficiency of the collector can be written as:

$$\eta_{\mathbf{cT}} = \eta_0 - \mathbf{U_L} \frac{(\mathbf{t_c} - \mathbf{t_a})}{\mathbf{I_c}} \quad (\mathbf{e11})$$

The efficiency can be estimated using the *Hottel-Wiss-Bliss* equation :

$$\eta_{cT}=F_R\tau\alpha-F_R U_L\frac{(t_i-t_a)}{I_c} \quad (e12)$$

F_R *collector heat removal factor*

τ *transmissivity, fraction of incoming solar radiation that reaches the absorbing surface*

α *absorptivity, fraction of solar energy reaching the surface that is absorbed*

t_i *temperature of fluid entering the collector* $[^\circ C]$

The collector heat removal factor compensates for the temperature difference between the heat transfer fluid and the absorber as well as for the temperature difference between the heat transfer fluid entering and leaving the collector. Thereby the inlet temperature t_i can be used instead of the collector temperature t_c.

Brunold (1994) writes the equation e12 slightly different as:

$$\eta_{cT}=F'(\tau\alpha)_e-F'U_L\frac{(t_m-t_a)}{I_c} \quad (e13)$$

F' *collector efficiency factor*

$(\tau\alpha)_e$ *effective transmissivity−absorbsorptivity product*

t_m *mean temperature of the heat transfer fluid* $[^\circ C]$

The main difference between equation e12 and e13 is that the inlet temperature of the fluid t_i is replaced by its mean temperature $t_m=(t_i-t_{out})/2$. As a result F_R is replaced by F' which compensates only for the temperature difference between the heat transfer fluid and the absorber. Another difference is that $\tau\alpha$ is replaced by $(\tau\alpha)_e$ (e=effective) which is usually 1 to 2 percent greater than the product $\tau\cdot\alpha$ (Tiwari 2003).

Brunold further states that in reality the heat loss coefficient U_L is not a constant but a function of the temperature of the absorber plate and the ambient temperature.

He therefore uses the following approach:

$$F'U_L=c_1+c_2(T_m-T_a)$$

c1 & c2 *constants*

The efficiency then becomes:

$$\eta_{cT}=F'(\tau\alpha)_e-c_1\frac{(t_m-t_a)}{I_c}-c_2\frac{(t_m-t_a)^2}{I_c} \quad (e14)$$

or with $c_0=F'(\tau\alpha)_e$ and $x=(t_m-t_a)/I_c$

$$\eta_{cT}=c_0-c_1x-c_2 I_c x^2 \quad (e15)$$

Note that e14 equals e13 when $c_2=0$. Thus e15 can be used to represent both e13 and e14.

Equation e11-e13 applies for non-concentrating collectors where $A_r=A_c$. When concentrators are used the receiver area of the collector through which heat is lost is smaller than the apparatus area of the collector over which solar radiation is collected. The efficiency e11 thus becomes (Goswami 1999):

$$\eta_{cT}=\eta_0-U_L\frac{(t_c-t_a)}{I_c}\times\frac{A_r}{A_c}=\eta_0-U_L\frac{(t_c-t_a)}{I_c}\times\frac{1}{RC}\quad(\mathbf{e16})$$

$$\mathbf{RC}=\mathit{Consentration\,ratio}$$

Equation e12 and e13 could thus be rewritten as:

$$\eta_{cT}=F_R\tau\alpha-F_R U_L\frac{(t_i-t_a)}{I_c}\times\frac{1}{RC}\quad(\mathbf{e17})$$

$$\eta_{cT}=F'(\tau\alpha)_e-F'U_L\frac{(t_m-t_a)}{I_c}\times\frac{1}{RC}\quad(\mathbf{e18})$$

Equations e14 and e15 need not to be rewritten since 1/RC can be included in c_1 and c_2. (Equations e12 and e13 can be seen as special cases of e17 and e18 in which RC=1.)

Both the simplified models, represented by e12, e13, e17 and e18, and the more advanced model represented by e14 are used for estimation of the collector efficiency. Whether the characteristics of a specific collector in an article, a test report or manufacture data sheet is defined with two constant (c_0,c_1) or three constants (c_0,c_1 , c_2) varies.

The model represented by e14 could be said to be the better model since it better models the temperature effect on the radiative losses. In e12, e13, e17 and e18 the radiative losses are treated together with the convectional as proportional to the temperature difference. The radiative losses are, however, exponentially related to the temperature difference rather than proportional. In most practical cases results obtained from e12,e13, e17, e18 and e14 seem anyway to be rather similar. Differences can, as illustrated in fig 2.63 (which illustrates the difference in result from the two models), be seen at high temperatures when radiation losses increase resulting in a faster decrease of efficiency than indicated by e12 and e13. At low temperature differences the opposite occurs.

The temperature at which the efficiency declines to zero (and thereby no energy is delivered from the collector) is called the *stagnation temperature*. The reason why the efficiency becomes zero is that the thermal losses from the collector are equal to the collected solar energy. The situation when the temperature of the collector reaches the collectors stagnation temperature is called *stagnation.* Not all collectors can reach stagnation without damages due to overheating of parts of the system. If so, there must be some kind of safety mechanism preventing stagnation. In figure 2.63 the stagnation temperature is where the efficiency curve intersects the vertical axis.

When constructing arrays or fields of solar collectors/reflectors there are often restrictions on how close to each other the collector/reflectors can be without extensively shadowing each other. In such cases it might be of interest to base the efficiency on the total needed gross area $\mathbf{A_g}$ rather than the collector apparatus area $\mathbf{A_c}$. This efficiency can be denoted $\boldsymbol{\eta}_{cTg}$. Naturally the efficiency based on the gross area is less than the efficiency based on the collector apparatus area. This is because the former also includes losses related to radiation that falls between the collectors, not collected, while the later only includes losses from the collector itself.

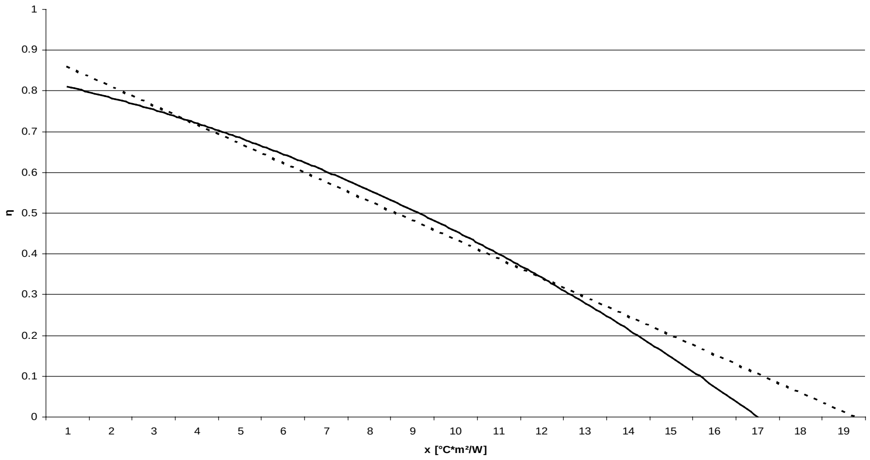


Fig. 2.63 Typical efficiency plot (for a flat CPC-collector) resulting from using e12, e13, e17 or e18 (dashed line) and e14 (solid line)

An other aspect to be taken into consideration is what the solar radiation variable $\mathbf{I_c}$ represents. As explained before there is a difference in the ability to collect diffuse radiation between concentrating (and in particular imaging concentrating) solar collectors and non-concentrating collectors. Highly concentrating collectors are poor in collecting diffuse radiation and therefore a higher amount of diffuse radiation can hardly improve the efficiency of the collector. Then you have to say that for highly concentrating solar collectors $\mathbf{I_c}$ represents the amount of direct radiation. When it comes to collectors that can collect diffuse radiation, then the efficiency improves with higher amounts of both diffuse and direct radiation and thus $\mathbf{I_c}$ represents the total amount of solar radiation. Because of this the efficiency values obtained from above equations for concentrating collectors is not fully comparable with the efficiency values obtained for non-concentrating collector.

The above efficiencies apply when the sun is shining straight into the collector. This is however seldom the case if the collector not actively tracks the sun along two axes. When the sun radiation comes from a angel rather than straight into the collector the efficiency of the collector usually (but not always) decreases. The ratio between actual efficiency and the efficiency if the sun would be shining straight into the collector is called the *incident angle modifier*.

Many collectors (for ex. evacuated tubes) are optically unsymmetric. Two separate incident angle modifiers, one referring to the longitudinal axis and one referring to the transverse axis, can then be used (Brunold 1994).

2.3.2.1High temperature collectors

To obtain high temperatures, solar energy must be concentrated to a high level, because at high temperatures, the radiation becomes a major mechanism for thermal losses from the receiver. The radiation losses are proportional to the aperture size of the receiver (Segal 1999).

Only the direct radiation can be effectively concentrated. High temperature collectors are therefore dependent on direct radiation and thereby on clear weather conditions to operate efficiently.

Since the high temperature collectors are highly focusing, continues solar tracking is necessary.

Central receiver systems

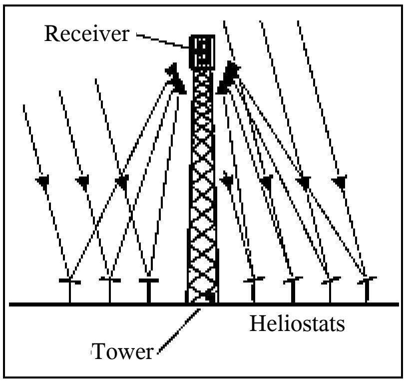


Fig. 2.64 Schematic heliostat field (SolarPACES 2001)

Fig. 2.65 Solar Two (Sun Labs 2003$^{©32}$)

The highest temperatures can, as seen from table 2.2, be obtained from central receiver systems. They consists of a central receiver, which receives solar radiation collected from a large number of reflector/collector units.

The most well known central receiver system is the *heliostat field* system, also called power tower system (PTS). A heliostat field consists of a large number of reflectors with two axis solar tracking. The reflectors are positioned individually so that each reflector reflects received solar radiation onto a central receiver mounted on a tower (fig 2.64). Figure 2.65 shows the Solar One/Two, a 10 MWe grid connected central receiver power plant, built and operated as a test facility in California. The Solar One operated from 1982 to 1988 and and the Solar two from 1996 to 1998 (NREL 2001).

Two possible designs of the receiver is illustrated in fig. 2.66. The *external receiver* (a), used in Solar One, consists of 24 panels with 70 vertical tubes of 12.7 mm diameter. The tubes are made of Incoloy 800 and coated with high absorptance black paint. They are connected at the top and the bottom by welded headers (Goswami 1999).

The *Captivity receiver* (b) directs the concentrated flux inside a small insulated cavity in order to reduce the radiative and convective heat losses. Typical designs have an aperture area about one-third to the internal absorbing surface area (Goswami 1999).

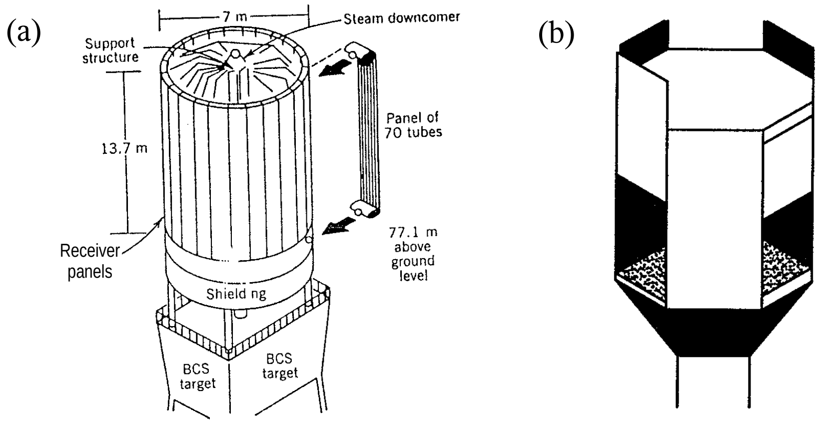


Fig 2.66 Central receivers (a) the external receiver of Solar One (b) a captivity receiver (Goswami 1999$^{\copyright 30}$)

To further increase concentration and thereby reduce losses at high temperature a *receiver concentrator* can be used. This second stage concentrator (usually a CPC-collector (see below)) mounted on the receiver concentrates the radiation received from the heliostats a second time, before it reaches the absorber. According to Segal (1999) a receiver concentrator is energetically justified when the receiver's working temperature is above approximately 730°C.

In a design originally presented by Rabl (1976) the central receiver is moved from the tower down to ground level. A hyperboloidal mirror mounted at the top of a tower reflects the radiation from the heliostats down to the receiver. In this case, the use of the receiver concentrators is absolutely necessary to obtain high concentrations because the hyperboloidal mirror magnifies the sun image (Segal 1999). Figure 2.67 illustrates this kind of system.

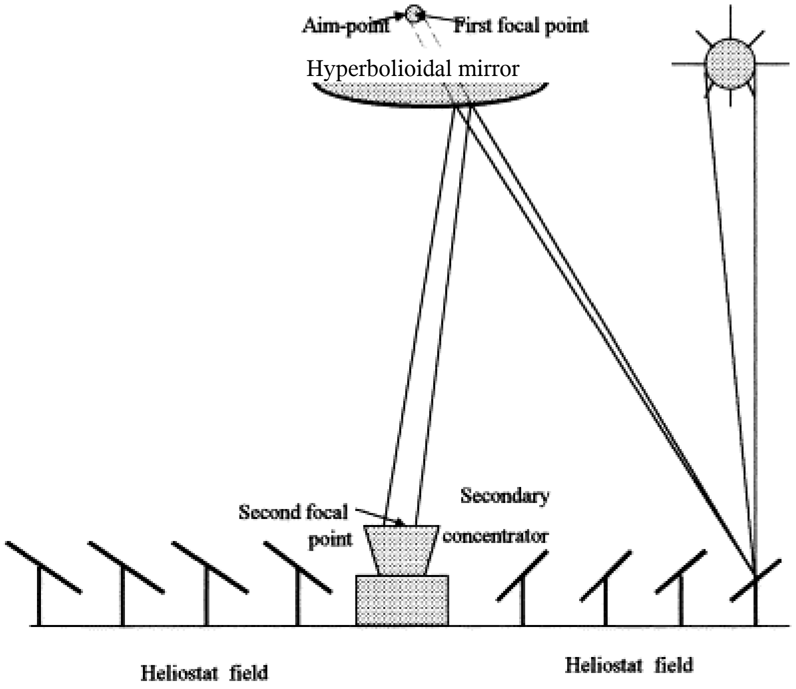


Fig. 2.67 Schematic representation of tower-reflector system (Segal 1999$^{\copyright 33}$)

Feuermann (1999) goes one step further and suggests removing the tower completely. Instead he suggests using a field of collector modules consisting of mini-(paraboloidal)-dishes (fig. 2.68 & 2.69). In the focus of every mini-dish is an optical fiber which receives the solar radiation concentrated from the dish. The fibers from all the dishes of all the collector modules are then brought together to a central receiver as illustrated in figure 2.70.

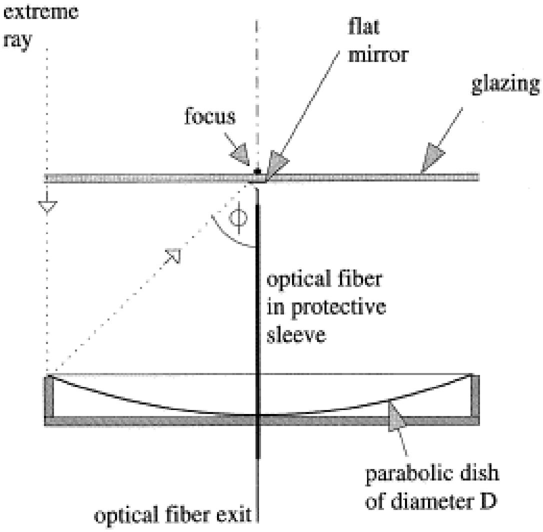


Fig. 2.68 Cross-sectional drawing of a solar fiber-optic mini-dish (Feuermann 1999©34)

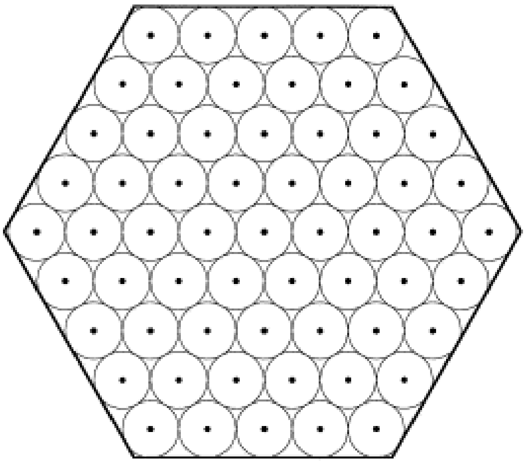


Fig 2.69 Schematic top view of a glazed module of circular mini-dishes. The small dots are the flat mirrors described in Fig. 2.68. (Feuermann 1999©34)

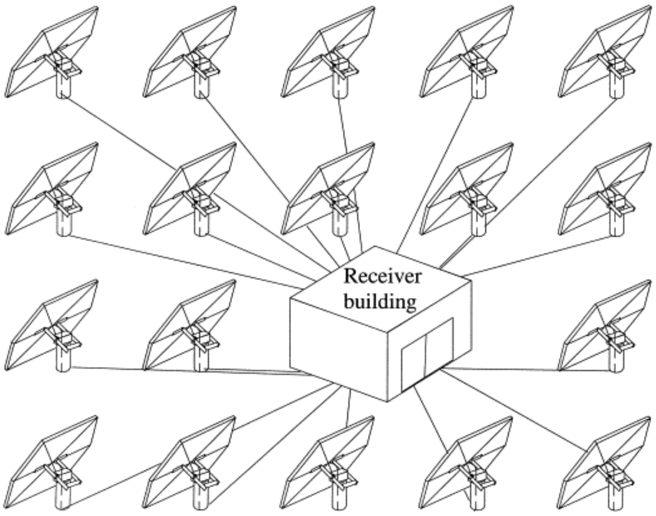


Fig 2.70 Schematic of part of a solar field comprised of many modules, with all optical fibers transporting concentrated sunlight to a remote protected receiver. (Feuermann 1999©34)

In the Solar One plant superheated steam used to drive a Rankine power cycle, was produced directly in the receiver. The plant lacked a good storage system. This meant that it could not generate electricity after sunset and since the highly concentrating system only works with direct radiation this caused some problems with interrupts as clouds passed by. In Solar Two, a development of Solar One, molten salt was used to remove and store heat from the receiver. The hot molten salt was used to produce steam through boilers/heat-exchangers. Yet another alternative is to use air-recivers in which air is used as heat transfer fluid. Air has the advantage that it can be heated to very high temperatures. However, air has rather low heat capacity and large volume and the heat is thus more complicated to transfer. Therefore it might be preferable to have the receiver at ground level, either by using a hyperboloidal mirror or by using optical fibers. The hot air can for example be used to drive a Brayton power cycle (gas-turbine) or a combined cycle (see section 2.2.2.3).

As mentioned is section 2.3.1, part of the spectrum of the collected radiation can be separated and used for photovoltaic collectors.

Parabolic dish collector

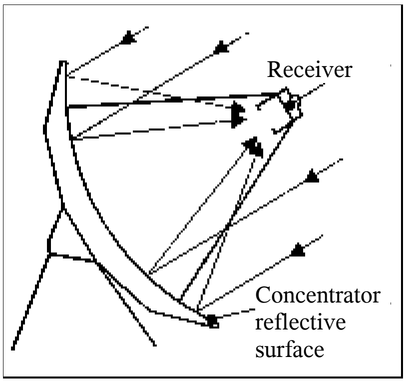


Fig 2.71 Schematic dish collector (SolarPACES 2001)

Fig 2.72 SAIC 20 kW dish Stirling System (Andrak 2003[©32])

A dish collector concentrates solar energy onto a receiver at its focal point (fig 2.71). The receiver absorbs the energy and converts it into thermal energy. This can be used directly as heat or can support chemical processes, but the most common application is power generation. The thermal energy can either be transported to a central generator for conversion, or it can be converted directly into electricity at a local generator coupled to the receiver (SolarPACES 2001).

Like the central receiver system dish collectors track the sun along two axes, and thus are the most efficient collector systems because they are always pointing at the sun. Concentration ratios usually range from 600 to 2000, and they can achieve temperatures in excess of 1 500°C (SolarPACES 2001). While Rankine-cycle engines (section 2.2.2.1), Brayton-cycle engines (section 2.2.2.3) and sodium-heat engines have all been considered for systems using dish-mounted engines, greatest attention has been paid to Stirling-engine systems (section 2.2.2.2). However, competition between Stirling and Brayton engines remains open (SolarPACES 2001). A free piston striling engine with built in receiver for dish collectors is shown in figure 2.73. Figure 2.72 shows a dish collector with a Stirling

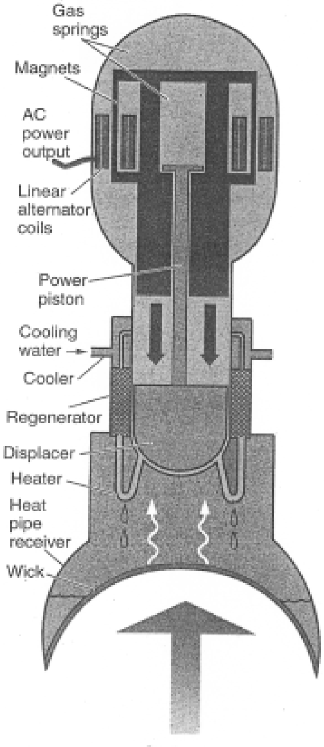


Fig 2.73 Combined receiver and Stirling heat engine (Goswami 1999[©30])

engine receiver.

The focus of current developments in the US and Europe are on 7.5 - 25kWe systems for remote applications, while developments in Australia take a different approach, with maximum activity centered on the demonstration of large dishes (400 m²), which provide steam for a central turbine (SolarPACES 2001).

Parabolic trough collector (PTC)

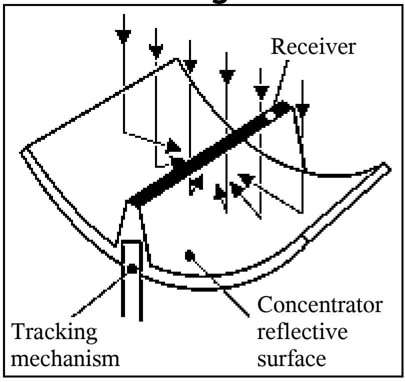


Fig 2.74 Schematic parabolic trough collector (SolarPACES 2001)

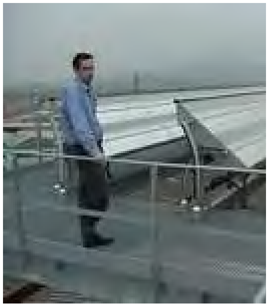


Fig 2.75 Parabolic trough collectors of different sizes. (Above Industrial 300, Right LS2) (Solel©35, Andrak 2003©32)

The parabolic trough collector or parabolic trough concentrator (PTC) is a commercially available high temperature collector. As illustrated by figure2.75, it is available both as large collectors for large scale power production (right image) and as smaller collectors for decentralized power production, process heat supply and hot water production (upper image).

Parabolic trough collectors assembled in collector fields are currently responsible for all commercially produced solar thermal (electrical) power, with a total installed capacity of more than 350 MWe in California (SolarPACES 2001).

Parabolic trough collectors usually track the sun along one axis using one of three orientations: east-west, north-south or polar. The east-west and north-south configurations are the simplest to assemble into large arrays, but the polar configuration is more effective (Goswami 1999).

The receiver of a parabolic trough collector normally consists of a pipe coated with a selective surface. Around the pipe there is a glass pipe to reduce convective and infrared radiative heat losses. To further reduce the heat losses the gap between the inner pipe and the outer glass pipe can be evacuated. Then no connective heat transfer can take place from the absorber surface to the glass. Fig. 2.76 shows the evacuated receiver of the LS2 collector (right image fig. 2.75).

Fig 2.76Evacuated receiver for the LS2 and the LS3 parabolic trough collectors (Solel 2003©35)

In large power production systems, using a synthetic oil as heat transfer fluid, temperature up to 400 °C can be achieved (Solel 2003). For lower temperature application water, if needed mixed with antifreeze (glycol), can be used (NREL 1998). A problem with water is its high pressure at high temperatures, especially since the collectors need to be able to rotate to track the sun. To enable the collectors to rotate the fluid flows into the collector through a flexible hose, and this seems to be the weak point at high pressure.

There is an ongoing project with the object of producing steam for power production directly in the receiver of the collector. Thereby the steam boiler and the expensive synthetic oil can be removed, and thus both cost reduction and efficiency improvement could be obtained. Some modification (for example to the flexible hose) is needed to handle the increased pressure (Price 1999).

NREL (1998) makes the conclusion that parabolic trough collectors can in some cases be attractive for hot water production. The key factors are size of the installation and availability of direct radiation. According to NREL (1998) they can be much less expensive than flat plate collectors if the system is large enough.

Figure 2.77 shows the (by the manufacture specified) efficiency of the smaller parabolic trough collector in figure 2.75 (upper image) from Solel, one of the two mayor manufactures of parabolic trough collectors (Solel and Industrial Solar Technologies IST).

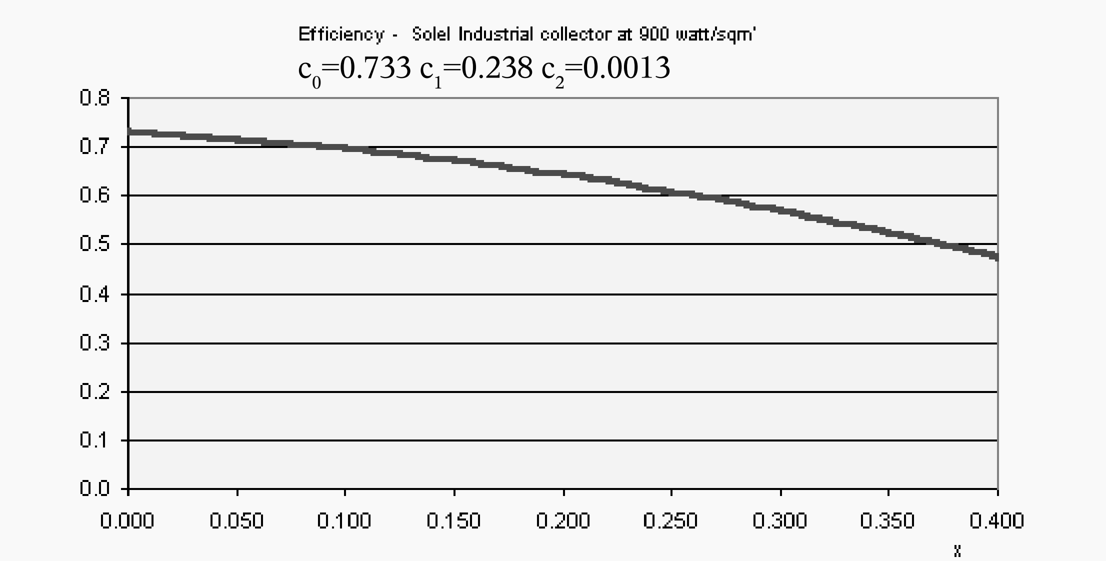


Fig. 2.77 Efficiency of a parabolic trough collector (Solel 2003©35*)*

Since the parabolic trough collector is useable also for rather low temperature application like tap water heating, and since water with antifreeze can be used as heat transfer fluid, it could be argued that it is not a pure high temperature collector. However, it has the characteristics of high temperature collectors like imaging concentration and tracking.

2.3.2.2 Medium-low temperature Hot water collectors

In a hot water collector, water is used as heat transfer fluid. Antifreeze, usually glycol, is often mixed with the water in areas where the temperature can dip below 0 °C. When antifreeze is used, a closed loop is formed which usually supplies heat to a sensible heat storage through a heat exchanger. The reason for not having a single loop with the storage tank included is that the antifreeze is expensive and not very environmentally friendly. Therefore it is desirable to keep the volume of antifreeze at a minimum. If no antifreeze is used the construction has to be such that the water can either be totally drained from all exposed parts, or that it can tolerate freezing.

Hot water collectors are generally non-tracking.

Evacuated tube collector

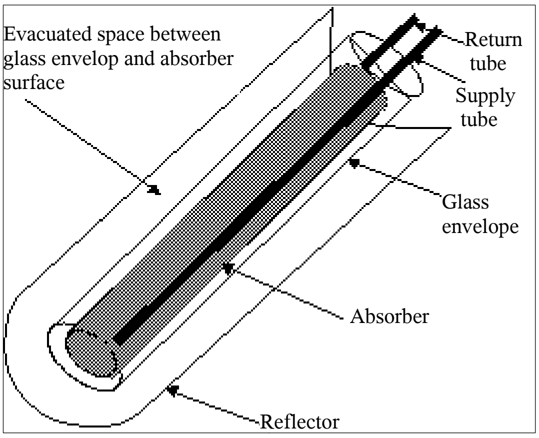


Fig. 2.78 Schematic ("Dewar-China") evacuated tube collector (NREL 1996)

Fig. 2.79 Evacuated tube module with 16 tubes (Oertli 2003©36)

The evacuated tube collector or vacuum tube collector is the most efficient collector in the upper part of the medium temperature range (x > 0.65-0.8). The reason for this is that no convective losses can occur through evacuated volumes in the collector. Since no convection can take place almost all losses in a evacuated collector is trough radiation. The glass envelope maintains the evacuated volume and reduces losses trough radiation. The tubular design makes the construction strong enough to withstand the subpressure within the collector.

The stagnation temperature of an evacuated tube collector can be more than 250 °C (Frei 2003). This can be a problem, as parts of the collector system may not be built to handle such high temperatures and the resulting high pressure.

One possible design of evacuated tube collectors has already been mentioned, the evacuated receiver tube of the parabolic through collector illustrated in figure 2.76. However, this is not the normal construction of a hot water evacuated tube collector.

Figure 2.80 illustrates the typical designs of evacuated tube collectors year 1980 (Frei 2003).

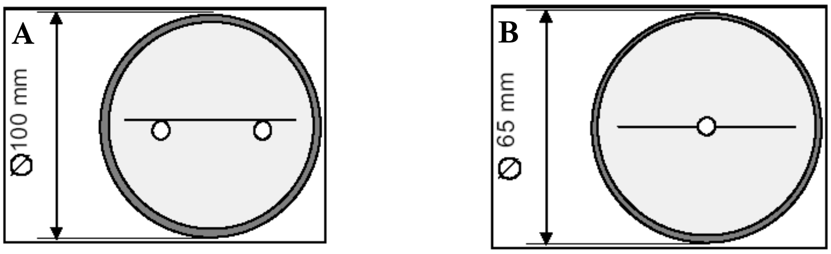


Fig. 2.80 Evacuated tubular collectors around 1980 (Frei 2003©39)

(These designs are still being manufactured). In design A (fig. 2.80) the heat transfer fluid (water) flows directly through a U-shaped loop connected to the metallic absorber inside the evacuated tube. This way of transferring the heat from the collector to the heat transfer fluid is called heat transfer through *direct flow*.

In design B (fig. 2.80) a *heat pipe* system is used. In a heat pipe system the heat transfer fluid does not enter the collector tube. Instead the heat pipe transports heat from the absorber to a heat exchanger in the manifold of the collector. On the other side of the heat exchanger flows the heat transfer fluid. The heat transport inside the heat pipe is primary as latent heat rather than sensible heat.

The heat pipe consists, as schematically illustrated in figure 2.81, of a circular pipe and (sometimes) an annular wick situated adjacent to the pipe wall. The inactive part of the circular pipe (between the evaporator and the condenser) is insulated to avoid thermal losses. A fluid in the evaporator is brought to boiling. As it evaporates and becomes vapor it takes up sensible heat from the absorber that is transformed into latent heat. The vapor migrates to the condenser where its latent heat is retransformed to sensible heat and transferred to the heat transfer media loop. After being condensed and having releasing its latent heat the heat-pipe fluid returns to the evaporator by capillary action in the the wick or by gravity and the cycle repeats. Gravity heat pipes can operate without a wick but cannot be operated horizontally (Tiwari 2003).

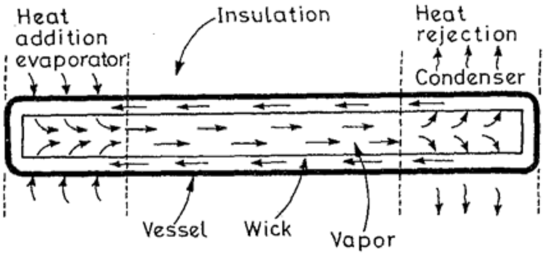


Fig. 2.81 Schematic drawing of a heat pipe (Tiwari 2003©37)

Figure 2.82 shows an example of a evacuated heat pipe collector module construction. Heat is transferred from the absorber to the evaporator, which runs along the entire absorber length, through heat conduction. The construction is often such that single tubes can be rotated or removed from the manifold and exchanged while the installation is in operation. This is called *dry connection.* The heat transfer fluid never enters the collector tuber. The receiver of the combined receiver Stirling heat engine in figure 2.73 also transfers heat through the heat pipe principle.

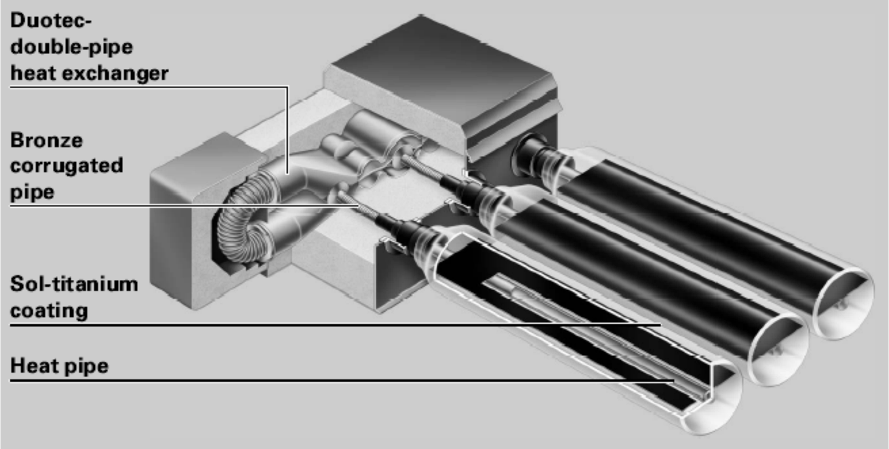


Fig. 2.82 Evacuated heat pipe collector construction (Viessmann 2003©38)

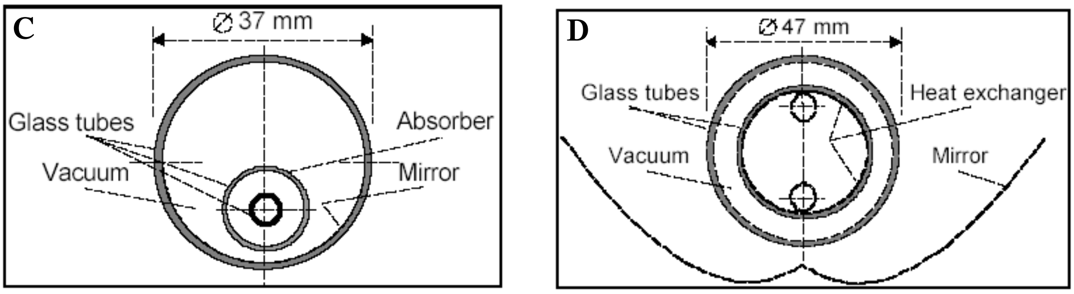


Fig. 2.83 "All-glass-design" evacuated tubular collectors (Frei 2003$^{©39}$*)*

During the 1980:s a memory-metal-based overheat protection system for collectors using the heat pipe system was introduced. It stops the vapor flow to the condenser when a desired maximum temperature is reached.

During then 1990:s new collector designs was introduced. The main feature of these new collectors was the "all-glass design". This means that the technically difficult task of sealing glass to metal (glass envelop-metal pipes) was eliminated, and the risk of vacuum loss reduced (Frei 2003).

Two kinds of "all-glass-design" collectors exist. Design C in figure 2.83, the "Schott"-collector, uses a 2 coaxial internal absorber situated at the focus of a mirror deposited on the inner surface of the outermost glass tube. To transfer heat from the absorber, the heat transfer media circulates through the internal glass tubes. The manifold including the sealing system is technically very demanding (Frei 2003).

The other "all-glass-design", design D (figure 2.83), is the "Dewar-China" or "Sidney"-collector. It was originally developed at the University of Sidney and later produced and commercialized on a large scale in China. It is constructed like a vacuum jug in which a vacuum gap between the outer and the inner glass tube effectively insulates the unevacuated interior. The inner glass tube, coated with a selective surface, is the absorber. Heat collection from the glass absorber is achive with a slip-in heat exchanger. Since this heat transfer is through a glass to metal interface that has only intermittent points of contacts, significant axial temperature gradients can develop, thereby causing stress on the glass tube. In addition, a large temperature difference can exist between the inner and the outer glass tubes. At the collector ends where the tubes are joined, a large temperature gradient and consequently thermal stress can exist (Goswami 1999). In the Dewar-China collector illustrated in figure 2.83 heat is collected through direct flow. Another options is to us a similar heat exchanger but with a heat pipe instead of a U-formed loop.

A third option which is very popular in China, where the Dewar-China collectors make up 65 % of the solar water heater market, is the "water-in-glass"-system illustrated in figure 2.84. The system consists of water-filled collectors with their upper open parts inserted directly into a storage tank. Silicon-rubber-ring-seals are used between the tank and the collectors. Heat extraction from a water-in-glass evacuated tube is achieved by heat-driven natural circulation of the fluid between the collector and the storage tank. The limitation of this concept is that it can only be used for a low-pressure system as the tubes can only withstand a few meters of water head (Budihardjo 2003).

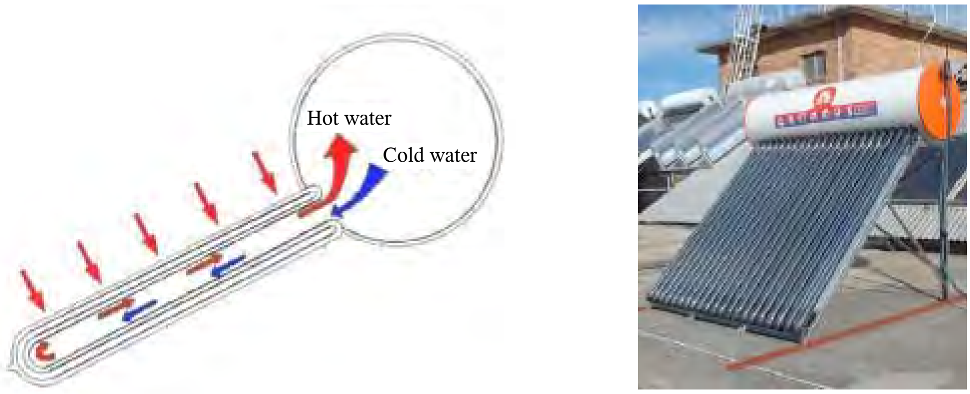


Fig. 2.84 "Water-in-glass" system (Budihardjo 2003©40)

The specially shaped mirror behind collector D in figure 2.83 is called *Compound Parabolic Concentrator* or in short CPC. This is a non-imaging concentrator that is able to concentrated radiation coming from different directions. Thus it does not need tracking like imaging concentrators and it is able to concentrate diffuse radiation. However, it is not as efficient in concentrating direct radiation as the imaging concentrators. The principle of a CPC-collector with tubular receiver is illustrated by figure 2.85

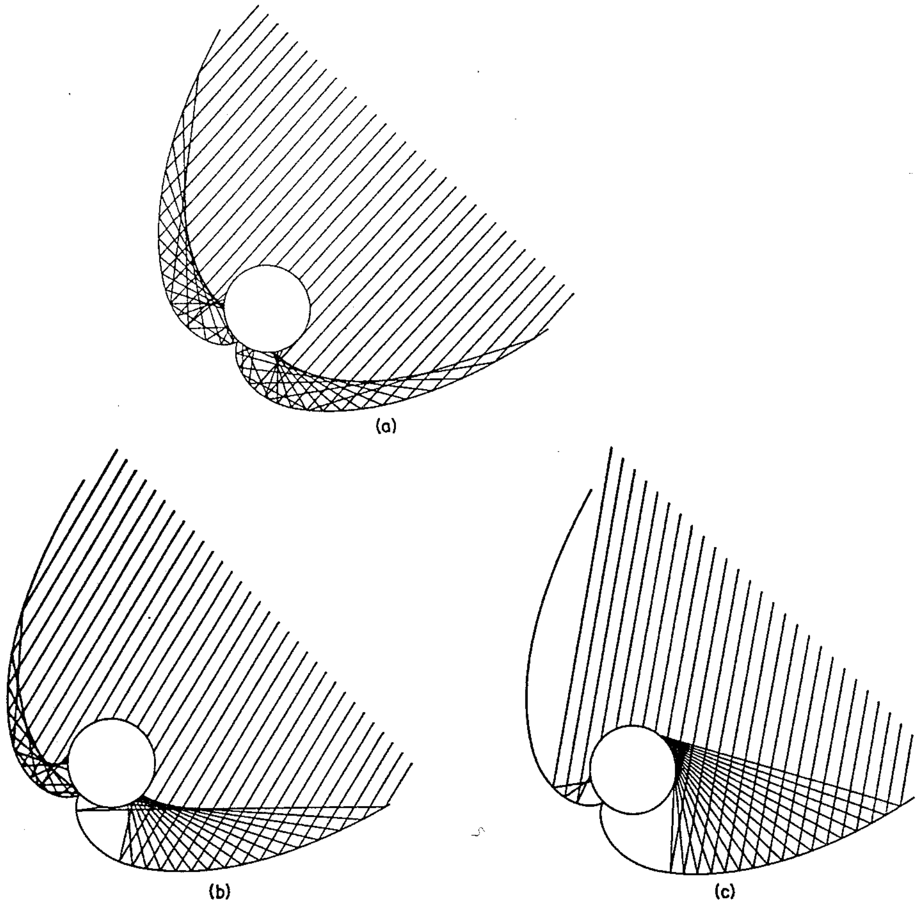


Fig 2.85 Ray trace diagrams of the tubular CPC collector at three values of incidence angle: (a) normal incidence; (b) intermediate; and (c) the limit of acceptance (Goswami 1999©30)

Reflectors are especially beneficial for tube collectors with absorption surfaces all around like the Dewar-China collector. This is because a large amount of the absorber surfaces will otherwise be unused. An alternative to the CPC-mirror is to use a moderately concentrating parabolic shape as in fig 2.78 or, less efficient but simpler, to use a flat white surface. Figure 2.86 illustrates the principle of a flat white reflector.

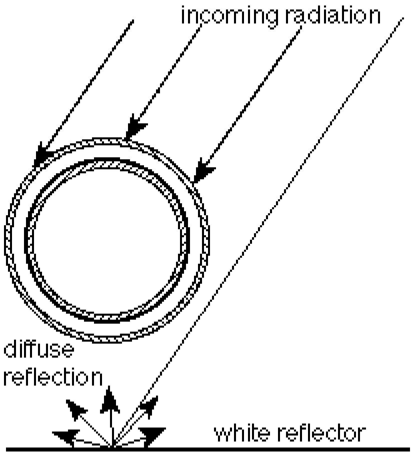


Fig. 2.86 Tubular collector with diffuse flat white reflector (based on Microtherm 2003a)

A CPC-collector for use at high temperatures, 160-180 °C, with a somewhat similar construction to the water-in-glass system has been suggested and tested by Microtherm (2003b). Like in the water-in glass design the heat extraction in tube takes place through natural circulation. To tolerate higher temperature and pressure the water circulates in a slip-in stainless raise tube instead of the tube itself. The tubes are connected to a header. The circulation in the header can be either through natural or forced circulating. According to Microtherm (2003b) an efficiency of over 50% at 170°C was achieved during testing in Rapperswil Switzerland. Fig 2.87 illustrates the system.

In the same study they also found some advantages in using an asymmetrical CPC-mirror instead of a symmetrical, especially at high latitudes.

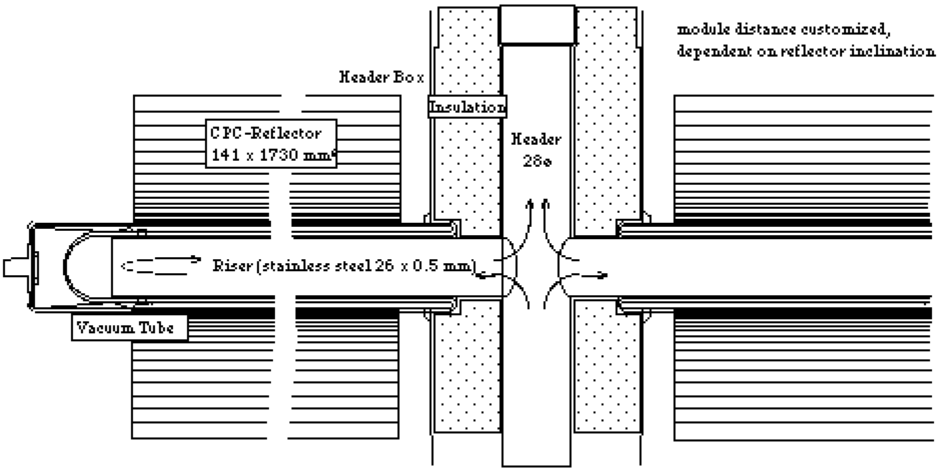


Fig. 2.87 Top view of the SK-400 high temperature CPC collector. Shown is one module of 0.5m^2. The arrows indicate the water flow. (Microtherm 2003b)

As mentioned before an efficiency-value based on the apparatus area is not always the best indicator of a collector's performance when they are put together in arrays or fields. This is true for the evacuated tube collector which is usually mounted in modules (fig. 2.79) of parallel tubes with spaces and sometimes reflectors in-between. If the space between the tubes is to narrow they might shadow each other at high incident angels and if "Dewar-China"-collectors with reflectors are used, then the back side absorber surfaces might not be used optimally. If the space is to wide useable radiation might be lost between the collectors.

For comparing evacuated tube collector efficiency it might therefore be better to use the gross area based efficiency value η_{cTg}. This also allows for a meaningful comparison of evacuated tubular collector designs with and without mirrors.

Figure 2.88 shows measured efficiency bases on absorber and gross area for two different evacuated tubular collector modules, one with direct flow through a absorber fin in the collectors (design A see also fig. 2.80) and the other with direct flow "Dewar-China" collectors with CPC-concentrators (design D see also fig. 2.83). Figure 2.89 shows the incident angel modifiers for the two collectors.

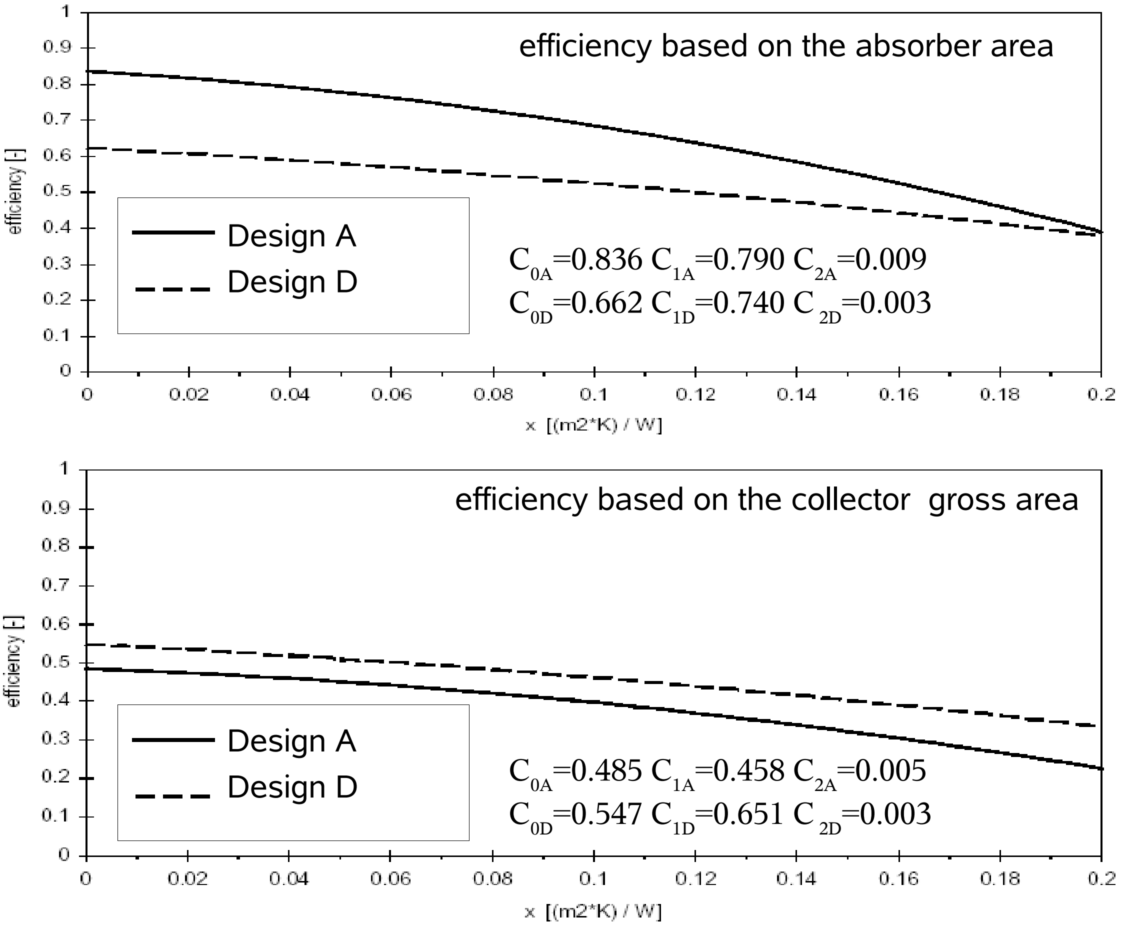


Fig. 2.88 Measured efficiencies for two evacuated tubular collectors of different design (based on Brunold 1994[©41]*)*

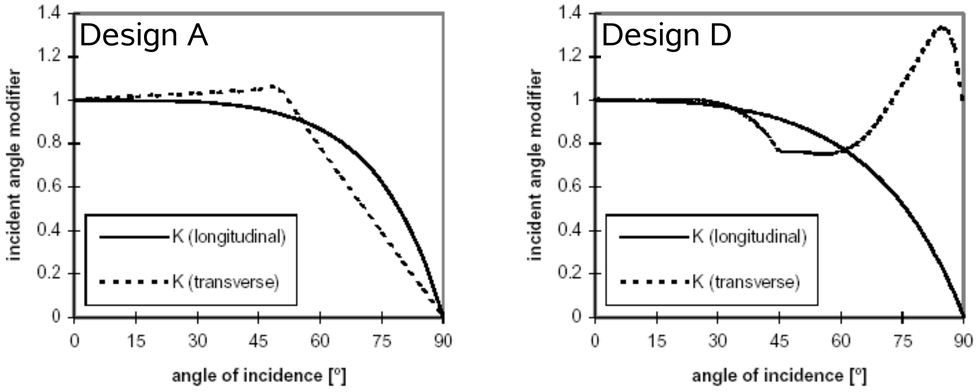


Fig. 2.89 Incidenct angel modifier for two evacuated tubular collectors of different design (based on Brunold 1994[©41]*)*

Flat plate collector

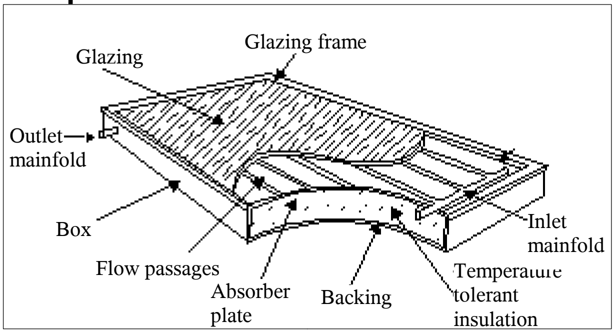


Fig. 2.90 Schematic flat plate collector (NREL 1996)

Fig. 2.91 Roof integrated flat plate collectors (SPF 2003©41)

The flat plate collector is the dominant solar thermal collector in Europe (80 %) (Frei 2003) and (as it seems) most of the world except China (where the evacuated tube collector dominates the market). The flat plate collector is available for a large range of temperatures and applications. As it is considered easier to manufacture it is usually cheaper than the evacuated tube collector. Another advantages is that it can be integrated into the roof (fig. 2.91) and walls of buildings.

The main component of the collector is the absorber plate. The most common forms of absorber plates in use is shown in figure 2.92. The absorber plate can be made of copper, aluminum, stainless steel, galvanized steel, plastic or rubbers. Copper seems (according to Goswami 1999) to be the most common material used for absorber plates and tubes because of its high thermal conductivity and high corrosion resistance. However copper is rather expensive. For low temperature applications (up to about 50°C) plastic absorber plates can be used. To compensate for the low thermal conductivity of plastic materials, larger surface is provided for heat transfer (for example through the use of a multichannel absorber, see fig. 2.92). The absorber plate is painted with a 'flat' black paint or (more efficient) coated with a selective surface (Goswami 1999).

Heat is collected from the absorber to the heat transfer fluid (water) that flows in tubes, either within or attached to the absorber plate. The tubes can be connected in parallel to an inlet and outlet manifold (as in fig. 2.90) or in serial.

Insulation is used to reduce heat losses through the back and sides of the collector. The insulation material has to be heat resistant enough to with stand stagnation without damage. A commonly used material is mineral wool.

To reduce upward losses the collector is normally glazed with one or more sheets of glass or plastic. As mentioned perviously glass is most efficient and common. Using more than one sheet reduces the upward heat losses but also the transmittance of the

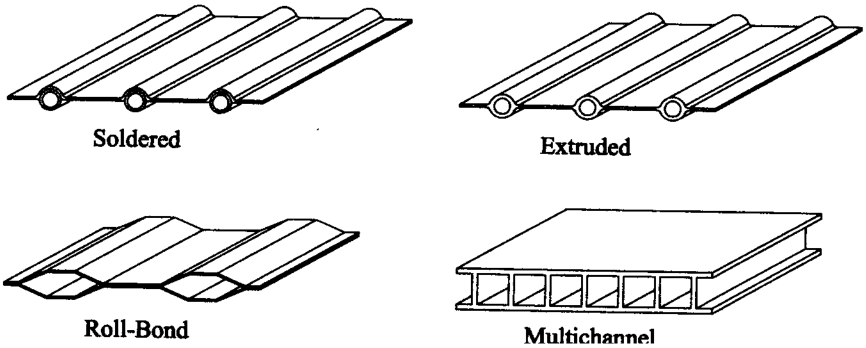


Fig. 2.92 Common types of absorber plates (Goswami 1999©30)

cover (Goswami 1999). Most collectors seem to use a single low-iron glass cover. To increase the transmittance, especially at high incident angles, the glass can be anti-reflectively treated. For low temperature applications the collators can be unglazed.

The box and backing, together with the glazing, protect the collector components from dust, moisture etc. (Tiwari 2003) Ventilation holes and passages for controlled ventilation are made through the casing and insulation, in order to prevent condensation on the inner side of the glass cover (Frei 2003).

Frei (2003) envision a hermetically sealed collector filled with krypton gas, pretty much like a gas-filled low energy window. This prevents any condensation of water vapor within the cover system and avoids the interaction of ambient air (including pollutants) with the material used inside the collector.

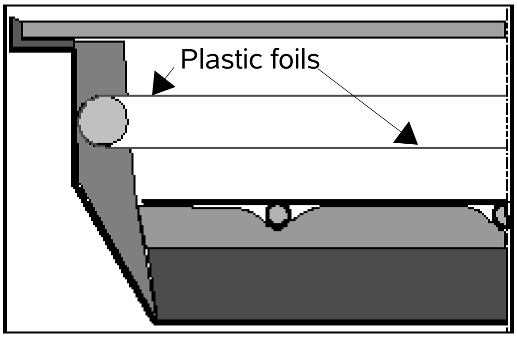


Fig. 2.93 Solar thermal flat collector envisioned for 2010 (Frei 2003©39)

The gap between the glazing and the absorber is not evacuated but air filled. Therefore heat will be lost through free convection from the absorber to the glass cover, and then through conduction to the outside. To reduce these losses different methods can be used. One method currently used in some collectors is to place a plastic (Teflon) foil in the air-filled gap between the absorber and the absorber, thereby splitting the single open volume into two. Frei (2003) suggests using two anti-reflectively treated plastic foils (fig 2.93).

Another method to suppress top losses, which is subject to research, is to use a transparent insulation material (TIM) layer between the glass and the absorber. The TIM (like the back and side insulation) has to be heat resistant, especially since the TIM material itself seriously increases the stagnation temperature. Therefore most plastic materials cannot be used. The most commonly used TIM materials for solar collectors seem to be glass honeycomb structures. According to Abdullah (2003), referring to an article of Rommel and Wagner (1992), a significant decrease in both radiative and convective losses can be achieved using TIM. Consequently, a flat plate collector equipped with TIM can have a performance comparable with an evacuated tube collector. However, since the TIM also decreases the optical efficiency and the angel incident modifier it is important to assure proper design or else the honeycomb may decrease the collector performance instead of increasing it. With proper design a reduction of the heat loss coefficient of up to 51 % can be achieved with only 12 % reduction in optical efficiency (Abdullah 2003). Figure 2.94 shows a commercial flat CPC-collector that has been modified by introducing a TIM layer laying directly against the glass cover and the reflector tops. This way of applying the TIM makes it easy to modify an existing collector. Abdullah (2003) quotes Edwards et al (1976) that leaving small air gaps above and below the honeycomb structure improves the efficiency.

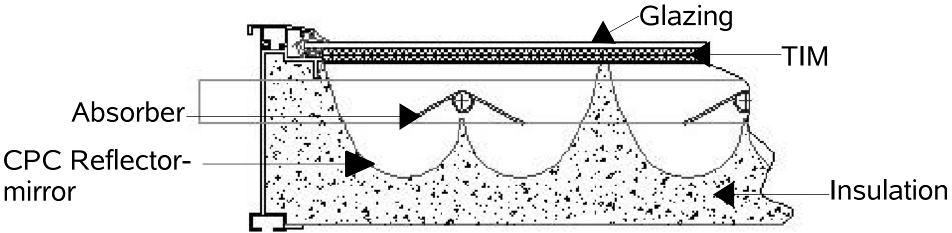


Fig. 2.94 Cross section of a flat plate CPC collector with honeycomb TIM (Pereira 2003©42)

The CPC-concentrator has already been dealt with in combination with evacuated tube collectors. A difference when used with flat collectors is that the concentrator is integrated into the collector within the insulation and glazing, whereas with the evacuated tube collector it was behind the actual collectors and exposed to the ambient air. The construction illustrated in figure 2.94 uses upside down V-shaped absorber plates with selective coating on both sides (Pereira 2003), thus both sides of the absorber plate is used instead of just the front. Another flat CPC-collector commercially available uses a round finned absorber (Rizzi 2003). Using a CPC-reflector in a flat plate collector means that the absorber area, and as it seems thereby heat losses, can be reduced. However, it also seems like the CPC-collector, because of less absorber area, is less efficient in collecting diffuse radiation.

Figure 2.95 shows typical efficiency and incident angle modifier for flat plate collectors 1980-2000 and by Frei (2003) predicted values for 2110.

Figure 2.96 shows measured efficiency for a flat (non CPC) TIM collector. For comparison the two efficiency curves for evacuated tube collectors from figure 2.88 is included in figure 2.96 (drawn in gray).

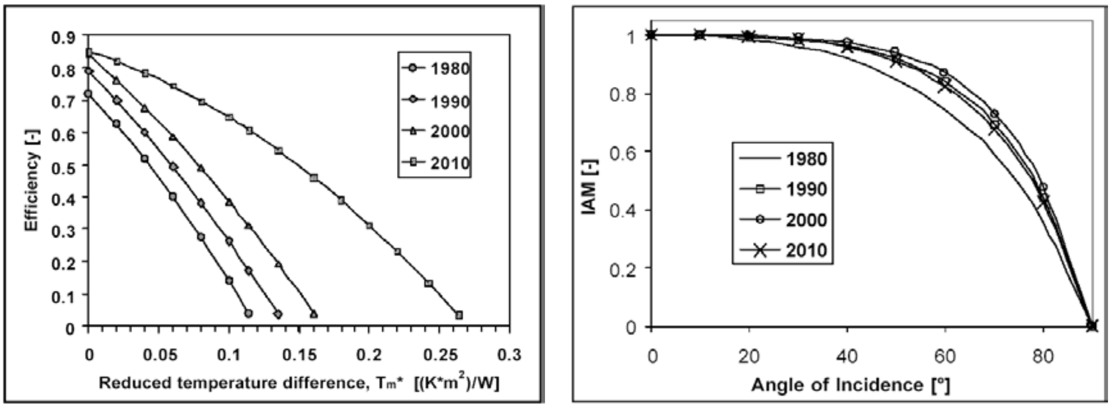


Fig. 2.95 Typical efficiency and incident angel modifier for flat plate collectors from 1980 to 2010 (Frei 2003©39)

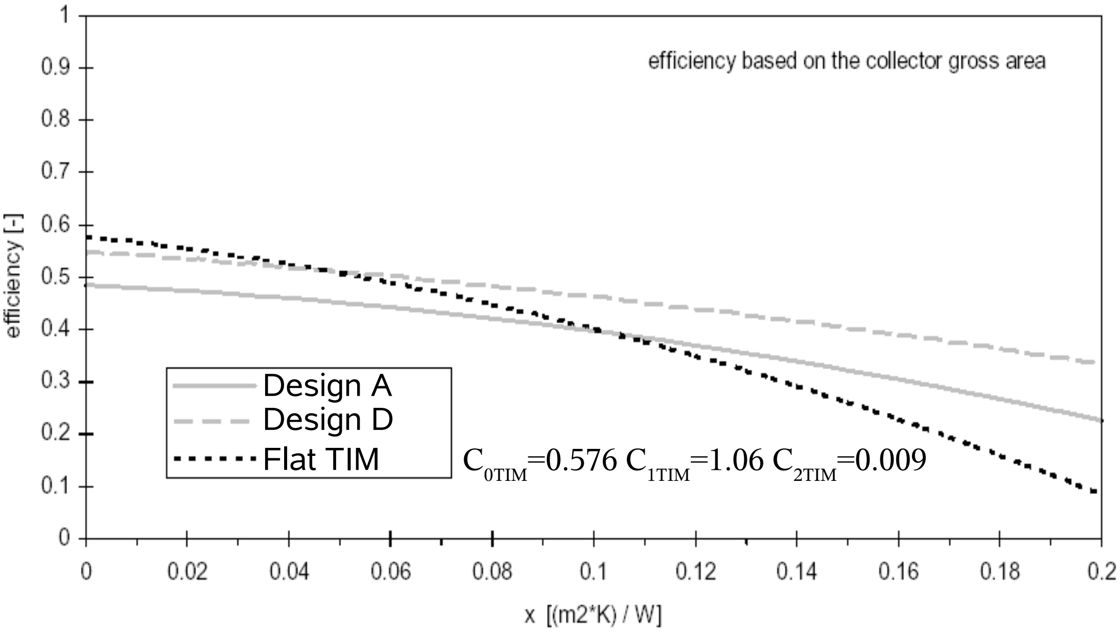


Fig. 2.96 Measured efficiencies for a flat plate TIM collector compared with two evacuated tubular collectors of different design (based on Brunold 1994©41)

Other hot water collectors

The *trickle* collector is a simple and inexpensive collector. It consists of a glass cover and a black painted corrugated metal absorber on which water flows down over, rather than enclosed in tubes. It is usually built on site. Although such a collector is simple and inexpensive, it has disadvantages; condensation on the glaze and a higher pumping power requirement (Goswami 1999).

The open and closed solar generator (section 2.2.3.3) are very similar in construction to the trickle collector. A fundamental difference, however, is that in the solar generator the whole idea is to evaporate water form the sorbent-solution, while in the trickle collector evaporation and thereby condensation is seen as a problem. In the closed solar generator the condensation of the evaporated water on the cover is also essential.

Another type of collector is *the integral coil storage (ICS)* system. These collectors incorporate thermal storage within the collector itself. The storage tank surface serves as the absorber. Most ICS systems use only one tank, but some use a number of tanks in series. As with the flat plate collectors, insulated boxes enclose the tanks with transparent coverings on the side facing the sun. While the simplicity of ICS systems is attractive, they are generally only suitable for applications in mild climates with small thermal storage requirements. Freeze protection is necessary in colder climates (ASHRAE 2000).

A variant of the ICS system is shown in figure 2.97. This system consists of a black tank, a CPC-reflector and a glass cover with TIM-insulation. The black walls of the tank works as an absorber. According to the TIM-manufacture Wacotech (2001) Storage collectors accordion to Wacotec constitute an ideal application field for TIM-honeycombs, as the company writes: "this is the way to construct low cost collectors with high power efficiency". Problems with overheating (stagnation), which could destroy the TIM, do not arise with storage collectors, in contrast to ordinary flat plate collectors (Wacotech 2001).

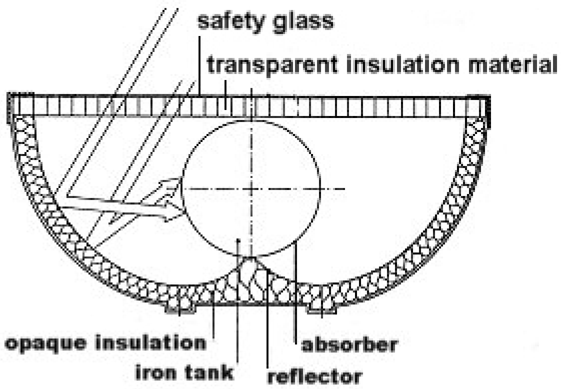


Fig. 2.97 ICS-collector with TIM and CPC-reflector (Wacotech 2001©43)

An other solar collection system, which also can be seen as a type of ICS, is the *non-convecting solar pond*. The nonconvecting solar pond (fig. 2.98) is a horizontal-surfaced solar collector using the absorption of solar radiation at the bottom of a 1- or 2-m-deep body of water to generate heat. To prevent the hot water generated in the bottom of the pond from convecting (through free convection) toward the cooler surface the bottom layer is made heavy by dissolving salts into it. Since stationary water is quite an effective insulator, it is possible for the lowest layer of a well- designed solar pond to boil. Boiling, of course, must be avoided, because it would destroy the stable density gradient. Heat is collected from the pond by removing the bottom layer slowly without disturbing the main water body. The removed water then passes through a heat exchanger in which heat is removed and the water is cooled. The cooled water is returned to the bottom of the pond. It is necessary to add water at the top of the pond to replace water that evaporates and to maintain the density gradient. It is also necessary to insure that the bottom layer is not diluted. This can be done either by adding salt to the bottom layer or by evaporating part of the water that is removed from the bottom layer. It has to be taken into consideration that the solar pond is exposed to the atmosphere with winds, rain, dirt and leafs affecting it. Also, since the solar ponds are horizontal and the sun is low in the sky north or south of the mid latitudes in the winter, the ponds need to be near the equator if winter yields are not to be curtailed sharply (Goswami 1999). The sharp curtail of the winter yields might not be a problem if used for air conditioning since air-conditioning loads are typically less during the winter.

According to Al-Marafie (1988) a solar pond with an efficiency of 20 % providing hot water with a temperature of 90 °C requires 268 % more collector-area than flat plate collectors. The land area occupied by a ground-mounted flat plate collector field (the gross area), however, may not differ much since flat plate collectors need nearly 2.5 times the actual collector area. (This means that the efficiency based on gross area η_{cTg} is similar to that of a flat plate collector field)

Two advantages of the solar pond over the flat plate collector, in addition to low capital cost, are (Al-Marafie 1988):

1. A total conversion of available solar radiation into useful thermal energy without any preheating requirement of the collector field in the morning.
2. Effective utilization of collected thermal energy without excessive losses in pipes or storage tanks.

These two advantages or at least the first seems to also apply for other type of ICS-collectors.

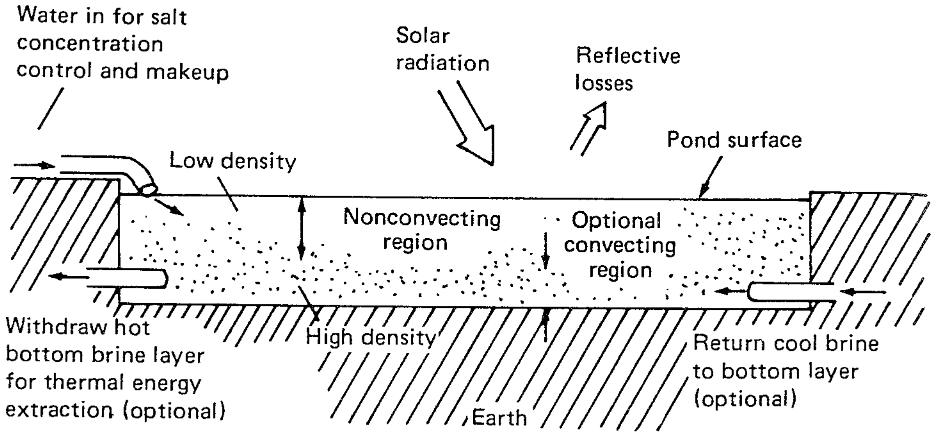


Fig. 2.98 Schematic diagram of a nonconvecting solar pond showing conduits for heat withdrawal; surface washing; and an optional convecting zone near the bottom (Goswami 1999©30)

2.3.2.3 Medium-low temperature Air collectors

Solar air collectors are commonly used for agricultural drying and space heating applications (Goswami 1999). It can also be used for providing hot air for regeneration of sorbents/desiccants in open generators and desiccant wheels. Their basic advantages are low sensitivity to leakage and no need for additional heat-exchangers and glycol/water-loops. The later means fewer components and less heat exchanger temperature losses.

Solar air collectors are similar in construction to the solar flat plate collector. As the flat plate collector it consists of an absorber plate contained in an insulated box covered with one or more sheets of glazing. The primary differences are the design of the absorber plate and flow passages. Because the heat transfer fluid (air) has poorer heat transfer characteristics, it flows over the entire absorber plate, and sometimes on both the front and the back, in order to make use of a larger heat transfer surface (ASHRAE 2000). Because of the low heat capacity of air a much higher flow is needed than in a hot water collector. This means large air ducts, at least from the collector. Supply-ducts to the collector can be avoided if ambient air is used.

The heat transfer area of the collector is most often increased with fins, matrices, or corrugated surfaces. Many of these designs increases air turbulence, which improves the collector efficiency (at the expense of increased fan power). Figure 2.99 shows cross sections of typical air collectors. Fins on the back of the absorber (A) increase the convection heat transfer surface. Air flowing across a corrugated absorber plate (B) creates turbulence along the plate, which increases the convective heat transfer coefficient. A box frame (C) creates airflow passages between the vanes. The vanes conduct heat from the absorbing surface plate to the back plate. Heat is transferred to the air by all of the surfaces of each boxed airflow channel. A matrix absorber plate (D) is formed by stacking several sheets of metal mesh such as expanded metal plastering lath. Placing the mesh diagonally in the collector forces the air through the matrix so it does not contact the glazing after being heated (ASHRAE 2000).

Unglazed, transpired solar air collectors offer a low-cost alternative for some applications such as preheating of ventilation air and agricultural drying and curing. Such collector consist of perforated dark colored absorber sheets exposed to the sun through which air is drawn (Goswami 1999) .

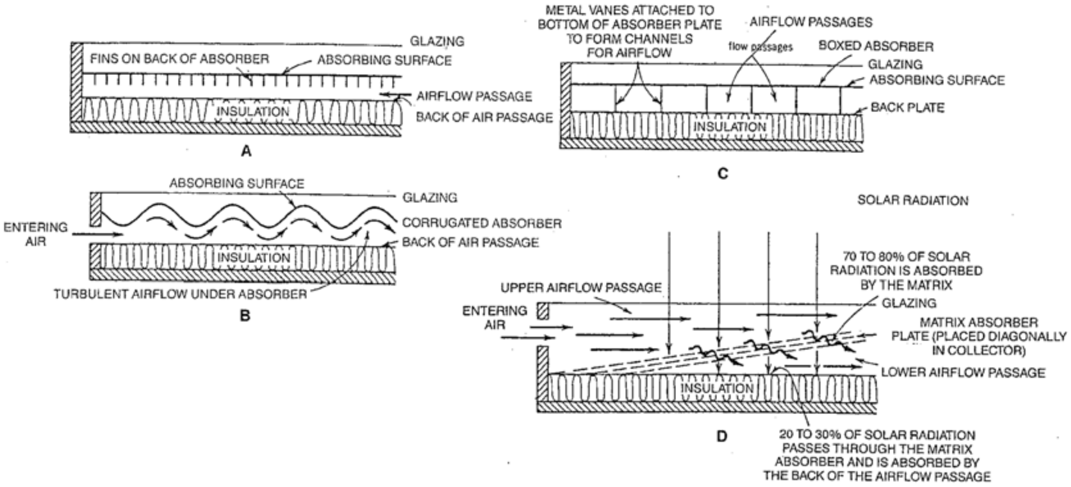


Fig. 2.99 Cross sections of typical air collectors (ASHRAE 2004[©1])

2.4 Heat rejection

In all kinds of cooling and heat engine processes (as seen from section 2.2) heat needs to be rejected. In some application this heat might be useable, but in most application it is considered as waste heat and needs to be discarded.

2.4.1 Air cooled systems

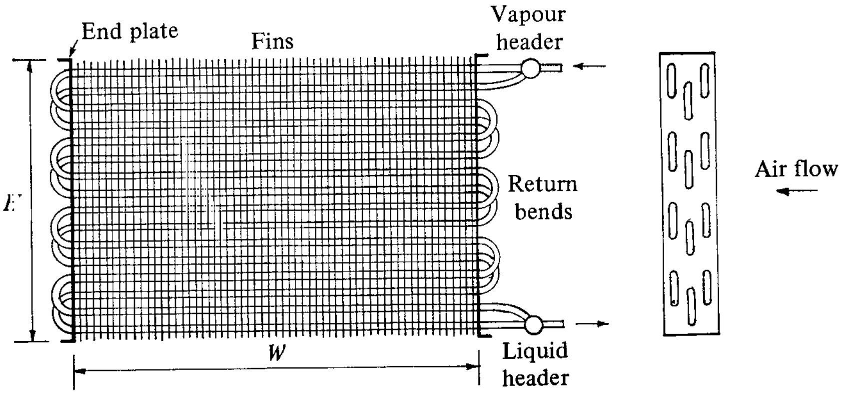


Fig. 2.100 Air-cooled condenser (Gosney 1982©8)

A rather common method of heat rejection for small or medium size application is to use finned *air-cooled condensers*. As illustrated by figure 2.100 they consist of pipes surrounded by fins. The purpose of the fins is to extend the heat transfer surface towards the ambient air. Refrigerant flows in to the entrance of the condenser as hot gas. Through heat exchange with the ambient air it looses heat and condenses. Fans are used to force ambient air throughout the condenser coil. At the end of the condenser the refrigerant has become a liquid with a temperature about 5-15 °C above the dry temperature of the ambient air. The window/wall air conditioner in figure 2.4 contains an air cooled condenser.

The system is rather simple and therefore inexpensive and has low maintenance requirement. However, it has some drawbacks. First of all it can only cool to a temperature some degrees above the dry bulb temperature of the ambient air. That might not be sufficient in some application, especially not for heat driven cooling processes, which as seen from section 2.2.3 requires heat rejection at a rather low temperature to be efficient.

An other disadvantage is that the condenser needs to be in contact with the ambient air. This is most often achieved by locating the condenser outdoors. Depending on the location of the cooling machine or heat engine this could result in long pipes, which would result in a large volume of refrigerant/working fluid in the system. As mentioned previously there might be reasons to reduce the amount of these substances in the system.

Somewhat similar to the air-cooled condenser is the *air-cooled absorber*. An example of an air cooled absorber is the absorber in the Platen Munters- heat pump in figure 2.33. A difference from the condenser is that two (three in the Platen Munters system) different fluids are involved, the refrigerant and the absorbent. These need to counterflow through the absorber.

2.4.2 Water cooled systems

In a water-cooled system condensation takes place in a heat exchanger. Either shell-and-tube heat (horizontal or vertical) or plate heat exchangers can be used. Figure 2.101 illustrates a horizontal shell-and-tube condenser. Since water has much better heat capacity than air the cooling water flow is only a fraction of the volumetric air-flow required in an air cooled condenser. The unit can thus be built much more compact than a air-cooled condenser.

Cooling water for cooling the condenser could be taken from ground water, from a groundwater loop, a lake, a pond or the sea. However, much more common is to take the cooling water through a closed loop from a special device that rejects the heat from the cooling water to the atmosphere.

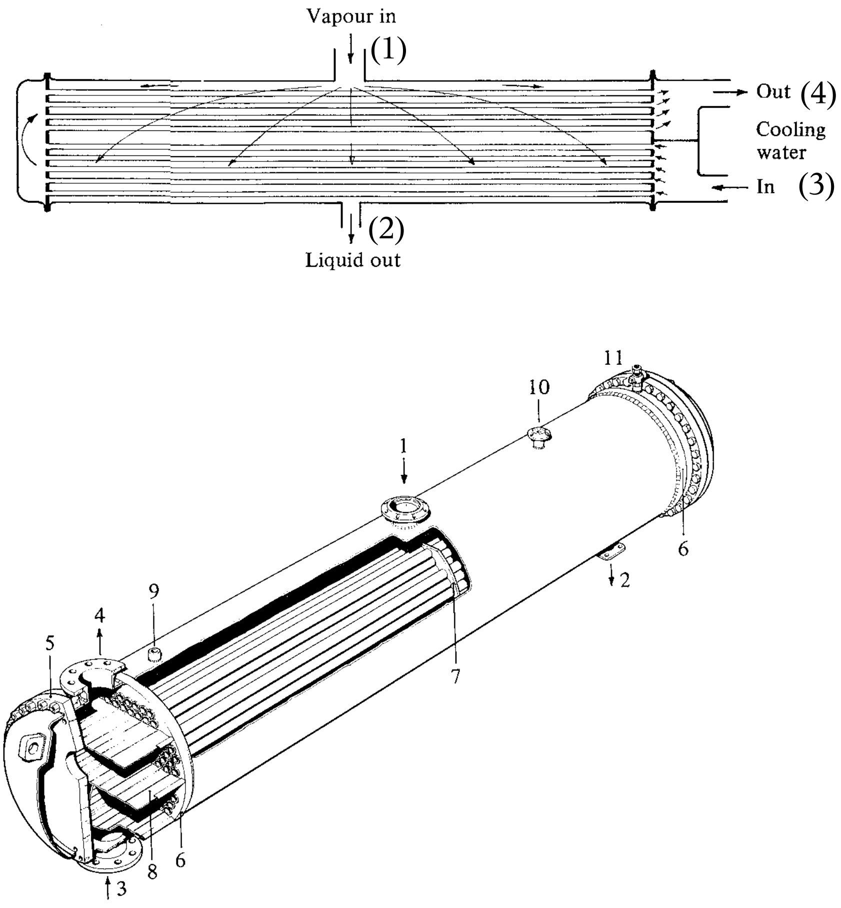


Fig. 2.101 Horizontal shell-and-tube condenser (Gosney 1982©8)

The simplest device for achieving this is the *dry cooler*. Its construction is almost identical to the air-cooled condenser. The main difference is that no phase change takes place. Warm water enters, cools down as heat is transferred through the tube walls of the coil to ambient air, and leaves with a temperature some degrees above the dry temperature of the ambient air. Thus it has the same advantages and main disadvantage as the air cooled condenser. Namely it is simple and cheaper but it might not be able to provide cooling water with a temperature low enough.

If a lower temperature cooling water is desired a *wet cooling tower* can be used. The cooling tower works on the same principles as evaporative cooling explained in section 2.2.4. The difference is that in section 2.2.4 it was sensible cooling of the air that was the object, here it is cooling of the water.

Figure 2.102a illustrates a *open* or *direct contact cooling tower*, often referred to as just cooling tower. Air, either outdoor air or exhaust air, is drawn from the bottom towards the top of the tower. Water is sprayed from the top over a packing and then flows downwards by gravity. As the water and the air gets in contact a portion of the water evaporates. The sensible heat required for the evaporation can be taken both from the water and the air. Since the evaporated water mixes with the air, the latent heat of the air is increased. Figure 2.103 illustrates the relationship between the temperature of the water and the wet bulb temperature of the air in a counterflow cooling tower. As explained in section 2.2.4, the low temperature limit for this process is the wet bulb temperature of the entering air, which often is considerably lower than its dry bulb temperature (see Mollier/psychometric chart appendix B). The temperature gap between the temperature of the water leaving the tower and the wet bulb temperature of the air entering is in figure 2.103 called *approach*. The difference in temperature between the water entering and leaving is called *range*.

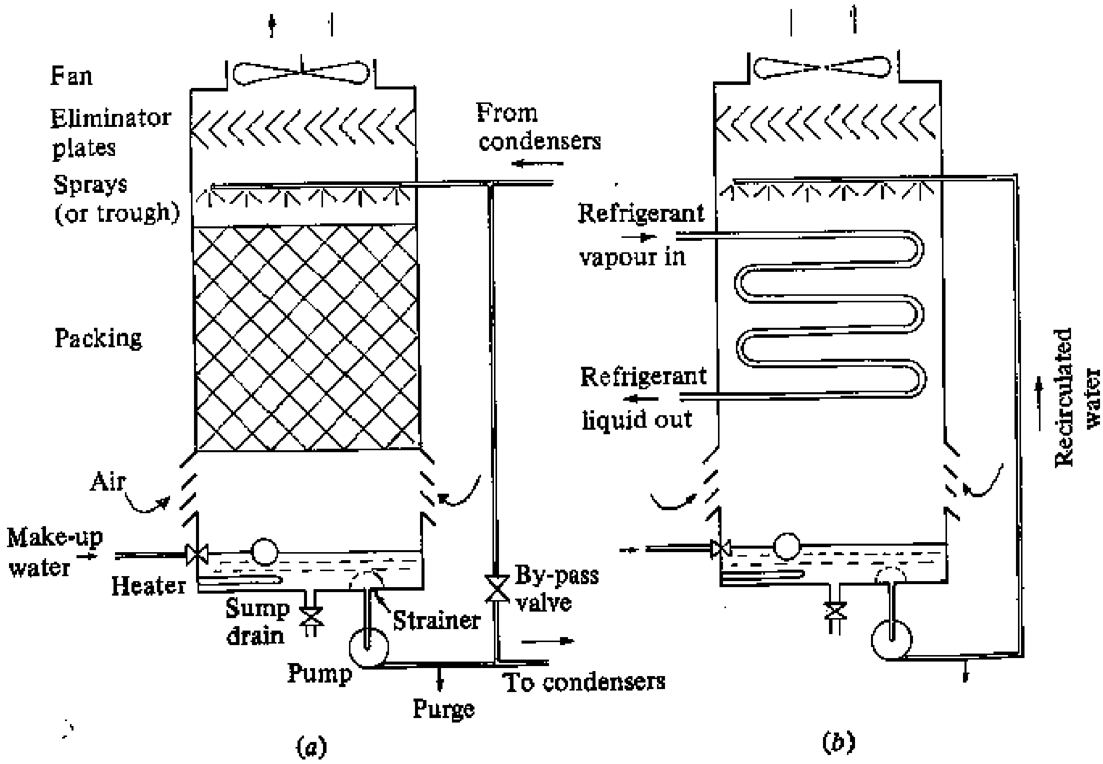


Fig. 2.102 Schematic drawing of (a) an open cooling tower and (b) an evaporative condenser (Gosney 1982©8)

As previously mentioned the heat required for evaporation can be taken both from the water and from the air. Thus the air exiting the cooling tower can actually have a lower dry bulb temperature than the air entering. On the other hand, if the dry bulb temperature of the air anywhere within the tower is lower than the water temperature, sensible heat will be transferred from the water to the air. Since the contact area between water and air is very large, due to the spray and the packing, the heat transfer is rather efficient. Figure 2.104 illustrated the two possibilities, that the air is sensibly cooled or sensibly heated, in a psycrometric plot. If the entering condition of the air is represented by A, then vector AB represents how it is heated both latently (vector CB) and sensibly (vector AC). If the entering condition of the air is represented by D (same wet bulb temperature but higher dry bulb temperature), then the air will still be latently heated (vector CB), but instead of being sensibly heated, it will be sensibly cooled (vector DE). Thus the air will leave the cooling tower with a lower dry bulb temperature than it entered with. This sensible cooling of the air increases the water consumption.

The evaporation rate at typical design conditions is approximately 1% of the water flow for each 7 °C of water temperature range. However, the average evaporation rate over the operating season is less than the design rate because the sensible component of the total heat transfer increases as the entering air temperature decreases (ASHRAE 2000).

As in an evaporative cooler some of the water needs to be purged to prevent build up of minerals. In addition to water losses from evaporation and purging, losses also occurs because of liquid carryover into the airflow. The lost water needs to be replaced by make-up water. Thereby a wet cooling tower give the whole system a water consumption in addition to energy consumption.

An issue of concern with open cooling towers in cold climates is freezing.

In large cooling towers (ex. for power stations) the airflow is induced by natural convection, but in the towers used for refrigeration plants the airflow is almost always assured by a fan (Gosney 1982).

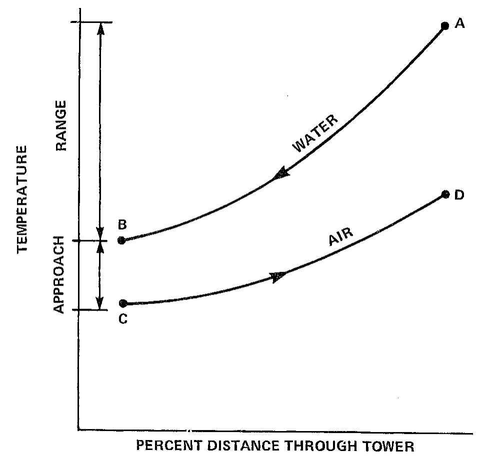


Fig. 2.103 Relationship between the temperature of the water and wet bulb temperature of the air in a counterflow cooling tower (ASHRAE 2004$^{©1}$)

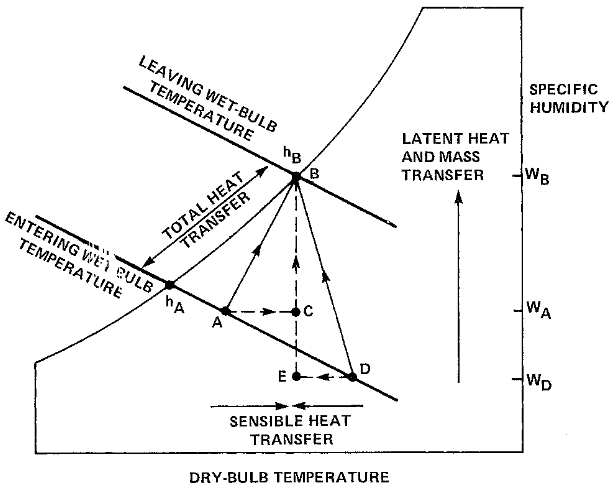


Fig. 2.104 Psycrometric analysis of air passing through a cooling tower. (ASHRAE 2004$^{©1}$)

Alternatives to the open cooling tower are the *evaporative condenser* and the *closed circuit* cooling tower. Both consist of a wetted tower similar to the above-described open cooling tower. The difference is that separate circuits are used to wet the tower and to remover heat. Figure 2.102b illustrates an evaporaportive condenser. The fluid to be cooled flows in tubes through a coil inside the tower and recirculated water wets the coil and the passing airflow. Heat flows from the the cooling fluid circuit, through the tube walls of the coil, to the recirculated water circuit and then, by heat and mass transfer, to the airflow and out into the atmosphere (ASHRAE 2000).

An advantage of the evaporative condenser or closed circuit cooling tower is that it can easily be converted to an air cooled condenser or dry cooler by simply shutting down the pump (and draining the water). This can be done at winter when the dry temperature of the ambient air is low enough to provide cooling by itself. Thereby the problem with freezing at low outdoor temperature can be eliminated (Gosney 1982, Lindholm 2003a).

The difference between the evaporative condenser and the closed circuit cooling tower is the same as between the air-cooled condenser and the dry cooler.

2.5 Storage

In general storage is useful when the demand is not well correlated with the supply. In solar air conditioning applications storage is useful when the air conditioning demand is not fully correlated with the solar radiation. In the early morning the cooling demand could be ahead in time (of the supply of solar energy), and in the afternoon the peak the demand could be some time after peak supply. Storage can increase the utilization of other components. For example solar collectors and chillers can be kept working at optimal performance even when there is lower demand (ex. lunch time). The chiller can also be kept working when there would otherwise be no natural energy supply (ex. nighttime or when the sun is behind). Installed capacity of different components can also be reduced since they do not necessarily have to cover peak demands. In systems where the chiller works discontinuously, storage is especially important as accessible energy supply would otherwise be wasted. The downside of energy storage is that it is impossible to store energy without losses.

Different methods of storage can be used.

2.5.1 Sensible heat storage

Sensible heat storage is most common as it is used for storage of hot tap water and hot water for space heating in all kinds of buildings, all over the world. It consists of a substance contained in an insulated tank or container. When used as a hot storage the substance is kept at a high temperature. It is charged by heating the substance with a heat source higher than its current average (from top to bottom) temperature and discharged by a fluid cooler than its current temperature. When used as a cool storage the opposite holds. The storage capacity is determined by the storage volume, the specific heat capacity of the substance and the temperature difference between the available heat source and the needed temperature level for the heat to be useful. (As seen in section 2.2 many processes work only with heat above a certain temperature).

Because of its relatively high heat capacity, water is a frequently used substance. For cold storage where the difference between charging and discharging usually is lower than 10 °C the energy density of the storage is in the order of 10 kWh/m^3. For heat storage this temperature difference can be greater, and the energy density larger, in the order of 40 kWh/m^3 (Setterwall 2003). For high temperatures the pressure in a tank containing water will have to be increased, this could be impractical and therefore other substances are often used.

In a rock-bed-storage rocks, kept in a container, are used as substance. Air is blown through the container to charge and discharge it. This can be practical if the heat source as well as the heat demand is in form of hot air. It seems like it could be used as a cool storage as well.

Energy losses from a sensible storage are in form of heat flow the storage to the cooler or warmer ambient environment outside the storage. This increases as the temperature difference increases. If additional heat exchangers are needed (to reduce the amount of antifreeze for example) there are also temperature losses.

For large-scale storage applications filled rooms below surface of the ground or bore holes into the bedrock can be used. (Note that a single bore hole will not do well as a storage because of extensive heat leakage. For a bore hole storage to work a group consisting of several bore holes is needed).

2.5.2 Latent heat storage

In a *latent heat storage* a fluid (substance) is kept at its melting point temperature. This gives two advantages. Firstly since melting and freezing are processes that occur at a constant temperature. The temperature in the storage is thus constant until all fluid is either melted or frozen (at which point it becomes a sensible storage if heated or cooled further). Therefore the storage can be charged and discharged at fairly constant temperatures. Secondly the heat of melting, latent heat, is very much higher than the sensible heat (Setterwall 2003).

The phase change with the highest storage potential is freezing and melting water. A storage where freezing and melting water is used to store energy is called an ice-storage. Since the melting temperature of ice is 0 °C, an ice-storage can only be used for cold storage. Even for cold storage one has to consider that cooling to temperatures below zeros reduces the efficiency of the chiller and thereby increased energy demand (Setterwall 2003). In applications where water is used as refrigerant it is impossible to achieve evaporator temperatures below 0 °C, thus ice-storage cannot be used.

Salt hydrates and eutectic salts have relatively high density. The latent heat based on volume of those substances is therefore in the same order of ice although the latent heat based on weight is considerably less. Paraffin however has low density and also lower latent heat of melting (Setterwall 2003).

The energy density for ice storage, depending on technology, varies between 40 and 70 $\mathrm{kWh/m^3}$. For salts the energy density are in the order of 40 $\mathrm{kWh/m^3}$ when used for cold storage and 70 $\mathrm{kWh/m^3}$ when used for heat storage (Setterwall 2003).

Losses for this storage technique are the same as for the sensible heat storage.

2.5.3 Chemical storage

Chemical storage uses the same principles as sorption heat pumps explained in section 2.2.3. In the closed liquid absorption cycle storage can be obtained by storing strong or even crystalized solution in (or in connection with) the generator and absorber and refrigerant in the condenser (or in a separate tank). In the solid sorption cycle, storage is obtained by simply holding the process after water has been desorbed from the sorbent.

In the open absorption and the desiccant systems, storage can be accomplished in the same way with the exception that no refrigerant needs to be stored.

Since the system is still essentially a heat pump the storage cannot be discharged without either input or rejection of heat. When heat is discharged, low-grade heat (low temperature heat) must be provided to an evaporator to create steam that can be absorbed or adsorbed. When cooling capacity is discharged, heat from the absorber/adsorber must be rejected. Rejecting heat should not be a problem when the storage (for example) is used for compensating for lack of solar radiation in a solar air conditioning (because of clouds or during the night). However, if used to cover peak demand due to high solar insolation, high outdoor temperature and high outdoor humidity, heat rejection could be a serious problem.

When the storage i being charged heat needs to be rejected from the condenser; this could also be problematic in a solar air conditioning system since at the time that the energy supply from the solar collectors to charge is most abundant, during peak noon, heat rejection is, as mentioned above, not uncomplicated. A suggested solution for these problems, to use a pond as heat sink, actually is the same as having it connected to a sensible cold storage. Thereby a kind of hybrid storage has bee created.

The need for heat rejection at storage is not present in an open system (see section 2.2.3.3).

Collier (1979) also claims that open absorption systems are more suitable for chemical storage than a closed, because in an open absorption system the generator is at atmospheric pressure while in a close system it is at subatmospheric pressure. Thus the strong solution in an open system can be stored in a simple tank while strong solution in a closed system would have to be stored in large pressure vessels. However, introducing an additional pump to pump the strong solution from the subpressure in the system to a storage tank at atmospheric pressure could easily solve this disadvantage for the closed system. The total lift of this additional pump and the systems normal solution pump together would then be the same as the pressure lift required by the single solution pump in an open system.

Adsorption of water in silica gel or on zeolites is a demonstrated technology leading to energy densities based on the volume of adsorbent in the order of 150 kWh/m^3. Reactions involving water and solid salts have been tested but have not yet reached commercial level (Setterwall 2003).

Losses in a chemical storage are connected to charging and discharging the storage. Unlike the sensible and latent storage there are no continues heat losses when not charged and discharged. The system could therefore be charged and then left charged for extended times without any energy being lost.

A disadvantage with chemical storage is that the sorptive substances seem to be rather expensive, and the storage is also rather complex compared to a sensible o latent storage. This could make it economically unattractive.

2.5.4 Electrical storage

Batteries can be used as storage in systems where the collected energy is in the form of electricity. An alternative if the system can be connected to a public electric grid, is to sell surplus electricity to the grid. When there is shortage electricity is bought back from the grid. Since there is a correlation between solar insolation and electricity demand in many countries, surplus electricity can be sold at peak load and then be bought back (not the same electricity of course but to the user this does not matter) if needed at off peak load, for example during the night. In countries where there are different tariffs for peak electricity or daytime electricity this could be profitable. This way of using the grid to “store” electricity could be seen as a *virtual storage.*

3. Solar air-conditioning systems

3.1 General

Solar air-conditioning system can be divided into two groups of systems: *solar-autonomous systems* and *solar-assisted systems.*

In a solar autonomous system "all" energy used by the air-conditioning system is solar energy. The word all is in between quotation marks because often systems called solar autonomous still use grid-provided electricity for powering fans and pumps. If the system is to be truly solar autonomous this energy should also be from solar energy.

Since the solar-autonomous system is dependent on solar energy to work, the indoor climate is limited by available solar energy (Henning 2000), either collectable or stored. The system design of a solar-autonomous system is based on indoor climate demands, expected internal and external heat loads and the time correlation between the loads and solar energy supply.

Two measures of interest in a solar-autonomous system are **COP$_{sol}$** and **SPF$_{sol}$.**

COP$_{sol}$ gives the momentary performance of the entire system and is defined as:

$$\mathbf{COP_{sol}} = \frac{\dot{\mathbf{Q}}_{\mathbf{0,\,sol}} \cdot \mathbf{10^3}}{\mathbf{I_c A}} \quad (\mathbf{e19})$$

$\dot{\mathbf{Q}}_{\mathbf{0,sol}}$ *heat removed* (*by use of solar energy*) *from conditioned space* [kW] (*delivered cooling capacity*)

$\mathbf{I_c}$ *solar radiation* [W/m^2]

A *solar collector* (*s*) *apparature area* [m^2]

SPF$_{sol}$ (Seasonal Performance Factor) give the performance over a time of operation e.g. the cooling season. It is defined as:

$$\mathbf{SPF_{sol}} = \int \frac{\dot{\mathbf{Q}}_{\mathbf{0,\,sol}} \cdot \mathbf{10^3}}{\mathbf{I_c A}}\,\mathbf{dt} = \int \mathbf{COP_{sol}}\,\mathbf{dt} = \frac{\mathbf{Q_{0,\,sol}} \cdot \mathbf{10^3}}{\int \mathbf{I_c A\ dt}} \quad (\mathbf{e20})$$

$\mathbf{Q_{0,\,sol}}$ *the amount of heat removed by means of solar energy over a season* [kWh]

In a *solar assisted system* the solar energy cover a certain fraction of the energy used by the air-conditioning system. This fraction is called the *Solar Fraction* (**SF**, SOLF). The rest of the energy is supplied through an auxiliary or backup system (Henning 2000). The auxiliary system can be a separate air-conditioning system working in parallel with the solar system, or it can be integrated into the solar air-conditioning system. Because of the auxiliary system, attainable indoor climate are not limited by the availability of solar energy. The focus when designing a solar-assisted air-conditioning system is the solar fraction (Henning 2000).

The solar fraction is defined as ([solar energy used]/[total energy used]):

$$\mathbf{SF} = \frac{\int \eta_c \mathbf{I_c A\ dt}}{\int \eta_c \mathbf{I_c A\ dt} + \mathbf{PE} \cdot \mathbf{10^3}} \quad (\mathbf{e21})$$

η_c *solar collecor efficency*

PE *primary energy* [*kWh*]

Where PE is the primary energy supplied to the system (to power the auxiliary system). Primary energy usually seems to be used do denote energy from fossil fuels.

Solar energy is not regarded as a primary energy as it is considered a 'free' and renewable resource.

Grid-electricity is not primary energy. The primary energy use (PE) for electricity produced by a fossil fired power plant is given by dividing the used electrical energy with the efficiency of the power plant and the efficiency of the transmission. The primary energy use for heat provided from a fossil fuel fired furnace is given by dividing the delivered heat with the efficiency of the furnace (and eventually multiplying it with a loss factor to compensate for transmission and storage losses), or simply by measuring the amount of fuel consumed and multiplying this with the energy content of the actual fuel.

None of the literature used for this report deals with how to regard energy from burning renewable fuels (bio-mass) or electricity produced from renewable sources. The grid-electricity provided in many countries (for example Sweden) consists of a mixture of electricity produce by fossil energy and renewable energy. The electricity produced with fossil energy comes from different plants with different efficiencies and with different transmission losses to consumer etc. All this has to be dealt with. An extensive treatment of this, however, falls outside the scope of this report.

Just as $\mathbf{COP_{sol}}$ was defined as the total COP from solar insulation to cooling effect $\mathbf{COP_{aux}}$ can be defined as:

$$\mathbf{COP_{aux}} = \frac{\dot{\mathbf{Q}}_{\mathbf{0,\,aux}}}{\dot{\mathbf{PE}}}$$

$\dot{\mathbf{Q}}_{\mathbf{0,aux}}$ *heat removed (by use of auxiliary energy) from conditioned space* [kW]
$\dot{\mathbf{PE}}$ *momentary primary energy consumption* [*kW*]

To easily be able to comparable a solar-assisted system with a conventional (non-solar-assisted) system or other solar assisted system, the measure primary energy factor (**PEF**) is useful. **PEF** is defined as:

$$\mathbf{PEF} = \frac{\mathbf{PE}}{\mathbf{Q_{0,sol}} + \mathbf{Q_{0,aux}}} = \frac{\mathbf{PE}}{\mathbf{Q_0}}$$

$\mathbf{Q_{0,\,aux}}$ *the amount of heat removed by means of auxiliary energy over a season* [kWh]
$\mathbf{Q_0}$ *the total amount of heat removed over a season* [kWh]

The importance of considering the primary energy factor is illustrated by figure 3.1 showing the primary energy factor of solar-assisted chillers working at different COP values plotted against the solar fraction of the system. The auxiliary system is assumed to have the same COP as the solar-driven system. This is the case in for example a heat driven system (ex. an absorption chiller) where the auxiliary system consists of a furnace for providing backup heat to the chiller.

The horizontal lines in figure 3.1 illustrate the primary energy factor of two electrically driven vapor compression systems with COP values of 4.5 and 2.5. The total efficiency of the power plant and the transmission is set to 0.36. Around 0.36 seem to be a rather common efficiency of older fossil fired power plant. However, in the 1990s (dual-cycle) power stations with electrical efficiencies of above 50% were put in to operation (Lamp 1998), decreasing the primary energy use, and thereby the primary energy factor, of electrically driven vapor compression system even further.

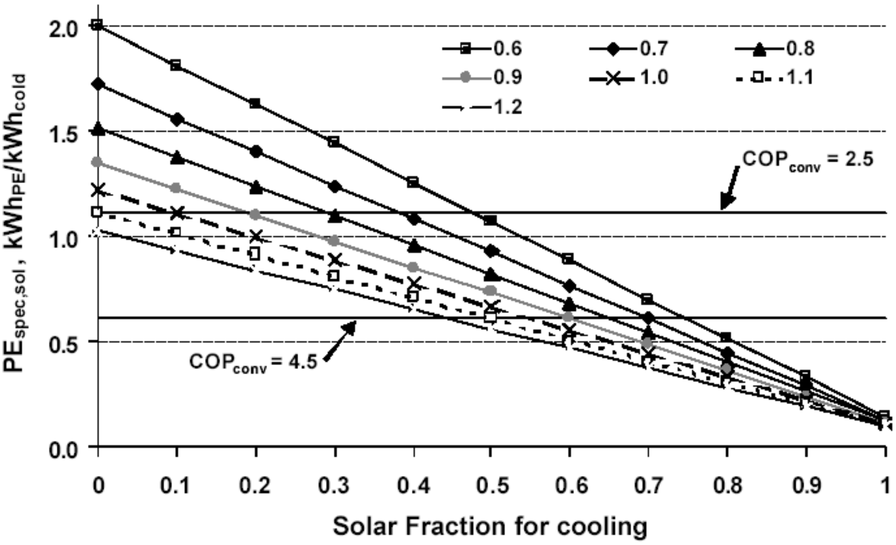


Fig. 3.1 Primary energy factor of solar assisted cooling systems as function of the solar fraction for different values of COP; also shown is the primary energy consumption of conventional chillers with two values of COP$_{conv}$ *(Henning 2003*©44*)*

As seen in figure 3.1 a substantial solar fraction is required to obtain any PE-savings at all, especially if a low COP chiller is used. For example, with a COP of 0.7 (typically for a single effect absorption chiller) the solar fraction would have to be above 0.4 or 0.7 for any primary energy saving to be achieved. If the solar fraction is less the system will in fact use more primary energy, and thereby possibly cause more environmental pollution, than an electrically driven vapor compression system.

A way to improve the primary energy saving can be to use a parallel auxiliary system with higher COP, for example a vapor compression chiller or a double effect absorption chiller. This will however result in a much higher first cost. An other approach briefly mentioned in section 2.2.3.1 is to use a chiller so constructed that firing it with auxiliary heat at high temperature will result in a substantial higher COP than firing it with lower temperature solar heat.

Other measures that can be used are life cycle energy use and life cycle polluting emissions. Biezanow (2003) has made a life cycle analysis of a gas/solar-fired single stage absorption chiller and a vapor compression chiller.

There are also different economical measures that can be used. Most common seems to be the life cycle cost of the system. Henning (2003) presents life cycle cost as cost per primary energy saved [€-cent/kWh] (cost since solar air-conditioning is to date usually not economical profitable). Syed (2002), Al-Marafie (1998) and Al-Homoud uses cost per kW cooling and life cycle cost of solar-air-conditioning systems in relation to cost of conventional vapor compression systems. Lindholm (2003b) presents life cycle cost as annual capital cost. A thorough economical analysis or methodology for such, although important, is outside the scope of this report. The reader is therefore refereed to previously mentioned articles for such.

An important aspect, previously mentioned in section 2.2.3.3, is how to treat water consumption. Many of the solar air-conditioning systems available consume water in one way or another. Often this water consumption does not seem to be treated at all. The water is seen as a free resource. This is however only true if the water is supplied from a private water resource with unlimited supply and all needed treatment such as desalinated is done within the system. In all other cases the water is a resource that cost both PE and money. This is especially true if the water needs to be desalinated. Al-Marafie (1998) and Al-Homoud present both cost and primary energy consumption for water consumption in Kuwait, where water needs to be desalinated, in an exemplary manner.

One might consider if it is wise to use such an essential substance as water for cooling in areas where there is shortage even if it is less costly economical and requires less primary energy.

3.2 System architecture

The aim of this section is to give some examples of how different solar air-conditioning systems can be but together with the components explained in section 2 and some thing to consider when putting them together. By no means should this section be seen as a full throughout survey of all possible systems or as a list of the best systems.

More systems can be found in an article by Syed 2002, in which he lists a large number of different possible solar cooling schemes.

3.2.1 Photovoltaic(PV.)-air-conditioning

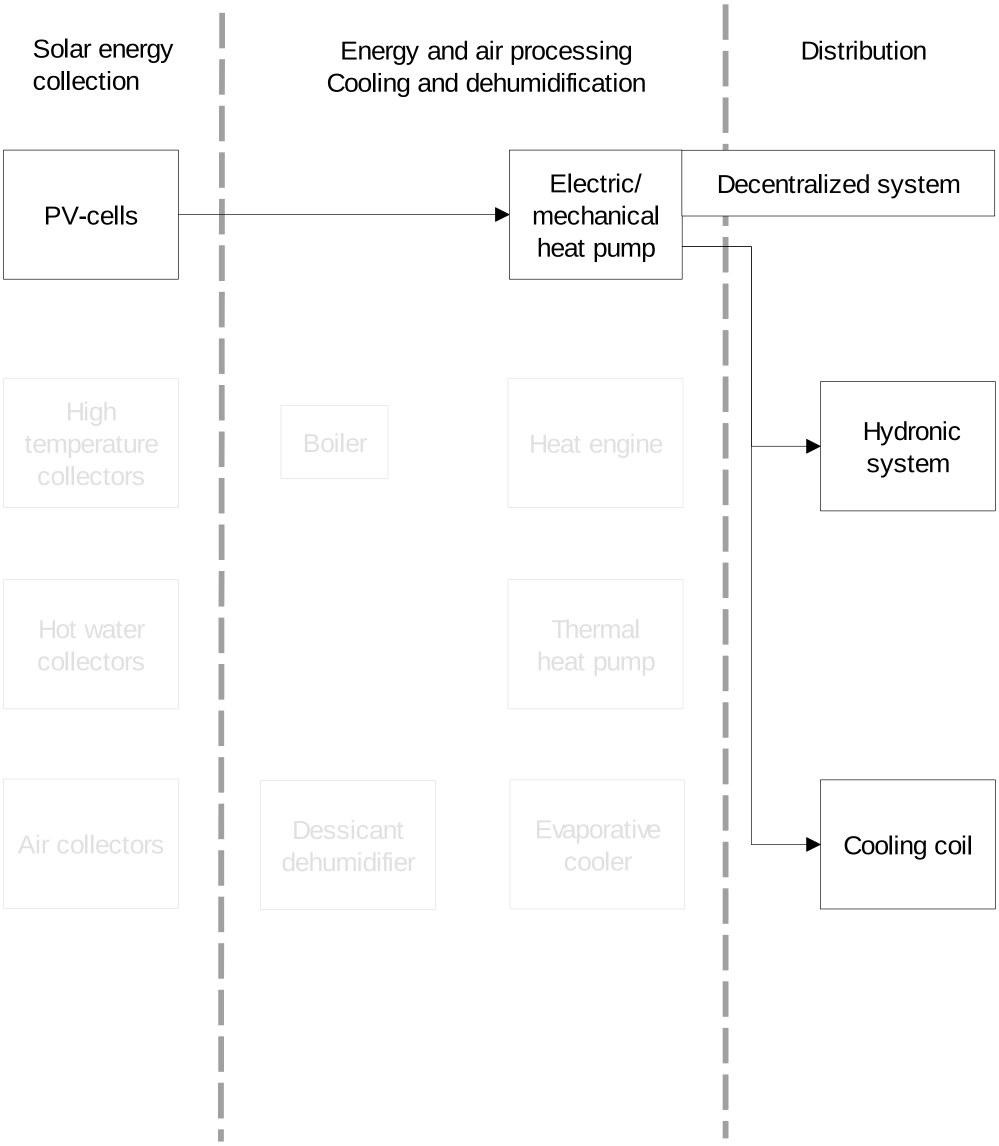


Fig. 3.2 Schematic Photovoltaic-air-conditioning systems

The PV.-air-conditioning system, illustrated schematically if figure 3.2, consists of PV.-cells (section 2.3.1) providing electrical energy to one or several electric heat pumps (section 2.2.1). The heat pump(s) either cools the space directly in a decentralized system (section 2.1.3) or serve cooling coil(s) (section 2.1.1) and sometimes a hydronic system (section 2.1.2).

A DC-to-AC or AC-to-DC converter might be needed, either to convert DC current from the PV.-cells to AC for driving the heat pump, or if a DC-powered heat pump is used, to convert AC grid electricity to DC.

Some advantages of this architecture are that off the shelf products can be used, even for very small systems. COP_{sol} can be rather high (0.25-0.56) and since they often use high COP vapor compression heat pumps their COP_{aux} is equal to the COP of a conventional vapor compression system. Excess power can easily be used in the internal grid or sold to the public grid. Decentralized systems can be used. An already existing conventional system, even if decentralized, can easily be converted to a solar-assisted system by simply adding PV.-cells to the internal grid.

Another advantage, if a vapor compression system is used, is that service for these is easily obtainable worldwide. The PV.-cell does not seem to require much service.

If a thermoelectric module is used the system will be completely without moving parts. However, to date the system will have a very poor efficiency.

The great disadvantage of PV.-air-conditioning is the high cost of the PV.-cells. At current prices this system cannot economically compete with heat driven systems other than for small-scale systems. For small systems they are almost equally expensive (Lindholm 2003b). This seem to be because small hot water or steam driven systems are not commercially available and therefore have to be custom made, while all components of a PV.-cell system are of the shell products. (For example the smallest commercial hot water fired absorption chiller has a capacity of 30 kW.)

The most attractive, where a public grid exists, seems to be to build the system as a solar-assisted system, especially since it would be very expensive to cover the whole load with PV.-cells because of their high cost. If the system would be built as a solar autonomous system some kind of storage, either batteries or cold storage, might be needed. Because a vapor compression system can produce temperatures below 0 °C with rather good efficiency an ice-storage (see section 2.5.2) could be considered.

Henning (2003) writes: In industrialized countries, which have a well-developed grid, the maximum use of photovoltaic is achieved by feeding the produced electricity into the public grid (rather than using it for air-conditioning).

This is worth considering, especially if one plans to install PV.-cell, and at the same time to continue to use electrically driven vapor compression systems for air-conditioning. Then one will in fact use some of the PV.-cell produced electricity that otherwise could have been feed to the grid to drive the air-conditioning.

3.2.2 Heat driven air-conditioning

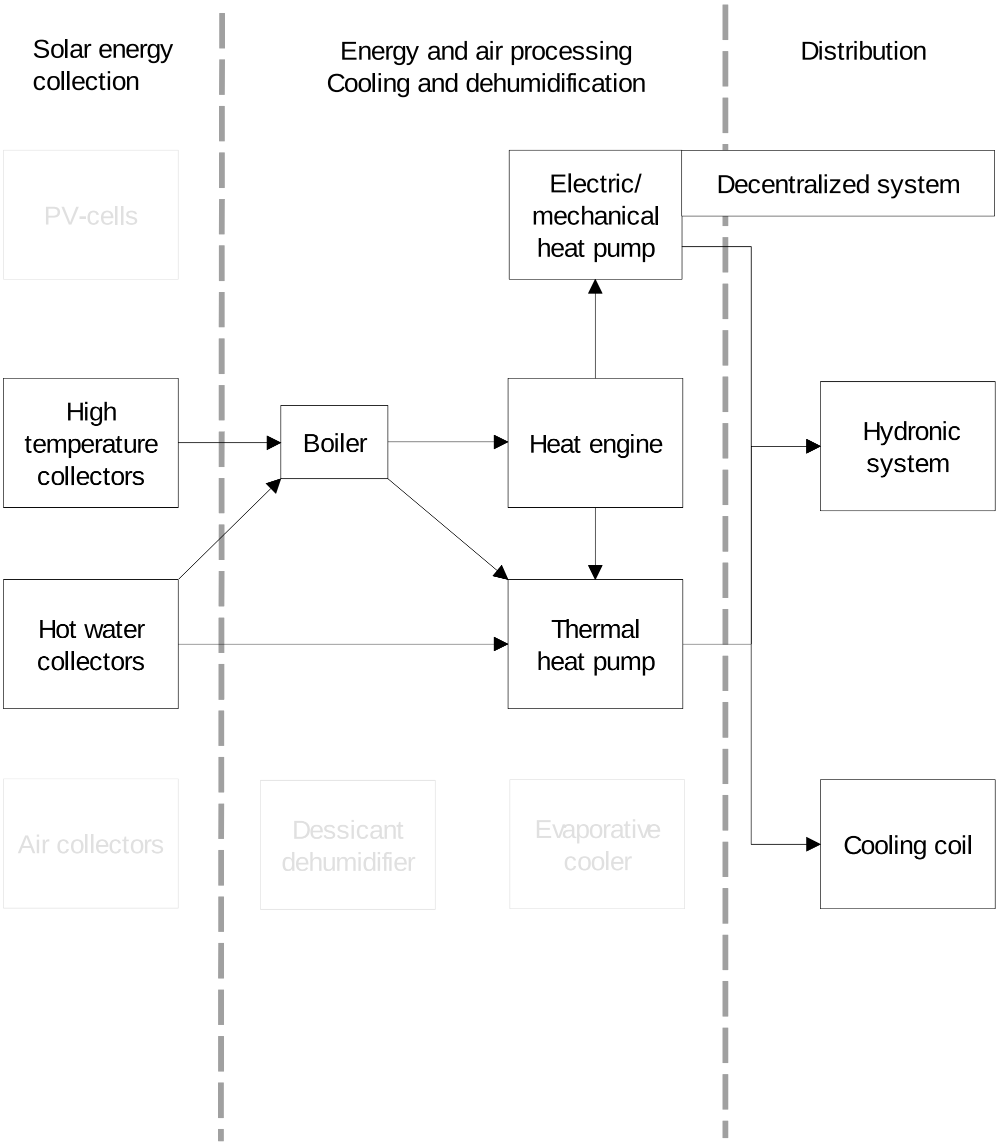


Fig. 3.3 Schematic heat driven solar air-conditioning systems

A solar heat driven air-conditioning system consists, as illustrated schematically in figure 3.3, of solar thermal collectors (section 2.3.2) (high temperature (section 2.3.2.1) or medium temperature hot water (section 2.3.2.2)) and some kind of system for converting the heat provided from the solar collectors to cooling capacity. This system can be either a heat engine (section 2.2.2) driving a mechanical or electrical heat pump (section 2.2.1), a thermal heat pump (section 2.2.3) or both.

An important aspect when coupling the thermal solar collectors to the thermal driven components is that they exhibit a reverse dependence of their efficiency from the operation temperature. The thermal solar collectors are most efficient at low temperature while both heat engines and thermal heat pumps are most efficient at high temperatures. The COP_{sol} will therefore have the characteristic shown in figure 3.4 with an optimum efficiency at some temperature. According to Henning (2003) this behavior requires a sophisticated control system. With an advanced control the actual maximum can be detected and a 'power-point-tracking' implemented, if the cooling power of the machine is sufficient to match the actual load.

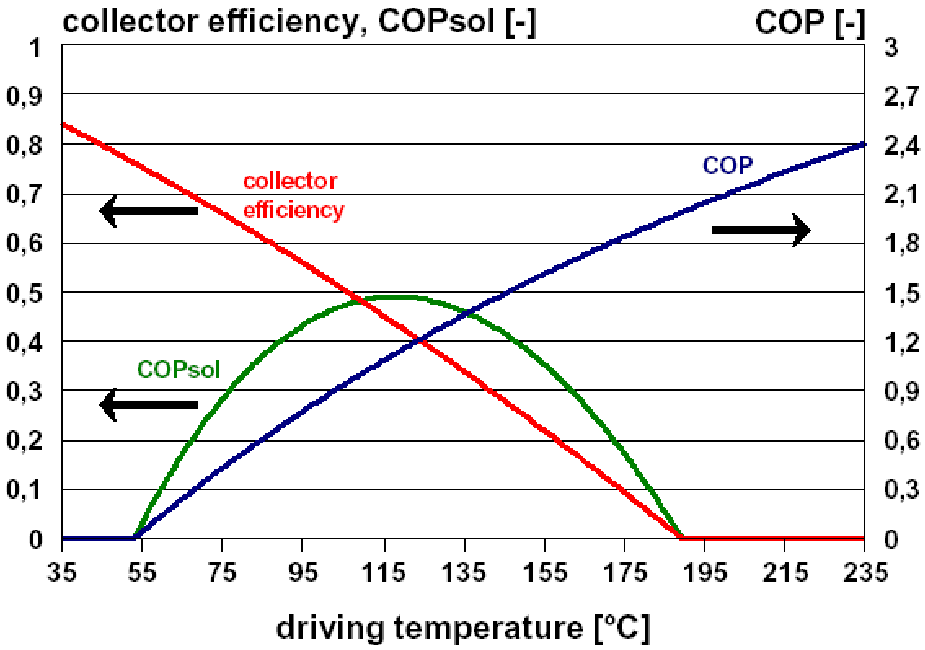


Fig 3.4 Dependence of COP, collector efficiency and COP_{sol} from the operation temperature (Henning 2000©44)

Storage seems to be implemented primary as hot storage or combined hot and cold storage. Heat engine driven systems can use ice-storage. Ice-storage does however not seem suitable for thermal heat pumps since their COP is drastically decreased at low temperature. Further more, many thermal heat pumps cannot reach temperatures below 0 °C.

3.2.2.1 Heat engine driven air-conditioning system

In the heat engine driven system the heat from the solar collectors is first transformed in to mechanical work (with the exception of the thermoelectric generator section 2.2.2.4). This mechanical work can either be used directly to power a mechanical heat pump or be converted into electricity. The electricity is then used to power a electrically driven heat pump. It can also be used to power a decentralized system. Excess electricity can just as in the PV.-cell air-conditioning system be feed to the grid.

Figure 3.5 illustrates a duplex Rankine-cycle (a Rankine power cycle driving a Rankine refrigeration cycle (vapor compression system)) in which an electric generator/motor can be connected to the power cycle, the refrigeration cycle, or both by means of clutches. In this way excess power can be taken out to be feed to the grid and auxiliary power can be feed to the refrigeration cycle. Just as the PV.-cell air-conditioning system this system has a high COP_{aux}.

An other option is to use a furnace as auxiliary power. Then the power cycle can be kept operational at all time and excess electricity sent to the grid.

Some systems using duplex Rankine-cycles have been built and tested. Two are described by Corcoleotes (1982).

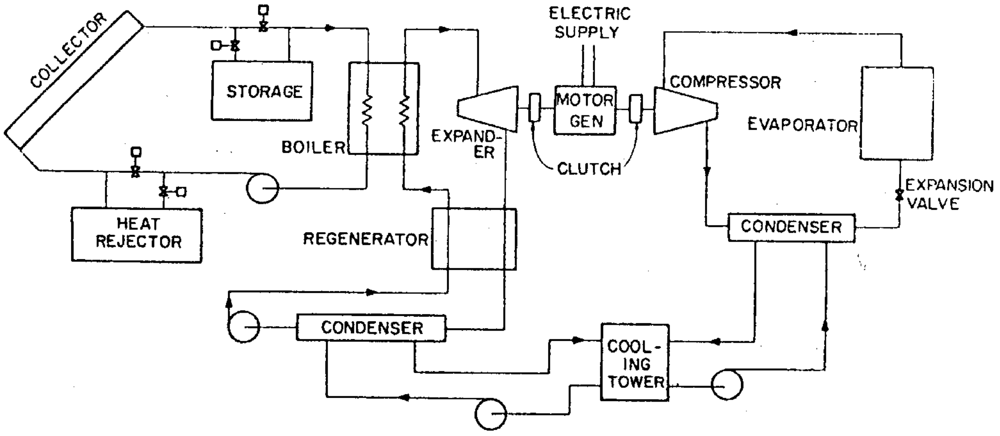


Fig. 3.5 Duplex Rankine cycle solar air-conditioning system (Logee 1983)

Since the system consists of many moving parts it can require rather much maintenance. Because of its complexity it also seem to be rather expensive compared with thermal heat pumps.

3.2.2.2 Thermal heat pump air-conditioning systems

In a thermal heat pump air-conditioning system heat from the solar collectors is used directly to remove heat from the conditioned space by one of the methods described in section 2.2.3.

The most commonly used thermal heat pump is the single effect H_20/LiBr absorption chiller, which seems to be used in the vast majority of all current operational solar air-conditioning systems.

Syed (2002) comes to the conclusion that a single effect H_20/LiBr absorption chiller heated with flat-plate collectors is the most economical option. It is, according to Syed, more economical than a double effect chiller because the heavy additional collector cost of evacuated tube collectors to provide high temperature heat.

Grossman (2002) compares a single effect H_20/LiBr absorption chiller heated with flat plate collectors, a double effect H_20/LiBr absorption chiller heated with flat-plate CPC collectors and a triple effect H_20/LiBr absorption chiller heated with PTC-collectors. He comes to the conclusion than a double effect chiller heated with Flat-plate CPC collectors is the most economical. Even if the collectors are a bit more expensive per m^2, the additional efficiency of the double effect chiller results in a smaller collector field as well as a lower chiller cost per produced cooling capacity. Therefore the first cost of the double effect system is about 3 % less than the first cost of the single effect system. In addition the heat rejection need for a double effect chiller is less than for a single effect and the COP_{aux} of a double effect chiller auxiliary heated with a furnace is much higher than for a single effect chiller auxiliary heated with a furnace.

Al-Marafie (1988) found that the life cycle cost for a solar air-conditioning system based on a H_20/LiBr single effect chiller in Kuwait can be reduced by ~25 % if a solar pond is used instead of flat-plate collectors. This is because the solar pond is simpler, cheaper and removes the need for additional heat storage, as heat storage is integrated into the pond.

In all cases it is important for improving the economy of the system that the collectors are also used off air-conditioning season, for example by using it for providing space and hot water heating.

Because of the COP of the absorption chiller sensitivity to low evaporator temperatures it might be well worth while considering using a separate hydronic system with higher temperature (~15 °C) for removing sensible heat loads. The efficiency could be increase with up to 30 %. This requires an additional system for handling the latent loads. This could be a traditional system with low temperature cooling coils chilled either by a small conventional vapor compression chiller (1/4-1/3 of the total cooling capacity) or by a smaller absorption chiller working at a lower temperature or a desiccant dehumidification system could be used (Olsen 1979).

3.2.2.3 Combined systems

It is also possible to design systems that include both a heat engine and a thermal heat pump. In such a system heat is delivered from the solar collection system to the heat engine where some of the heat is used to produce mechanical work. The rest of the heat, which is rejected from the heat engine, is used to power a thermal heat pump.

For example, heat rejected from a Brayton heat engine (which rejects heat at high temperature,) can be used to drive a double effect absorption chiller. Heat rejected from a Rankine steam engine can be used to drive a single effect absorption chiller.

The mechanical work delivered by the heat engine can either be used to provide cooling together with the thermal heat pump, or it can be used provide electricity to local and public grid, while the thermal heat pump provides all required cooling capacity.

Gordon (2000) has investigated a system, illustrated in figure 3.6, consisting of a cascade coupling of a Brayton heat engine and a double effect chiller heated by a solar mini-dish (central receiver). It uses the mechanical energy to power a vapor compression system that either provides chilled water 5 °C or is used for an ice-storage. The double effect chiller also provides chilled water with a temperature of 5 °C. With the article of Olsen (1979) in mind (see above section 3.2.2.2) one can question if this is optimal, or if the chilled water from the absorption chiller could be dedicated for sensible cooling and thereby raise its temperature. The COP for the investigate cascade couple is found to be 1.4.

Another way to integrate a heat engine and a thermal heat pump has been investigate by Tamm and Goswami (2002). This cycle is not cascade coupled but a hybrid between a Rankine heat engine and NH_3/H_2O absorption chiller. It is intended to be driven by medium temperature heat, around 100 °C. The cycle is illustrated in figure 3.7. The left part of the cycle is the same as a NH_3/H_2O absorption chiller. However, when the NH_3 vapor leaves the rectifier it flows to a superheater and then to a turbine instead of to a condenser. In the turbine mechanical work is taken out. When it leaves the turbine its pressure and temperature has decreased to such a low level that it can be used for refrigeration. The vapor is then absorbed into absorbent solution in the absorber where all heat rejection seems to take place (since there is no condenser and rectifier losses are recycled).

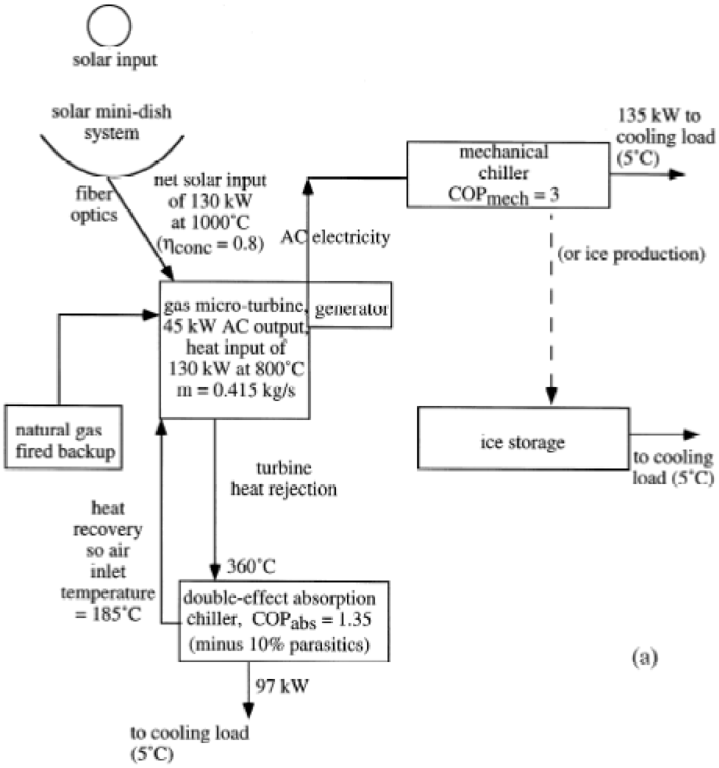


Fig. 3.6 Schematic high efficient solar cooling plant consisting of a Brayton heat engine driving a mechanical vapor compression chiller and a double effect absorption chiller (Gordon 2000[©45])

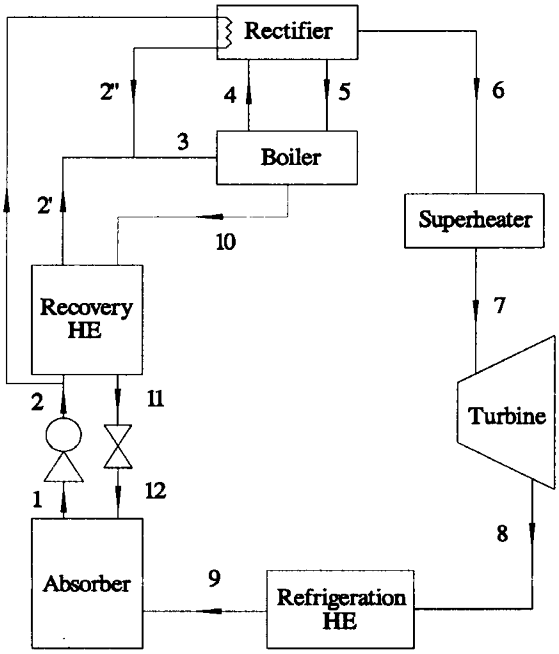


Fig. 3.7 Schematic Rankine heat-engine absorption-chiller cycle hybrid (Tamm 2003)

3.2.3 Desiccant cooling

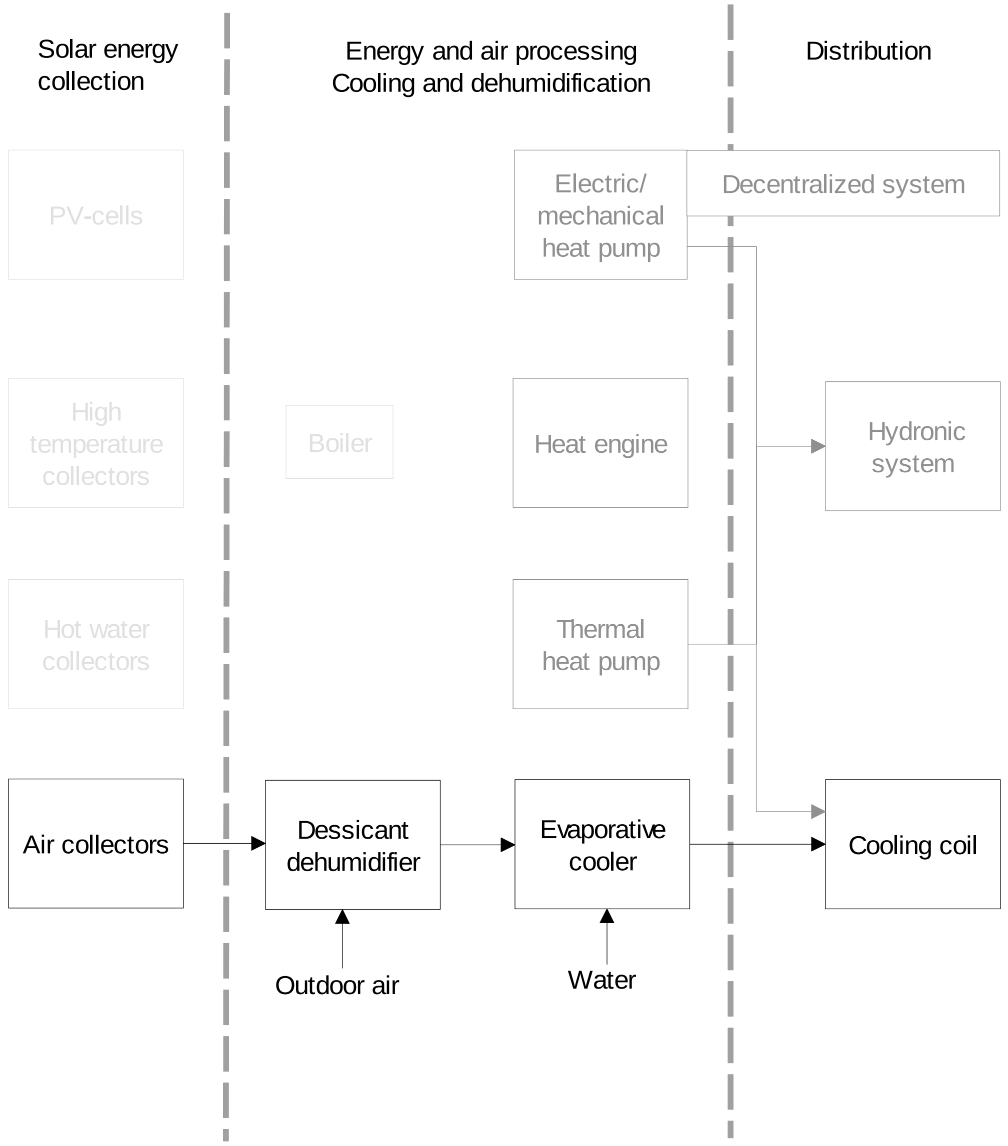


Fig. 3.8 Schematic solar driven desiccant cooling system

In a desiccant cooling system, illustrate schematically in figure 3.8, desiccant dehumidification is combined with an additional cooling system. This may be a conventional cooling coil, or evaporative cooling.

A system using evaporative cooling is called an *all-desiccant cooling systems* (AGCC 1999). The dehumidification unit is used to improve the cooling potential for direct evaporative cooling.

A system using a conventional coiling coil to remove sensible heat is called a *hybrid-desiccant cooling system* (AGCC 1999). The dehumidification unit is used to remove the latent load of the supply air while the coiling coil removes the sensible load.

Figure3.9 illustrates the most common all-desiccant cycle, the so-called Pennington cycle. It works as follows:

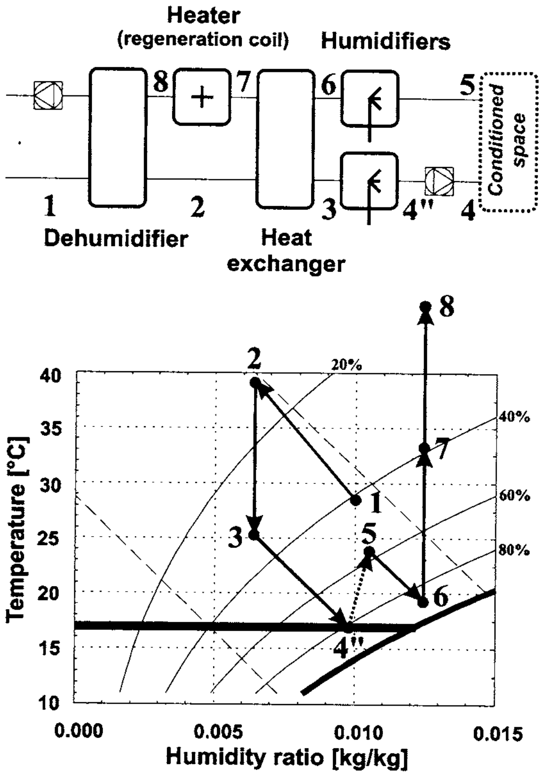


Fig. 3.9 Pennington all-desiccant cycle. Illustrated schematic and in a Mollier chart (Lindholm 2000[©27])

[1-2] Warm and humid air enters the process side of the desiccant wheel from outside the building. The air is dehumidified, latent heat is turned into sensible heat, and in the process its dry-bulb temperature increases (see section 2.2.5.1).

[2-4] The air leaves the desiccant wheel hot and dry. To reduce the dry-bulb of the air it is cooled through indirect and direct evaporative cooling (section 2.2.4.3)

[4-5] As the air passes trough the conditioned space both latent and sensible heat is taken up.

[5-7] The reactivation air first passes the indirect evaporative cooler in which it takes up sensible heat from the process air (see [2-4] above). At [7] it has been heated both latently by humidification [5-6] and sensibly [6-7] by heat transferred from the process air.

[7-8] The reactivation air is heated sensibly to a high enough temperature for reactivation. It then flows through the reactivation side of the desiccant wheel to remove the ab/adsobed water.

In a hybrid cycle the evaporative, especially the direct evaporative cooling, is replaced by sensible cooling by a cooling coil. The air leaving the desiccant dehumidifier can then be more humid. The hybrid system thus separates dehumidification and sensible cooling of the air. Doing this, can as seen above (section 3.2.2.2), be very beneficial to the efficiency of the chiller as its evaporator temperature can be raised . Separating sensible and latent heat removal also seems to make the system easier to control than the all-desiccant system.

Control of the all-desiccant system is complicated by the fact that the evaporative cooler can often not be quickly turned off. An evaporative humidifier continues to cool and humidify the air for a while after the water pump is shut down because of remaining water in the wetted material.

An aspect sometimes seen as an advantage and sometimes as a disadvantage is that the desiccant system described above uses 100 % outdoor air. This can be seen as an advantage because the ventilation of the conditioned space is improved, however, it also means that more outside air needs to be dehumidified and cooled than if some of the air can be recirculated.

A way to circumvent this is to use a dedicated outdoor air system (DOAS), which only provides for ventilation and removal of latent heat loads. Sensible heat loads are handled by a separate system, for example a hydronic system. For the DOAS system desiccant cooling is used. A hybrid-desiccant system might be the best choice since the supply air of an all-desiccant system is normally rather humid because of the direct evaporative cooling. Since only enough air for ventilation is provided the airflow is rather small and it might need to be very dry to handle internal latent loads.

In warm humid climates it can be difficult to achieve supply air with low enough temperature and humidity with an all-desiccant system.

In a hybrid system the heat rejected from the chiller can be used to regenerate the desiccant. There are commercial systems that use heat rejected from vapor compression chillers to regenerate the desiccant. There have also been some studies on using heat rejected from absorption chillers for regeneration. Wilkinson 1988, 1990 and 1991, Gari 1990, Fathala 1996 and Edwards 1999 have done such studies.

To solar heat the reactivation air in an Pennington cycle flat plate or hot air solar collectors can be used. Pesaran and Wipke (1994) investigates whether unglazed transpire air solar collectors can be used. They come to the conclusion that it is possible and that the collectors of a system using these collectors costs 45 % less than glazed air collectors, still the system does not seem to be able to compete economical relative to a gas-fired system.

When air collectors are used ambient air is sometimes used instead of exhausts air for reactivation. Advantages of using ambient air is that the need for ductwork is reduced and that ambient air in some cases can be less humid than the exhaust air, which has been humidified in the indirect evaporative cooler.

If unglazed transpire air solar collectors is used ambient air is always used for reactivation (because of the collectors construction).

The liquid desiccant systems, basically working on the same principles as the Pennigton cycle, generally require a lower reactivation temperature. Since this improves collector efficiency some (for example Grossman (2002)) see them as more suitable than solid desiccants systems. In a solar-driven liquid desiccant system the desiccant can be regenerated in a closed single or multiple effect generator (of the same type as used in absorption chiller section 2.2.3.1) or a flow through open generator (section 2.2.3.3) heated with hot water collectors. It could also be regenerated in an open or closed solar generator (section 2.2.3.3).

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Appendix A

Concepts

Mechanical Work and *Heat* are two forms of energy. Mechanical work is considered to be a higher form of energy than heat since all mechanical work can be converted to heat, but not all heat can be converted to mechanical work.

Vapor is gas at a temperature below the critical temperature, so that it can be liquefied by compression, without lowering the temperature (AccesScience 2003).

Partial pressure is the pressure that would be exerted by one component of a mixture of gases if it were present alone in a container (AccesScience 2003). Dalton's law gives the total pressure of a ideal gas mixture as the sum of its components partial pressure (Gordon J. 1985).

Vapor pressure, (saturation vapor pressure, or *saturation pressure)* defines the pressure that is exerted by a vapor when the vapor is in equilibrium with a liquid or solid form, or both, of the same substance (Encyclopedia Britannica online 2003). The saturation vapor pressure of a substance is a function of the temperature. The liquid substance can be part of a solution, the vapor saturation pressure then also depends on the concentration of the substance in the solution. Note that the definition above is the thermodynamic definition of the vapor pressures. There is also a meteorological definition of the term vapor pressure that is: The partial pressure of water vapor in the atmosphere (AccesScience 2003).

A system in which only one substance exists as vapor and in which the saturation vapor pressure of the substance equals the absolute pressure, is in balance. The same applies for a system consisting of several vapors and in which the partial pressure of every vapor equals its saturation vapor pressure.

If the absolute pressure of a system is decreased, or the system is heated, so that the saturation vapor pressure of the substance exceeds that of the absolute pressure of the system, then the balance is broken. To reestablish the balance matter will go from liquid phase to vapor. This is called *evaporation* through *boiling.* The temperature at which this occurs at a given pressure is called *boiling point.*

Evaporation requires heat, this heat is called *heat of evaporation* or *latent heat (of evaporation).* If it is not added from an external source, it will be taken from the *sensible heat* of the system. The sensible heat of the system is proportional to the temperature of the system. The temperature of the system will therefore decrease if its sensible heat is reduced.

If the system is sealed in a container, the absolute pressure of the system will increase as the amount of vapor in the container increases. Note that if the unbalance was caused by a pressure reduction (for example due to volume increase) this pressure increase will be less than the initial pressure decrease. The total absolute pressure change will thus be negative.

The evaporation will, as long as there is some substance left in its liquid phase, continue until the saturation vapor pressure of the substance and the absolute pressure of the system is equal. The pressures (return to) be equal because the substance's saturation vapor pressure has decreased (because temperature or concentration of the substance has changed), the absolute pressure of the system has increased, or a combination of both.

If the absolute pressure of a system is increased, or the system is cooled, so that the saturation vapor pressure of the substance becomes below the absolute pressure of the system, then substance will go from vapor to liquid phase. This is called *condensing* and the temperature at which this occurs at a given pressure is called *dew point (temperature).*

When vapor condenses, the latent heat (or heat of evaporation) of the vapor is released. If this heat is not removed to an external heat sink, the heat will remain in the system as sensible heat and the temperature of the system will increase.

If the system is sealed in a container, the absolute pressure of the system will decrease as the amount of vapor in the container decreases. Note that depending on the reason for the unbalance, (similar to above) this does not mean that the final pressure will necessary be lower than the original pressure. If, for example, the reason for the unbalance was a volume reduction of the system the final pressure will still be higher than the original.

Condensation will continue until the saturation vapor pressure of the substance and the absolute pressure of the system is equal, because the saturation vapor pressure of the substance has increased, because the absolute pressure has decreased or a combination of both.

If a liquid or solid substance is in contact with a gas mixture and the absolute pressure is higher than the saturation vapor pressure of the substance, boiling can per definition not occur. However, if the partial pressure of the specific substance in the gas mixture is lower than the saturation vapor pressure of the substance, then the substance will evaporate **from the surface** of the liquid or solid (where it is in contact with the gas mixture). In the same way if the partial pressure of the substance in the gas mixture is higher than the saturation vapor pressure of the substance, then vapor will condense **on the surface** of the liquid or solid phase of the substance. If the partial pressure of a substance in a gas mixture rises to above its saturation vapor pressure it will condense even if not in contact with any substance in liquid or solid phase. It will condense on a surface, as fog (liquid) or desublimate as snow(solid). Like above, heat of evaporation will be required or released during these processes.

The ratio [partial pressure]/[saturation vapor pressure] is called *relative vapor pressure* and is denoted φ. In metrology it is often referred to as *relative humidity* denoted Rh. It is commonly used together with temperature to specify the state of air. (Ex 30 °C, 70 %)

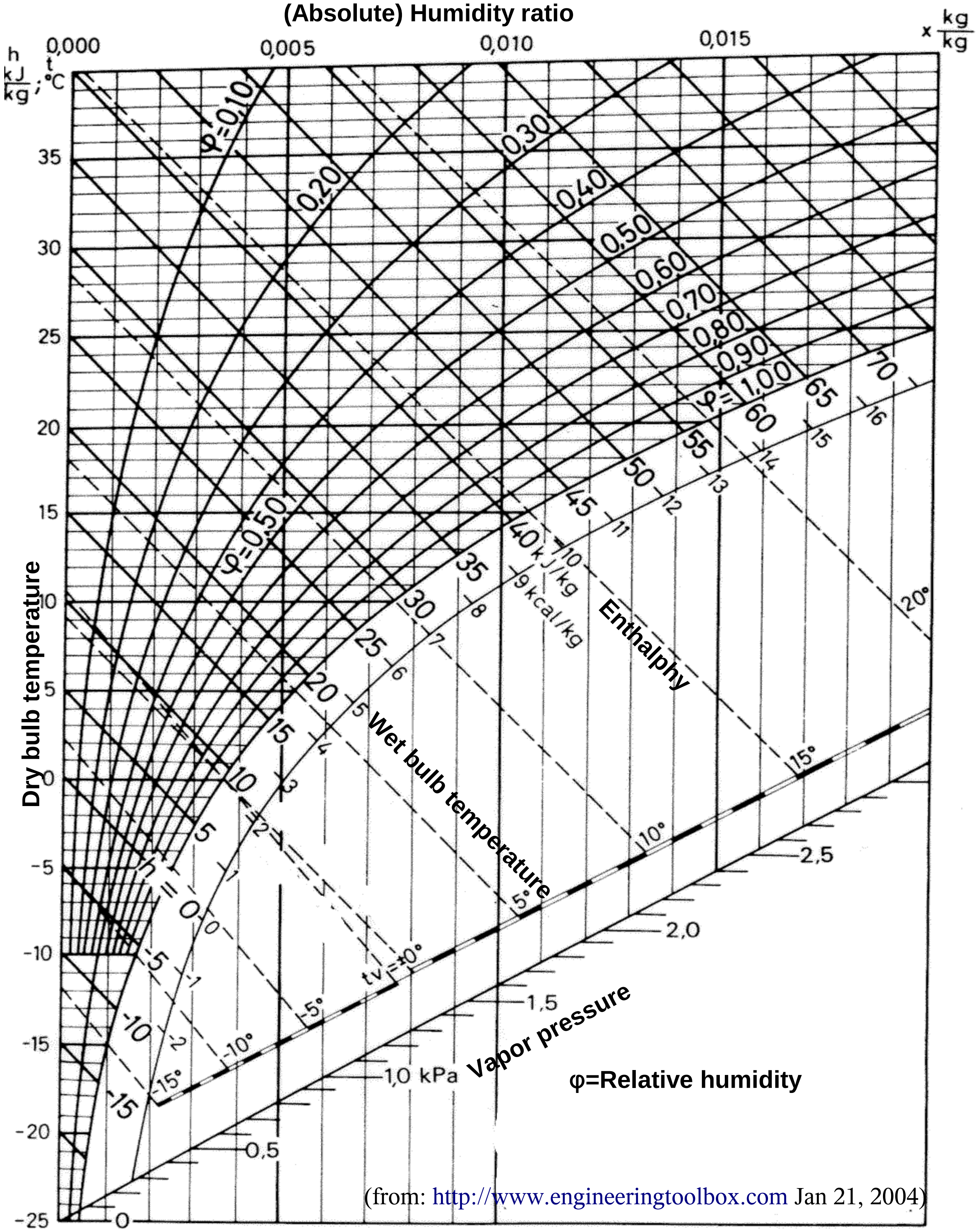
When the absolute pressure of a vapor is below the saturation vapor pressure of the vapor it is called a s*uperheated vapor.* When the absolute pressure of a liquid is above the saturation vapor pressure of the liquid it is called a *subcooled liquid.* When the saturation vapor pressure of a substance in a gas mixture is equal to its partial pressure the gas mixture is said to be *saturated* with this substance. Likewise if the saturation vapor pressure of a solution is equal to its absolute pressure the solution is called saturated. In both cases it means that they cannot take up more of the substance in the mixture or solution. (Ex. Saturated air cannot take up more water from a wet surface.)

The approach of working with partial pressure versus saturation vapor pressure gives a considerably accurate model, when the following can be assumed (Gordon J 1983):

1. The solid or liquid phase contains no dissolved gases.
2. The gaseous phase can be treated as a mixture of ideal gases.
3. When the mixture and the condensed phase are at a given pressure and temperature, the equilibrium between the condensed phase and its vapor is not influenced by the presence of the other components. This means that when equilibrium is achieved the partial pressure of the vapor will be equal to the saturation (vapor) pressure corresponding to the temperature of the mixture.

Appendix B Mollier and Psychrometric charts

Mollier chart



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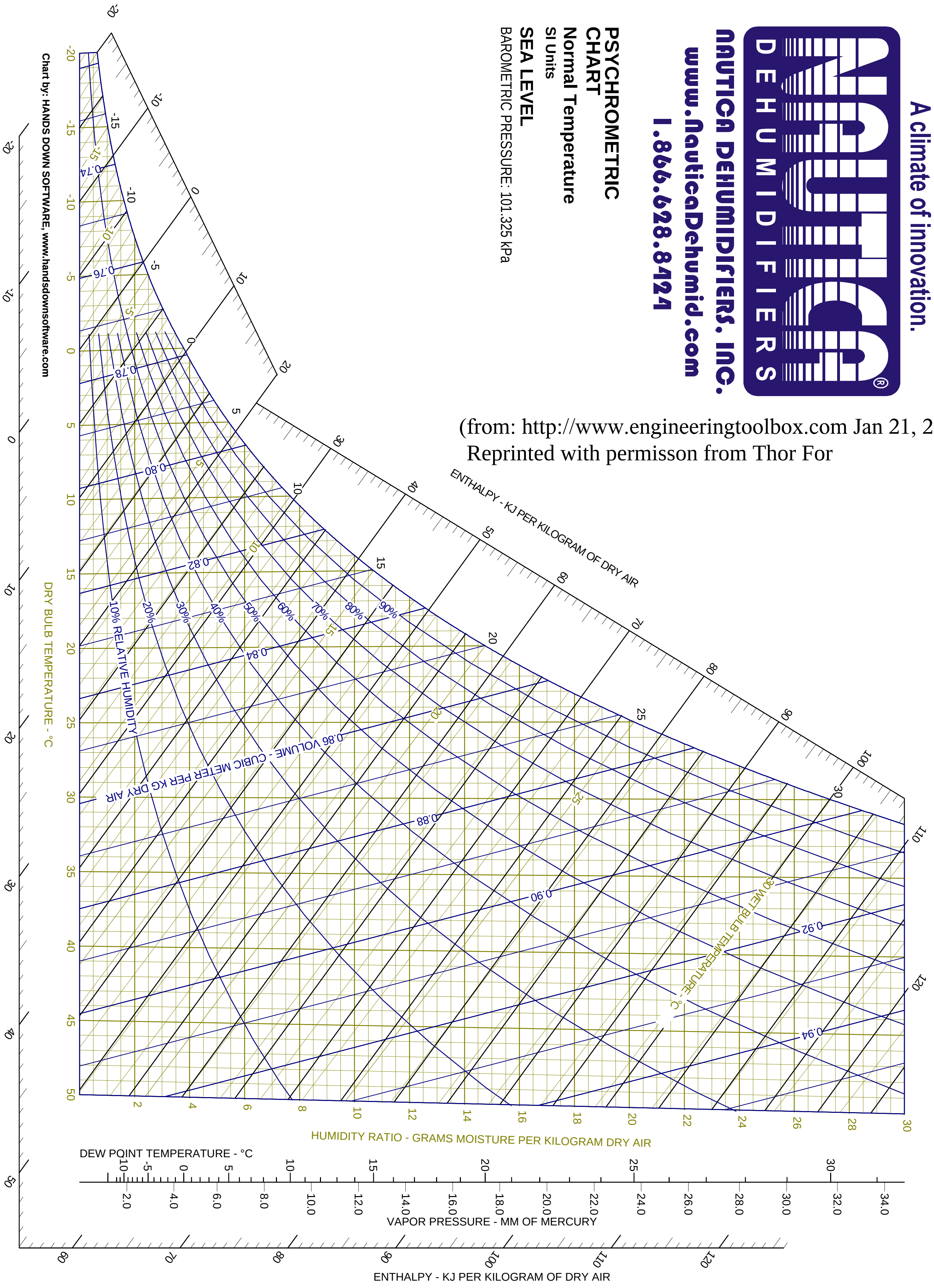

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How to transform between the psycrometric chart and the Mollier chart
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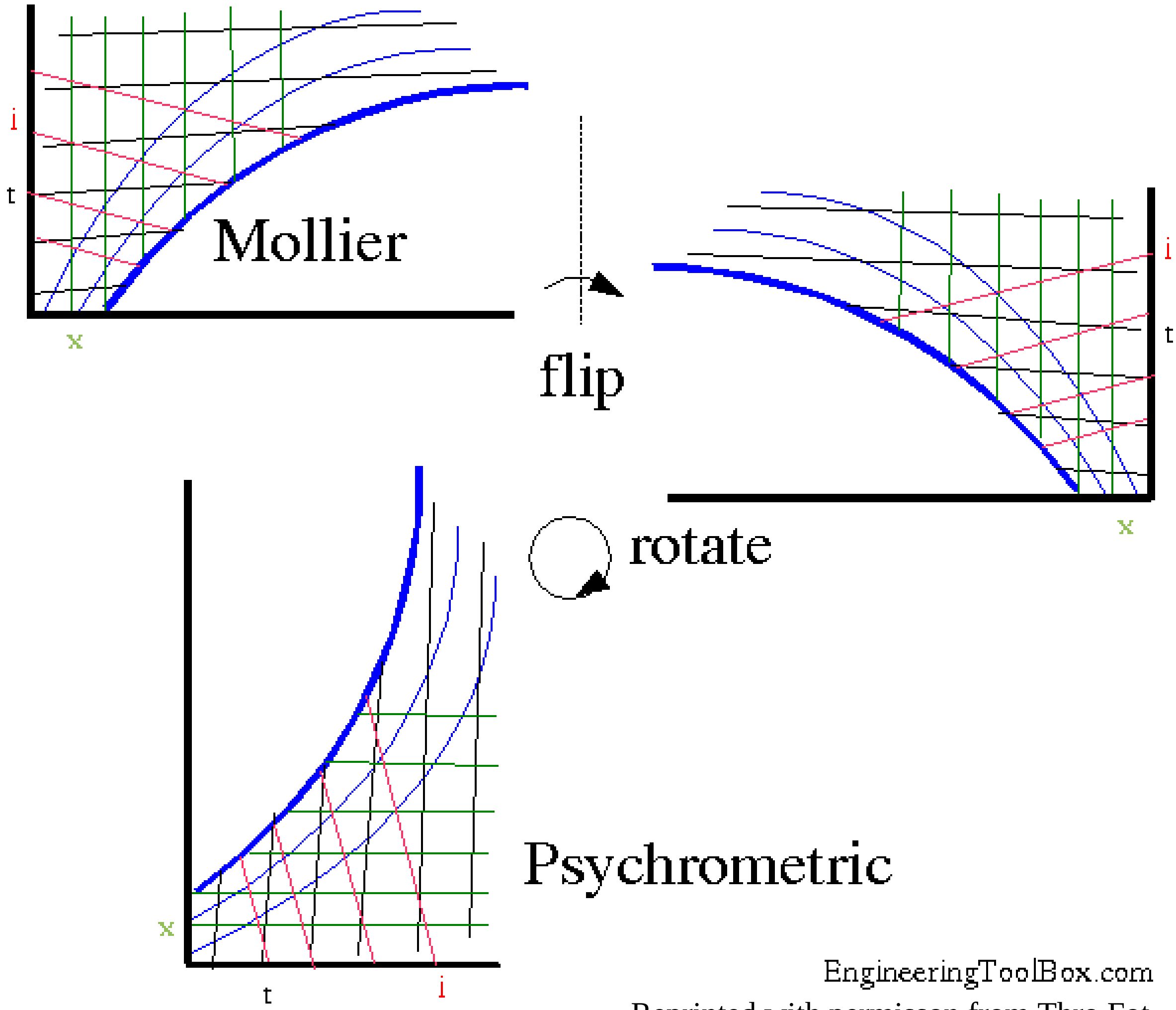
The Mollier diagram is the same as the Psychrometric Chart. It can be transformed by first be reflected in a vertical mirror and then rotated through 90 degrees.



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Appendix C

COP and volume of refrigerant vapor per cooling power for ejector-chiller for some different refrigerants.

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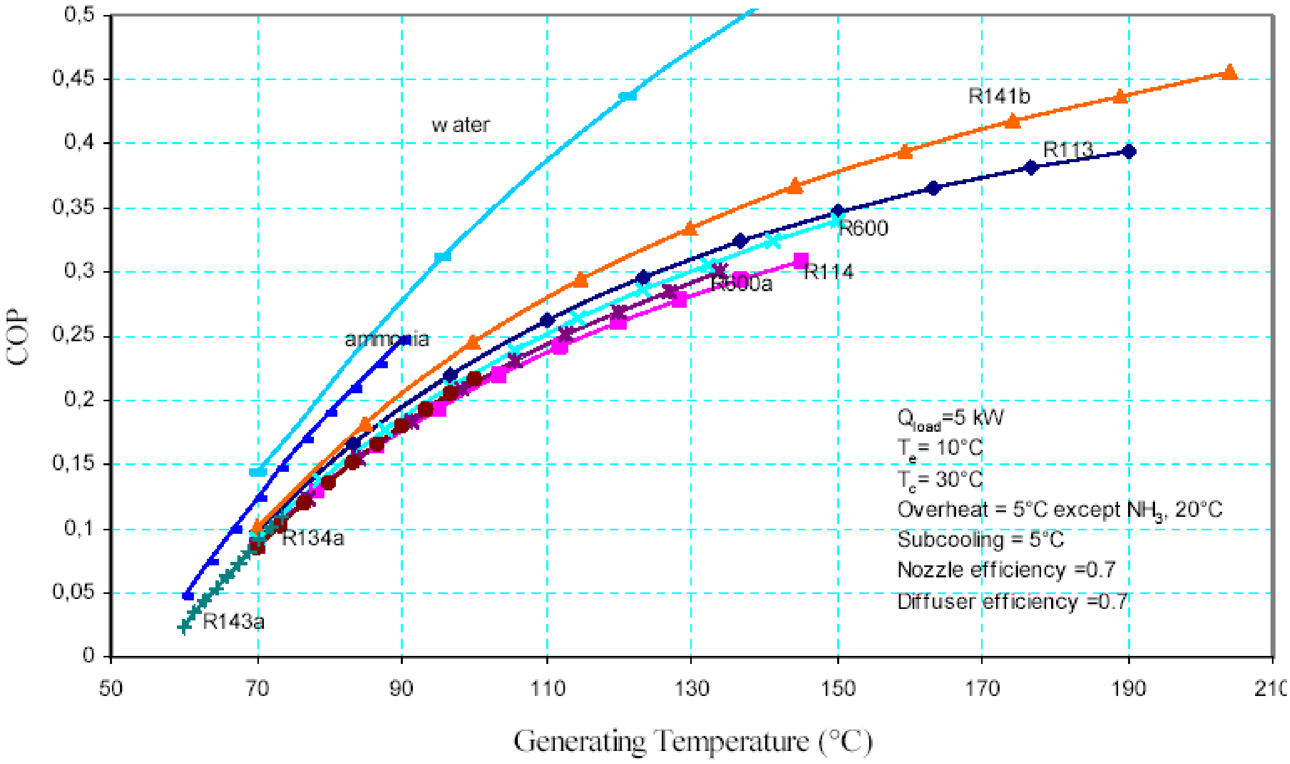

Fig. C1 COP of an ejector refrigeration cycle at refrigeration load 5 kW and condensing temperature 30 °C

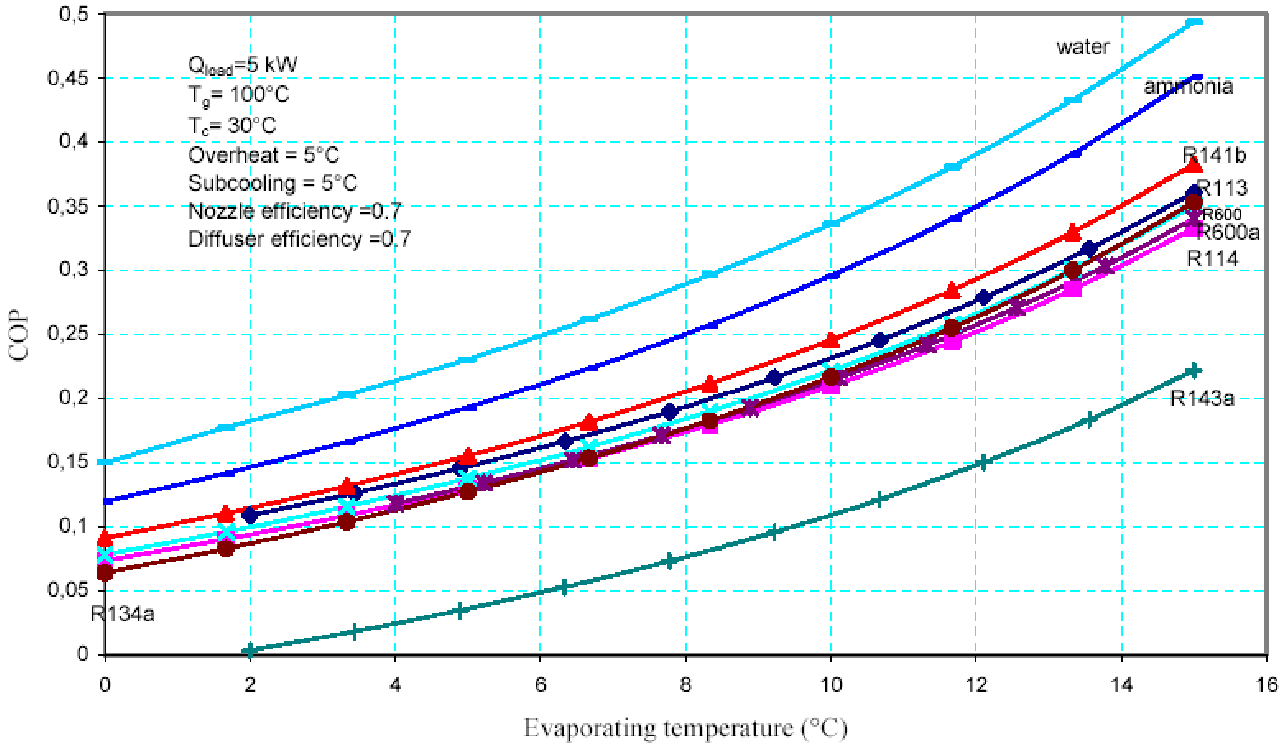

Fig. C2 Effect of evaporating temperature to COP

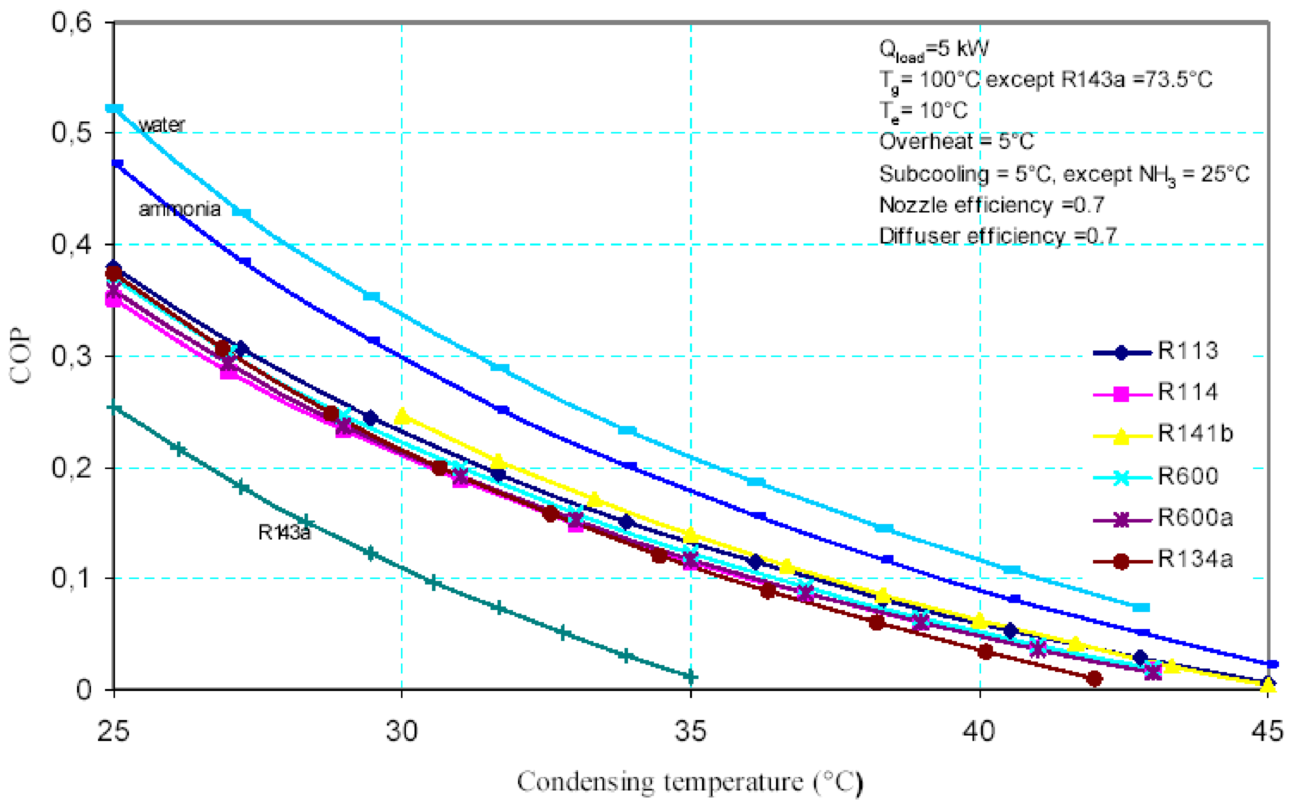

Fig. C3 Effect of condensing temperature to COP

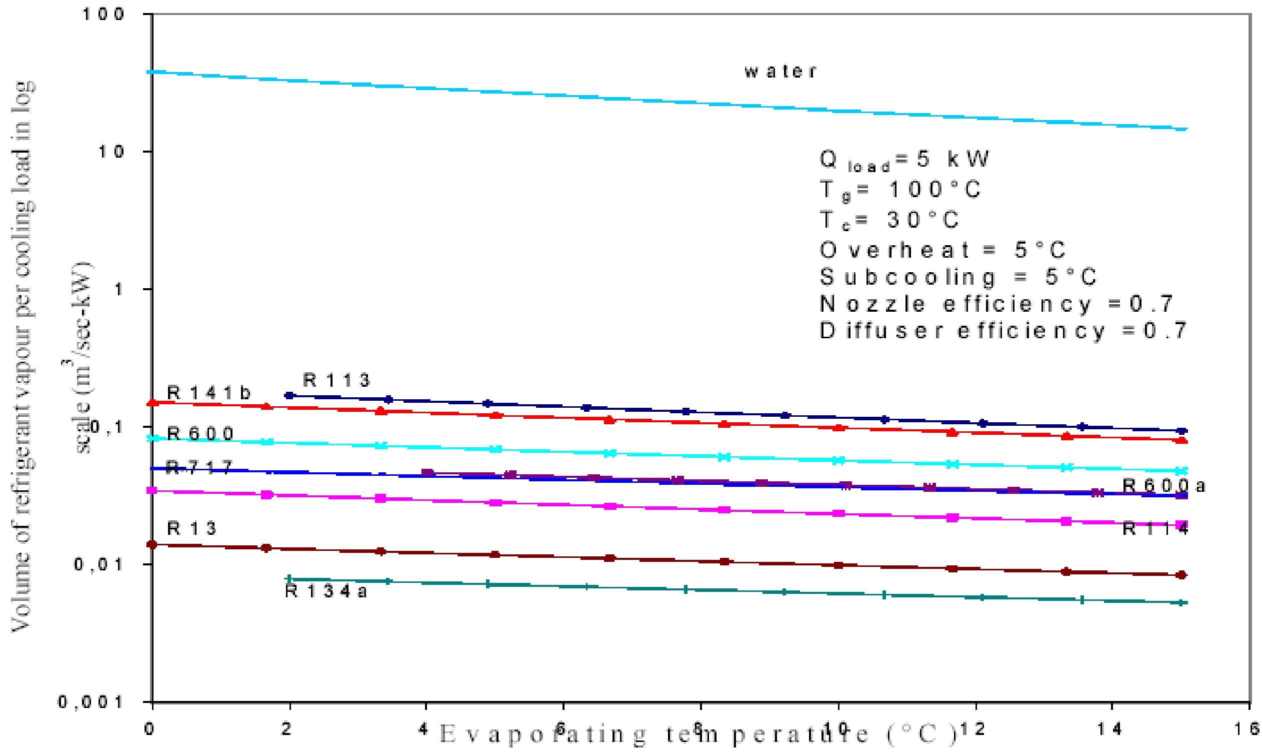

Fig. C4 Volume of refrigerant vapour per cooling power at different evaporating temperature in a logaritm scale.

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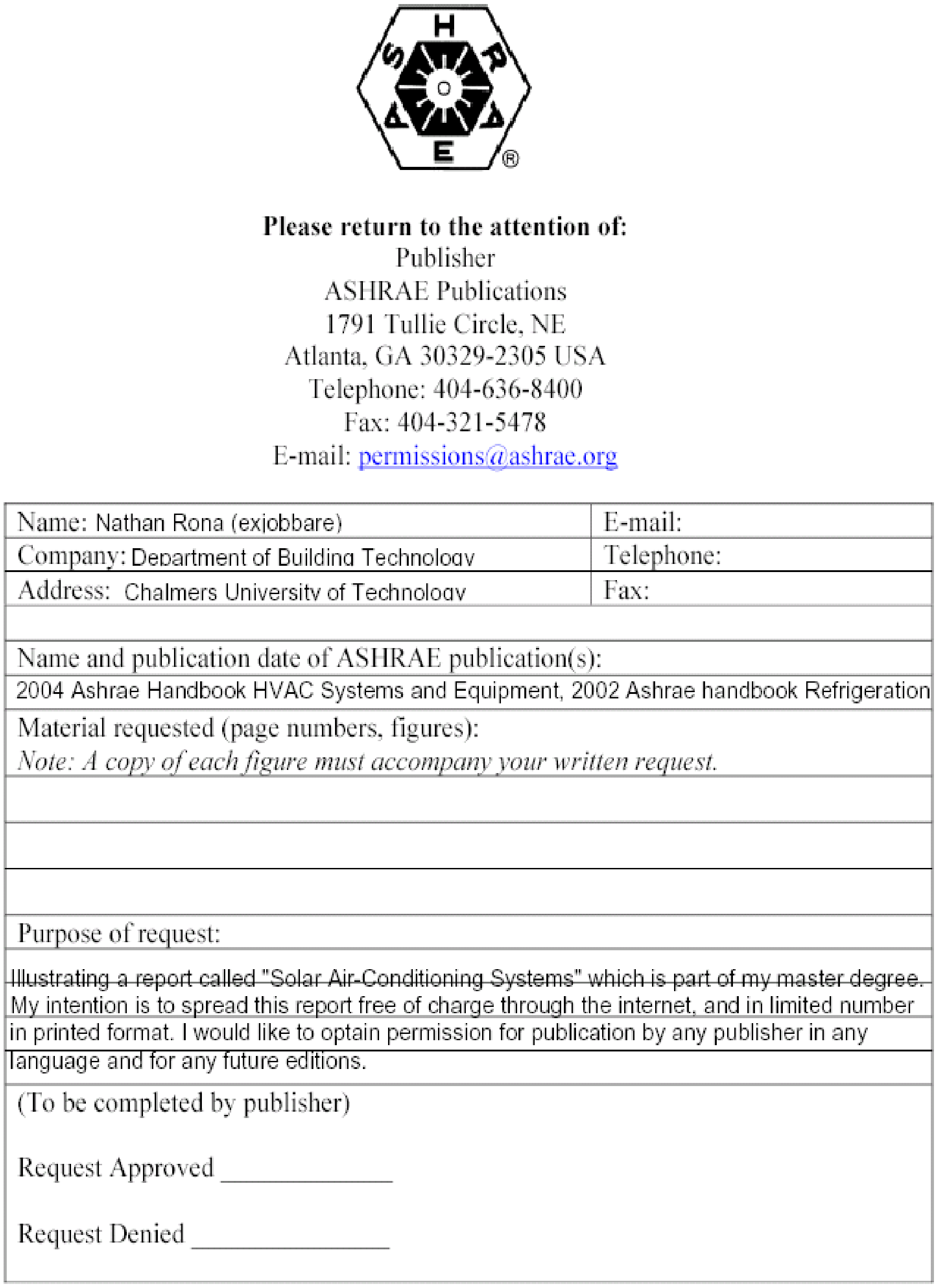

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