Thermal Energy Storage for Sustainable Energy Consumption

Fundamentals, Case Studies and Design

Edited by

Halime Ö. Paksoy

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Sustainable Energy Consumption

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PREFACE

We all share a small planet. Our growing thirst for energy already threatens the future of our earth. Fossil fuels—energy resources of today—are not evenly distributed on the earth. 10% of the world's population exploits 90% of its resources. Today's energy systems rely heavily on fossil fuel resources which are diminishing ever faster. The world must prepare for a future without fossil fuels. Sustainable energy consumption—thereby maintaining continuity and security of energy resources—has become urgent matter for all countries. Concerns over an imminent climate change have increased as a consequence of global warming and many recent storms, "heat waves," as well as "black-outs" experienced around the world.

Thermal energy storage provides us with a flexible heating and/or cooling tool to combat climate change through conserving energy and increasing energy efficiency. Utilizing local and renewable sources can be greatly improved and made more productive when coupled with thermal energy storage systems. Thermal energy storage can also be used to level out both diurnal and seasonal peaks occurring in energy demand curves, especially those which result from cooling demand.

Thermal storage applications have been proven to be efficient and financially viable, yet they have not been exploited sufficiently. For widespread applications of properly designed and high-efficiency thermal energy storage systems, one must consult the scientific and technical expertise of scientists and engineers in this field. There is an urgent need for an educational opportunity to serve this purpose.

Çukurova University, Turkey, in collaboration with Ljubljana University, Slovenia, and the International Energy Agency Implementing Agreement on Energy Conservation Through Energy Storage (IEA ECES IA) has organized this *NATO Advanced Study Institute on "Thermal Energy Storage*" held in Cesme-Izmir, Turkey, on June 6–17, 2005 (http://www.tessec.org) to fulfill this purpose.

Eminent experts who have worked in a number of Annexes of IEA ECES IA were among the lecturers of this Advanced Study Institute. 24 lecturers from Canada, Germany, Japan, Netherlands, Slovenia, Spain, Sweden, Turkey, and USA have all enthusiastically contributed to the scientific programme. In Çeme, Turkey, 65 students from 17 countries participated in this 2-week summer school. This book contains the manuscripts prepared based on the lectures given during the summer school. I am most grateful to the contributors.

I also appreciate the support given by the International Scientific Committee members, Professor Lynn Stiles, Dr. Luisa Cabeza and especially Dr. Uros Stritih who was also the Co-director.

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PART I. INTRODUCTION

1. HISTORY OF THERMAL ENERGY STORAGE

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Abstract. This chapter discusses the history of thermal energy storage focusing on natural energy sources. Links are made to recent trends of using renewable energy to achieve greater energy efficiencies in heating, cooling and ventilating buildings. The Deep Lake Water Cooling development in Toronto is presented as a typical modern interpretation of past practices with an integration of municipal services of water supply and district cooling. Environmental concerns and restrictions have also stimulated thermal energy storage developments. Cold storage in aquifers originated in China where excessive groundwater extraction related to industrial cooling had resulted in significant land subsidence. To rectify the subsidence problem, cold surface water was injected into the aquifers. Subsequently, it was observed that the injected and "stored" water had maintained its cool temperature for months and was suitable for industrial cooling. Thus, aquifer thermal energy storage (ATES) was born. The Netherlands restricted groundwater mining for industrial cooling but left an exemption if reinjection using ATES for cooling were implemented. This stimulated interest in ATES and led to many implemented projects. In some areas such as Winnipeg, Manitoba, the natural groundwater temperature (6 °C) is suitable for direct cooling. The reinjection of warm waste energy results in a gradual warming of the aquifer, ultimately leading to lower system efficiency for cooling. Using the aquifer for ventilation air preheating in winter helped the WINPAK plant maintain the natural groundwater temperature. Ground source technologies combined with underground thermal energy storage are seen as the best current method of combining natural energy sources with modern energy efficient building design. The latest technical findings have been incorporated into codes, standards and guidelines. Some of these are briefly described. Storing freely available energy to meet the requirements of a later season is "seasonal storage". Three principal stimuli to the development of large-scale seasonal energy storage are: (1) the decoupling of electricity and heat production in cogeneration plants with heat storage increasing the fraction of the annual heat demand met by cogeneration; (2) seasonal storage-assisted central solar heating plants to enable solar energy to supply winter heating demands; (3) the storage of ambient winter air temperatures for summer cooling. Thermal storage associated with cogeneration

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and district heating is a standard application. Central solar heating plants have been investigated, constructed and monitored as part of the International Energy Agency, Task VII of the Solar Heating and Cooling Implementing Agreement "Central solar heating plants with seasonal storage". The storage of ambient winter air temperature is particularly suited to continental climates characterized by long cold winters with brief hot summers. Ice and snow are practical latent energy storage media for cold winter air. Snow and ice may be fabricated or gathered from natural sources. In larger commercial buildings, particularly those of energy efficient design, the energy expended for cooling can be a major proportion of the total energy requirement. This combination of a suitable ice-making climate with significant building cooling demands stimulated interest in seasonal thermal energy storage. Various design alternatives were investigated, tested, evaluated and demonstrated. These efforts originated in the USA and Canada but now have been applied successfully in Sweden.

Keywords: ATES; aquifer thermal energy storage; borehole thermal energy storage; BTES; building cooling; chiller; district cooling; hypolimnion water; ice; ice storage; lake water; PCM; phase change materials; seasonal energy storage; snow; thermal energy storage; TES; underground thermal energy storage; UTES.

1.1. Introduction

The history of thermal energy storage is a rich tale dating back to ancient civilizations. It is based on natural sources of energy complemented by human ingenuity. These natural sources include ambient air, sky, ground, and the evaporation of water with the storage materials of building mass, rocks, water, ground and phase change materials. Heat transfer includes radiation, convection and evaporation. This limited review of some historical themes may assist in current attempts to reduce energy intensities by taking greater advantage of natural energy sources.

Perhaps the oldest form of energy storage is the harvesting of natural ice or snow from lakes, rivers and mountains for food preservation, cold drinks and space cooling. The following extract from 350 years ago illustrates the popularity of ice in Persia:

> They use abundance of Ice in Persia, as I have been observing; in Summer especially every one drinks with Ice: But that which is most remarkable, is, That tho' at Ispahan, and even at Tauris, which is further North, the Cold is dry and penetrating more than it is in any part of France or England, yet the greatest part of the People drink with Ice as well in the Winter as the Summer.

> Ice is sold in the out-parts of the City in open places: Their way of making it is thus . . . In less than eight Days Working after this Manner, they have Pieces of Ice five or six Foot thick; and then they gather the People of that Quarter together, who with loud Shouts of joy, and Fires lighted upon the Edges of the Hole, and with the Sound of Instruments to Animate them, go down into it, and lay these Lumps of Ice one upon the other [1].

Some regions of Japan (Kyoto, Kanazawa and Kusatsu) still celebrate the Himuro tradition as the Ice House Festival on June 1. Natural ice was collected and stored in Himuro, an icehouse, for use in the summer. It is thought the ice was used for cold drinks and to relieve the summer heat. The townspeople of Kusatsu collected and stored the precious ice in Himuro, a kind of icehouse, for the summer. It was customary to offer guests some of the natural ice with a decorative rhododendron flower. This frozen offering was thought to ward off sickness and disease for the entire year.

Before mechanical refrigeration systems were introduced, people cooled their food with ice and snow, found or made on-site or gathered in the mountains. This practice survives today in the Taurus Mountains of Turkey. Ice was stored in icehouses usually partially buried in the ground and lined with straw or sawdust. Remains of these structures survive on many farms in Europe and North America. Ice has long been used for space comfort conditioning. In the early nineteenth century, ice was placed in air ducts to cool and dehumidify warm air blown by fans.

Conventional building HVAC design focuses on the peak load conditions to enable equipment sizing and the system efficiencies. The total energy usage determines energy costs but is not a major risk factor. But energy storage design must consider both the peak delivery of energy and the total amount of energy that can be stored. This is especially true in some recent applications in the Netherlands that have no conventional backup.

The first ground-source heat pump was installed in Indianapolis, Indiana in the home of Robert C. Webber, an employee of the Indianapolis Power and Light Company. It was a 2.2 kW compressor hooked to a direct expansion ground coil system in trenches supplying heat to a warm air heating system. The installation was monitored beginning October 1, 1945 and this is the first day of ground source heat pump operation documented in literature [2].

Although groundwater cooling has a long history, the deliberate storage of cold water in an aquifer for later use or Aquifer Thermal Energy Storage (ATES) has a history of about forty years. It originated in China where excessive groundwater extraction related to industrial cooling had resulted in significant land subsidence. To rectify the subsidence problem, cold water (from surface water) was injected into the aquifers. Subsequently, it was observed that the injected and "stored" water had maintained its cool temperature

Figure 1. Scarborough Town Centre: first building ATES combined heating and cooling in Canada.

for a long period of time and was suitable for industrial cooling. The development of ATES in North America and Europe focused on the independent storage of cold and heat energy, both through experimentation and industrial applications. Environmental impacts related to aquifer warming, as well as the need for both heating and cooling, called for a technology advancement that would allow the effective storage of both warm and cold energy at different times of the year. The first application in Canada of a combined heating and cooling ATES for a new building was at the Scarborough Centre [3] building of the Government of Canada (Figure 1).

In many areas, the natural groundwater temperature is suitable for direct cooling. For example, in Winnipeg, Manitoba, the natural groundwater temperature is 6 °C. However, the reinjection of warm waste energy may result in a gradual warming of the aquifer, ultimately leading to aquifer degradation and lower system efficiency for cooling. ATES can avoid the gradual warming of the aquifer by using the waste heat for ventilation air preheating in winter.

1.2. Seasonally-Charged Deep Lake Water Cooling (DLWC) For Downtown Toronto [4]

Toronto is located on the northern shore of Lake Ontario. This lake contains a large volume of seasonally replenished 4°C water some six kilometers from shore below the 80-m level in the hypolimnion layer. Toronto is the largest metropolitan area in Canada with a downtown core of high rise buildings adjacent to the lake. This downtown core has a large cooling requirement (720 MW thermal) supplied primarily by on-site vapour compression chillers. The Deep Lake Water Cooling (DLWC) project will tap this lake water for cooling downtown Toronto buildings through a district cooling network. The district cooling network has been constructed and the DLWC implemented beginning in 2001–2002. Extending the municipal water intake for the DLWC project provides Toronto with a new water supply source and solves summer taste and odour problems. DLWC has made a major impact on replacing electrically-driven vapour compression chilling, much of it still using CFC refrigerants, and has cut electrical use to power these chillers. DLWC is unique in the size of the cooling demand that can be served; in the integration with municipal services, specifically the water supply for metropolitan Toronto; and the positive impact on energy efficiency and reduced greenhouse gas emissions. The project has the capacity to cool 100 office buildings or 3 million square meters of building space eliminating about 60 MW from the Ontario electrical grid.

1.2.1. GENERAL DESCRIPTION OF DLWC

Toronto is located on the northern shore of Lake Ontario. The high average monthly maximum temperature occurs in July at 26.5 °C with an extreme historical high of 40.6 °C. Relative humidity levels of greater than 80% often accompany the maximum temperatures and determine the low chilled water temperatures needed for dehumidification. The mean daily temperature averaged over the year is 8.9 °C.

Early in this century some Toronto buildings used lake water for cooling. Toronto is now a large metropolitan area with a downtown core of high-rise buildings having a large cooling requirement peaking at roughly 720 MW thermal. Pepco became the first summer peaking utility in 1942. This is now the common situation even for utilities such as Toronto Hydro and Hydro Ottawa. This core is within close proximity to the lake. The cooling requirement is provided by on-site, electrically-driven vapour compression chillers requiring approximately 200 MW peak electrical demands. This electricity usage gives off 150,000 tonnes annually of carbon dioxide. The lake water pumps require only 5% of the electricity currently used by the electrically-driven chillers that would be replaced [5].

1.2.2. FREECOOL FEASIBILITY STUDY

Freecool was the original concept involving the use of permanently cold water from the 80-m depth of Lake Ontario as a substitute for water that is cooled mechanically by electric chillers. A feasibility report [6] was completed for the Canada Mortgage and Housing Corporation in 1982.

The study found that Freecool could result in significant energy conservation in electricity for cooling and in natural gas for steam generation. The general concept is applicable in whole or in part to other major metropolitan areas in Canada and the USA where the natural cooling source could be a lake, river, sea or aquifer. The results of this preliminary feasibility study indicated that the Freecool concept was both technically and economically sound. Negative reactions to the Project Freecool proposal included the view that it was an expensive megaproject; that it ignored cooling demand reduction opportunities at source; that it lacked an identified proponent; and that the dumping of large volumes of return water in the harbour would have negative environmental consequences [7].

1.2.3. DISTRICT COOLING

District cooling is prompted by two factors. One is the large and growing demand for cooling within modern large buildings, caused by growth in the use of heat-producing office equipment and trends towards greater occupant densities in buildings. The other factor is growing difficulty in the provision of on-site cooling posed by refrigerant restrictions. Large modern buildings in Toronto require at least some cooling for 365 days a year to offset heat build-up in their cores.

There are environmental and economic benefits from district cooling. There is the avoidance of use of chemicals that destroy the stratospheric ozone layer such as CFCs. Another advantage is avoidance of the increases in ambient temperature, humidity, and noise that occur outside buildings that have on-site chillers, largely on account of their fans and cooling towers. The major environmental and economic benefits of DLWC arise because pumping cold water in from the lake requires only 5% of the energy required to produce the same amount of cold water using conventional chillers. The adverse environmental impacts are negligible when DLWC is integrated with the municipal water supply.

1.2.4. DLWC AND THE MUNICIPAL WATER SYSTEM

The exciting feature of the DLWC's district cooling system is that it makes use of the huge reservoir of cold water at the core of Lake Ontario. The lake is more than 250 m deep in places. Below about 80 m in the hypolimnion layer, reached within five kilometers of downtown Toronto, the water is permanently at 4 °C. This is the result of a natural phenomenon present in all large deep bodies of water where winters are cold. Surface water sinks when it is cooled to

4 °C because water is at its most dense at this temperature. Summer warming penetrates only to about a 60-m depth. Thus, a deep lake such as Lake Ontario has within it a very large volume of naturally cold water that is seasonally replenished each winter.

1.2.5. FUTURE EXPANSION OPTIONS

DLWC will gradually expand its cooling customer base. As the district cooling demand exceeds the DLWC capacity other options are available. The existing steam network is available during the cooling season to produce ice from steam-driven compressors cascading to absorption chillers. A number of large distributed ice storages are foreseen. They would be charged during the off peak evening hours and discharged during the day to meet the peak cooling demands.

The first phase of DLWC has the potential to meet more than half the annual cooling demand for the whole of Toronto's downtown, and about a fifth of the peak demand. The chilled water will cost less to produce than by conventional means, but the overwhelming advantages are environmental. As noted, the downtown will be quieter and less humid in summer because there will be no need for noisy chillers or vapour-producing cooling towers, which can also be a source of bacterial contamination and cause a form of pneumonia known as Legionnaires' disease.

The DLWC concept has sparked interest elsewhere. Cornell University has proposed a $55-million project called Lake Source Cooling [8] would reduce Cornell's air-conditioning energy usage by 80% by tapping nearby Cayuga Lake.

1.2.6. DLWC CONCLUSIONS

The DLWC project was a success for several reasons. It finally gained a clear proponent; it is integrated into the existing municipal services of water supply and steam and chilling services. It also contributes in a major fashion to the environmental goals of the city of Toronto [9]. These aspects were missing in the original proposals that were large, expensive and add-ons. Future growth of the district cooling network will be determined by customer response, but environmental and economic trends favour a rapid growth. Potential applications at other lake sites exist, but are limited in number.

1.3. Ice For Thermal Energy Storage

The storage of ambient winter air temperature is, however, particularly suited to continental climates characterized by long cold winters with brief hot summers are ideal for the seasonal storage of cold to offset summer cooling peaks. Ice and snow are practical latent energy storage media for cold winter air.

Long-term storage of latent energy in Canada has involved the seasonal storage of winter's cold for use in cooling buildings. This combination of a suitable ice-making climate with significant building cooling demands stimulated a seasonal thermal energy storage program to investigate, test and evaluate various design alternatives and to demonstrate potentially cost-effective storage schemes.

Ice, including both natural and artificial snow, is the choice for the storage of cold as latent energy. In areas of heavy snowfall, where it is collected and transported to snow dumps, an existing supply system is in place. In this respect, the situation is very similar to refuse collection and disposal. Any use that can be made of the refuse or snow that reduces the volume and frequency of hauls to dumps has a cost saving. Thus a building-related snow storage facility could earn revenue from dumping charges. This revenue in some Canadian cities would be comparable to the energy savings. Snow making is also a possibility, as it extends the range beyond those cities receiving heavy snowfall and eases the problems related to annual variation of snowfall and temperature. However artificial snow making involves operational expenses. It was decided to begin with ice formed in situ based on the wider applicability of this approach, i.e., only cold temperatures are required, the volume of storage is less (about one-half that of snow), the storage is not contaminated with salt and dirt as is street snow, and the possibility of a completely automated process exists.

The objective was to develop an automated, efficient ice freezing, storage, and utilization technique with minimal operating and maintenance costs; to determine the height of ice that could be efficiently produced as a function of winter temperatures; and to develop a standard modular approach applicable at any site in a cost-effective manner. The application was to cool commercial and industrial buildings. A construction cost goal of \$150–\$200 (Canadian 1985) per cubic meter of ice was set for the system based on expected energy and chiller savings in comparison with conventional cooling techniques. All of the major technical objectives were achieved. Agricultural applications with a modified design were evaluated over several seasons with positive results.

1.3.1. FABRICATING ICE

Interest in long-term latent energy storage in the form of ice and snow dates back to 1975 [10]. The first field experiment was conducted in the winter of 1979–1980, referred to as Project Icebox because of the wooden structure in which the ice was formed, stored, and melted. The water layer was simply exposed to ambient conditions and allowed to freeze. Results indicated that

Figure 2. Fabrikaglace during cooling extraction.

a substantial volume of ice could be formed, but manual intervention was needed. The meltdown response of the ice to a varying cooling load was completely successful.

The height of ice that can be formed in any location is dependent on many factors, but is directly related to winter temperatures and the method employed. The ice is formed of thin water layers continuously refreshed as the previous layer is frozen. The frequency of water spraying is dependent on the outside temperature. The ice is formed at the point of demand, eliminating all collection and transportation from the production site to the point of use. The size of such a storage should be large enough to spread the costs over a sufficient volume to reduce the first costs to an acceptable level. This method of fabricating a large ice block layer-by-layer using a water spray and outside air distributed over the water surface was called *Fabrikaglace* (Figure 2).

The usable cooling energy content of the cubic meter of ice is about 100 kWh. Therefore, the cost avoidance based on Canadian commercial electricity rates was about \$5–\$50 per cubic meter of ice. Farmers pay about \$25 per cubic meter for delivered ice used in harvesting operations. Fishermen pay about \$10 per cubic meter for ice made in large quantifies, and cube ice sold primarily in summer for chilling of drinks sells for an equivalent of about \$200 per cubic meter. Cost considerations led to an initial cost goal of \$120 (1979 \$) per cubic meter of ice [11].

A field trial during the winter of 1981–1982 employed a fully automated enclosed design. It utilized fans to deliver cold outside air onto the water surface. The air is conducted to the water surface within polyethylene tubes sewn from sheeting. The outside temperature controls the frequency of water spraying.

Spraying was used to form a thin water layer. The water spray was formed by directing water onto an attached plate, which spreads a fountain of water over the ice. This splasher provides an even distribution of water controlled by a timer to give the desired thickness. This same splasher was used to distribute water in the meltdown phase. The water layer thickness used was from 1 to 3 mm. Then the ice-making potential at any location can be estimated based on local temperature records. The overall system coefficient of performance, including formation and storage, of the ice and delivery of useful cooling to the building, was estimated to be 50–100.

Freezing degree minutes or FDM are the product of the temperature in degrees below 0 °C times the duration of the temperature in minutes. The original scheme of three temperature ranges controlling the frequency of spraying resulted in about 300 FDM needed on average to freeze a layer of water. The method of sensing temperature every minute gave a 150 FDM requirement, a doubling of efficiency.

In a typical Quebec City winter, Fabrikaglace based on FDM control, produced a 20-m height of ice. The system coefficient of performance for a reasonably sized application is many times that of competing conventional or other renewable designs. This conclusion is based on data from several seasons of operation. It may be possible to produce significantly more ice in exchange for a decrease in the coefficient of performance. This field trial was successful and supplied cooling to a nearby building.

1.3.2. TECHNICAL GOALS

The goal of a completely automated ice making technique was achieved gradually over five years. First, fans were introduced to bring in cold air when required. Air tubes were used to carry the air to the water surface. These tubes were originally shortened manually as the ice surface rose. The spraying of water interfered with these tubes, and ice formed where it was not wanted. A better method of spraying water was developed and multiple generations of "splashers" eventually led to an acceptable system that distributed water without problems. The plastic air tubes were replaced with collapsible tubing. They were fully extended at the beginning of the freezing cycle and gradually shortened during the winter. Air tubes and splashers were later integrated into one unit that was hoisted from the roof. The control originally was based on reading outdoor air temperatures and basing frequency of spraying on temperature. A method based on sensing temperature every minute and accumulating freezing degree minutes doubled the efficiency. Later, the separate tubes bringing air vertically onto the water surface were integrated into a header that uses angled vents to distribute air horizontally over the water surface.

1.3.3. COST GOALS

Costs estimates for a full-sized application, based on experimental costs, were competitive with conventional cooling. As technical development progressed, costs increased. While the goal of minimum operation and maintenance seemed secure, it was achieved by increased sophistication. Also, as the technique increased in efficiency and the height of ice that could be made increased, the importance of land area decreased. The demonstration chosen involved a building complex scheduled for replacement of the chiller for a cost of about $250,000. The annual energy cooling costs had been dramatically reduced to about $20,000 through modifications in building operation. Therefore, the combined capital cost and energy cost avoidance over ten years for the introduction of a Fabrikaglace would be about $350,000 or $115 per cubic meter of ice. Special R&D requirements for instrumentation and evaluation of about $200,000 gave a rough cost of $550,000. Higher cost for a first trial could only be justified if there was a reasonable expectation that future project costs could be reduced based on the experience gained in constructing and operating the demonstration. Major assumptions are that the designs are reliable enough to replace part or all of the conventional chilling capacity and that no additional maintenance and operational expenses are required. Commercialization was envisioned by involving prefabricated building manufacturers in the design of the demonstration. Such manufacturers may have general contracting capabilities and can therefore deliver the entire project. Local teams formed of such manufacturers with engineers and architects would be able to market the technology. Cost-effective trials have been completed for chilling of vegetable crops [12].

1.3.4. SWEDISH SNOW STORAGE

The Swedish tradition of using stored snow and ice for cooling is old. Ice barns were used for centuries for storage of food. The barns were normally well-insulated buildings kept cold by the stored ice. In some cases the barns were merely a storage room for ice in which case the ice was stored under a layer of sawdust. There are 6th century ice storage pits, which according to archaeologists were used for storage of meat and fish in summer time. Some old ice barns are still in operation. During the 19th century ice cooling became common in the Swedish cities. Ice was harvested from lakes and stored for later use. Ice cooling was a large industry and Swedish and Norwegian ice was exported to New York, London, and India. Norway exported more than 300,000 tons of ice to London in the year 1900.

Today there is a renewed interest in snow and ice cooling. The first large-scale plant for space cooling is now in its sixth year of operation. Several similar projects are now under evaluation. The Sundsvall Hospital [13] snow cooling plant, mainly used for comfort cooling, is located in central Sweden, where the annual mean temperature is 6 °C. The plant is designed for a cooling load of 1,000 kW (1,000 MWh), which requires 30,000 m^3 of snow. It consists of a watertight, slightly sloping asphalt surface, 140 m × 60 m. Cold melt water is pumped from the storage to heat exchangers in the hospital building. While cooling the building the water is warmed. The water is re-circulated to the storage to melt more snow, i.e., to produce cold water. The storage has no power limit. If a higher cooling power is needed more water is re-circulated and more melt water is formed.

Most of the snow is collected from nearby streets. If necessary, snow guns are used to produce additional snow. In April the snow deposit is thermally insulated by 0.2 m of wood chips. Some meltwater is evaporated through the sawdust, which gives an evaporative cooling effect that corresponds to 25% of the extracted cold. The wood chips in Sundsvall are reused several seasons before being burnt in a nearby cogeneration plant.

The operation of the snow cooling plant has so far been successful. During the year 2000—the first short cooling season—about 19,000 m^3 of snow was sufficient to supply 93% of the cooling demand. In 2001 about 27,000 m^3 of snow met 75% of the cooling demand. Some modifications were made on the storage as a result of experiences from the first two years of operation.

1.4. Underground Thermal Energy Storage (UTES)

1.4.1. INTRODUCTION TO ATES AND BTES

Underground Thermal Energy Storage (UTES) has been used to store large quantities of thermal energy to supply process cooling, space cooling, space heating, and ventilation air preheating, and can be used with or without heat pumps. UTES is used as an energy sink and source when supply and demand for energy do not coincide. Recognized energy sources include winter ambient air, heat-pump reject water (from cooling and heating mode), solar energy, process heat, etc. UTES may be used to meet all or part of the heating or cooling requirements of the building or process. Heat pumps may be employed to decrease or increase the storage temperature for cooling or heating.

Applications for UTES include space heating and cooling of all building types, agriculture (e.g., greenhouses), and industrial process cooling. UTES can be applied at sites presently using groundwater for cooling. UTES would assist in keeping aquifer temperatures from increasing and ensure that environmental safeguards such as aquifer recharge are employed. The cost-effectiveness of UTES is based on the capital cost avoidance of conventional heating or chilling equipment and energy savings. Heat rejection equipment, such as cooling towers, may be avoided if the earth is used as an energy store.

UTES encompasses both aquifer thermal energy storage (ATES) and borehole thermal energy storage (BTES).

ATES may be used on a short-term or long-term basis

(a) as the sole source of energy or as a partial storage;
(b) at a temperature useful for direct application or needing upgrade; and
(c) in combination with a dehumidification system, such as desiccant cooling.

Cold storage underground is now a standard design option in several countries. The duration of storage depends on the local climate and the type of building or process being supplied with cooling or heating. Aquifers, necessary for the implementation of ATES, can be discharged effectively through production wells to meet large cooling and heating demands. Aquifers are underground, water-yielding geological formations, and can be unconsolidated (gravel and sand) or consolidated (rocks). The temperature of aquifer water is related to, but usually slightly warmer than, the mean annual air temperature.

In Alabama, where the natural groundwater temperature (about 18 °C) is too high for direct cooling, ATES is used to store chilled water produced during the winter months. The wells are separated by a critical distance to ensure that the warm and cold stores remain separate and that thermal breakthrough does not occur within one season. This critical distance is primarily a function of the well production rates, the aquifer thickness, and the hydraulic and thermal properties that control the storage volume. A minimum separation distance is 30 m and distances of 100–200 m are common for commercial building applications. Multiple-well configurations, involving more that two wells, have been employed where large volumes of water are required and in systems where individual well yields are low. Single-well applications have also been employed using vertical separation of hot and cold groundwater where multiple aquifers exist. A number of software programs are available to simulate UTES systems.

The increasing use of groundwater source heat pumps for heating and cooling residential and commercial buildings has stimulated the demand for ATES applications with heat pumps. Facilities in northern latitudes may have roughly equal heating and cooling energy requirements. A groundwater source heat pump connected to a cold well and a warm well is a rudimentary ATES. The warm well can be used as a heat source for the heat pump evaporator in the heating season and the by-product chilled water stored in the cold well. The chilled water is stored until the cooling season when, for a time, it can be used directly, without the heat pump, to provide space and process cooling.

Cold storage water is normally injected at temperatures of 2–5 °C, with cooling power typically ranging from 200 kW thermal to 20 MW thermal, and with the stored cooling energy extending to 20 GWh. At a cogeneration waste heat project supplying direct heating to the Utrecht University campus in the Netherlands, hot wells have successfully stored water up to a temperature of 90 °C. Attempts to store water at temperatures of 250 °C have not been successful due to adverse water–rock geochemical interactions. A typical single-well flow rate is 3 l/s for a small application and 50 l/s for larger applications. Cost-effectiveness is enhanced when both heating and cooling energy are supplied.

UTES systems fall into two categories depending on whether the stored energy is actively gathered (e.g., the facility at the University of Alabama building in Tuscaloosa where the cooling tower was used in winter as a collector of chilled water) or whether the system stores waste or by-product energy (e.g., groundwater heat pump projects that store the by-product chilled water from the heating season). These double-effect storage projects are more likely to be economical.

Chemical changes in groundwater due to temperature and pressure variations associated with aquifer thermal energy storage may result in operational and maintenance problems. Fortunately, these problems are avoidable and manageable within the operating range of most common applications. Explicit guidelines on proper design, materials selection, and operation, which would decrease the likelihood of such problems, are now available. To maintain well efficiency, back flushing is recommended.

BTES applications involve the use of boreholes and are operated in closed loop (i.e., there is no contact between the natural groundwater and the heat-exchange fluid). Typically, a BTES system will include one or more boreholes equipped with borehole heat exchangers (e.g., U-tubes) through which waste heat or cold energy is circulated and transferred to the ground for storage. Borehole thermal energy storage has been extensively exploited in Sweden. Large systems are operating at Richard Stockton College in New Jersey and at the new University of Ontario Institute of Technology in Oshawa, Ontario.

Seasonal storage began as an environmentally sensitive improvement on the large-scale mining of groundwater. It has the potential to save large amounts of energy. The modern approach is characterized by the reinjection of all extracted water, minimal water treatment, and the attempt to achieve annual thermal balancing. The recent research, which is focused on community-based and aquifer-based use of aquifer thermal energy storage, perhaps integrated with other community services such as drinking water supply or aquifer remediation, will lead to large-scale storage implementation.

1.4.2. GUIDELINES AND STANDARDS

1.4.2.1. *C448.3-02—Design and Installation of Underground Thermal Energy Storage Systems for Commercial and Institutional Buildings [14]*

This Standard covers minimum requirements for equipment and material selection, site survey, system design, installation, testing and verification, documentation, and commissioning and decommissioning of Underground Thermal Energy Storage Systems for Commercial and Institutional Buildings. It is not a design guideline but rather a provision that the user is obliged to satisfy in order to comply with the Standard. It makes clear who is responsible for what and what expertise is needed.

The Standard covers Equipment and Materials, e.g., acceptable grouting materials; Site Survey Requirements, e.g., all wells shall be tested for their recharge rate up to the maximum recharge rate required; Design of UTES Systems, e.g., an ATES system shall be designed with due consideration being given to significant chemical changes due to warming or cooling based on water chemistry analysis and modeling; Installation of Systems, e.g., the engineer shall ensure that the water-supply and recharge well(s) and pumping equipment are installed, hydraulically tested, and sealed and grouted by a qualified contractor.

1.4.2.2. *Environmental Checklist for Earth Energy Heat Pumps and Underground Thermal Energy Storage (UTES) Systems*

Potential environmental concerns over the use of earth energy heat pumps and UTES are

- the possibility of leakage of the heat-exchange fluid into the natural environment;
- thermally induced biochemical effects on groundwater quality;
- ecological distress due to chemical and thermal pollution; and
- external contaminants entering the groundwater.

Potential environmental concerns should be addressed through system design and monitoring. The following items are provided as a guide:

(a) The antifreeze solution used for heat exchange shall not be toxic, as determined by the occupational safety and health provisions of the *Canada Labour Code*, Workplace Hazardous Materials Information System (WHMIS) Material Safety Data Sheets, and standard test methods.
(b) The antifreeze solution shall not adversely affect the physical, metallurgical, or chemical integrity of the piping system.
(c) The physical piping connections shall not allow leakage of the antifreeze solution under the anticipated maximum internal pressure of the system.

(d) A system-monitoring procedure shall be designed (including an emergency response plan) to handle any accidental leakages safely and effectively. Before installation, the entire system or randomly selected critical components of the system shall be tested under conditions resembling those that will prevail after installation of the system, to determine any flaws or shortcomings in design and manufacturing. The installation of the system shall be accompanied by a means for system integrity monitoring and for remediation in case of failure.
(e) The monitoring system shall be managed properly to prevent any leakages from spreading beyond property limits.
(f) Heat-exchange fluid spills or leakages in a soil-based heat sink whose base is located at least 1 m (3 ft) above the highest anticipated or known groundwater level are generally not a concern, except in those cases where the heat sink consists of soils with Darcy permeability coefficients greater than 10 cm/s. In such cases, consideration may be given to lining the heat sink base and sides with clay or artificial impermeable barriers.
(g) Where the heat sink consists of a prism of soil-containing vertical boreholes that penetrate the groundwater, some of the holes may be designed as pumping wells to confine accidental leaks and recover lost fluids.
(h) Vertical and/or inclined boreholes shall be properly sealed against contamination from external sources or from any contaminated stratigraphic units intersected by the boreholes.
(i) Any soil-based installation shall not permit the flushing of soil into aquatic habitat.
(j) Antifreeze released from a submerged earth energy heat pump system will be diluted immediately, but it will also change the quality of the surrounding water. The system design shall include mitigating measures to protect aquatic habitat and water quality.
(k) Potential thermal pollution of a water body shall be analyzed with respect to its impact on aquatic habitat and on the local or regional ecosystem. Temperature rises above certain critical limits can promote the growth of certain parasites or bacteria, which can affect fish. It has been suggested that the columnaris disease of salmon in the Columbia River was facilitated by a slight elevation in temperature.
(l) During well drilling, well development, water sampling, and in situ testing, care shall be taken to prevent the entry of contaminants from the surface and/or the subsurface stratigraphic units intersected by the well bore.
(m) Well bores and vertical boreholes shall not penetrate designated geological formations.

(n) The velocity and direction of groundwater flow shall be determined in order to maximize the thermal store retention time.
(o) Pumping from existing or new wells and injecting water into existing or new wells shall not affect the water rights of other well owners/users, and in no case should such withdrawal or injection affect the pre-use quality of the groundwater, except by mutual consent of the parties involved (and then only if the change in water quality satisfies local regulations and can be shown to be beneficial for the purpose intended).
(p) The UTES storage temperature should not result in unacceptable temperature changes in the groundwater being withdrawn from the storage aquifer by neighbouring groundwater users. In some jurisdictions a maximum change of +1 °C is considered acceptable.
(q) Groundwater treatments designed to improve the efficiency of UTES systems and/or components should not result in such changes to the groundwater quality as may be considered unacceptable for the supply aquifer or the aquifer into which the treated groundwater is injected.
(r) Groundwater injected into an aquifer shall meet the water quality criteria established for that aquifer.
(s) The impact of groundwater or ground-temperature changes on soil micro-organisms shall be investigated to ensure that pathogens (human or animal disease-bearing or—causing micro-organisms) are not encouraged to multiply.
(t) Ground-temperature changes shall not adversely affect surrounding structures and the natural habitat.
(u) The earth energy heat pump and UTES systems shall be designed to prevent the entry of allochthonous (external origin) bacteria, viruses, fungi, and spores into the human or animal hygiene domain served by the system.
(v) The earth energy heat pump and UTES system design shall include antiseptic measures and treatments that preserve and enhance hygiene, to counteract possible contaminants. For example, bacteria of the genus *Legionella* are associated with cooling towers.
(w) During extraction of groundwater, the quantity of dissolved gases released shall not be sufficient to change the groundwater quality from the standard established for the supply aquifer.
(x) During well drilling and/or development, the drilling muds used should not be of the type that can serve as nutrients to indigenous bacteria or viral species that may be present in a dormant state within the subsurface environment. Where such drilling muds are required to be used by the nature of the geological formations that are intersected by the

well bore, the well-development phase should include a thorough flushing of the well bore and its immediate surroundings to remove potential nutrients.

1.4.2.3. *Direct Use of the Underground as Heat Source, Heat Sink, Heat Storage Guideline VDI 46401 [15]*

The guideline is intended for planning and construction firms, manufacturers of components (e.g. for heat pumps, piping, thermal insulation materials, etc), licensing authorities, energy consultants and for persons providing instruction/training in this field. Proceeding from the current state of the art, the goal of the guideline is to ensure correct design, suitable choice of materials, and correct production of boreholes for UTES systems as well as their proper installation and incorporation into energy supply facilities. This will ensure that these systems operate trouble-free in an economically and technically satisfactory manner, also over long periods of time, without having a negative impact on the environment.

Direct Use of the Underground as Heat Source, Heat Sink, Heat Storage *GuidelineVDI 4640 Part4:* Thermal Use of the Underground: Direct Uses 02.08.2004 Editor: VDI Verein Deutscher Ingenieure (The Association of Engineers), Release: September 2004 Price: EUR 65,80 Released in German and English. Available from: Beuth Verlag GmbH, 10772 Berlin www.beuth.de.

The new publication of part 4 “Direct Uses” of the guideline series “Thermal Use of the Underground” describes how the groundwater or the underground can be used thermally as heat source, heat sink and heat storage without operating a heat pump or a cooling device in this process. Part 4 treats the direct thermal use of the groundwater, its design and installation, water economic and water legal aspects as well as environmental aspects. The direct thermal use of the underground with borehole heat exchangers, energy piles, etc forms a further main emphasis. The earth-to-air heat exchangers for heating up and/or cooling the air are described in detail. System descriptions with their design, installation, environmental and economical aspects are some of the topics of this chapter.

Under consideration of the status quo of the technological development, the guideline series of VDI 4640 “Thermal Use of the Underground” presents in four different parts a proper design, a suitable material choice and a correct execution of drillings, installation and system integration of plants for the thermal use of the underground:

Part 1: Fundamentals, approvals, environmental aspects.
Part 2: Ground source heat pump systems (GSHP).
Part 3: Thermal underground energy storage (UTES).
Part 4: Direct uses.

1.5. Conclusions

Thermal energy storage has had a long history extending from thousands of years ago to the present. These traditions have been modified to assist in the current need to achieve energy efficient building technologies. Thermal energy storage ranges through scales from the personal to the municipal. Integration of storage with other municipal services such as snow collection and water supply has been seen as giving promising results. Natural energy sources seem ready again to play a major role in the effort to achieve large reductions in energy intensity in existing and new buildings [16]. This requirement will be clear as the response to the post-2012, post-Kyoto world is formulated.

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2. ENERGETIC, EXERGETIC, ENVIRONMENTAL AND SUSTAINABILITY ASPECTS OF THERMAL ENERGY STORAGE SYSTEMS

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Abstract. Thermal energy storage (TES) systems and their applications are examined from the perspectives of energy, exergy, environmental impact, sustainability and economics. Reductions possible through TES in energy use and pollution levels are discussed in detail and highlighted with illustrative examples of actual systems. The importance of using exergy analysis to obtain more realistic and meaningful assessments, than provided by the more conventional energy analysis, of the efficiency and performance of TES systems is demonstrated. The results indicate that cold TES can play a significant role in meeting society's preferences for more efficient, environmentally benign and economic energy use in various sectors, and appears to be an appropriate technology for addressing the mismatches that often occur between the times of energy supply and demand.

Keywords: Energy, exergy, environment, efficiency, sustainability, thermal energy storage.

2.1. Introduction

Thermal energy storage (TES) is considered an *important energy conservation technology* and, recently, increasing attention has been paid to its utilization, particularly for HVAC applications [1, 2]. Economic factors involved in the design and operation of energy conversion systems have brought TES forefront. It is often useful to make provisions in an energy conversion system for times when the supply of and demand for thermal energy do not coincide. Research conducted by several researchers (e.g., [1–4]) has revealed a wide range of practical opportunities for employing TES systems in industrial applications. Such TES systems provide significant potential from an economic perspective for more effective use of thermal equipment and for facilitating large-scale energy substitutions. A coordinated set of actions is required in

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 23–46.

several sectors of the energy system for the maximum potential benefits of TES to be realized. TES appears to be an advantageous option for correcting the mismatch between the supply and demand of energy, and can contribute significantly to meeting society's needs for more efficient, environmentally benign energy use. TES is a key component of many successful thermal systems. An effective TES incurs minimum thermal energy losses, leading to energy savings, while permitting the highest possible recovery efficiency of the stored thermal energy.

Today, several energy storage technologies exist that can be used in combination with on-site energy sources to economically buffer variable rates of energy supply and demand. TES is considered by many to be one of the most important of these energy technologies and, recently, increasing attention has been paid to utilizing TES in a variety of thermal engineering applications, ranging from heating to cooling and air conditioning.

Although TES is a somewhat mature technology, it is receiving renewed consideration today in commercial and institutional building applications. In a recent study, Dincer [2] points out that TES technology has been successfully applied throughout the world, particularly in developed countries, and states that advantages of TES exceed disadvantages. Some of the advantages of utilizing TES often are [1–2]:

- reduced energy consumption and hence costs,
- reduced initial and maintenance costs,
- reduced equipment size,
- increased flexibility of operation,
- improved indoor air quality,
- conservation of fossil fuels, by facilitating more efficient energy use and/or fuel substitution,
- reduced pollutant emissions (e.g., CO_2 and CFCs),
- increased efficiency and effectiveness of equipment utilization.

2.2. TES Methods

TES systems contain a thermal storage mass, and can store heat or cool. In many hot climates, the primary applications of TES are cold storage because of the large electricity peak demands and consumptions for air conditioning. TES can be incorporated relatively straightforwardly into building air-conditioning or cooling systems. In most conventional cooling systems, there are two major components: a chiller, which cools water or some other fluid, and a distribution system, which transports the cold fluid from the chiller to where it is needed for cooling air for building occupants. In conventional systems, the chiller is operated when cold air is required. In a TES-based cooling system, the chiller can operate at times other than when cooling is needed, so that cooling capacity can be produced and stored during off-peak hours (generally at night) and the cooling capacity used during the day.

There are two principal types of TES: sensible (e.g., water, rock) and latent (e.g., water/ice, salt hydrates). The selection of a TES is mainly dependent on the storage period required (i.e., diurnal, weekly or seasonal), economic viability, operating conditions, etc. The use of TES in thermal applications can facilitate efficient energy use and energy conservation. Efficient TES systems minimize thermal energy losses and attain high energy recovery during extraction of the stored thermal energy with little degradation in temperature. Many researchers have cited exergy as the most appropriate tool for analyzing TES efficiency and performance (e.g., [2–9]).

2.3. Economic Aspects of TES

TES-based systems are usually economically justifiable when the annualized capital and operating costs are less than those for primary generating equipment supplying the same service loads and periods. TES is often installed to reduce initial costs of other plant components and operating costs. Lower initial equipment costs are usually obtained when large durations occur between periods of energy demand. Secondary capital costs may also be lower for TES-based systems. For example, the electrical service equipment size can sometimes be reduced when energy demand is lowered.

In comprehensive economic analyses of systems including and not including TES, initial equipment and installation costs must be determined, usually using manufacturer data, or estimated. Operating cost savings and net overall costs should be assessed using life cycle costing or other suitable methods for determining the most beneficial among multiple systems.

TES use can enhance the economic competitiveness of both energy suppliers and building owners. For example, one study for California [10] indicates that, assuming 20% statewide market penetration of TES, the following financial benefits can be achieved in the state:

- For energy suppliers, TES lowers generating equipment costs (by 30–50% for air-conditioning loads), reduces financing requirements (by US$1–2 billion), and improves customer retention.
- For building owners, TES lowers energy costs (by over one-half billion US dollars annually), increases property values (by US$5 billion), increases financing capability (by US$3–4 billion), and increases revenues.

Comprehensive studies are needed to determine details for the selection, implementation and operation of a TES system since many factors influence these design parameters. Studies should consider all variables which impact the economic benefits of a TES. Sometimes, however, not all factors can be considered. The following significant issues should be clarified and addressed before a TES system is implemented (for details, see [1, 2]):

- management objectives (short- and long-term),
- environmental impacts,
- energy conservation targets,
- economic aims,
- financial parameters of the project,
- available utility incentives,
- the nature of the scenario (e.g., if a new or existing TES system is being considered),
- net heating and/or cooling storage capacity (especially for peak-day requirements),
- utility rate schedules and associated energy charges,
- TES system options best suited to the specific application,
- anticipated operating strategies for each TES option,
- space availability (especially for a storage tank),
- the type of TES (e.g., short- or long-term, full or partial, open or closed).

2.4. Environmental Aspects of TES

TES systems can help increase efficiency and reduce environmental impacts for energy systems, particularly in building heating and cooling and power generation. By reducing energy use, TES systems provide significant environmental benefits by conserving fossil fuels through increased efficiency and/or fuel substitution, and by reducing emissions of such pollutants as CO_2, SO_2, NO_x and CFCs.

TES can impact air emissions in buildings by reducing quantities of ozone-depleting CFC and HCFC refrigerants in chillers and emissions from fuel-fired heating and cooling equipment. TES helps reduce CFC use in two main ways. First, since cooling systems with TES require less chiller capacity than conventional systems, they use fewer or smaller chillers and correspondingly less refrigerant. Second, using TES can offset the reduced cooling capacity that sometimes occurs when existing chillers are converted to more benign refrigerants, making building operators more willing to switch refrigerants.

The potential aggregate air-emission reductions at power plants due to TES can be significant. For example, TES systems have been shown to reduce CO_2 emissions in the UK by 14–46% by shifting electric load to off-peak periods [11–13], while an EPRI co-sponsored analysis found that TES could reduce CO_2 emissions by 7% compared to conventional electric cooling technologies [11–13]. Also, using California Energy Commission data indicating that existing gas plants produce about 0.06 kg of NO_x and 15 kg of CO_2 per 293,100 kWh of fuel burned, and assuming that TES installations save an average of 6% of the total cooling electricity needs, TES could reduce annual emissions by about 560 tons of NO_x and 260,000 tons of CO_2 statewide [10].

2.5. Sustainability Aspects of TES

A secure supply of energy resources is generally agreed to be a necessary but not sufficient requirement for development within a society. Furthermore, sustainable development demands a sustainable supply of energy resources. The implications of these statements are numerous, and depend on how sustainable is defined. One important implication of these statements is that sustainable development within a society requires a supply of energy resources that, in the long term, is readily and sustainably available at reasonable cost and can be utilized for all required tasks without causing negative societal impacts. Supplies of such energy resources as fossil fuels and uranium are generally acknowledged to be finite; other energy sources such as sunlight, wind and falling water are generally considered renewable and therefore sustainable over the relatively long term. Wastes (convertible to useful energy forms through, for example, waste-to-energy incineration facilities) and biomass fuels are also usually viewed as sustainable energy sources. A second implication of the initial statements in this section is that sustainable development requires that energy resources be used as efficiently as possible. In this way, society maximizes the benefits it derives from utilizing its energy resources, while minimizing the corresponding negative impacts (such as environmental damage). This implication acknowledges that all energy resources are to some degree finite, so that greater efficiency in utilization allows such resources to contribute to development over a longer period of time, i.e., to make development more sustainable. Even for energy sources that may eventually become inexpensive and widely available, increases in energy efficiency will likely remain sought to reduce the resource requirements (energy, material, etc) to create and maintain systems and devices to harvest the energy, and to reduce the associated environmental impacts.

TES systems can contribute to increased sustainability as they can help extend supplies of energy resources, improve costs and reduce environmental and other negative societal impacts.

Sustainability objectives often lead local and national governments to incorporate environmental considerations into energy planning. The need to satisfy basic human needs and aspirations, combined with increasing world population, make successful implementation of sustainable development increasingly needed. Requirements for achieving sustainable development in a society include [14]:

- provision of information about and public awareness of the benefits of sustainability investments,
- environmental education and training,
- appropriate energy and energy storage strategies,
- availability of renewable energy sources and clear technologies,
- a reasonable supply of financing, and
- monitoring and evaluation tools.

2.6. Thermodynamics and Energy Technologies

Thermodynamics is broadly viewed as the science of energy, and energy engineering is concerned with making the best use of energy resources and technologies. Thermal systems include power generation, refrigeration, TES.

2.6.1. THE DOMAIN OF THERMODYNAMICS

The science of thermodynamics is founded primarily on two fundamental natural principles, the first and second laws. The first law of thermodynamics expresses the conservation of energy principle, and asserts that during an interaction energy can change from one form to another but the total amount of energy remains constant. The second law of thermodynamics asserts that energy has quality as well as quantity, and actual processes reduce the quality of energy (e.g., high-temperature heat is degraded when it is transferred to a lower temperature body). Quantification of the quality or "work potential" of energy in the light of the second law has resulted in the definition of the quantities entropy and exergy.

The scope of thermodynamic concepts is schematically illustrated in Figure 1, where the domains of energy, entropy and exergy are shown. This paper focuses on the intersection of the energy, entropy and exergy fields as given in Figure 3 [15]. Note that entropy and exergy are also used in other fields (e.g., economics, management and information theory), and therefore are not subsets of energy. Some forms of energy such as shaft work are entropy-free, and thus entropy subtends only part of the energy field. Likewise, exergy subtends only part of the energy field since some systems (such as air at atmospheric conditions) possess energy but no exergy. Most thermodynamic systems possess energy, entropy and exergy, and thus appear at the intersection of these three fields.

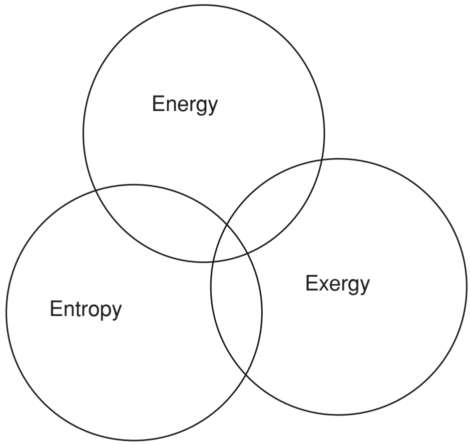

Figure 3. Interactions between the domains of energy, entropy and exergy

2.6.2. ENERGETIC AND EXERGETIC ASPECTS AND SUSTAINABILITY

Energy is a key element in interactions between nature and society and is considered a key input for economic development. Environmental issues span a continuously growing range of pollutants, hazards and eco-system degradation factors that affect areas ranging from local through regional to global. Some of these concerns arise from observable, chronic effects on human health, while others stem from actual or perceived environmental risks such as possible accidental releases of hazardous materials. Many environmental issues, e.g., acid rain, stratospheric ozone depletion and global climate change, are caused by or relate to energy production, transformation, transport and use. Energy, consequently, is a key consideration in discussions of sustainable development.

Energy use is governed by thermodynamic principles and, therefore, an understanding of thermodynamic aspects of energy can help us understand pathways to sustainable development. The impact of energy resource utilization on the environment and the achievement of increased resource-utilization efficiency are best addressed by considering exergy. The exergy of an energy form or a substance is a measure of its usefulness or quality or potential to cause change, and provides the basis for an effective measure of the potential of a substance or energy form to impact the environment. In practice, a thorough understanding of exergy, and the insights it can provide into the efficiency, environmental impact and sustainability of energy systems, are required for the engineer or scientist working in the area of energy systems and the environment. During the past decade, the need to understand how exergy and energy are linked to environmental impact has become increasingly

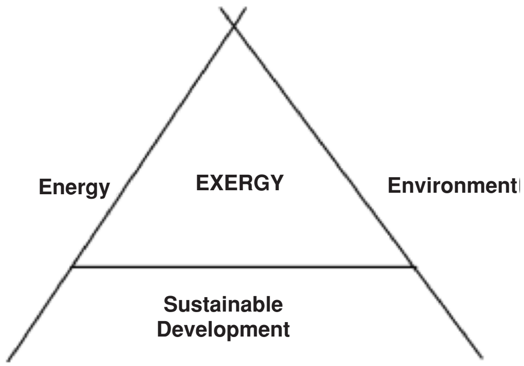

Figure 4. The interdisciplinary triangle of exergy

significant [15–17]. Generally, exergy can be viewed as the confluence of energy, environment and sustainable development (Figure 4), mainly due to its interdisciplinary character.

2.6.3. SUSTAINABLE DEVELOPMENT AND THERMODYNAMIC PRINCIPLES

Thermodynamic principles can be used to assess, design and improve energy and other systems, and to better understand environmental impact and sustainability issues. For the broadest understanding, all thermodynamic principles must be used, not just those pertaining to energy. Thus, many researchers feel that an understanding and appreciation of exergy is essential to discussions of sustainable development.

An inexpensive and stable energy supply is a prerequisite for social and economic development, in households as well as at the national level. Indeed, energy is essential to human welfare and quality of life. However, energy production and consumption generate significant environmental problems (at global, regional and local levels) that can have serious consequences and even put at risk the long-term sustainability of the planet's ecosystems. The relationship between energy consumption and production and sustainability is, therefore, complex [16]. Decisions by the individual, and society regarding how to meet energy needs require careful thought about many issues, including energy resource selection, efficiency and the role of hydrogen and fuel cell technologies.

We consider sustainable development here to involve four key factors (Figure 5): environmental, economic, social and resource/energy sustainability. The connections in Figure 3 illustrate that these factors are interrelated and thus each must taken into consideration to increase sustainable development.

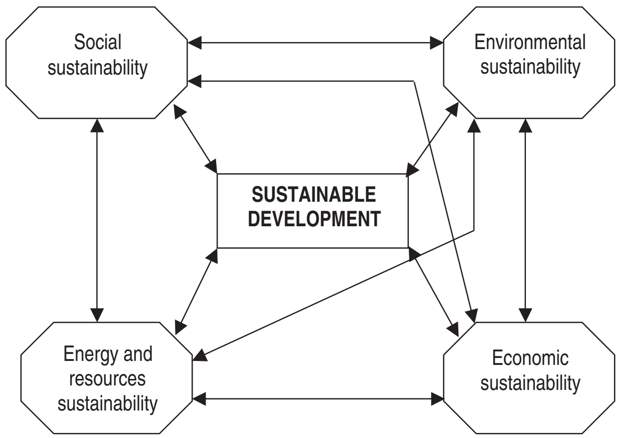

Figure 5. Four key factors for sustainability

2.6.4. EXERGY AND THE ENVIRONMENT

Increasing energy efficiency reduces environmental impact by decreasing energy losses. From an exergy viewpoint, such activities lead to increased exergy efficiency and reduced exergy losses (both waste exergy emissions and internal exergy consumption). Thus, a thorough understanding of the relations between exergy and the environment may reveal underlying fundamental patterns and forces affecting changes in the environment, and help researchers deal better with environmental damage.

The second law of thermodynamics is instrumental in providing insights into environmental impact. The most appropriate link between the second law and environmental impact has been suggested to be exergy, in part because the magnitude of the exergy of a system depends on the states of both the system and the environment and because exergy is a measure of the departure between these states. This departure is zero only when the system is in equilibrium with its environment. The authors have discussed this concept extensively previously [1, 2, 9, 14–16].

2.6.5. EXERGY AND SUSTAINABILITY

Mass and energy balances are normally evaluated prior to performing an exergy analysis. The energy information quantifies only the energy transfers and conversions in a system or process, whereas the exergy results, since exergy is a measure of the quality of energy, quantify the degradation of energy or material in the system. Exergy is conserved only for reversible or ideal processes. Exergy analysis uses the first and second law of thermodynamics to pinpoint the losses of quality, or work potential, in a system. Exergy analysis is consequently linked to sustainability because to increase the sustainability of energy use, we must be concerned not only with loss of energy, but also loss of energy quality (or exergy).

One principal advantage of exergy analysis over energy analysis is that the exergy content of a process flow is a better valuation of the flow than the energy content, since the exergy indicates the fraction of energy that is likely useful and thus utilizable. This observation applies equally on the component level, the process level and the life cycle level. Application of exergy analysis to a component, process or sector can lead to insights into how to improve the sustainability of the activities comprising the system by reducing exergy losses.

Sustainable development requires not just that sustainable energy resources be used, but that the resources be used as efficiently as possible. Many feel that exergy methods can help improve sustainability. Since energy can never be "lost" as it is conserved according to the first law of thermodynamics, while exergy can be lost due to internal irreversibilities, exergy losses which represent potential not used, particularly from the use of non-renewable energy forms, should be minimized when striving for sustainable development.

Furthermore, some studies (e.g., [15–17]) show that some environmental effects associated with emissions and resource depletion can be expressed based on physical principles in terms of an exergy-based indicator. It may be possible to generalize this indicator to cover a comprehensive range of environmental effects, and such research is ongoing.

The relation between exergy, sustainability and environmental impact is illustrated in Figure 6. There, sustainability can be seen to increase and environmental impact to decrease as the exergy efficiency of a process increases. Two limiting efficiency cases, as shown in Figure 5, are of practical significance:

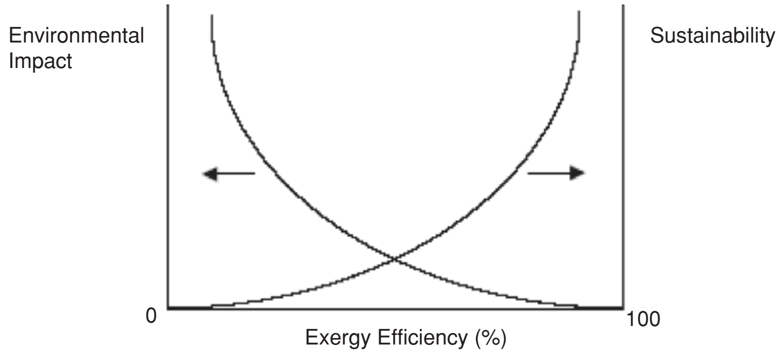

Figure 6. Qualitative illustration of the linkages between the environmental impact and sustainability of a system or process, and its exergy efficiency

- As exergy efficiency approaches 100%, the environmental impact associated with process operation approaches zero, since exergy is only converted from one form to another without loss. Also sustainability approaches infinity because the process approaches reversibility.
- As exergy efficiency approaches 0%, sustainability approaches zero because exergy-containing resources (fuel ores, steam, etc) are used but nothing is accomplished. Also, environmental impact approaches infinity because, to provide a fixed service, an ever-increasing quantity of resources must be used and a correspondingly increasing amount of exergy-containing wastes are emitted.

Research into the benefits of using thermodynamic principles, especially exergy, to assess the sustainability and environmental impact of energy systems is relatively new, and further research is needed to ascertain a better and more comprehensive understanding of the potential role of exergy. Required research includes (i) better defining the role of exergy in environmental impact and design, (ii) identifying how exergy can be better used as an indicator of potential environmental impact, and (iii) developing holistic exergy-based methods that simultaneously account for technical, economic, environmental and other factors.

2.7. Energy and Exergy Analyses of Cold TES

An important application of TES is in facilitating the use of off-peak electricity to provide building heating and cooling. Recently, increasing attention has been paid in many countries to cold TES (CTES), an economically viable technology that has become a key component of many successful thermal systems (Figure 7). Although CTES efficiency and performance evaluations are conventionally based on energy, energy analysis itself is inadequate for complete CTES evaluation because it does not account for such factors as the temperatures at which heat (or cold) is supplied and delivered. Exergy analysis overcomes some of these inadequacies in CTES assessments.

Here we assess using exergy and energy analyses CTES systems, including sensible and/or latent storages (e.g., [1, 2]). Several CTES cases are considered, including storages which are homogeneous or stratified, and some which undergo phase changes. A full cycle of charging, storing and discharging is considered for each case. This section demonstrates that exergy analysis provides more realistic efficiency and performance assessments of CTES systems than energy analysis, and conceptually is more direct since it treats cold as a valuable commodity. An example and case study illustrate the usefulness of exergy analysis in addressing cold thermal storage problems.

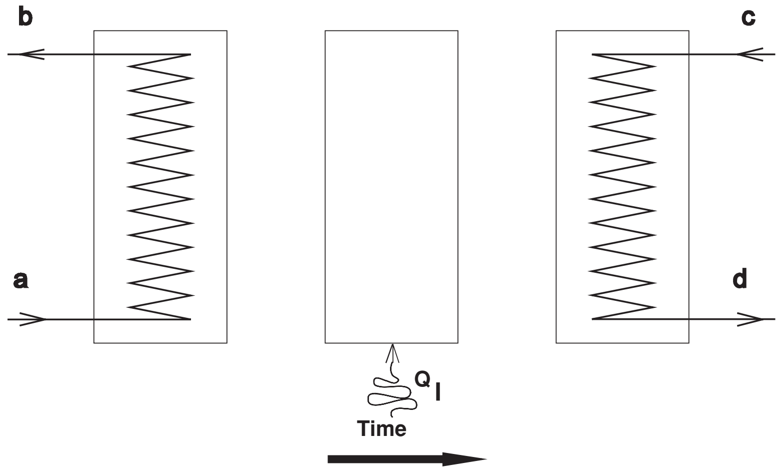

Figure 7. The three processes in a general CTES system: charging (left), storing (middle), and discharging (right). The heat leakage into the system Q_1 is illustrated for the storing process, but can occur in all three processes

2.7.1. ENERGY BALANCES

Consider a cold storage consisting of a tank containing a fixed quantity of storage fluid and a heat-transfer coil through which a heat-transfer fluid is circulated. Kinetic and potential energies and pump work are considered negligible (see [1] for details). An energy balance for an entire cycle of a CTES can be written in terms of "cold" as follows:

$$\textit{Cold input} - [\textit{Cold recovered} + \textit{Cold loss}] = \textit{Cold accumulation}. \quad (1)$$

Here, "cold input" is the heat removed from the storage fluid by the heat-transfer fluid during charging; "cold recovered" is the heat removed from the heat transfer fluid by the storage fluid; "cold loss" is the heat gain from the environment to the storage fluid during charging, storing and discharging; and "cold accumulation" is the decrease in internal energy of the storage fluid during the entire cycle. The overall energy balance for the simplified CTES system illustrated in Figure 5 becomes

$$(H_b - H_a) - [(H_c - H_d) + Q_l] = -\Delta E \quad (2)$$

where H_a, H_b, H_c and H_d are the enthalpies of the flows at points a, b, c and d in Figure 5; Q_l is the total heat gain during the charging, storing, and discharging processes; and ΔE is the difference between the final and initial storage-fluid internal energies. The terms in square brackets in Equations (1) and (2) represent the net "cold output" from the CTES, and $\Delta E = 0$ if the

CTES undergoes a complete cycle (i.e., the initial and final storage-fluid states are identical).

The energy transfer associated with the charging fluid can be expressed as

$$H_b - H_a = m_a c_a (T_b - T_a) \tag{3}$$

where m_a is the mass flow of heat-transfer fluid at point a (and at point b), and c_a is the specific heat of the heat transfer fluid, which is assumed constant. A similar expression can be written for $H_c - H_b$. The energy content of a storage which is homogeneous (i.e., entirely in either the solid or the liquid phase) is

$$E = m(u - u_o) \tag{4}$$

which, for sensible heat interactions only, can be written as

$$E = mc\,(T - T_o) \tag{5}$$

where, for the storage fluid, c denotes the specific heat (assumed constant), m the mass, u the specific internal energy and T the temperature. Also, u_o is u evaluated at the environmental conditions.

For a mixture of solid and liquid, the energy content of the solid and liquid portions can be evaluated separately and summed as follows:

$$E = m[(1 - F)(u_s - u_o) + F(u_t - u_o)] \tag{6}$$

where u_s and u_t are the specific internal energies of the solid and liquid portions of the storage fluid, respectively, and F is the melted fraction (i.e., the fraction of the storage fluid mass in the liquid phase).

For a storage fluid which is thermally stratified with a linear temperature profile in the vertical direction, the energy content can be shown with Equation (5) to be

$$E = mc\left(\frac{T_t + T_b}{2} - T_o\right) \tag{7}$$

where T_t and T_b are the storage-fluid temperatures at the top and bottom of the linearly stratified storage tank, respectively.

The change in CTES energy content from the initial (i) to the final state (f) of a process can be expressed as

$$\Delta E = E_f - E_i \tag{8}$$

where E_i and E_f denote the initial and final energy contents of the storage. In the case of identical initial and final states, $\Delta E = 0$ and the overall energy balance simplifies.

2.7.2. EXERGY BALANCES

An exergy balance for a CTES undergoing a complete cycle of charging, storing and discharging can be written as

$$\begin{aligned}&\textit{Exergy input} - [\textit{Exergy recovered} + \textit{Exergy loss}] - \textit{Exergy consumption}\\&\quad = \textit{Exergy accumulation}\end{aligned} \tag{9}$$

or

$$(\in_a - \in_b) - [(\in_d - \in_c) + X_l] - I = \Delta\Xi \tag{10}$$

where $\in_a$, $\in_b$, $\in_c$ and $\in_d$ are the exergies of the flows at states a, b, c and d, respectively; and X_l denotes the exergy loss associated with Q_l;I is the exergy consumption; and $\Delta\Xi$ is the exergy accumulation. In Equation (10), $(\in_a - \in_b)$ represents the net exergy input and $(\in_d - \in_c)$ is the net exergy recovered. The quantity in square brackets represents the net exergy output from the system. The terms I,X_l and $\Delta\Xi$ are given respectively by

$$I = \sum_{j=1}^{3} I_j \tag{11}$$

$$X_l = \sum_{j=1}^{3} X_{l,j} \tag{12}$$

$$\Delta\Xi = \Xi_f - \Xi_i \tag{13}$$

where, I_1, I_2 and I_3 denote respectively the consumptions of exergy during the charging, storing and discharging periods; $X_{1,1}$,$X_{1,2}$ and $X_{1,3}$ denote the exergy losses associated with heat losses during the same periods; and Ξ_i and Ξ_f denote the initial and final exergy contents of the storage. When the initial and final states are identical, $\Delta\Xi = 0$.

The exergy content of a flow of heat transfer fluid at state k (where k = a, b, c, or d in Figure 4) can be expressed as

$$\in_k = (H_k - H_o) - T_o\,(S_k - S_o) \tag{14}$$

where $\in_k$, H_k and S_k denote the exergy, enthalpy and entropy of state k, respectively, and H_o and S_o the enthalpy and the entropy at the temperature T_o and pressure P_o of the reference environment. The exergy expression in Equation (14) only includes physical (or thermomechanical) exergy. Potential and kinetic exergy components are, as pointed out earlier, considered negligible for the devices under consideration. The chemical component of exergy is neglected because it does not contribute to the exergy flows for sensible CTES systems. Thus, the exergy differences between the inlet and outlet for the charging and discharging periods are, respectively:

$$\in_a - \in_b = (H_a - H_b) - T_o\,(S_a - S_b) \tag{15}$$

and

$$\in_d - \in_c = (H_d - H_c) - T_o\,(S_d - S_c) \tag{16}$$

where it has been assumed that T_o and P_o are constant, so that H_o and S_o are constant at states a and b, and at states c and d.

The exergy loss associated with heat infiltration during the three storage periods can be expressed as

$$X_{l,j} = \left(1 - \frac{T_o}{T_j}\right) Q_{l,j} \tag{17}$$

where j represents the particular period, and T_1,T_2 and T_3 are constant during the respective charging, storing and discharging periods. Sometimes T_j represents a mean temperature within the tank for period j.

The thermal exergy terms are negative for sub-environment temperatures, as is the case here for CTESs, indicating that the heat transfer and the accompanying exergy transfer are oppositely directed. That is, the losses associated with heat transfer are due to heat infiltration into the storage when expressed in energy terms, but due to a cold loss out of the storage when expressed in exergy.

The exergy content of a homogeneous storage can be expressed as

$$\Xi = m[(u - u_o) - T_o(s - s_o)] \tag{18}$$

where s is the specific entropy of the storage fluid and s_o is s evaluated at the environmental conditions. If only sensible heat interactions occur, Equation (18) can then be written as

$$\Xi = mc[(T - T_o) - T_o \ln(T/T_o)]. \tag{19}$$

For a mixture of solid and liquid, the exergy content can be written as

$$\Xi = m\{(1 - F)[(u_s - u_o) - T_o(s_s - s_o)] + F[(u_t - u_o) - T_o(s_t - s_o)]\} \tag{20}$$

where s_s and s_t are the specific entropies of the solid and liquid portions of the storage fluid, respectively.

Consequently, the exergy content of a storage which is linearly stratified can be shown as

$$\Xi = E - mcT_o\left[\frac{T_t(\ln T_t - 1) - T_b(\ln T_b - 1)}{T_t - T_b} - \ln T_o\right]. \tag{21}$$

The change in TES exergy content can be expressed as in Equation (13).

2.7.3. ENERGY AND EXERGY EFFICIENCIES

For a general CTES undergoing a cyclic operation, the overall energy efficiency η can be evaluated as

$$\eta = \frac{\textit{Energy in product outputs}}{\textit{Energy in inputs}} = 1 - \frac{\textit{Energy loss}}{\textit{Energy in inputs}} \tag{22}$$

where the word energy represents the cold. Then, following Figure 5, the overall and charging-period energy efficiencies can be expressed as

$$\begin{aligned}\eta &= \frac{\textit{Energy recovered from TES during discharging}}{\textit{Exergy input to TES during charging}} \\ &= \frac{H_d - H_c}{H_a - H_b} = 1 - \frac{Q_l}{H_a - H_b}\end{aligned} \tag{23}$$

$$\eta_1 = \frac{\textit{Energy accumulation in TES during charging}}{\textit{Energy input to TES during charging}} = \frac{\Delta E_1}{H_a - H_b}. \tag{24}$$

The energy efficiencies for the storing and discharging subprocesses can be written respectively as

$$\eta_2 = \frac{\Delta E_1 + Q_l}{\Delta E_1} \tag{25}$$

$$\eta_3 = \frac{H_c - H_d}{\Delta E_3} \tag{26}$$

where ΔE_1 and ΔE_3 are the changes in CTES energy contents during charging and discharging, respectively.

The exergy efficiency for the overall process can be expressed as

$$\begin{aligned}\psi &= \frac{\textit{Exergy recovered from TES during discharging}}{\textit{Exergy input to during charging}} \\ &= \frac{\in_d - \in_c}{\in_a - \in_b} = 1 - \frac{X_l + I}{\in_a - \in_b}.\end{aligned} \tag{27}$$

If the TES is adiabatic, $Q_{l,j} = X_{l,j} = 0$ for all j. Then the energy efficiency is fixed at unity and the exergy efficiency simplifies to

$$\psi = 1 - \frac{I}{\in_a - \in_b}. \tag{28}$$

The exergy efficiencies for the charging, storing and discharging processes, respectively, can be expressed as

$$\psi_1 = \frac{\textit{Exergy accumulation in TES during charging}}{\textit{Exergy input to TES during charging}} = \frac{\Delta \Xi_1}{\in_a - \in_b} \tag{29}$$

$$\psi_2 = \frac{\textit{Exergy accumulation in TES during charging and storing}}{\textit{Exergy accumulation in TES during charging}} = \frac{\Delta\Xi_1 + \Delta\Xi_2}{\Delta\Xi_1}\mathcal{M} \tag{30}$$

$$\psi_3 = \frac{\textit{Exergy recovered from TES during discharging}}{\textit{Exergy accumulation in TES during charging and storing}} = \frac{\in_d - \in_c}{\Delta\Xi_1 + \Delta\Xi_2}. \tag{31}$$

Further information on energy and exergy analyses of TES and CTES systems can be found in Dincer and Rosen [1].

2.8. Illustrative Example

2.8.1. CASES CONSIDERED AND SPECIFIED DATA

Four different CTES cases are considered as given in [1]. In each case, the CTES has identical initial and final states, so that the CTES operates in a cyclic manner, continuously charging, storing and discharging. The main characteristics of the cold storage cases are as follows:

(I) Sensible heat storage, with a fully mixed storage fluid.
(II) Sensible heat storage, with a linearly stratified storage fluid.
(III) Latent heat storage, with fully mixed storage fluid.
(IV) Combined latent and sensible heat storage, with fully mixed storage fluid.

The following assumptions are made for each of the cases:

- Storage boundaries are nonadiabatic.
- Heat gain from the environment during charging and discharging is negligibly small relative to heat gain during the storing period.
- The external surface of the storage tank wall is at a temperature 2 °C greater than the mean storage-fluid temperature.
- The mass flow rate of the heat transfer fluid is controlled so as to produce constant inlet and outlet temperatures.
- Work interactions, and changes in kinetic and potential energy terms, are negligibly small.

Specified data for the four cases are presented in Table 1 and relate to the diagram in Figure 7. In Table 1, T_b and T_d are the charging and discharging outlet temperatures of the heat transfer fluid, respectively. The subscripts 1, 2 and 3 indicate the temperature of the storage fluid at the beginning of charging,

TABLE 1. Specified temperature data for the cases in the CTES example

Temperature (°C)	Case I	II	III	IV
T_b	4.0	15	−1	−1
T_d	11.0	11	10	10
T_1	10.5	19/2*	0 (t)	8
T_2	5.0	17/–7*	0 (s)	– 8
T_3	6.0	18/–6*	0 (t&s)	0 (t&s)

* When two values are given, the storage fluid is vertically linearly stratified and the first and second values are the temperatures at the top and bottom of the storage fluid, respectively.

storing or discharging, respectively. Also, t indicates the liquid state and s the solid state for the storage fluid at the phase-change temperature.

In addition, for all cases, the inlet temperatures are fixed for the charging-fluid flow at $T_a = -10$ °C and for the discharging-fluid flow at $T_c = 20$°C. For cases involving latent heat changes (i.e., solidification), $F = 10\%$. The specific heat c is 4.18 kJ/(kg K) for both the storage and heat-transfer fluids. The phase-change temperature of the storage fluid is 0 °C. The configuration of the storage tank is cylindrical with an internal diameter of 2 m and internal height of 5 m. Environmental conditions are 20 °C and 1 atm.

2.8.2. RESULTS AND DISCUSSION

The results for the four cases are listed in Table 2, and include overall and subprocess efficiencies, input and recovered cold quantities, and energy and exergy losses. The overall and subprocess energy efficiencies are identical for Cases I and II, and for Cases III and IV. In all cases the energy efficiency values are high. The different and lower exergy efficiencies for all cases indicate that energy analysis does not account for the quality of the "cold" energy, as related to temperature, and considers only the quantity of "cold" energy recovered.

The input and recovered quantities in Table 2 indicate the quantity of "cold" energy and exergy input to and recovered from the storage. The energy values are much greater than the exergy values because, although the energy quantities involved are large, the energy is transferred at temperatures only slightly below the reference-environment temperature, and therefore is of limited usefulness.

The cold losses during storage, on an energy basis, are entirely due to cold losses across the storage boundary (i.e., heat infiltration). The exergy-based

TABLE 2. Energy and exergy quantities for the cases in the CTES example

Period or quantity	Energy quantities				Energy quantities			
	I	II	III	IV	I	II	III	IV
Efficiencies (%)								
Charging (1)	100	100	100	100	51	98	76	77
Storing (2)	82	82	90	90	78	85	90	85
Discharging (3)	100	100	100	100	38	24	41	25
Overall	82	82	90	90	15	20	28	17
Input, recovered and lost quantities (MJ)								
Input	361.1	361.1	5,237.5	6,025.9	30.9	23.2	499.8	575.1
Recovered	295.5	295.5	4,713.8	5,423.3	4.6	4.6	142.3	94.7
Loss (external)	65.7	65.7	523.8	602.6	2.9	2.9	36.3	48.9
Loss (internal)	—	—	—	—	23.3	15.6	321.2	431.4

cold losses during storage are due to both cold losses and internal exergy losses (i.e., exergy consumptions due to irreversibilities within the storage). For the present cases, in which the exterior surface of the storage tank is assumed to be 2°C warmer than the mean storage-fluid temperature, the exergy losses include both external and internal components. Alternatively, if the heat transfer temperature at the storage tank external surface is at the environment temperature, the external exergy losses would be zero and the total exergy losses would be entirely due to internal consumptions. If heat transfer occurs at the storage-fluid temperature, on the other hand, more of the exergy losses would be due to external losses. In all cases, the total exergy losses, which are the sum of the internal and external exergy losses, remain fixed.

The four cases demonstrate that energy and exergy analyses give different results for CTES systems. Both energy and exergy analyses account for the quantity of energy transferred in storage processes. Exergy analyses take into account the loss in quality of "cold" energy, and thus more correctly reflect the actual value of the CTES.

In addition, exergy analysis is conceptually more direct when applied to CTES systems because cold is treated as a useful commodity. With energy analysis, flows of heat rather than cold are normally considered. Thus, energy analyses become convoluted and confusing as one must deal with heat flows, while accounting for the fact that cold is the useful input and product recovered for CTES systems. Exergy analysis inherently treats any quantity which is out of equilibrium with the environment (be it colder or hotter) as a valuable commodity, and thus avoids the intuitive conflict in the expressions associated with CTES energy analysis. The concept that cold is a valuable commodity is both logical and in line with one's intuition when applied to CTES systems.

2.9. Conclusions

TES systems generally are attracting increasing interest in several thermal applications, e.g., active and passive solar heating, water heating, cooling and air conditioning. Also, TES is presently identified as an economic storage technology for building heating, cooling and air-conditioning applications. The main conclusions of the present study follow:

- TES can play a significant role in meeting society's preferences for more efficient, environmentally benign energy use in various sectors, and appears to be an appropriate technology for addressing the mismatch that often occurs between the times of energy supply and demand.
- Using TES systems substantial energy savings can be obtained when implementing appropriate demand side management strategies and emissions, e.g., CO_2, SO_2 and NO_x, can significantly be reduced.
- For complete performance and efficiency evaluation of TES systems, both energy and exergy analyses should be undertaken. Exergy analysis often provides more meaningful and useful information than energy analysis regarding efficiencies and losses for TES systems, partly because the loss of low temperature in cold TES is accounted for in exergy-based performance measures, but not in energy-based ones.
- Assessments of the sustainability of processes and systems, and efforts to improve sustainability, should be based in part upon thermodynamic principles, and especially the insights revealed through exergy analysis.
- To realize the energy, exergy, economic and environmental benefits of TES technologies, an integrated set of activities should be conducted including research and development, technology assessment, standards development and technology transfer. These can be aimed at improving efficiency, facilitating the substitution of these technologies and other environmentally benign energy currencies for more harmful ones, and improving the performance characteristics of these technologies.

Consequently, exergy analysis can likely assist in efforts to optimize the design of CTES systems and their components, and to identify appropriate applications and optimal configurations for CTES in general engineering systems.

Acknowledgement

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Nomenclature

A	surface area
c	specific heat
c_p	specific heat at constant pressure
c_v	specific heat at constant volume
C	heat capacity rate
e	specific energy
E	energy
f	fraction; mean height fraction
F	fraction of storage-fluid mass in liquid phase
h	specific enthalpy; height
H	enthalpy; TES fluid height
I	exergy consumption due to irreversibilities
k	thermal conductivity
ke	specific kinetic energy
l	distance of plates
L	latent heat
m	mass
$\dot{m}$	mass flow rate
N	moles
pe	specific potential energy
P	absolute pressure, perimeter
Q	heat
R	thermal resistance
s	specific entropy
S	entropy
t	time
T	temperature
u	specific internal energy
U	internal energy
v	specific volume; velocity
V	volume
W	shaft work
x	coordinate; distance
X	thermal exergy (i.e., exergy associated with heat Q)
y	mole fraction; coordinate

Greek Symbols

α	constant parameter
ε	specific flow exergy

$\in$ flow exergy
η energy efficiency; parameter ($= s/l$)
θ parameter; temperature; dimensionless T temperature ($= (T_m - T_s)/(T_m - T_\infty)$)
μ chemical potential; dynamic viscosity
ζ specific exergy; parameter ($= x/l$)
Ξ exergy
$\prod$ entropy production
ρ density
ϕ zone temperature distribution; general dependent variable
ψ exergy efficiency
ΔT temperature difference $= T_w - T_m$

Subscripts

a inlet flow during charging; adiabatic; parameter
amb ambient
b outlet flow during charging; bottom; parameter
c injected during charging period; charging; inlet flow during discharging
d recovered during discharging period; discharging; outlet flow during discharging
e exit; equivalent; effective
f final
i initial
ini initial
k number of zones
min minimum
kin kinetic component
l loss; liquid; liquid phase
m mixed; melting; melting point
n nonadiabatic; phase
net net
o environmental state; chemical exergy; outlet
oo dead state
p product
PCM phase change material
ph physical component
pot potential component
r region of heat interaction
s solid; solid phase; solid state; storage fluid

st	storage (overall)
t	threshold; top; liquid state
T	total
th	thermocline zone (zone 2)
w	working fluid; wall
1	charging period
2	storing period
3	discharging period

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PART II. CLIMATE CHANGE AND THERMAL ENERGY STORAGE

3. WHAT ENGINEERS NEED TO KNOW ABOUT CLIMATE CHANGE AND ENERGY STORAGE

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Abstract. Climate change is increasingly apparent. Regional impacts of climate change are being observed. Those commonly cited include extended growing seasons; shifts of plant and animal ranges; earlier flowering of trees, emergence of insects and egg-laying in birds; and local temperature, humidity and wind-speed anomalies. Air temperatures in Alaska and western Canada have increased as much as 3–4 °C in the past 50 years. Engineers who design infrastructure for predicted future conditions face challenges due to these shifts in climate. Building codes already specify minimum health and safety requirements for some key climate variables such as heating and cooling design temperatures; heating and cooling degree days; rainfall and snow loads; and wind pressures. Predicted changes in these variables at specific locations are not usually available. Regional scenarios give a general trend but lack precision and verification. Eco-conscious clients and a concerned public are causing manufacturing and construction firms to adopt more environmentally sound engineering practices. Proactive members of every important industry are getting involved with education and research into new technologies and approaches to address design problems with sustainable solutions. The demand for "green" innovations in design is growing. Even with the mitigation measures underway to cut net emissions of greenhouse gases and so reduce climate change, current predictions see more frequent and more severe extreme weather events. As climate change continues, the prediction and mitigation of climate related hazards will ultimately require adaptation across the entire construction sector. Another response to climate change is the development of regulations and standards of professional practice designed to protect the environment while protecting the public and its infrastructure from increased weather hazards. Approaches to the prediction of weather trends, the reduction of human impacts on the climate, and the mitigation of the effects of changes beyond our control require integrated global efforts. Adaptation can keep up with the predicted shifts in conditions if it is begun well before it is forced by natural disasters. The predictions for most communities in Canada include more violent winter storms, high intensity rainfalls of short-duration,

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and extended heat waves with the accompanying increased risk of smog, wildfires, tree parasites, severe thunderstorms and tornadoes. Current structural design, farming, and forestry practices as well as water resource management, health standards, land use planning, power supply, and insurance policies were developed for the existing conditions. All these aspects of our society and infrastructure will have to change along with the climate. Water and energy conservation are of primary importance, followed by pragmatic and future-oriented reviews of standards, codes, regulations and other practices. The climate is changing at an unprecedented rate and in ways that are not yet fully understood, hence the difficulty and urgency of adaptation. This chapter focuses on the building industry.

Keywords: Adaptation; climate change; global warming; greenhouse effect; infrastructure; mitigation; thermal energy storage; weather hazard.

3.1. Introduction

Climate change is increasingly apparent. Regional impacts of climate change are being observed. Those commonly cited include extended growing seasons; shifts of plant and animal ranges; earlier flowering of trees, emergence of insects and egg-laying in birds; and local temperature, humidity and wind-speed anomalies. Air temperatures in Alaska and western Canada have increased as much as 3–4 °C in the past 50 years. According to the Third Assessment Report (TAR) of the Intergovernmental Panel on Climate Change (IPCC), the Earth's average surface temperature increased $0.6 \pm 0.2\,^{\circ}\mathrm{C}$ in the 20th century. This trend is expected to continue, with an increase of 1.4–5.8 °C by 2100. Even with "best case" mitigation efforts, some climate change cannot be avoided due to the inertia of the global climate system. Warming will vary by region and be accompanied by significant changes in precipitation patterns as well as changes in the frequency and intensity of some extreme events. Average global sea levels will rise between 9 and 88 cm by 2100. Fifty to seventy per cent of the world's population currently live in low-lying coastal areas.

Even with the mitigation measures underway to cut net emissions of greenhouse gases and so reduce climate change, current predictions see more frequent and more severe extreme weather events. As climate change continues, the prediction and mitigation of climate related hazards will ultimately require adaptation across the entire construction sector. Another response to climate change is the development of regulations and standards of professional practice designed to protect the environment while protecting the public and its infrastructure from increased weather hazards. Approaches to the prediction of weather trends, the reduction of human impacts on the climate, and the mitigation of the effects of changes beyond our control require integrated global efforts.

The adaptation reflects the need to prepare for and respond to the impacts of climate change, and the corresponding recognition that any future global climate change regime will need to address adaptation in a more prominent manner. Defining a new approach to addressing adaptation in a post-2012 regime will be challenging, in part because the international community is only beginning to understand how to effectively respond to the complex socio-economic and environmental impacts that will result from climate change. Adaptation to human-induced climate change is a new process for all countries and concrete experience in applying an integrated approach to adaptation is limited.

Increasing evidence of human contributions to climate change is coming to light. The market pressure of eco-conscious clients and a concerned public is causing manufacturing and construction firms to adopt more environmentally sound engineering practices as a result. Proactive members of every important industry are getting involved with education and research into new technologies and approaches to address design problems with sustainable solutions. The demand for "green" innovations in design is bound to grow with our awareness of our impacts on nature in an increasingly industrialized world.

Even with the preventive measures being developed to protect the environment, the current research and climate models show that we can expect global warming to accelerate causing more frequent and more severe extreme weather events. For professionals involved in construction, the designing of structures for predicted future conditions is a challenge but also an investment that pays off with reduced maintenance and retrofit costs. As climate change continues, the prediction and mitigation of climate related hazards will ultimately require adaptation across the entire construction sector.

Approaches to the prediction of weather trends, the reduction of human impacts on the climate, and the mitigation of the effects of changes beyond our control require integrated global efforts. The sharing of the latest studies, defined risks and new solutions through communication between different governments and various professional disciplines is essential to timely, economical progress. Adaptation can keep up with the predicted shifts in conditions if it is begun immediately and before it is forced by natural disasters.

The predictions for almost all communities in Canada include more violent winter storms, very high intensity rainfalls of short-duration, and extended heat waves with the accompanying increased risk of smog, wildfires, tree parasites, severe thunderstorms and tornadoes. Current structural design, farming, and forestry practices as well as water resource management, health standards, land use planning, power supply, and insurance policies were developed for the existing conditions. All these aspects of our society and infrastructure will have to change along with the climate.

Individuals, enterprises and policies related to planning and infrastructure development need to address climate change with strategies and measures that offset or reduce its effects. Diversified and integrated approaches involving all stakeholders improve adaptive capacities, reduce vulnerability to climate change, and reduce the costs associated with impacts. Successful adaptation is contingent on proper communication and collaboration, sufficient funding and technological capability.

3.2. Climate Change

The Earth has experienced many different climate regimes throughout geological history and will undoubtedly experience them in the future. Carbon dioxide levels have been much greater during past epochs and temperatures have been much higher. Climate change is a naturally occurring phenomenon at a geological time scale and more or less hospitable to varying life forms, including human beings. The pace of climate change is the danger for modern societies.

Key climate variables:

- Temperature.
- Precipitation and atmospheric moisture.
- Extent of land and sea ice and permafrost.
- Sea level.
- Snow cover.
- Patterns in atmospheric and oceanic circulation.
- Extreme weather and climate events.
- Climate variability.
- Habitat.

Potential impacts of the key climate variables:

- Increased average and peak temperatures.
- More varied and severe weather—greater storm related damage and insurance claims.
- Faster sea level rise (5 centuries of rise have already been experienced in Europe).
- Less pack ice in polar regions and permafrost.
- Decreased snow cover harms agricultural areas.

- Increased snow cover increases roof loads and increases building collapses.
- Changed agricultural zones.
- Habitat modification—pressure on fragile species, changed disease vectors.

A change in the net radiative energy available to the global Earth-atmosphere system is termed a “radiative forcing”. Positive radiative forcings tend to have a warming effect while negative radiative forcings have a cooling effect. Greenhouse gases are positive; aerosols and volcanic eruptions have a negative effect. Greenhouse gases persist for decades while aerosols act over weeks.

According to the IPCC the global average surface temperature has increased by $0.6 \pm 0.2\,^{\circ}\mathrm{C}$ since the late 19th century. A causal relationship is suggested between increased greenhouse gas emissions and the observed temperature increase. Current CO_2 and CH_4 concentrations are higher than at any time during past 420,000 years, based on the Vostok Ice Core. Present N_2O concentration has not been exceeded during the past 1,000 years. Halocarbons are anthropogenic compounds and are potent GHG’s in addition to their ODP. However, Montreal Protocol has been successful in curbing their emission.

The IPCC “Hockey Stick” model of temperature increase has been challenged as faulty (e.g., 15th century is warmer than 20th century if data is properly analyzed). Predicting future climate states is like predicting the weather—very difficult especially because of non-linearity. The argument that human activity is causing climate change is one of circumstantial evidence. The correlation between GHG’s and change in temperature is not causation. What causes what? Use of General Circulation Models (GCM’s) to predict future climate states and to postdict past climate states is the basis of predictions. GCM’s are developing as the understanding of natural systems and level of computing power increases.

“The dominant issue in global warming, in my opinion, is sea level change and the question of how fast ice sheets can disintegrate. A large portion of the world’s people live within a few meters of sea level, with trillions of dollar in infrastructure. The need to preserve global coastlines sets a low ceiling on the level of global warming that would constitute dangerous anthropogenic interference” [3].

The Planetary Energy Balance [3] of Incoming Solar (340 W/m^2) minus Reflected (101 W/m^2) minus Radiated (238 W/m^2) = 1 W/m^2. This excess energy warms the oceans and melts glaciers and ice sheets. The GHG component is 2 W/m^2. The amount of heat required to melt enough ice to raise sea level 1 m is about 12 Watt-years (averaged over the planet)—energy that could be accumulated in 12 years if the planet is out of balance by 1 W/m^2 per year.

Earth's climate has swung repeatedly between ice ages and warm interglacial periods. We are currently in the Holocene interglacial (12,000 yrs old), at the peak temperature. The previous interglacial period (Eemian) was warmer than the Holocene with a sea level 5–6 m higher. One additional W/m^2 of forcing, over and above that of today, will take global temperature approximately to the maximum level of the Eemian—thus providing a proxy estimate of sea level considered by Hansen to be the most critical metric of climate change impact.

Based on paleoclimate evidence, the highest prudent level of additional warming is not more than 1 °C. This means additional climate forcing should not exceed 1 W/m^2. Another way to work with climate scenarios is to examine changes in individual climate forcing agents. In the 1980s there was a growth rate of 0.5 W/m^2 per decade that declined in the 1990s to 0.3 W/m^2 supposedly due to a decline in CFCs. CO_2 is growing between 1.5 and 2 ppm per year although the CH_4 growth rate is down by 2/3 in the past 20 years.

The long-term reduction of CO_2 is a greater challenge, as energy use will continue to rise. Continued efficiency improvements are needed along with more renewable energy and new technologies that produce little or no CO_2 or that capture and sequester it. Study of the Earth's climate suggests that small forces maintained long enough can cause large climate change (nonlinear effects, 29th day). The positive aspect of this situation is that mitigation efforts can make a real difference in the rate and magnitude of climate change.

Climate change is happening. If current climate change is natural, then all we can do is try to adapt. If human activity is causing or contributing to climate change, then mitigation measures can make a difference.

3.3. Climate Change Implications for Engineers

Engineers involved with providing society's infrastructure requirements face new challenges and new opportunities due to shifts in climate occurring around the world today. Building codes already specify minimum health and safety requirements for some key climate variables such as heating and cooling design temperatures; heating and cooling degree days; rainfall and snow loads; and wind pressures. Predicted changes in these variables at specific locations are not usually available. Regional scenarios give a general trend but lack precision and verification. Eco-conscious clients and a concerned public are causing manufacturing and construction firms to adopt more environmentally sound engineering practices. Proactive members of every important industry are getting involved with education and research into new technologies and approaches to address design problems with sustainable solutions.

Every engineering discipline has specific climate change issues, preventive measures and adjustments to impacts. Some of these are summarized in

Appendix. For example, lighting accounts for roughly a third of the electricity demand in buildings. Many energy efficient lighting products and ballasts are available. Better lighting controls are also effective in reducing electricity use. New design standards will also be effective in creating better illumination while increasing energy efficiency.

Effective mitigation of and adaptation to the impacts of climate change require a common set of response priorities. Water and energy efficiency are of primary importance, followed by pragmatic and future-oriented reviews of standards, codes, regulations and other practices. Measures to assess risk and manage durability need to be developed and integrated into practice. Finally, emergency preparedness and response programs need to be further improved as extreme weather risks are becoming less predictable. The climate is changing at an unprecedented rate and in ways that are not yet fully understood, hence the difficulty and urgency of adaptation.

Another response to climate change is the development of regulations and standards of professional practice designed to protect the environment at the same time as protecting the public and its infrastructure from increased weather hazards. When these codes of conduct come into effect they force changes that must legally be implemented immediately. Although they usually represent the minimum required for due diligence, they may be more expensive to implement at the last minute than changes responsibly planned ahead based on predictions of future conditions.

Adaptation can keep up with the predicted shifts in conditions if it is begun immediately and before it is forced by natural disasters. The predictions for most communities in Canada include more violent winter storms, high intensity rainfalls of short-duration, and extended heat waves with the accompanying increased risk of smog, wildfires, tree parasites, severe thunderstorms and tornadoes. Current structural design, farming, and forestry practices as well as water resource management, health standards, land use planning, power supply, and insurance policies were developed for the existing conditions. All these aspects of our society and infrastructure will have to change along with the climate.

Energy is used in the construction, operation, maintenance and renewal of buildings. Energy efficient building technologies from construction through to deconstruction are under development. Some emerging and advanced technologies are listed on the Greentie web site [5]. Some of the technologies are listed below.

- Agriculture.
- Buildings.
- Coal combustion.
- Combined cycle.
- Combined heat & power.

- Combined renewable energy technologies.
- Electrical.
- Forestry & energy crops.
- Fuel cells.
- Gas cleaning systems.
- Geothermal energy.
- Heat recovery & storage.
- High temperature technologies.
- Hydroelectricity.
- Hydrogen.
- Industrial technologies.
- Industrial waste.
- Landfill gas.
- Lighting.
- Municipal waste.
- Nuclear technology.
- Ocean energy.
- Solar energy (heat).
- Solar power.

3.3.1. EMERGING AND ADVANCED TECHNOLOGIES [5]

Short descriptions of some building related technologies are given. Application of these technologies will make a significant contribution to mitigation efforts.

Buildings use about 30% of the total world's energy resources. Existing buildings can typically cut energy usage by 20% or more by intelligent energy management. New buildings can be built to cut energy usage by more than 50% and remain economical.

Renewable electricity technologies can supply energy to remote or off grid communities. They can also feed electricity into the grid. Technologies include photovoltaic, wind and low head hydro.

Fuel cells can be highly efficient in generating electricity on-site as well as providing waste heat. Building applications are in the demonstration phase and offer security of electricity supply.

Geothermal energy or ground source heat pumps extract energy from the earth for heating and cooling. Combined with energy storage this technology offers one of the more efficient means of supplying building energy requirements.

Hydrogen is emerging as a potential future energy medium with applications to all energy sectors. It is used to feed fuel cells. The major question is the source of the hydrogen and the energy required to produce it. Ultimately it is hoped to produce hydrogen from renewable energy sources.

Landfill gas is produced as a result of organic wastes decomposing in landfill sites. It can be recovered at an inexpensive cost of for direct use as a boiler fuel, converted into electricity with a microturbine or upgraded to a higher value fuel gas. This is a very effective mitigation technology since methane is 21 times as powerful as carbon dioxide as a greenhouse gas.

Lighting is one of the single largest end uses for electricity and thus greenhouse gases. Energy efficient design and products combined with intelligent controls can reduce electricity demands and cut cooling loads.

Solar thermal energy can also be used to economically provide heat for a variety of applications. Solar cooling technologies are also available. Through the use of climate-sensitive building design, solar energy can be used for space heating, natural daylighting and even ventilation.

3.4. Climate Change: Dealing with the "Inevitable"?[2]

Dealing with the inevitable is adapting or adjusting to the effects of climate change to reduce the consequences.

3.4.1. ASPECTS OF ADAPTATION

- Develop approach and practices for protecting and improving existing construction against effects of climate change.
- Develop approach and practices for design, operation and maintenance of buildings (such as additional cooling requirements in the summer and heating in the winter).
- Revise codes, such as flood plain mapping and climate data and return frequencies for hazard-prone areas, adjusting to new realities, i.e., 100-year floods become 500-year floods, higher snow and wind loads.
- Consider land use restriction on new construction, especially for floodplains, coastal shoreline, landslide prone areas.

Three-step approach for protecting existing buildings

- Screening—to set priority (ranking) for detailed evaluation needs (based on building's location, type and use of the building, building age, A/M/E systems, etc) .
- Evaluation—to determine a building's deficiency against effects of climate change.
- Retrofitting—improve a building's performance against effects of climate change.

Aspects of Mitigation

- Develop and adapt energy-saving technologies.
- Construction: material selection (minimum use of natural resources), design for disassembly, efficient and durable envelope, durability.
- O&M: clean renewable energy, energy efficient HVAC and lighting systems.
- Building renewal/deconstruction: waste management, deconstruction practice, effective and efficient recycling.

Environmental Goals for Energy Efficient Buildings

- Energy and resource efficiency.
- Water conservation.
- GHG reduction.
- Waste reduction.
- Recycling construction materials.
- Safety and health (Indoor Environmental Quality).
- Community sustainability.

The Passive Energy Saver

- Building materials—minimum use of natural resources.
- Building structure—durability, adaptability.
- Building envelope (walls, roofs, windows)—adequate insulation, air leakage prevention, durability, etc to minimize energy consumption during O&M of buildings, while maintaining interior environment healthy and comfortable for the occupants.
- Waste reduction at the design stage (design for disassembly) and the deconstruction stage.

The Active Energy Saver

- Clean and renewable electricity.
- Efficient heating, ventilating and air conditioning.
- Energy efficient lighting and office equipment.
- Ultra-low energy new buildings.
- Water conservation.

Building Materials: Use of Industrial By-Products

- Supplementary cementing materials.
- Producing 1 tonne of cement leads to 0.9 tonne of GHG emission.
- 2,200,000 tonnes of fly ash available, only 450,000 tonnes used.
- In Ontario, 40% of concrete produced, 20% of fly ash used (used by 60% contractors).
- Reduce use of natural resource and GHG emission.

Waste Reduction: Resource Recovery

- *Lumber:* mainly reused, re-graded by lumber grader.
- *Steel:* mostly recycled, not reused (no way of qualification for reuse—opportunity: can it be stamped at production or a re-grading standard for reuse?).
- *Concrete:* 50% of construction & demolition waste is concrete.
- Economic benefits and reuse versus recycling.

Performance Requirements for Buildings

- Health .
- Safety.
- Security.
- Energy efficiency.
- Sustainability.

3.5. Concluding Remarks

Climate Change is changing Engineering Practice. Already mitigation efforts have been applied to energy efficiency improvements in the building stock (see Chapter on Sustainable Buildings). Adaptation efforts are beginning. Many of these efforts will be carried out through national building codes as the design values for key climatic variables change. Proactive designs will also be necessary for predicted changes.

Thermal energy storage has obvious mitigation benefits. It is necessary when using renewable or natural energy. It can tap a vast quantity of waste and natural energy sources. It will be even more important during post-Kyoto mitigation efforts.

Although mitigation efforts may influence the timing and severity of climate change, adaptation efforts are also necessary. Climate change will lead to inevitable natural hazards, but need not necessarily lead to natural disasters.

Appendix: Impacts of Climate Change on Architectural and Engineering Practices

1. *Disciplines:* All
   - Climate Change Issue.
     - Global Warming and increased Climatic Variability due to Greenhouse Gas (GHG) Accumulation in the Atmosphere and Increased Radiation due to Ozone Depletion by Chlorofluorocarbons (CFC).

- Preventive Measures.
  - GHG and CFC Reduction Research.
  - Energy Conservation, Exceeding Minimum Code Requirements, & Development of New Sustainable Technologies.
  - Broadened Education (e.g., experience with Engineers Without Borders).
  - Interdisciplinary and International Consultation (as recommended in the "Climate Change Strategy and Technology Innovation Act"—USA).
  - Consideration of Indirect and Delayed Effects (such as those from GHGs and CFCs).
  - Seeking "Spill-Over" Technologies from other fields as Solutions or to Derive more immediate Benefits from Environmental Technologies.
  - Political Initiatives to promote all the above (e.g., Canada's $1 billion in Kyoto-compliance incentives and the "Clear Skies and Global Climate Change Initiatives" of USA).
- Adjustments to Impacts.
  - Knowledge Sharing, Consultation, and Research into Sustainable Climate Change Adaptations such as Water Conservation and Safer Structures, e.g., the Centre for Sustainable Infrastructure which studies:
    1. efficient motors & transformers (India and China).
    2. eco-homes for energy-efficient, affordable housing (South Africa).
    3. decentralized energy-efficient wastewater treatment (USA).
    4. storm water management and erosion control for transportation systems (USA).
    5. earth brick technology transfer for low-income house construction (South Africa).
    6. blended cements to reduce energy consumption (USA).
    7. transportation tech.to reduce GHC emissions (India).
    8. community transportation strategies for GHC reduction (Pennsylvania).
    9. conversion of refineries to unleaded fuel production (South Africa).
    10. International Network & Standard Method for Natural Disaster Investigation.
- Climate Change Issue.
  - Climate Change in general.
- Preventive Measures.
  - Participatory Approaches inclusive of all stakeholders—both before an activity is initiated and throughout (e.g., women in developing countries).

    - Long-Term Engagement.
    - Consideration of Multiple Scales—local, landscape, national, regional, and global.
    - Involvement of Other Disciplines in teams.
    - Appropriate Technology that is Adapted to the social, political, and cultural Context.
    - Engineering Systems that consider not only Supply Management but also Demand Management, e.g., we need to focus on using water and electricity efficiently as well as producing more
      1. designs that are Easily Maintained.
      2. designs that Consider the Technical Capacity of People who must Maintain them, and that have a Capacity-Building Component.
      3. Advancing Local Adaptation by Learning from the "Spatial Analogues" of Other Communities and Commercial Operations that have already Adapted Successfully and Economically to the Anticipated Climate
2. *Discipline:* Development of Standards
   - Climate Change Issue.
     - Climate Change in general.
   - Preventive Measures.
     - New & Stricter Environmental Standards based on UN Protocols: Kyoto (GHGs) & Montreal (ozone-depleting chemicals: CFCs, etc) (e.g., Canada's New Model Energy Code for Housing that reduces GHGs).
     - Guidelines for Sustainable.
     - Professional Practice (e.g., ASCE Policy Statement 418 "The Role of the Civil Engineer in Sustainable Development" & "The 12 principles of Green Engineering").
   - Adjustments to Impacts.
     - Standards for Durability including resistance to long-term effects and extreme weather impacts.
     - Revision of Safety and Fire Codes for buildings and other structures.
3. *Discipline:* Climatology/Meteorology
   - Climate Change Issue.
     - Climate Change in general.
   - Preventive Measures.
     - Continued Detailed Study of the Changing Situation.
     - Improved methods of Sharing Information.
   - Adjustments to Impacts.
     - Development of more powerful Predictive Tools (e.g., for thunderstorms, tornadoes & hailstorms, & ones that help prepare for the impacts of events like "Ice Storm'98").

    - Contribution to an International Network and a Standardized Methodology for the Investigation of Natural Disasters to Learn from Rare Severe Weather Events (like "Ice Storm'98").
4. *Discipline:* Infrastructure Design
    - Climate Change Issue.
        - Climate Change in general.
    - Preventive Measures.
        - Accelerated Technology Development.
        - Immediate Implementation of Adaptive Measures.
        - Encouragement of Win–Win Actions.
        - Contingency Planning Assuming GHG Emissions Are Costly.
        - Experimentation with Land Use and Pricing to affect demand.
    - Adjustments to Impacts
        - Accelerated Technology Development.
        - Immediate Implementation of Adaptive Measures.
        - Seeking Win–Win Solutions.
5. *Discipline:* Civil Engineering
    - Climate Change Issues.
        - Global Warming and the accompanying increase in climatic extremes in a relatively short period of time and causing a disproportionate increase in disaster losses.
    - Preventive Measures.
        - Life Cycle Analysis of projects to develop Design and Construction processes that address Energy Efficiency and the Reduction of GHG Production.
        - Immediate Start to a Steady process of Adaptation while there is Still Time to do it Economically.
    - Adjustments to Impacts.
        - Civil Infrastructure made to withstand the best predictions of likely Extreme Weather Events and with adequate Durability for their exposure to Worsened Average Conditions (CAN/CSA S478 is the Canadian Standards Association's "Guideline for Durability in Buildings").
        - Emergency Service Procedures and Facilities undergoing ongoing modification, as extreme weather risks are too unpredictable to be fully contained (IDNDR's research units for hazard mitigation and/or disaster preparedness are in place at the Université de Québec à Rimouski, the University of Manitoba, & the University of British Columbia).
6. *Discipline:* Geotechnical Engineering
    - Climate Change Issues.
        - Weakening of Foundations, Diminishing Slope Stability, Erosion & Landslides from Increased Rainfall.

  - Reduced Bearing Capacity from Lowered Water Tables due to Increased Evaporation and/or Reduced Precipitation.
  - Erosion of Coasts and Inland Shores by Rises in Water Levels, by Increased Waves due to Higher Winds, and by Floods (in combination with high tides in the case of Howe Sound, British Columbia).
- Preventive Measures.
  - Retaining Walls, Sea Walls, etc or carefully selected Planted Trees on slopes, shores and coasts for Soil Retention.
- Adjustments to Impacts.
  - Awareness of changed climate and Increased Vigilance of changed Hazards in Applying Existing Technologies to Mitigate New Risks.
- Climate Change Issue.
  - Weakening of Foundations by irreversible Permafrost Thaw due to Higher Average Temperatures in the North.
- Preventive Measures.
  - Protection Against Melting in the short term (one attempt in Alaska, painting highways white to reflect the sun's heat, failed because of the distracting glare).
- Adjustments to Impacts.
  - Monitoring of older structures.
  - Projecting areas of greatest threat.
  - Budgeting for the expensive changes.
  - Retrofitting or Rebuilding as needed.
  - Designing structures to Allow for the Thaw.

7. *Discipline:* Municipal Engineering
   - Climate Change Issues.
     - Altered Precipitation Patterns causing More Severe Storm Damage.
     - Worse Snowstorms with potentially Reduced total Snowfalls.
     - Rising or Falling Water Levels—even to the point of Flooding or Drought.
   - Preventive Measures.
     - Combined Municipal Water and Sewage Treatment Plants in the interest of Water Conservation and Reduction of Pollution of Natural Waterways.
     - Integrated Solutions that consider the water needs of all communities Sharing given Water Resources, leading to Improved Irrigation and Management of Wells (including the decommissioning of old wells to prevent contamination of the water table).
   - Adjustments to Impacts.
     - Reduced Need for Snow Removal.
     - Increased Need for Emergency Preparedness (in case of events like Ice Storm'98).

- ◦ Contribution to an International Network and a Standardized Methodology for the Investigation of Natural Disasters to Learn from Rare Severe Weather Events (like "Ice Storm'98").
- ◦ Improved Drainage.
- ◦ Higher Capacity Sewers and Treatment Plants.
- ◦ Separate Storm Sewers.
- ◦ Protection of the Drinking Water Supply: this is the most pressing concern especially in developing island nations like Tuvalu in the Pacific but even in the Netherlands where Protection Against Sea Water Encroachment in the Water Supply is harder to ensure than Prevention of the Flooding of Land near or below sea level.

- Climate Change Issues.
  - Global Warming from Vehicle-Generated Greenhouse Gases.
- Preventive Measures.
  - Experimentation with New Solutions that Anticipate Future Conditions.
  - Reduction of Travel Demand by improved Street Layouts, creative Land Use solutions, & better Transit.
  - Increased Transit Use & Cycling through the use of Tolls for Motor Vehicles & improved Convenience of Bicycle and Transit Routes.
- Climate Change Issues.
  - Increased Heat Waves and More Severe Precipitation Events due to Global Warming.
- Preventive Measures.
  - Increased Parkland through zoning, and incentives for buildings to have "Green Roofs" with water retention and heat reflection.
- Adjustments to Impacts.
  - Adaptation of better Emergency Preparedness Measures (e.g., to increased incidents of wildfires near residential areas, increased heat-related illnesses, & more severe summer and winter storms).

8. *Discipline:* Hydrotechnical Engineering
   - Climate Change Issues.
     - Increased Extreme Precipitation Events leading to Worse Flooding (e.g., China 1995 & Cadiz, Spain 1990s).
     - Raised or Lowered average Precipitation Rates (e.g., central prairies 1999).
     - Changes in Floodplains.
     - Lowered Water Levels from Evaporation (possibly affecting shipping, water supply & quality, & hydropower production in the Great Lakes—St. Lawrence system).
     - Droughts (e.g., Spain 1992, Portugal 2005).

  - Severe Snowstorms and Ice Storms (e.g., Eastern Canada & USA 1998).
  - Higher Waves due to Stronger Winds.
  - Accelerated Sea Level Rise due to Ice Cap and Glacier Melt.
- Preventive Measures.
  - Improved Predictive Tools both for Design Tolerances and the Management of the Impacts of Natural Disasters.
  - Hydrological Studies to predict Water Supply and Flood Protection Needs, and to Avoid exacerbating water problems with Poorly Conceived Solutions (e.g.: desertification in one area caused by channeling water to irrigate another—such as in Iran; or flooding worsened downstream in urban areas by dykes protecting farmland—like Poland's 1997 flood).
  - Waterway Designs that do not narrow, block or accelerate natural flows, causing erosion or silting of channels.
- Adjustments to Impacts.
  - Development of Reliable Predictions of water-related Threats.
  - Continued Research into Mitigation Measures against effects of Precipitation, Waves, Floods and the Sea Level's Rise (e.g., the North Sea drilling platforms that are designed by oil companies for a one-meter rise in sea level).
  - Identification and Protection of Coastlines most sensitive to Erosion (e.g., Charlottetown, P.E.I. & near Vancouver, B.C.).
  - Development of an International Network and a Standardized Methodology for the Investigation of Natural Disasters to Learn about the Impacts of Rare Severe Events (like the flood in Quebec in 1996).
  - Predictions of possible Advantages (such as a lengthened shipping season in the Great Lakes).

9. *Discipline:* Structural Engineering
   - Climate Change Issue.
     - Changes in Local Climatic Averages and in Frequency and Severity of Extreme Weather Events, mostly due to Global Warming.
   - Preventive Measures.
     - Exceeding the Current Minimum Requirements of Codes to Adapt to the Latest Understanding of Climate Change.
     - Reduced Life Cycle Impact of structures on the environment, especially by Choices of Materials (e.g., depending on local availability and forest management practices wood can be a versatile, sustainable alternative) and Construction Methods that Minimize the Production of GHGs:—Construction Waste Management including Recycling and Reuse.
     - Life Cycle Costing to Optimize Durability and Waste Reduction.

  - Consideration of Embodied Energy of Materials including processing and transportation.
- Adjustments to Impacts.
  - Exceeding the Current Minimum Requirements of Codes to Adapt to the Latest Understanding of Climate Change.
  - Selection from existing or innovative Construction Methods, Schedules and Designs to Adapt projects to the Latest Understanding of the Changing Risks due to Climate at each individual site (e.g., the passive damping system at the Taipei Financial Center).
  - Targeted Durability based on Life Cycle Analyses considering the best Predictions of Future Conditions (in terms of changed averages and new extremes).
  - Possible Longer Building Season with the warming trend (e.g., in Ontario and Quebec).
  - Monitoring of Structures during extreme weather (e.g., Ice Storm'98) to Fine-Tune Climate Design Data and Contribute to the International Investigation of Natural Disasters.
  - Developing broadly applicable Emergency Procedures and Facilities.

10. *Discipline:* Materials Engineering
    - Climate Change Issues.
      - Global Warming Caused by GHGs from Power Generation and Materials Processing.
    - Preventive Measures.
      - Improved Materials Processing and Recycling Techniques that Reduce Energy Consumption and GHG Production (e.g., reduction of Portland cement use by substitution of industrial waste products such as fly ash, which has several side-benefits).
11. *Discipline:* Mining Engineering
    - Climate Change Issues.
      - Indirect Climatic Effects of Deforestation and Pollution.
    - Preventive Measures.
      - Continued Efforts to Develop Practices to Protect the Environment such as Reforestation of old mine sites, Confinement of Tailings and Minimized Disruption and Contamination of Natural Streams.
    - Climate Change Issues.
      - Warming of the Climate Causing Permafrost Thaw and Reduced Winter Ice in Arctic Waters.
    - Adjustments to Impacts.
      - Opportunities for easier Mining and Oil Exploration in Permafrost Areas.
      - Reduced Need for Ice-Breakers in the Northwest Passage.

  - Adaptations to potentially Increased Risks of Landslides, Flooding and Erosion.

12. *Discipline:* Mechanical Engineering
    - Climate Change Issues.
      - Global Warming from GHGs.
      - Increased Need for Air Conditioning and Refrigeration resulting from the warming trend.
      - Ozone Depletion by CFCs and other chemicals, especially from AC and refrigeration systems.
    - Preventive Measures.
      - Improved Efficiency of Motors and Mechanical Systems to Reduce Energy Consumption and Harmful Emissions from the use of hydrocarbon-fuels (e.g., natural gas, biodiesel & hybrid engine vehicles; improved, popular mass transit; "Energuide-Award"-winning cars that are cheaper to run; Stirling engines or microturbines as auxiliary CHP—combined heat and power—generators running on natural gas or waste gases from landfills, pipeline gas-flaring locations and farms).
      - Adoption of Technologies that Do Not Contribute at all to Global Warming and Ozone Depletion, especially for Sustainable Electrical Power Generation (e.g., 100% renewable resources such as geothermal, wind, solar, wave & tidal power, & hydro-dams; or abundant, non-polluting energy sources like nuclear fission reactors, fuel cells and eventually fusion reactors), Transportation (e.g., fuel-cell vehicles, & alternative fuels such as biodiesel & coconut oil), & HVAC (e.g., passive solar heating/cooling, heat-driven absorption coolers & CHP).
    - Adjustments to Impacts.
      - Generators Powered by Waves, Tides or Wind to Harness the Impacts of Climate Change where the Effects have made such installations Feasible in New Locations.
13. *Discipline:* Refrigeration Engineering
    - Climate Change Issues.
      - Ozone Depletion by CFCs and other chemicals.
    - Preventive Measures.
      - Designing systems that use Ozone-Friendly Refrigerants.
    - Adjustments to Impacts.
      - Designing More Efficient Refrigeration systems.
14. *Discipline:* Industrial Design of aerosol cans, fire extinguishers and foam products
    - Climate Change Issues.
      - Ozone Depletion by CFCs and other chemicals.

- Preventive Measures.
  - Switching to Ozone-Friendly Compressible Gases and Propellants
  - Using Atomizers for Sprays.

15. *Discipline:* Environmental Engineering
  - Climate Change Issues.
    - Climate Change in general.
  - Preventive Measures.
    - Ongoing Studies of Environmental Impacts from Engineering Practices and the Provision of Infrastructure.
    - Rules or Laws of Professional Conduct that Protect the Environment from Causes of Climate Change (e.g., ASCE Policy Statement 418 "The Role of the Civil Engineer in Sustainable Development").
    - Guidance on Sustainable Practices (e.g., the 12 Principles of Green Engineering).
16. *Discipline:* Waste Management Engineering
  - Climate Change Issue.
    - Global Warming from GHGs.
  - Preventive Measures.
    - Development of Re-use, Recycling (e.g., redirected construction wastes) and Composting Programs to Reduce Landfill Loads.
    - Research into Improved Landfill Efficiency, Accelerated Waste Degradation (e.g., leachate recycling & bioreactor landfills), the Generation of Electricity from Landfill Gases (e.g., microturbines & Stirling engines), & "Mining" Landfills for Raw Materials.
17. *Discipline:* Chemical Engineering
  - Climate Change Issues.
    - Climate Change in general.
  - Preventive Measures.
    - Reduction of GHGs, CFCs and other climate-affecting Pollutants generated in the Delivery of Civil Infrastructure Projects and elsewhere.
    - Development of Alternate Fuels for Transportation, Electrical Power Generation and various Industrial Processes (e.g., biodiesel from renewable sources such as vegetable oils).
    - Engineering of Existing Fuels to make them Less Harmful.
18. *Discipline:* Natural Resource Management
  - Climate Change Issues.
    - Climate Change in general.
  - Preventive Measures.
    - Development of Integrated Systematic Engineering Approaches to Sustainable Resource Exploitation (e.g., life-cycle analysis, soft-systems analysis) in fields such as Mining, Forestry, and Agriculture,

especially using Geotechnical and Hydrotechnical Expertise and the latest Climate Information Systems (refer to the relevant areas for more details).
    - Exploitation of "Spill-Over" Technologies from other sectors (e.g., from mining or defence to agriculture or forestry).
    - Involvement in S&T at the International Decision-Making Level and at the Local Resource Management Level.
    - Focus on Technologies Adapted and Integrated into the Local Social, Political, and Cultural Contexts as well as meeting the Physical Constraints while not Relying on Prolonged Outside Involvement.
  - Adjustments to Impacts.
    - Case by Case Adapted Solutions, Taking Advantage of Knowledge Gained from Experience Worldwide shared through Information Technologies such as Databases and through Direct Consultation Between different Disciplines and Governments.
    - Adaptation of Some Traditional Practices to the changed climates and Abandonment of Others (such as slashing and burning).

19. *Discipline:* Agriculture
  - Climate Change Issues.
    - Climate Change and its Worse Effects such as Erosion and Desertification.
    - Increased Evaporation from the Warming Trend in North America possibly causing the Moisture Balance to Decrease 35% this century despite increased precipitation (during the same timeframe, GIS Studies and Computer Modeling of Global Warming show the Okanagan Valley's crop water demand could potentially increase by more than 35%).
    - Increased Risks of Pests, Diseases, and Wildfires in Canada, from the Hotter Climate.
  - Preventive Measures.
    - Development of Agriculture Techniques that have Less Impact on Ecosystems.
    - Capturing and Processing Livestock Methane (e.g., to generate electricity).
    - Exhausting Carbon Dioxide from Engines or Heating (or CHP) into Greenhouses to Reduce GHG Contribution and Help Plants Grow.
    - Elimination of Demand for the Slashing and Burning of Forests through Maintenance of Soil Fertility (e.g., through crop rotation, irrigation).
    - Reversal of some of the Effects of Deforestation by the Afforestation of Unused Farmland.

    - Containment of Desertification by Irrigation, Limiting Grazing, and Planting Grasses and Bushes that Hold the Soil.
  - Adjustments to Impacts.
    - Selection of Crops and Farming Methods Adapted to Altered Climates.
    - Adaptation of Traditional Methods and New Approaches to Water Conservation.
    - Farming Opportunities due to an Extended Agricultural Growing Season in Ontario and Quebec and perhaps Atlantic Canada (e.g., allowing the selection of alternative crops).
    - Possible Benefits from Improvements in Plants' Water Utilisation and Overall Yields for some crop types due to the Increase in Atmospheric CO_2 Concentrations.
    - Possible Agricultural Opportunities in the Mackenzie Basin and other previously Less-Developed Areas.

20. *Discipline:* Forest Management
  - Climate Change Issue.
    - Increased Heat Waves.
  - Preventive Measures.
    - Reforestation/Afforestation.
  - Adjustments to Impacts.
    - Emergency Preparedness Measures adapted to increased incidents of forest fires, insect infestations and the impacts of soil erosion.
    - Potentially Longer Growing Season.
    - Studies and Measures to Ensure Adequate Water Supplies to Forests.
21. *Discipline:* Building Engineering
  - Climate Change Issue.
    - Global Warming and Ozone Depletion.
  - Preventive Measures.
    - R&D and Investment in Energy-Efficient and Ozone-Friendly Building Technologies that also Save Money in the Life-Cycle Analysis and Predict the Trend of Environmental Standards (e.g., energy-smart buildings that use daylighting, other renewable energy resources, energy-efficient technologies and other sustainable features in new constructions, rehabilitations and retrofits).
    - CHP Generation with optional Grid-Independence.
  - Adjustments to Impacts.
    - Designing for Controlled Durability Adapted to the best Predictions of Future Weather Conditions.
22. *Discipline:* Illumination Engineering
  - Climate Change Issue.
    - Global Warming.

- Preventive Measures.
  - Higher-Efficiency Lighting that uses Less Power—reducing GHG emissions—and generates Less Heat—reducing cooling load (e.g., CMHC's publication of the first book on passive solar house designs for Canada; reduction of unneeded illumination in offices by the installation of individual lighting controls).
- Adjustments to Impacts.
  - Application of the latest illumination standard of the IESNA (www.iesna.org).
  - ANSI/IESNA-RP-04.

23. *Discipline:* HVAC (Heating, Ventilation and Air Conditioning) Technology
    - Climate Change Issues.
      - Global Warming, which is Increasing Weather Variability and Cooling Loads.
    - Preventive Measures.
      - Advanced Controls.
      - Innovative Duct Layouts.
      - Zoning of Modular Systems (e.g., individual environmental controls at each workstation).
    - Adjustments to Impacts.
      - HVAC Systems that can handle a Wide Range of Temperatures.
      - Increased Cooling Capacity.
      - Decreased Cooling Demand.
24. *Discipline:* Building Envelope Design
    - Climate Change Issues.
      - Global Warming and Weather Variability.
    - Preventive Measures.
      - Development of Durable & Healthful Building Enclosures that Reduce Energy Costs.
      - Building Envelope Research (e.g., at proposed centre at Oak Ridge National Laboratory, USA).
      - Long-Term Monitoring of untried building Products & Methods.
      - Consideration of the Embodied Energy of the Materials while Extending the Useful Life of Structures.
      - Providing for Change of Use at some later stage.
      - Research on Flexibility in Design, allowing Economical Rearrangement and Reuse of Components and Assemblies.
      - Increased Weather-Tightness for Fuel Economy.
    - Adjustments to Impacts.
      - Protection from the Humidity Trapped by Weather-Tightness (including threats to occupant health & building durability).

  - ◦ Designing for Extreme Climatic Events to Provide Safe Structures but Designing Components & Assemblies for Climatic Averages to Cope with their Gradual Deterioration.
  - ◦ Retention of the Economic Benefit of Lowered Energy Consumption by an accurate Life Cycle Analysis.
  - ◦ Consideration in LCA for Climate Change Effects on Humidity Control (i.e., sizing of heating and cooling systems made to handle extreme ranges of heat, cold and wind), Cladding Ventilation/ Dehumidification Measures, Weather-Dependent Attacks on Cladding (from Radiation, Moisture and Pollutants such as acid rain), Correct Materials Choices, & Periodic Maintenance.

25. *Discipline:* Real Property Management
    - Climate Change Issues.
      - ◦ Threats caused by Climate Change in general and Impacts of Infrastructure Design on the Environment.
    - Preventive Measures.
      - ◦ Life Cycle Analysis to help Lower Energy Consumption and Global Warming through Considerations for Embodied Energy and Waste Reduction.
      - ◦ Development of new and existing Risk Assessment Tools (e.g., Natural Hazards Electronic Map and Assessment Tools Information System—NHEMATIS), Risk Management Tools inclusive of all stakeholders in the process (e.g., CAN/CSA Q850-97 for risk management in Canada, CAN/CSA-Z763-96 specifically for environmental concerns) & Effective Risk Communication.
    - Adjustments to Impacts.
      - ◦ Life Cycle Costing to Reduce Economic Impact of Adaptation.
      - ◦ Risk Assessment, Management & Communication.
26. *Discipline:* Architecture
    - Climate Change Issue.
      - ◦ Global Warming.
    - Preventive Measures.
      - ◦ Priority on Environmental Protection as a Design Criterion.
      - ◦ Staying Informed as to the Latest Environmentally-Friendly Technologies to Make Responsible Choices (e.g., "green roofing", natural & recycled cladding, high-efficiency lighting, photoluminescent emergency lighting, natural ventilation, individualized controls for lighting and HVAC systems in office spaces, passive solar heating, daylighting windows and floor-plans: the CMHC has published the first book on passive solar house designs for Canada, and the American DOE promotes "Zero Energy Homes").

- Adjustments to Impacts.
  - Continued Suiting of the Design to the Environment, as always, but with an Awareness now of How that Environment is Changing.

27. *Discipline:* Information Technology
    - Climate Change Issue.
      - Climate Change in general.
    - Preventive Measures.
      - Continued Development of Computerized Modeling (e.g., to be able to include tornadoes, thunderstorms, hailstorms, & iceberg migration).
      - Monitoring and Measuring Technologies using the latest in Satellite-Based Remote Sensing (e.g., NASA's EOSDIS).
      - Geomatics, and High-Speed Communications to Gather, Process (e.g., by Distributed computing) and Disseminate (e.g., via the WFEO's virtual engineering library & the iiSBE web site) useful Climate Information Globally (especially including developing countries, thanks in part to the Clean Energy Initiative of the World Summit on Sustainable Development) to assist with Risk Analysis, Design (e.g., of the Confederation Bridge over ice-covered waters), Emergency Preparedness and Post-Disaster Studies.
28. *Discipline:* Port Authorities
    - Climate Change Issue.
      - Higher Sea Levels and More Severe Storms due to Global Warming.
    - Adjustments to Impacts.
      - Retrofit of Seaports to Allow for Higher Water Levels and Worse Storms.
29. *Discipline:* Shipping Regulation
    - Climate Change Issue.
      - Shortened Period of Ice Cover, Thinner Ice, and Higher Sea Levels.
    - Adjustments to Impacts.
      - Possible Longer Shipping Season in the Great Lakes, the St. Laurence Seaway, and the Northwest Passage.
30. *Discipline:* Shipping Regulation and Coast Guards
    - Climate Change Issue.
      - Worse and More Frequent Storms.
    - Adjustments to Impacts.
      - The development of more reliable Weather Advisory Systems.
31. *Discipline:* Fishing Quota Regulation
    - Climate Change Issue.
      - Warmer Waters and Longer Summers.
    - Adjustments to Impacts.
      - Possible Higher Sustainable Fishing Quotas.

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4. GLOBAL WARMING IS LARGE-SCALE THERMAL ENERGY STORAGE

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Abstract. The purpose of this paper is to present a controversial and CO_2 free explanation to global warming and to show that global warming means that large-scale thermal energy storage. Global warming is here explained by dissipation of heat from the global use of non-renewable energy sources (fossil fuels and nuclear power). Resulting net heat is thus released into the atmosphere. A minor part of this heat is emitted to space as outgoing long-wave radiation while the remaining is heating the Earth. Some of this heat is accumulated as sensible, i.e., by heating air, ground, and water. The rest is also stored as latent heat, i.e., in the form of vapor in the air and in the melting of the large ice fields of the planet.

Keywords: global warming, heat accumulation, net heating, sensible, latent, storage

4.1. Introduction

A global rise in temperatures is undoubtedly real according to IPCC, the Intergovernmental Panel on Climate Change (Macilwain, 2000). An increasing body of observations gives a collective picture of a warming world and other changes in the climate system (IPCC, 2001). The estimated temperature increase during the past century was between 0.4 °C and 0.8 °C with the ten warmest years all occurring within the last fifteen years (EPA, 2001).

Even though there is a scientific consensus about an ongoing global warming there is no consensus about its cause. Most studies, however, assume that it is a result of the increasing greenhouse gas concentrations into the atmosphere, i.e., the greenhouse effect. The greenhouse explanation is based on the fact that the global mean temperature increase coincides with increasing emissions of carbon dioxide (and other greenhouse gases) into the atmosphere, which has been increasing since 1800, from about 275 ppm to 370 ppm today

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 75–86.

(CDIAC, 2002). It is presumed that increases in carbon dioxide and other minor greenhouse gases will lead to significant increases in temperature. It is generally believed that most of this increase is due to the increased burning of fossil fuels. This theory is adopted by international environmental politics though there is a growing scientific scepticism about the greenhouse explanation. The main absorbers of infrared in the atmosphere are water vapour and clouds. Even if all other greenhouse gases were to disappear, we would still be left with over 98 percent of the current greenhouse effect. Another reason to scepticism is that existing models cannot be used to forecast climate (Lindzen, 1992). The IPCC's own Third Assessment Report included an entire chapter in its science report assessing the regional climate information from climate models. It concludes that a "coherent picture of regional climate change via available regionalization techniques cannot yet be drawn (IPCC, 2001)."

Another explanation to global warming is that it is a result of natural variations in solar irradiance, see e.g. Lean and Rind (2001); and Mende and Stellmacher (2000).

Before global warming Earth's mean temperature was 13.6 °C at which temperature it was in thermal equilibrium (NOAA-NCDC, 2001). During a global mean day, incoming short-wave radiation (SWR) was heating the ground surface. Later that day it was cooled off as the same amount of energy was re-emitted to space as outgoing long-wave radiation (OLR). The ground surface was then back at its mean temperature at which the OLR was limited only by the geothermal heat flow rate ($\sim$0.07 W m^{-2}) from the interior of the Earth.

Before global warming the geothermal heat flow was the only net heat source on Earth. Since then heat dissipation from the global use of non-renewable energy has resulted in an additional net heat source.

Genchi et al. (2000) showed that heat dissipation from traffic, air conditioning, and other human activities, during a warm day in Tokyo, adds up to a heat production of 140 W m^{-2} in Tokyo with a resulting air temperature increase of about 3 °C. A similar estimation for Stockholm results in 70 W m^{-2}. The Swedish energy consumption, mainly based on fossil fuel and nuclear power, corresponds to a national heat generation of 0.16 W m^{-2} though the country is sparsely populated (20 person km^{-2}).

There are several studies, e.g. Lachenbruch (1986) and Beltrami (2001), in which the global temperature change is evaluated from measured temperature profiles in the upper hundred meters of boreholes. These temperature profiles reflects long-term changes in ground surface temperature and can thus be used to analyse the changes in climate. Here, the analysis starts with the mean temperature of the ground surface. At this balance temperature occurring net heat (geothermal heat and thermal pollution) is emitted to space as OLR. The aim of this study was to analyse to what extent heat

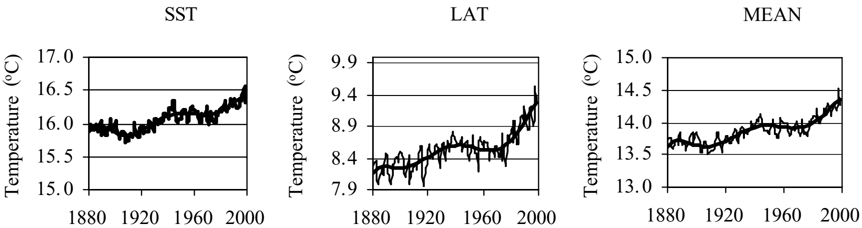

Figure 8. Sea surface temperature (SST), Land area temperature (LAT), and Combined global mean temperature (MEAN) (NOAA-NCDC, 2001)

dissipation, from the global use of non-renewable energy, contributes to global warming.

4.2. Global Temperature

This study is based on global monthly temperature data, from 1880 until today (NOAA-NCDC, 2001). These temperatures are separated into land area temperature (LAT), sea surface temperature (SST), and a combined global mean temperature (MEAN). MEAN is calculated by area weights corresponding to the global sea area (71%) and land area (29%). During the year SST is rather constant while the LAT varies considerably. LAT is about 3–12 °C lower than SST during the year. Figure 8 indicates the LAT increase since 1880 to be 1.2 °C (to 9.3 °C) while the SST increase is 0.5 °C (to 16.4 °C). The corresponding combined global mean temperature has increased by 0.7 °C (to 14.3 °C). Before that, during the years 1856–1880, the global mean temperatures were almost constant (Jones et al., 2001).

4.3. Net Heat Sources

There are two major net heat sources on Earth; the geothermal heat flow and net heating generated by human activities. Thermodynamics tells us that all energy will eventually dissipate into heat. The increasing utilization of non-renewable energy sources, mainly fossil fuels and nuclear power, has thus resulted in additional net heating on Earth. The utilization of renewable energy, i.e., solar energy in some form, also results in heat dissipation but does not cause any additional heating.

There are also a number of less continuous but still natural net heat sources e.g. the heat released from volcanoes, earth quakes, and meteorites. There are also some anthropogenic sources form nuclear bomb tests, conventional bombs and explosives.

4.3.1. GEOTHERMAL HEAT FLOW

Continental geothermal heat flow measurements are made in boreholes usually drilled to a depth of a few hundreds metres. Measurements on the ocean floor are made in the sediments (IHFC, 2001). Pollak and Chapman (1977) made the first comprehensive evaluation which included 5,500 measurements of which 70% were oceanic data and resulted in a global average close to 0.060 W m^{-2}. Parasnis (1985) later showed that the very high values occurring along the mid-Oceanic ridges give an oceanic mean heat flow of 0.0798 W m^{-2}. By compensating for oceanic heat flow that originates from sub-sea lava eruptions, the geothermal heat flow is 0.045–0.065 W m^{-2} for both continental and oceanic measurements.

The most recent compilation of heat flow data (Pollak et al., 1993) numbers almost 25,000 measurements. On a $5° \times 5°$ longitude-latitude grid, 62% of the Earth's surface was covered by measurements, while the heat flow of the remaining area of the planet was estimated. The resulting heat flows for the continents and the oceans were 0.065 W m^{-2} and 0.101 W m^{-2}, respectively, with a global mean value of 0.087 W m^{-2}.

4.3.2. THERMAL POLLUTION

The global annual use of fossil energy incl. nuclear power (IEA, 1999; BP-AMOCO, 2002) is almost 9,000 Mtoe (million metric ton oil equivalent). All of this energy, which corresponds to 11.63 TWh $Mtoe^{-1}$, will dissipate into heat. The heat of not fully combusted fuel will also be released, when this organic substance is decomposed. Consequently, the total amount of heat generated by fossil fuels is 10^{14} kWh. By distributing this energy over the total area of the Earth, an additional 0.02 W m^{-2} is heating the planet.

4.3.3. ADDITIONAL NET HEAT SOURCES

Additional net heat sources are studied in ongoing research at LTU. The preliminary results show that the nuclear testing did contribute much though such bombs are powerful. The studied wars (II WW and the war in Kuwait (1990) also indicates that the bombings did not contribute much to the net heating. However, the secondary effects of the wars like the burning of oil fields in Kuwait meant a considerable net heating.

The main less continuous net heat sources are from volcano eruptions, earth quakes and also the fall meteorites. The preliminary additional net heating (excl. meteorites) during the last 120 years are listed in Table 3. It is seen that volcanoes corresponds to 16% of the global energy consumption.

TABLE 3. Preliminary net heat generation 1880

Source	Released energy	
	kWh	%
Energy consumption	3.47×10^{15}	83.6
Volcanoes	3.95×10^{14}	9.5
Earthquakes	2.7×10^{14}	6.5
Meteorites	?	(?)
Wars	1×10^{13}	0.24
Nuclear test	6.64×10^{12}	0.16
Σ	4.15×10^{15}	100

4.4. Earth's Radiation Balance

The driving force of the Earth's atmosphere is the absorption of solar energy at its surface. Over long time-scales, compared to those controlling the redistribution of energy, the planet is in thermal equilibrium because the absorption of SWR from the sun is balanced by OLR, from Earth to space, at exactly the same rate. During clear days the atmosphere is transparent to SWR, which passes through without energy loss. At the same time this atmosphere is almost opaque to OLR. Because of the very small mean temperature gradient through the atmosphere convective heat transfer is not induced. Thus, the OLR is heating the atmosphere while radiated layer-by-layer through the atmosphere. At the top of the atmosphere, the OLR, q (W m^{-2}), emitted from Earth to space as given by the Stefan–Boltzmann law:

$$q = \sigma T_s^4. \tag{1}$$

Here, T_s (K) is Earth's effective mean temperature and the Stefan–Boltzmann constant $\sigma = 5.6697 \times 10^{-8}$ (W m^{-2} K^{-4}).

Measurements show that the mean OLR from Earth is 237 W m^{-2} (Salby, 1996). Equation (1) indicates that our planet is then in thermal equilibrium at an effective temperature of 254.2 K. This blackbody temperature of the atmosphere is the effective mean temperature of Earth and its atmosphere. It corresponds to a shell-like layer of temperature, Te, surrounding the planet at a mean altitude of approximately 6.5 km.

4.4.1. NET OLR

If Earth was monitored from outer space the annual mean values would show an effective temperature of 254.2 K, where the incoming SWR is balanced by the OLR, i.e., a zero net heat flow. Detailed measurements on this long

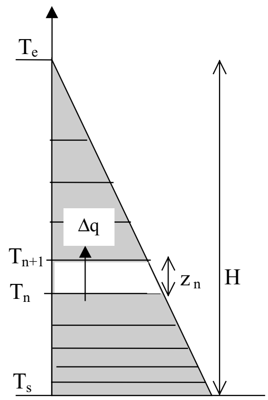

Figure 9. The net OLR is emitted layer-by-layer through the atmosphere. T_s = ground surface temperature; T_e = effective temperature of the atmosphere

time-scale would, however, show a net outgoing heat flow that originates from geothermal heat and the use of non-renewable energy sources.

The global mean temperature at ground surface is 33.1 K warmer than the effective Earth temperature, demonstrating the thermally insulating qualities of the atmosphere. This temperature difference drives the global net heat flow from the Earth. This is also the basis of performed calculations, i.e., by using annual mean temperatures of the ground surface and the atmosphere the resulting radiation is the outgoing net heat radiation. The suggested approach implies that the geothermal heat flow < net OLR < (geothermal heat flow + thermal pollution) until the global equilibrium temperature is reached.

The global mean temperature gradient through the atmosphere is approximately linear between the ground surface of temperature T_s (K) and the Earth's blackbody layer of temperature T_e (K). Clouds always cover about half of the planet's area, which means that the global mean day is partly cloudy with light rain. During the day SWR heats the ground surface, which is later cooled off by the same amount of OLR. When all incoming energy has been emitted the only remaining energy sources are the geothermal heat flow and thermal pollution. This constant heat flow is radiated layer-by-layer through the atmosphere (Figure 9). By using the Stefan–Boltzmann law, between the N atmospheric layers the net OLR,Δq (W m^{-2}), becomes

$$\Delta q = \frac{\sigma}{N}\sum_{1}^{N}(T_n^4 - T_{n+1}^4) = \frac{\sigma(T_1^4 - T_{N+1}^4)}{N} = \frac{\sigma(T_s^4 - T_e^4)}{N} \qquad (2)$$

For net OLR rates <0.1 W m^{-2} Equation (2) can be simplified as

$$\Delta q = \frac{\sigma(T_s^4 - T_e^4)}{N} \approx \sigma(T_s - T_e)^4. \qquad (3)$$

The error of this approximation is less than 0.01% for relevant global mean temperatures (see the Appendix). The thickness of the atmospheric layers, which is increasing with the altitude, is correlated to the optical depth of the atmosphere.

By calculating the net OLR since 1880, based on global mean monthly temperatures (separated in LAT and SST), it is seen that the global temperature rise has resulted in a net OLR increase from 0.068 W m^{-2} in 1880 to 0.074 W m^{-2} in 1999. The insignificant use of fossil fuel at the end of the 19th century denotes that the occurring global net heat outflow at that time (0.068 W m^{-2}) is equivalent to the geothermal heat flow. This heat flow rate agrees with earlier heat flow estimations by Parasnis (1985), but is lower than the most recent estimation by Chapman (1998). A more detailed separation in land and sea areas of different mean monthly temperatures would give a somewhat higher geothermal heat flow value.

Consequently, the net OLR has increased by 0.006 W m^{-2} since 1880. This means that about one-third of today's thermal pollution (0.02 W m^{-2}) is emitted from Earth. In the long term our use of non-renewable energy will cause a global temperature increase up to a point where the net OLR balances the net heat generation. Meanwhile, nature has some means of delaying global warming.

Result from our most recent and still ongoing studies on additional net heating is not included in performed calculations.

4.5. Global Warming is Large-Scale Thermal Energy Storage

At present the Earth's temperature is not in thermal equilibrium, i.e., the net OLR is still not as high as generated net heat. Natural cold sinks in water, ground, and atmosphere slow down the effect of thermal pollution. The main sources of natural cold are in water and ice. The total volume of global water is 1.4×10^{18} m^3, of which 94% is seawater while 3×10^{16} m^3 (2%) of the water is ice in the form of glaciers and ice fields (Singh and Singh, 2001).

The water of the oceans reduces the global warming by getting warmer. The melting of permafrost and ice means no temperature increase, but increasing volumes of melt water. Ice fields and glaciers have a large cooling capacity. The melting of 3×10^{16} ton of ice requires about 3×10^{18} kWh of energy. Since the annual total heat dissipation is 10^{14} kWh the ice fields and glaciers would last 30,000 years, with present use of non-renewable energy and no other cold sink.

A continental warming means that the underground is also warming up, which eventually will show by a reduced geothermal heat flow as the geothermal gradient decreases (Chapman, 1998).

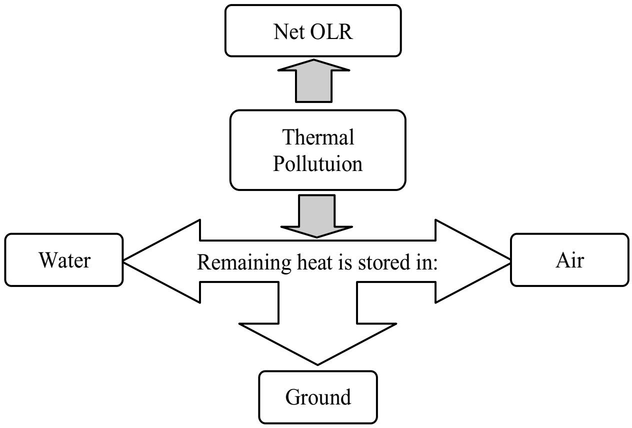

Figure 10. Global net heat balance 1880–2000

The atmosphere itself also has a huge heat storage capacity. In this case, part of the thermal energy is stored in the heating of air, i.e., global warming. Subsequent increase in evaporation means that additional energy is stored in the atmosphere.

The effects of the increasing humidity of the atmosphere are complex. It increases its heat storage capacity, i.e., it absorbs more heat without getting warmer. The resistance to OLR will also increase which indicates an increasing temperature gradient. On the other hand the increasing humidity of the atmosphere will reflect a greater part of incoming solar radiation. Consequently less OLR is emitted which must reduce Earth's effective temperature, T_e.

The global heat accumulation is summarized in Figure 10. The basic idea is that thermal pollution is released on Earth. Part of this heat is emitted to space as OLR. The remaining heat is first released into the atmosphere. Part of it will accumulate in the air, which in its turn warm the ground and water. In the air heat is stored as both latent and sensible heat. In the ground heat is stored as sensible heat. It is easy to calculate heat that is accumulating in air and ground. The problem is to estimate the heat that is stored in water.

This ongoing research at Luleå University of Technology will be completed during 2005. Preliminary results are shown below.

Global Heat Accumulation due to Global Warming 1880–2000

Thermal Pollution (Global energy consumption + other net heat sources)
–OLR

Σ *Globally accumulated net heat (in air, ground and water)*

Preliminary estimations of accumulated heat
≈40% in air
≈15% in ground
≈45% in water (by heating or melting of ice)
= 100%.

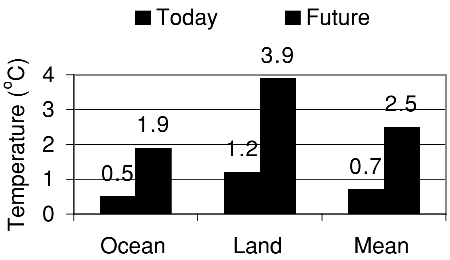

Figure 11. Calculated global temperature increase (ocean, land and mean) today and in the future (at thermal equilibrium)

4.6. Steady-State Global Warming

The future steady-state global temperature was estimated by assuming that continued warming follows the same pattern as during the last century. This means that the previous LAT/SST increase ratio of 2.4 was assumed constant. Therefore, the land area temperature increase was assumed a factor 2.4 greater than the sea surface temperature increase. Today's consumption of non-renewable energy was also assumed unchanged.

In a steady-state situation all net heat generated is emitted to space. Therefore, by adding the calculated geothermal mean heat flow (0.068 W m^{-2}) and the total thermal pollution (0.020 W m^{-2}), the total net OLR becomes 0.088 W m^{-2}. Equation (3) then gives that this net OLR requires a SST of 17.8 °C and a LAT of 12.0 °C resulting in a future global mean temperature of 16.1 °C.

Thus, the global thermal pollution will at steady state have increased the sea surface temperature by 1.9 °C, the land area temperature by 3.9 °C and the global mean temperature by 2.5 °C. Since part of this heating has already begun, further temperature increases of 1.4 °C (Ocean), 2.7 °C (Land), and 1.8 °C (Mean) should be expected (Figure 11).

4.7. Discussion and Conclusions

Around 1880, before global warming, Earth was in thermal equilibrium at a mean temperature of 13.6 °C. The occurring net OLR was then equal to the geothermal heat flow. Calculations based on the undisturbed global temperatures (monthly mean values of sea and land temperatures) indicate that the geothermal mean heat flow is 0.068 W m^{-2}, which is slightly lower than the most recent estimations. A more detailed calculation, by separating sea and land areas in several areas of different temperatures, would result in a slightly greater geothermal heat flow.

Since 1880 the increasing use of non-renewable energy has resulted in a thermal pollution, which today corresponds to a global heating of 0.02 W m^{-2}. This heating has so far resulted in a global temperature increase of 0.7 °C. As a consequence the net OLR has increased and in 1999 one-third of the thermal pollution was emitted as OLR. The current use of non-renewable energy requires a further global temperature increase of 1.8 °C, until Earth is again in thermal equilibrium. Then the total net heat generation (0.088 W m^{-2}) will be emitted as OLR.

Earth counteracts global warming by its natural cold sources. This has started to show as a temperature increase of ground, air and water. Ice fields and glaciers offer another huge cold reserve and the world's total non-renewable energy use would annually melt only about 0.003% of current ice, with present use of non-renewable energy and no other cold source.

Increasing concentrations of gases, aerosols, and humidity into the atmosphere will increase the Earth's albedo. This would mean that less solar energy reaches the Earth's surface and thus less OLR to be emitted. Therefore, the Earth's effective temperature should decrease.

What can we then do to put a stop to global warming? Today's policies are aiming at reducing the CO_2 emissions. There are also ideas of storing CO_2 in deep deposits. Some countries plan to expand their nuclear power industry. All these ideas would reduce the CO_2 emissions but would not reduce the global net heat generation. In the case of nuclear power it would become even worse because of the large amounts of heat generated by nuclear power production. There are also visions of importing clean energy from space. Even if this energy would be renewable on Mars it would cause global warming since such systems would release net heat on Earth. The only sustainable way is to use our own renewable energy. With renewable energy systems it is not even possible to disturb the energy balance of Earth.

Appendix

The Net OLR, Δq (W m^{-2}), from the ground surface to the atmosphere is given by the Stefan–Boltzmann law. The linear temperature change between the surface temperature of the Earth (T_s) and the effective temperature of the atmosphere (T_e) indicates that the radiation occurs layer-by-layer through the atmosphere (Figure 9). Since the net OLR is constant through the atmosphere the net OLR through layer n is

$$\Delta q = \sigma(T_n^4 - T_{n+1}^4) = \sigma\left[T_n^4 - (T_n + \Delta T_n)^4\right] \qquad \text{(A1)}$$

where σ is the Stefan–Boltzmann constant. This expression can be rewritten as

$$\Delta q = \sigma \cdot \Delta T_n^4 \left[4\left(\frac{T_n}{\Delta T_n}\right)^3 - 6\left(\frac{T_n}{\Delta T_n}\right)^2 + 4\left(\frac{T_n}{\Delta T_n}\right) - 1 \right]. \tag{A2}$$

If $\Delta T_n \ll T_n$ then

$$\begin{aligned}\Delta q &= \sigma \cdot \Delta T_n^4 \left[4\left(\frac{T_n}{\Delta T_n}\right)^3 - 6\left(\frac{T_n}{\Delta T_n}\right)^2 + 4\left(\frac{T_n}{\Delta T_n}\right) - 1 \right] \\ &\approx 4\sigma \cdot \left(\frac{T_n}{\Delta T_n}\right)^3 \Delta T_n^4. \end{aligned} \tag{A3}$$

The linear temperature gradient through the atmosphere gives that

$$\Delta T_n = \frac{z_n}{H}(T_s - T_e) \tag{A4}$$

where z_n is the thickness of layer n and H the distance to the effective atmospheric temperature as shown in Figure 2. Equations (4) and (5) give that

$$\Delta q \approx 4\sigma \cdot \left(\frac{T_n}{\Delta T_n}\right)^3 \Delta T_n^4 = 4\left(\frac{T_n}{\Delta T_n}\right)^3 \cdot \left(\frac{z_n}{H}\right)^4 \cdot \sigma (T_s - T_e)^4. \tag{A5}$$

It can be shown that

$$4\left(\frac{T_n}{\Delta T_n}\right)^3 \cdot \left(\frac{z_n}{H}\right)^4 = 1.000 \tag{A6}$$

for $\Delta q < 0.1$ W/m^2, i.e., for $\Delta q <$ total net heat generation

$$\Theta \Delta q \approx \sigma \cdot (T_s - T_e)^4. \tag{A7}$$

This approximation gives an error <0.01% for relevant global mean temperatures.

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5. ENERGY STORAGE FOR SUSTAINABLE FUTURE—A SOLUTION TO GLOBAL WARMING

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Abstract. The most dramatic challenge we are facing today is climate change induced to a considerable degree by human originated greenhouse gas emissions, especially the correlation between CO_2 concentrations and temperatures. The reduction of energy consumption and greenhouse emissions is among the issues of greatest interest for the prevention of global warming and climate change. Hence, developing and deploying more energy efficient and environmentally friendly energy technology is critical to achieving the objectives of **E**nergy security, **E**nvironmental protection, **E**conomic growth and social development known as three **E**'s. Energy Storage systems—a tool in forming the structure of sustainable energy for sustainable future—can play a key role in decreasing emissions that lead to global warming.

Keywords: Climate change; global warming; greenhouse effect; emissions; energy storage; energy conservation; sustainable future

5.1. Introduction

There is little doubt that we live in an age of rapid and accelerating change. Indeed, in today's world everything at times seems to be in flux.

The break of the third millennium coincided with important challenges for the energy sector. The demand for energy is increasing and so is the notion that we should take care of the environment. Our challenge today is to solve this apparent contradiction. Environmental concerns gain ground mostly on economic considerations, but above all decision makers now face crucial long-term energy policy choices. The opportunity therefore should be the increasing role of energy policy using renewable energy sources and production on global level.

For decades people have tried to achieve more of everything: more mobility, more comfort and more consumption. The signs are becoming increasingly evident that this cannot go on forever; too much burden is put on the environment and natural sources are becoming exhausted (Figure 12). Human

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 87–99.

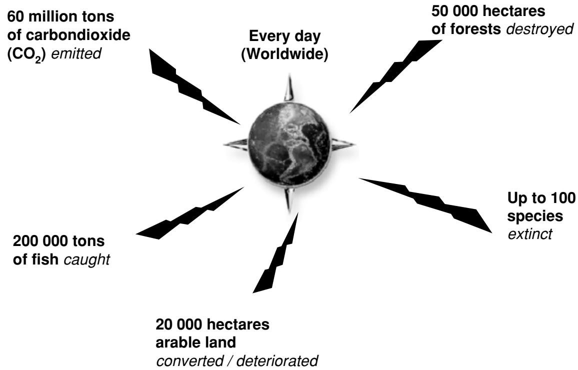

Figure 12. The daily toll

kind is approaching a critical turning point. We are already beyond limits of the Earth's carrying capacity. The new target is to change this tendency from "more" to "better", from "quantity" to "quality".

Mankind's impact on the global climate and whether pollution from modern energy use is indeed warming the Earth have become important issues for national and international policy makers. Political pressure and public sentiment are based on complex data sets that, alone, cannot tell the whole story. The ultimate question is whether our climate is becoming warmer because of the slow build-up in atmospheric greenhouse gas concentrations (1). The answer is not clear, because much of what we know about global climate change is inferred from historical evidence of uncertain quality.

5.1.1. CLIMATE CHANGE—WHAT IS THE PROBLEM?

Lack of reliable measurement is the first reason, as reliable ground-based measurements by scientific instruments have been made just in this century. These measure conditions only at the location of each instrument, and they are usually land-based, although 75% of the Earth is covered with water. We have been able to take precise, direct measurements only in the last four decades, and not until the advent of precision space borne instruments in the 1970s were we able to measure global temperatures at a range of altitudes across the entire atmosphere.

The presence of water as solid, liquid, and gas is a feature that makes Earth unique in the solar system and that makes life possible as we know it. The transport of water and the energy exchanged as it is converted from one state to another are important drivers in our weather and climate. One of the key missions is to develop a better understanding of the global water cycle at a variety of scales so that we can improve model forecasts of climate trends, predictions of short term and regional weather events, and even their impacts on society's regional and global activities.

Although water vapor is the most important greenhouse gas, we have not adequately sampled its four-dimensional variability—across space and time—over the globe. Recent analyses of global satellite data have produced many new observational data sets that describe the vertical and horizontal structure of moisture over the last 10–20 years. These data sets are being analyzed and validated to document the satellite measurements' capability to describe accurately, various components of the hydrologic cycle. In some cases, the trends observed by satellites are consistent with predictions by computer models, while in others, there are significant discrepancies. Understanding when and why the models are right or wrong and the nature and limitations of satellite data is key to their intelligent use in climate studies

The most dramatic challenge we are facing today is climate change induced to a considerable degree by human originated greenhouse gas emissions, especially the correlation between CO_2 concentrations and temperatures. The reduction of energy consumption and greenhouse emissions is among the issues of greatest interest for the prevention of global warming and climate change.

According to the National Academy of Sciences, the Earth's surface temperature has risen with accelerated warming during the past two decades (2). There is new and stronger evidence that most of the warming over the last 50 years is attributable to human activities. Human activities have altered the chemical composition of the atmosphere through the build up of greenhouse gases—primarily carbon dioxide, methane, and nitrous oxide. The heat-trapping property of these gases is undisputed although uncertainties exist about exactly how earth's climate responds to them.

5.2. Global Climate Change

Climate is the average pattern of weather over the long term. The earth's climate has warmed and cooled for million years since long before we appeared on the scene. There is no doubt that the climate is growing warmer currently; indications of that change are recorded and discussed by scientists.

Though climate change is not new, the study of how human activity affects the earth climate is. The exploring climate change encompasses many fields, including physics, chemistry, biology, geology, meteorology, oceanography and even sociology. I shall try to present you some evidences, uncertainties and conclusions about climate change.

The earth receives tremendous amount of energy from the sun. The land, sea and air absorb some of this energy and reflect back into space. The over all description of this process is called the earth's energy budget. The balance between energy absorbed by the earth and energy reflected back into space is fundamental in determining how warm or cool the planet becomes. The proportion of radiation reflected away by a surface is called its albedo. Albedo can range between 0 (no reflectance) and 1 (complete reflectance like a mirror) The earth's average albedo is 0.31 which means that overall, the planet reflects about 31% of incoming solar radiation back into space. But forests and deserts, oceans, clouds, snow and ice all have different albedos—and changes in these types of ground cover can therefore affect how much solar radiation the earth receives. For example, the albedo of forests lies in the range 0.07–0.15, while deserts have an albedo of around 0.3 (3).

The albedo of earth surface varies from about 0.1 for the oceans to 0.6–0.9 for ice and clouds which mean the clouds, snow and ice are good radiation reflectors while liquid water is not. In fact, snow and ice have the highest albedos of any parts of the earth's surface: Some parts of Antarctic reflect up to 90% of incoming solar radiation.

5.2.1. GREENHOUSE EFFECT—A PROBLEM OR ACTUALY ESSENTIAL TO OUR EXISTENCE?

The green house effect is one aspect of the energy budget. Solar radiation passes through the clear atmosphere: some solar radiation is reflected by the earth and the atmosphere and most radiation is absorbed by the earth's surface while some of the infrared radiation passes through the atmosphere and some is absorbed and reemitted in all directions by greenhouse gas molecules. The sun warms the earth and certain gases (CO_2 and water vapor) act like a screen, hence, trapping heat and keeping the planets surface at a hospitable 15 °C warm enough to support life (with no greenhouse effect the earth's average temperature would stabilize at about −18 °C).

5.2.2. WHAT ARE GREENHOUSE GASES?

Some greenhouse gases occur naturally in the atmosphere, while others result from human activities. Naturally occurring greenhouse gases include water vapor, carbon dioxide, methane, nitrous oxide, and ozone. Certain human activities, however, add to the levels of most of these naturally occurring gases:

Carbon dioxide is released to the atmosphere when solid waste, fossil fuels (oil, natural gas, and coal) and wood and wood products are burned.

Methane is emitted during the production and transport of coal, natural gas, and oil. Methane emissions also result from the decomposition of organic wastes in municipal solid waste landfills, and the raising of livestock. More information on methane.

Nitrous oxide is emitted during agricultural and industrial activities, as well as during combustion of solid waste and fossil fuels.

Very powerful greenhouse gases that are not naturally occurring include hydrofluorocarbons (HFCs), perfluorocarbons (PFCs), and sulfur hexafluoride (SF6), which are generated in a variety of industrial processes.

Each greenhouse gas differs in its ability to absorb heat in the atmosphere. HFCs and PFCs are the most heat-absorbent. Methane traps over 21 times more heat per molecule than carbon dioxide, and nitrous oxide absorbs 270 times more heat per molecule than carbon dioxide. Often, estimates of greenhouse gas emissions are presented in units of millions of metric tons of carbon equivalents (MMTCE), which weighs each gas by its GWP value, that is, Global Warming Potential.

Problems may arise when the atmospheric concentration of greenhouse gases increases. Since the beginning of the industrial revolution, atmospheric concentrations of carbon dioxide have increased nearly 30%, methane concentrations have more than doubled, and nitrous oxide concentrations have risen by about 15%. These increases have enhanced the heat-trapping capability of the earth's atmosphere.

Studies show that most of the warming over the last 50 years is attributable to human activities which have changed the chemical composition of the atmosphere through the built up of greenhouse gases, resulting in increased global temperatures—or as scientists call "global warming" (4).

A key issue in understanding global climate change is measuring humanity's effect on the concentration of green house gases. Scientists generally believe that the combustion of fossil fuels and other human activities are the primary reason for the increased concentration of carbon dioxide which may cause a significant warming trend. Plant respiration and the decomposition of organic matter release more than 10 times the CO_2 released by human activities; but these releases have generally been in balance during the centuries.

Once, all climate changes occurred naturally. However, during the Industrial Revolution, we began altering our climate and environment through changing agricultural and industrial practices. Additionally, through population growth, fossil fuel burning, and deforestation, we are affecting the mixture of gases in the atmosphere.

What has changed in the last few hundred years is the additional release of carbon dioxide by human activities. Fossil fuels burned to run cars and trucks, heat homes and businesses, and power factories are responsible for about 98% of carbon dioxide emissions, 24% of methane emissions, and 18% of nitrous oxide emissions. Increased agriculture, deforestation, landfills, industrial production, and mining also contribute a significant share of emissions (5). For example, in 1997, the United States emitted about one-fifth of total global greenhouse gases.

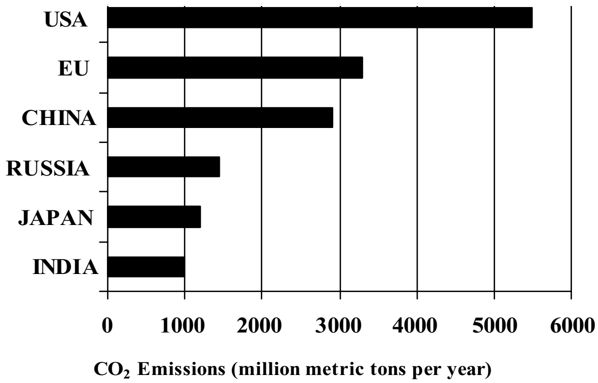

Figure 13. Total CO_2 emission (according to UNFCCC)

Estimating future emissions is difficult, because it depends on demographic, economic, technological, policy, and institutional developments. Several emissions scenarios have been developed based on differing projections of these underlying factors. For example, by 2100, in the absence of emissions control policies, carbon dioxide concentrations are projected to be 30–150% higher than today's level.

The Intergovernmental Panel on Climate Change (IPCC), a group established by the World Meteorological Organization (WMO) and the United Nations Environment Program (UNEP) reports that the average surface temperature of the earth has increased during the twentieth century by about 0.60 ± 0.20. This may seem like a small shift, but although regional and short-term temperatures do fluctuate over a wide range, global temperatures are generally quite stable. In fact, the differences between today's average global temperature and the average global temperature during the last Ice Age are only about 5 °C. Indeed it is warmer today around the world than at any time during the past 1,000 years, and the warmest years of the previous century have occurred within the past decade. But climate change is a better description as some areas may cool.

The total CO_2 emissions of some countries are given in Figure 13.

The US pollutes more, absolutely and per head, than any other country (it also produces more wealth). Its greenhouse emissions have raised by more than 11% since 1990: its Kyoto commitment was to reduce them by 6%. It is the only country to have signed the protocol and then to have repudiated it. President Bush said in March, 2005 the US would not ratify Kyoto, because he thought it could damage the US economy and because it does not yet require developing countries to cut their emissions. His domestic and foreign critics think the US will lose economically by staying alone.

The EU wants a rigorous application of Kyoto, allowing only restricted use of its flexibility mechanisms: these allow countries to go some way to meeting their pollution reduction targets by paying for improvements beyond their frontiers. The EU says countries should meet at least half their targets by cutting emissions at home. It also opposes widespread use of forests and other carbon "sinks" to absorb pollution. While the green blocs in several northern European governments remain rigorous, the UK is trying to rebuild bridges with the US. But the EU will probably remain united in seeking Kyoto's early entry into force.

China is an Annex II country, not yet required to cut its emissions. But it is reported to have cut its emissions of the main gas listed in the protocol, carbon dioxide, by 17% since the mid-1990s. In the same period its economy has grown by one-third. Accounting for a fifth of the world's population, with hopes of a better life, China could obviously soon emit enough to dwarf any reductions agreed by the Annex I countries. But its leaders recognise that climate change could devastate their society. China encourages the protocol's supporters to believe that Kyoto is already helping to make a difference.

Russia, a developed country, is part of the "Annex 1" bloc of countries committed to cutting emissions under the protocol. But its economy has shrunk so drastically since 1990 that it cannot afford to burn the fuel that would produce the emissions Kyoto entitles it to. Its emissions have fallen by almost 40% in a decade. So it favours emissions trading, selling its unused entitlement to developed countries wanting to emit more than the protocol allows them. Russia will ratify Kyoto, because it recognises it as a way of earning desperately needed money. It plans to use the cash for energy efficiency projects.

A major world economic power, Japan is a leading Annex I member of Kyoto, committed to cutting emissions. It feels an attachment to the protocol, named after the Japanese city where it was concluded. It recognises the argument that its economy could gain from seeing the treaty in force, as Japanese companies could capture markets for new, clean technology. But Japan is very reluctant to ratify Kyoto unless the Americans do so as well. Without Japanese ratification the protocol is very unlikely to attract the support it needs to become international law.

Developing countries like India are listed under Kyoto as Annex II countries, and they are not obliged to make any cuts in greenhouse emissions reduction yet. But as they raise living standards their emissions will obviously increase: India's have risen by more than 52% since 1990. Under Kyoto, they will have to accept reduction targets in a few years from now. The protocol's architects say it is fair to allow them a grace period, because the problem has been caused by the industrialised countries. But India, with more than 1 bn people, will soon be a major polluter.

What proof do we have? The proofs are listed below:

- It is getting warmer.
- Long-term temperature changes.
- Rising waters.
- Arctic sea ice decrease.
- Fossil fuel emissions increase.

Measurements from a variety of sources have suggested that the earth's average atmospheric temperature has raised over the last several hundred years—but by how much? Taking the average temperature of the earth's atmosphere is a very difficult measurement problem. First, measurements must be taken in a large and diverse enough range of locations to ensure that their average is truly a measure of global temperature and is not biased toward one region or another. Second, those locations must be chosen so that individual measurements are not thrown off by sources of unusually high or low temperatures, such as cities (which tend to be "heat islands" warmer than the surrounding landscape). Third, no measuring device is perfect—all measurements include some amount of error, or "noise." Understanding the kinds of errors associated with different measurement techniques is a key element in evaluating the accuracy of a given temperature value. In addition, the study of climate requires measurements over very long time periods, so sources of paleoclimate data (data on climate from the distant past) are key to understanding climate change.

Two kinds of problems make this an exceptionally difficult question to answer.

First, the enormous complexity of the earth's dynamic climate system including the interacting air masses, winds and, ocean currents, and patterns of evaporation and precipitation makes long-term climate prediction extremely problematic. Estimates drawn from reports by the Intergovernmental Panel on Climate Change (IPCC) project increases in average global temperatures ranging from 1.4 °C to 5.8 °C by the year 2100. These numbers may seem small, but because average global temperatures are actually remarkably stable over long periods, this range actually represents a very significant rise in the earth's temperature over a very short time.

A second problem complicating the picture is the unpredictability of human behavior. At what rate will the human population—and its production of carbon dioxide—grow? As formerly undeveloped countries expand their industry, often using cheaper (and more polluting) fossil–fuel technology, their contributions to greenhouse gases will rise and add to the problem—but by how much? To what extent will new, cleaner technologies (such as cars powered by hydrogen fuel cells) be developed and adopted by countries around the world? These kinds of uncertainties make the tough problem of predicting climate change all the more difficult.

Thirdly the oceans play a key role in regulating the earth's climate—and yet they remain mysterious, because so many of the basic processes underlying ocean dynamics are still poorly understood. Their fundamental role in climate is based largely on their storage and transport of heat around the globe. The oceans store vast amounts of heat, much more than the heat stored by the atmosphere, because water is 1,000 times more dense and has a heatholding capacity four times that of air. Ocean currents are primary highways for the transport of heat around the globe.

Fourthly living things do not just respond to the climate—they affect it as well. Plants consume carbon dioxide and produce oxygen through photosynthesis. Earthbound plants take carbon dioxide directly from the air; drifting photosynthetic micro-organisms called phytoplankton use carbon dioxide dissolved in water.

It is estimated that photosynthesis is a "sink" for around 60 billion tons of carbon every year, by far the strongest mechanism for carbon dioxide removal from the atmosphere. (This removal is almost exactly balanced by the respiration of animals, which combines oxygen with hydrocarbons to produce carbon dioxide and water vapor.)

Increases in the level of carbon dioxide in the atmosphere could promote plant growth. If the planet's vegetation grows stronger and more widespread, it could take in more of the atmospheric carbon dioxide, preventing a runaway greenhouse effect. This controversial "greening hypothesis" has led to more research exploring the connections between global climate and the earth's biological systems.

The extremely complex interrelations between human activity and natural forces—air masses, winds, ocean currents, evaporation, and precipitation—means that researchers from many fields pool their efforts in an attempt to understand how the climate is reacting to changes. But this complexity also means that knowing what the climate will be like in 50 or 100 years is among the most challenging problems in science.

Some of the changes researchers in all these areas are exploring may seem small, especially in relation to the typical temperature changes associated with daily and seasonal cycles. But although regional and short-term temperatures do fluctuate over a wide range, global temperatures are generally very stable. Indeed, during the last Ice Age (about 20,000 years ago), the average global temperature was only about 5 °C cooler than it is today.

- Greenhouse effect is essential to our existence.
- Oceans cover more than 70% of the earth's surface play a fundamental and complex role in regulating climate.

- Changes in climate dramatically alter the planets snow and ice covered cryosphere.
- The effects of climate change on plants and animals are difficult to measure, but potentially dramatic.
- Interpreting past and present climate data is difficult, but predicting future climate change—and its possible effects—is even more challenging.

The relationship between greenhouse gases and temperature in Antarctica over a 420.000 year period confirms global warming. Although these measurements cannot prove what factors caused climate changes, these data do strongly suggest that atmospheric gas concentrations and temperatures are related.

There is scientific consensus on global warming, but no consensus about its cause (6).

The fact that seemingly small changes can have dramatic effects is one reason why an understanding of the data, techniques, and controversies of global climate research is so fundamental to understanding the phenomenon itself.

5.3. Why Store Energy?

World wide perspective of energy storage is that:

- Existing energy systems are not sustainable.
- Diversification of supply is essential (a solution only to a part of the problem).
- Efficiency with renewables is the strategic mission to sustainability.

5.3.1. MOTIVATION AND CHALLENGES

Considerable efforts are being made to economize non-renewable energy resources by sensitizing experts in the field of energy who in turn will look for alternatives, thereby removing the dependency on non-renewable resources. The motivation and challenges for storing energy are focused mainly on three important facts.

- Energy security/reliability—using new energy technology.
- Environmentally friendly techniques for climate protection, hence, contribution to environmental conservation—commitment for reduction of CO_2—obligations of Convention on Climate Change, Kyoto Protocol.
- Economic feasibility—using market principles.

Developing and deploying more efficient and environmentally friendly energy technology is critical to achieving the objectives of **E**nergy security,

Environmental protection, **E**conomic growth and social development known as three **E**'s. The mission is to implement an environmentally friendly energy system. If we are to achieve sustainable development, we will need to display greater responsibility for energy, economy and environment.

The environmental benefits of TES is utmost important and discussed within the framework of UN Convention on Climate Change and Kyoto Protocol which is a significant issue of critical debate among countries nowadays (7).

5.3.2. WHY ENERGY EFFICIENCY AND CONSERVATION?

Energy efficiency and conservation are necessary, because energy consumption is a major cause of environmental degradation. All types of energy use result in environmental costs, it's just a matter of degree. And most modern activities seem to involve energy consumption. Transportation, food production, manufacturing, governments, recreation and household management all consume energy.

At the same time, our major energy supplies (oil, coal, and gas) are finite. They are not renewable, yet we burn through these fuels as if there were no tomorrow. The energy supplies which are renewable (solar, wind, thermal) are not being used as widely or thoughtfully as they should be.

Given these facts, we need to reduce our energy consumption and environmental damage to the extent we can, and come into balance with natural energy recovery and production processes. We need to develop truly sustainable energy consumption practices for a sustainable future (8).

Therefore, Storage systems can play a key role in decreasing emissions that lead to global warming and ozone depletion. TES (Thermal Energy Storage) contributes significantly to energy efficiency (9). Consequently storage techniques can be used as a tool in forming the structure of sustainable energy for sustainable future.

Energy storage technologies are a strategic and necessary component for the efficient utilization of renewable energy sources and energy conservation. There is a great technical potential to substitute burning fossil fuels by using stored heat that would otherwise be wasted and using renewable generation resources.

5.3.3. EXPECTED BENEFITS

- Better management of energy resources.
- Increase energy efficiency.
- Use alternative energy systems.
- Reduction of harmful emissions.

- Decreasing dependency on fossil fuels.
- A brake to global warming.

Creation of a new understanding how to make TES happen is crucial, hence the introduction of the correct economic and environmental instrument is the single most important factor controlling the sustainable growth of renewable technology, hence effective international collaboration and awareness have an increasingly important role.

5.4. Conclusion

Many governments have committed to reduce CO_2 emissions into the atmosphere. They have decided to strengthen their national efforts to increase the deployment of energy conservation technologies and the utilization of renewable energy sources. So far in most industrialized countries, renewable energy sources contribute only marginally to satisfy energy demand. This is due to several reasons, in particular because some new energy systems are not yet economically competitive with the combustion of fossil fuels, long-term reliability is not yet proven, and there are still some regulatory and market barriers which have to be overcome. Therefore, further attempts are being made to resolve these issues. This is especially true for many new energy storage technologies and concepts that have not yet been implemented on a large scale in the market Research into the future alternatives must therefore be urgently conducted. The principal task is to secure energy supplies for both the short and long terms. In order to cope jointly with energy problems many countries have come together for research to conserve energy, to reduce dependence on oil, to reduce the environmental emissions associated with energy and to pursue research and development. The mission is to implement an environmentally friendly energy system.

Over the last few years, the emphasis in research has shifted towards storage technologies that improve the manageability of energy systems or facilitate the integration of renewable energy sources.

If fully exploited, storage systems can play a key role in decreasing emissions that lead to global warming and ozone depletion. Additionally the usage of such techniques will lead to energy conservation in the range of 40–80% in terms of fossil fuel and electricity, and improves energy efficiency when compared with conventional systems. Consequently storage techniques can be used as a tool in forming the structure of sustainable energy for sustainable future.

We have not fully integrated energy with the economic and environmental pillars of development. If we are to achieve sustainable development, we shall need to display greater responsibility for energy, economy and environment. Achieving sustainable development is not an easy task. Significant changes will be needed—in decision making at the highest levels and in day-to-day behavior—if we wish to reach our goal of development that meets the needs of today without sacrificing the ability of future generations Tens years ago in Rio de Janeiro, governments committed themselves to just such transformations, but commitments alone have proven insufficient to the task. I see ourselves, as members of IEA ECES, a mediator between research, policy, business and the public to end the "cold war" between the three E's.

Taking this into account I look forward to a productive NATO Summer School with new motivation to shift to energy conservation techniques of the new millennium in meeting Kyoto Targets for the reduction of greenhouse gas emissions in order to combat climate change.

We are all on the same boat—the Earth, the important thing is what we do together for the optimization of the worlds conditions through environmental measures and energy conservation.

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PART III. ENERGY EFFICIENT DESIGN AND ECONOMICS OF TES

6. ENERGY EFFICIENT BUILDING DESIGN AND THERMAL ENERGY STORAGE

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Abstract. This chapter discusses the potential for cost-effectively reducing the energy intensity of office buildings by applying proven technologies, especially the use of ground source systems with thermal energy storage. It is shown that significant energy use reductions are possible without increases in capital costs and that reductions of more than 50% are possible within normal investment criteria. Energy storage techniques need to be adapted to these reduced energy levels. Energy efficient buildings are better suited to energy storage. Low-energy building design can contribute to dramatically reduced energy usage and can be applied to all new building projects. The example of a small office building located in Canada is used to illustrate this potential. The reference energy level is that specified by the minimum (or prescriptive) requirements of the Canadian Model National Energy Code for Buildings (MNECB) 1997. Costs and savings evaluated include energy, capital and maintenance. Typical small and large office buildings were analyzed. The role of energy storage was also considered. The results indicate that energy savings greater than a 50% reduction can be achieved with attractive economic returns through careful selection and application of existing technologies. Energy savings greater than 50% were achieved in four cases for the small office building with discounted payback periods between 2.5 and 6 years. The 50% energy reduction relative to the MNECB is the threshold for high performance buildings. It is possible to achieve 25% reduction compared to the base case building with no incremental cost. With careful selection and application of efficient building technologies at the early stages of design, and adjustment of equipment sizing to account for reduced demands, many designs result in energy savings of 30–40% with no incremental cost. A brief guide to design options for the building side of ground source heat pump systems is then presented. Ground source heat pump systems can significantly lower heating and cooling operating costs and can qualify designs for energy efficiency and renewable energy credits under various building rating schemes. Both distributed or incremental heat pumps and central heat pumps as applied to closed loop ground source systems are considered. Available products are identified for each design option as well as pumping arrangements, system

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schematics showing components for each design option, and outdoor air systems. A table comparing the design options is also provided. In addition to a schematic drawing of each system, elevation sketches showing zone component arrangements for both distributed and under floor designs are given. Some recent Canadian building designs are then summarized. They show a range of retrofit and new building options using innovative approaches and cost-effective results.

Keywords: Building design; Canadian Model National Energy Code for Buildings; cost-effectiveness; DOE 2.1 E; energy storage; energy code; energy efficiency; energy simulation; ground source heat pumps; low-energy; MNECB; TES; thermal energy storage; water loop heat pump.

6.1. Introduction

The potential for cost-effectively reducing the energy intensity of office buildings by applying proven technologies is calculated. The use of ground source systems with thermal energy storage is included. It is shown that significant energy use reductions are possible without increases in capital costs and that reductions of more than 50% are possible within normal investment criteria. Energy storage techniques need to be adapted to these reduced energy levels. Energy efficient buildings are better suited to energy storage. Heating reductions in Canada are greater than cooling and this tends to make heating and cooling loads more comparable. Low-energy building design can contribute to dramatically reduced energy usage and can be applied to all new building projects. The example of a small office building located in Canada is used to illustrate this potential. The reference energy level is that specified by the minimum (or prescriptive) requirements of the Canadian Model National Energy Code for Buildings [1] (MNECB) 1997. Costs and savings evaluated include energy, capital and maintenance. A typical small and large office building were analyzed. The results indicate that energy savings greater than a 50% reduction can be achieved with attractive economic returns through careful selection and application of existing technologies. Energy savings greater than 50% were achieved in four cases for the small office building with discounted payback periods between 2.5 and 6 years. The 50% energy reduction relative to the MNECB is the threshold for high performance buildings. It is always possible to achieve 25% reduction compared to the base case building with no incremental cost. With careful selection and application of efficient building technologies at the early stages of design, and adjustment of equipment sizing to account for reduced demands, many designs result in energy savings of 30–40% with no incremental cost. A brief guide to some design options for the building side of ground source heat pump systems is then presented. Ground source heat pump systems can significantly lower heating and cooling operating costs and can qualify designs for energy efficiency and renewable energy credits under various building rating schemes. Both distributed or incremental heat pumps and central heat pumps as applied to closed loop ground source systems are considered. Available products are identified for each design option as well as pumping arrangements, system schematics showing components for each design option, and outdoor air systems. A table comparing the design options is also provided. A schematic drawing of each system is given. Some recent Canadian building designs are then summarized. They show a range of retrofit and new building options using innovative approaches and cost-effective results.

6.2. Low Energy Building Design Economics and The Role of Energy Storage

6.2.1. INTRODUCTION TO LOW-ENERGY DESIGN

MNECB 1997 contains cost-effective minimum requirements for energy efficiency in new buildings. The MNECB provides maximum thermal transmittance levels for building envelope components per type of energy for different regions of Canada. These levels were determined using regional construction and heating energy costs in a life cycle cost analysis. As well, the MNECB gives regional U-values for windows, references energy efficient equipment standards, and identifies when heat recovery from ventilation exhaust is required for dwelling units. To allow flexibility in achieving a minimum level of energy efficiency, the code offers three compliance approaches: a Prescriptive Path, a Trade-off Path, and a Performance Path. The Prescriptive Path was used as the reference level in this study.

Two generic office building models, the smaller having a floor area of 4,200 m^2, the larger having a floor area of 24,300 m^2 were simulated using the DOE 2.1 E software. The two buildings provided different opportunities to significantly reduce energy use. These buildings were the base cases to which subsequent design modifications to the buildings were compared. Full details on both the small and large buildings in various Canadian locations can be found in the project report [2].

Some basic data on the small building is given below:

- Natural gas-fired central boiler heating.
- Individual zone packaged rooftop DX air cooled (EER = 8.9) with economizer cooling.
- Walls—Brick, batt and rigid insulation.

- Built-up roof, rigid insulation.
- Double-glazed windows, grey tint, aluminum frame with thermal break.
- Solar Heat Gain Coefficient (SHGC) = 0.54.
- Lighting load = 17.8 W/m^2.
- Equipment Appliance Load = 7.5 W/m^2.
- Elevator load = 1 $\times$ 30 kW.
- Occupant density = 25 m^2/person.
- Percent fenestration = 40%.
- Fenestration U-value (W/m^2 C) = 3.2.
- Opaque wall U-value (W/m^2 C) = 0.55.
- Roof U-value (W/m^2 C) = 0.47.
- No below grade wall insulation.
- No perimeter floor insulation.
- Infiltration = 0.25 l/s/m^2 exterior wall.
- Outdoor air = 0.4 l/s/m^2 floor area.

The MNECB provides the energy reference level. A reference construction budget was developed by applying and costing conventional equipment to satisfy the energy reference level. The small building has a packaged rooftop, direct expansion cooling unit with an economizer and a natural gas-fired central boiler. The large building has an individual floor variable–air-volume system with central make up air, a cooling tower and economizer with hydronic radiation heating supplied by a natural gas-fired central boiler.

The list of technologies considered is described in rows S1 to S26 in Table 1 and includes envelope, lighting, smart controls, HVAC, renewable energy and equipment aspects. Each of the 26 measures was analyzed individually in terms of energy reduction and life cycle cost in comparison to the base case. The results are shown in Table 4 as percentage energy reduction and payback in years. Individual measures were then grouped into 13 measure sets and shown as columns SA through SM in Table 4. For example, measure set SA contains individual measure S1, S7 and S10. Energy savings and payback of the measure sets are shown in the last two rows of Table 4. Note that measure set energy reduction may be different that the simple sum of the savings associated with individual measures. For example, the savings of S1 (3.7%) + S7 (13.2%) + S10 (13%) equal to 29.9%, while the simulated energy reduction of SA is 28%. The grouping into measure sets attempts to account for such interactions among measures.

Significant energy savings with attractive economic returns were attained. Energy savings over 50% were achieved in five measure sets (SG, SJ, SK, SL, SM) in the small office building and four of these had discounted payback periods between 2.5 and 6 years. The 50% savings relative to the MNECB is considered as the high performance building threshold. Thus, the small

TABLE 4. Individual measures and measure sets with energy and cost comparisons to the base case

Measure*	Energy savings	Payback	SA	SB	SC	SD	SE	SF	SG	SH	SI	SJ	SK
S0	0.0												
S1	3.7	2.5	*	*	*	*	*	*	*	*	*	*	*
S2	2.5	2.8			*		*	*		*	*		*
S3	3.1	5							*			*	
S4	1.1	137											
S5	8.0	7.7						*			*		
S6	9.3	9.3											
S7	13.2	8.5	*	*	*	*							
S8	21.0	10.4							*			*	
S9	3.2	6.1				*		*	*		*	*	
S10	13.0	5.6	*	*									
S11	4.9	6.3		*	*			*	*		*	*	
S12	2.8	28.6											
S13	0.1	8.2											
S14	0.0	Never											
S15	16.4	0			*								
S16	16.7	0								*	*	*	
S17	34.0	14.2				*							*
S18	18.6	0					*	*	*				
S19	0.8	0			*	*	*	*	*	*	*	*	*
S20	2.1	8							*			*	
S21	2.0	24.6											
S22	2.2	244											
S23	–	?											
S24	1.8	0					*	*	*	*	*	*	*
S25	1.6	0					*	*	*	*	*	*	*
S26	1.0	10.9							*			*	
Energy savings (%)			28	32	40	46	31	44	56	30	43	53	52
Simple payback (years)			12	6	0.5	15	0	0	3	0	0	5	6

* S0: Base case, S1: Lighting power density of 11.5 W/m^2, S2: Perimeter daylighting with light dimming, S3: Occupancy sensors for lighting, S4: Active solar shading, S5: Add low-emissivity coating to windows, S6: Add low-emissivity coating and argon fill to windows, S7: Add low-emissivity coating, argon fill, and vinyl framed windows, S8: Triple-glazed low-e coated, argon filled, vinyl framed windows, S9: Increase wall insulation by ΔRSI = 0.9, S10: Condensing boiler (thermal efficiency = 95%), S11: Central air-to-air heat recovery 60% annual effectiveness, S12: Solar air preheating system, S13: Install high efficiency motors on supply fans, S14: Variable speed pump on heating loop, S15: WLHP system with condensing boiler and cooling tower, S16: WLHP system (same as S15) plus thermal storage, S17: WLHP system with ground source, S18: Radiant panel heating and cooling with displacement ventilation, S19: Low flow faucets, S20: Heat pump water heaters, S21: Solar thermal domestic hot water system, S22: Photovoltaic electric array, S23: Microturbine with heat recovery, S24: Low-energy office equipment, S25: Elevator efficiency measures, S26: Increase roof insulation by ΔRSI = 0.9.

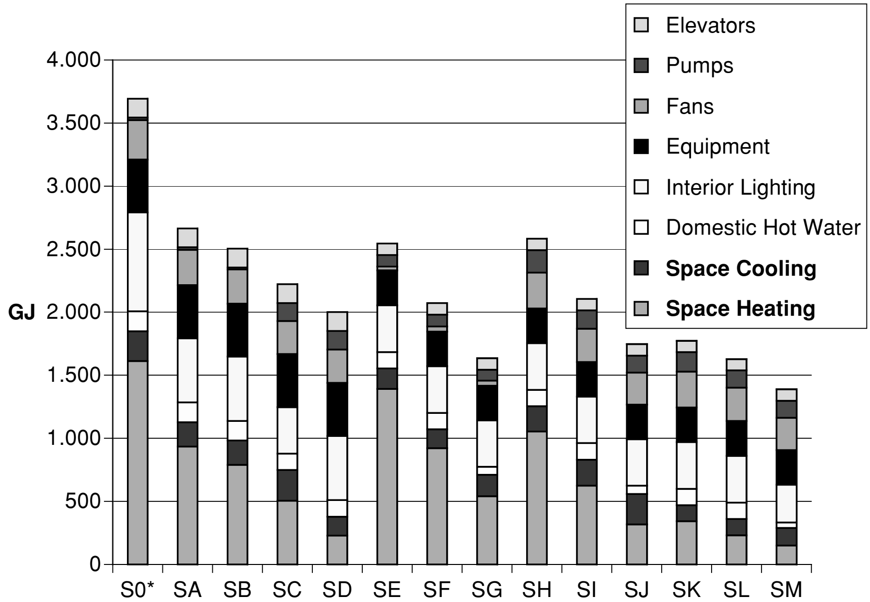

Figure 14. Energy usage versus measure sets for a small office building in Ottawa

office can apply a variety of measure sets to achieve this high performance level.

6.2.2. SIMULATION RESULTS

The DOE 2.1 E model was used to simulate the various measures and measure sets. Figure 14 presents the energy usage for the small office building in Ottawa, Canada. Energy usage is disaggregated into eight end-uses to indicate where measure sets impact energy reduction. These simulation results show that applying proven technologies can reduce energy consumption from the reference MNECB level by 25–65%.

6.2.2.1. *The Role of Energy Storage*

Measure S15 is a water loop heat pump system and reduces energy by 16% with an immediate payback. When included as part of measure set SC it achieves a 40% energy reduction at a payback of 0.5 years. Measure S16 is S15 with an increased storage capacity of about 30%. The cost is roughly \$50 per kW cooling capacity. This gives a 16.7% energy savings with an immediate payback. When S16 is included in measure sets SH, SI and SJ it can achieve 3, 43 and 53% energy reductions at 0, 0 and 5 years paybacks, respectively. This storage measure (S16) does not have much of an impact

TABLE 5. Space heating and cooling versus measure sets for Ottawa

	Measure sets													
	S0	SA	SB	SC	SD	SE	SF	SG	SH	SI	SJ	SK	SL	SM
Space heating (MJ)	1,6	935	791	503	227	1,3	922	541	1,0	623	318	344	230	148
Reduction heating (%)	—	42	51	69	86	14	43	66	35	61	80	79	86	91
Space cooling (MJ)	235	192	190	244	152	162	149	168	199	208	240	125	131	140
Reduction cooling (%)	—	18	19	−4	35	31	37	28	15	12	−2	47	44	41

on energy reduction, but when grouped with compatible measures in SI and SJ it is effective. Measure S17 is a ground source heat pump system and individually saves 34% energy at a payback of 14.2 years. When included in measure sets SK, SL and SM is reduces energy usage by 52, 56 and 65% at 6, 6 and 22 years paybacks. In actual projects small storages also serve to reduce the capital costs of ground source heat pump systems and can be used as a backup with standby electric or natural gas-fired boilers.

Table 5 gives the heating and cooling requirements for each measure set and the percentage reduction from the base case. It is seen that reduction in heating requirements is generally greater than cooling requirements, in many measures twice as great. Cooling is generally electrically driven and more expensive per unit of heating or cooling delivered. Cooling storage has always been more popular in Canada and the increase in energy efficient building designs should maintain this preference. For storage systems supplying both heating and cooling, energy efficient designs are better balanced in heating and cooling requirements. A complication is the low temperature needed for latent cooling due to the high summer humidity. Separate latent and sensible cooling need to be considered.

6.2.3. ECONOMIC ANALYSIS

A life cycle cost analysis was done to evaluate the economic attractiveness of the various measure sets. Included in the analysis is the impact on equipment sizing, usually a saving. The sizing changes can result in a significant cost reduction for the measure sets. In order to realize the payback periods shown, equipment must be sized in accordance with load reductions.

The analysis used projected average annual escalation rates of commercial sector electricity and fuel input prices supplied by Natural Resources Canada.

A real discount rate of 10% was used to convert all future expenses and savings into current dollars. This allowed calculation of the net present value of the savings and costs. Also calculated were the discounted payback period and the simple payback period. Each of these quantities was calculated over a life cycle analysis period of 20 years, the assumed life of the mechanical system. Maintenance costs were considered to remain constant in real terms.

There are several measure sets with immediate payback; SE, SF, SH, SI. Measures SE and SF are radiant panel systems with displacement ventilation. These systems have a similar cost to the base case, but they offer energy savings. Furthermore, significant sizing reductions, mainly in the cooling tower and chiller sizes, offset the incremental cost of the envelope and heat recovery measures. Because the elevator efficiency measures offer a net savings in capital cost, the capital cost of the other measures is further offset.

Measures SH and SI are hydronic water loop systems with water-to-air heat pumps. These systems also offer energy savings over the base case, but have an incremental cost over the base case. Other than the system type, these measures are the same as SE and SF, so either payback period is immediate for similar reasons.

A microturbine with heat recovery was one of the measures modeled. It has not been included in any of the measure sets. A more comprehensive analysis was done that took account of the different treatments of electricity and thermal energy and the effects of varying electricity and natural gas prices. This analysis is available in a separate report [6].

6.2.4. LOW-ENERGY BUILDING DESIGN CONCLUSIONS

The performance and economics of the measure sets applied to the small office building have demonstrated that significant energy reductions with attractive economic returns, are possible through careful selection and application of individual measures. Energy savings over 50% were achieved in five measure sets (SG, SJ, SK, SL, SM) in the small office building and four of these had discounted payback periods between 2.5 and 6 years. The 50% savings relative to the MNECB is considered as the high performance threshold. Thus, the small office can apply a variety of measure sets to achieve this high performance level.

It is possible to achieve 25% reductions compared to the base case building with no incremental cost (SE, SF, SH, SI). With careful selection and application of efficient building technologies at the early stages of design, and adjustment of equipment sizing to account for reduced demands, designs can result in energy savings of 30–40% with no incremental cost.

To successfully apply energy efficient designs, it is helpful to use an integrated design process involving energy simulation specialists to facilitate and support the architect and the mechanical and electrical engineers. The energy specialist works as an integral part of the design team to ensure that measures are properly planned and implemented and that their impacts are accounted for in HVAC equipment sizing. This process has now been applied in numerous designs where the energy reductions have confirmed simulation results.

The existing stock of buildings represents a much larger opportunity than new buildings for energy savings. Many of the measures and results presented here would be applicable to existing buildings when major system upgrades, replacements or building retrofits are undertaken. There may even be cases where pre-mature retrofits could be justified on a life cycle cost basis.

6.3. Design Options for Ground Source Heat Pump Systems [5]

6.3.1. INTRODUCTION

This section is a brief guide to design options for the building side of ground source heat pump systems for office buildings. Ground source heat pump systems can significantly lower heating and cooling operating costs and can qualify designs for renewable energy credits under several sustainable building rating programs.

There are a variety of design approaches that can be taken. Both distributed or incremental heat pumps and central heat pumps as applied to closed loop ground source systems are considered. Available products are identified for each design option as well as pumping arrangements, system schematics showing components for each design option, and outdoor air systems. Table 6 compares design options.

In addition, a schematic drawing of each system, elevation sketches showing zone component arrangements for both distributed and under floor designs are given. Drawings of two approaches to vertical ground heat exchanger layout are also shown.

6.3.2. DISTRIBUTED WATER LOOP HEAT PUMP (WLHP) SYSTEMS

This is a conventional water loop heat pump system using a boiler and cooling tower to maintain the water loop temperature (see Measures S15–S17 in Section 6.2). Since outside air handling unit and other terminal devices like unit heaters, wall fin convectors, etc which are often used with this water loop heat pump system need a different operating water temperature than that of the water loop, a plate heat exchanger is used between the primary heating circuit

TABLE 6. Comparison of ground source heat pump system types

Features	Distributed water loop heat pump with ground loop	Central reversing heat pump withground loop	Central multiple heat pumps with ground loop
1 Simultaneous heating & cooling	The system can provide simultaneous heating and cooling to individual spaces inside the building as heat pumps operate under control of zone thermostats.	The system can provide simultaneous heating and cooling with the help of 4 pipe fan coil units. During heating season, central heat pump maintains the temperature of the heating loop. The cooling loop exchanges heat with the low temperature ground loop that passes out of the central heat pump. During summer, the central heat pump reverses its cycle, rejecting heat to the ground loop. The cooling loop bypasses the ground loop and passes through the evaporator of the central heat pump.	The system can provide simultaneous heating and cooling by the use of both cooling & heating side of heat pumps. During the heating season, if there is no demand for cooling a diverting valve connects the low temperature side of the heat pump loop to ground loop. During the summer, heat pumps reject heat to ground loop. When there is simultaneous demand for heating & cooling, the water loops on both low and high temperature side are isolated from the ground loop and the cooling and heating water is supplied by the respective secondary pumps.
2 Heat pump system capacity	The total installed capacity of heat pumps will be more than the central heat pump.	The size of the central heat pump will be smaller when compared to the total installed capacity of heat pumps in the distributed water loop heat pump system. This is because the central heat pump is sized to meet the block load of the entire building.	The size of the central heat pump will be smaller when compared to the total installed capacity of heat pumps in a distributed water loop heat pump system. This is because the central heat pump is sized to meet the block load of the entire building.

3 Space requirements	Space savings are not expected at the zone level. However space required for mechanical room components may be lower when compared to central heat pump system.	Space savings arising from the smaller size of boiler and cooling tower is offset by the addition of central heat pump. Use of fan coil units at the zone level is not expected to result in space savings because of larger requirement of air flow arising from lower temperature difference across the heating coil.	Space savings are expected with no need for a boiler and cooling tower. However this may be offset by the presence of a few larger water-to-water heat pumps.
4 Use with radiant floor heating and cooling	Radiant floor heating and cooling cannot be done with this system.	The system appears to be quite compatible with radiant floor heating and cooling. Since the supply temperature of water will be relatively lower, it looks ideal for use with radiant floor heating and cooling as the control of surface temperature will be easier.	This system can be used with radiant floor heating & cooling. During mid-season when there is demand for simultaneous heating and cooling, the low temperature side of heat pumps may have to depend on ground loop more often for heat source. In the case of only primary pumps, control of floor temperature during cooling may be difficult due to lower cooling supply temperature.
5 Humidification	Humidification can be done at the system level by having humidifier section at the air-handling unit supplying fresh air. The humidifier can be electric or gas type. Another method is to use evaporative pads such as GLASdek from Munters. This eliminates the use of gas or electricity for humidification.	Humidification can be done at the system level by having humidifier section at the air-handling unit supplying fresh air. The humidifier can be electric or gas type. Another method is to use evaporative pads such as GLASdek from Munters. This eliminates the use of gas or electricity for humidification.	Humidification can be done at the system level by having humidifier section at the air-handling unit supplying fresh air. The humidifier can be electric or gas type. Another method is to use evaporative pads such as GLASdek from Munters. This eliminates the use of gas or electricity for humidification.

(*cont.*)

TABLE 6.6. (*Continued*)

Features	Distributed water loop heat pump with ground loop	Central reversing heat pump withground loop	Central multiple heat pumps with ground loop
6 Noise, maintenance, first cost	Noise at the occupied spaces can be a problem if proper care is not taken during design. Maintenance may become laborious and expensive. The first cost is expected to be less than a central heat pump system using ground loop.	Substantial reduction in noise level can be expected in the conditioned spaces by eliminating heat pumps at the zones. However central heat pump may require special attention to reduce the noise level. The maintenance is expected to be lower when compared to the conventional distributed water loop systems. A fan coil unit requires less maintenance than a heat pump due to fewer moving parts. The first cost of this system is expected to be higher than the distributed water loop heat pump system. Addition of a central heat pump and separate piping for hot and cold water should result in higher first cost.	Substantial reduction in noise level can be expected in the conditioned spaces by eliminating heat pumps at the zones. However central heat pumps may require special attention to reduce noise level in vicinity of mechanical room. This system when used with VAV air handling systems will further reduce the noise levels in occupied spaces. Maintenance may not be lower due to the presence of more rotating machinery and controls. The first cost of this system is expected to be higher than the distributed water loop heat pump system. Addition of secondary pumps and separate piping for hot and cold water will result in higher first cost. The cost involved with the ground loop may result in a higher first cost.
7 Outdoor air	Outdoor air is treated and pre-conditioned by a central water-to-air heat pump. The outdoor air is delivered into the return air plenum of the zone heat pump units. A total enthalpy wheel can reduce the energy consumed for treating outdoor air.	Outdoor air is treated and supplied by a central air-handling unit. The required amount of air is delivered into the return air plenum of the fan coil units. The air-handling unit is provided with a hot water coil and chilled water coil. Both the coils are served by central reversing heat pump. A total enthalpy wheel can reduce the energy consumed for treating outdoor air.	Outdoor air is treated and supplied by a central air-handling unit. The required amount of air is delivered into the return air plenum of the fan coil units. The air-handling unit is provided with a hot water coil and a chilled water coil. Both the coils are served by central reversing heat pump. A total enthalpy wheel can reduce the energy consumed for treating outdoor air.

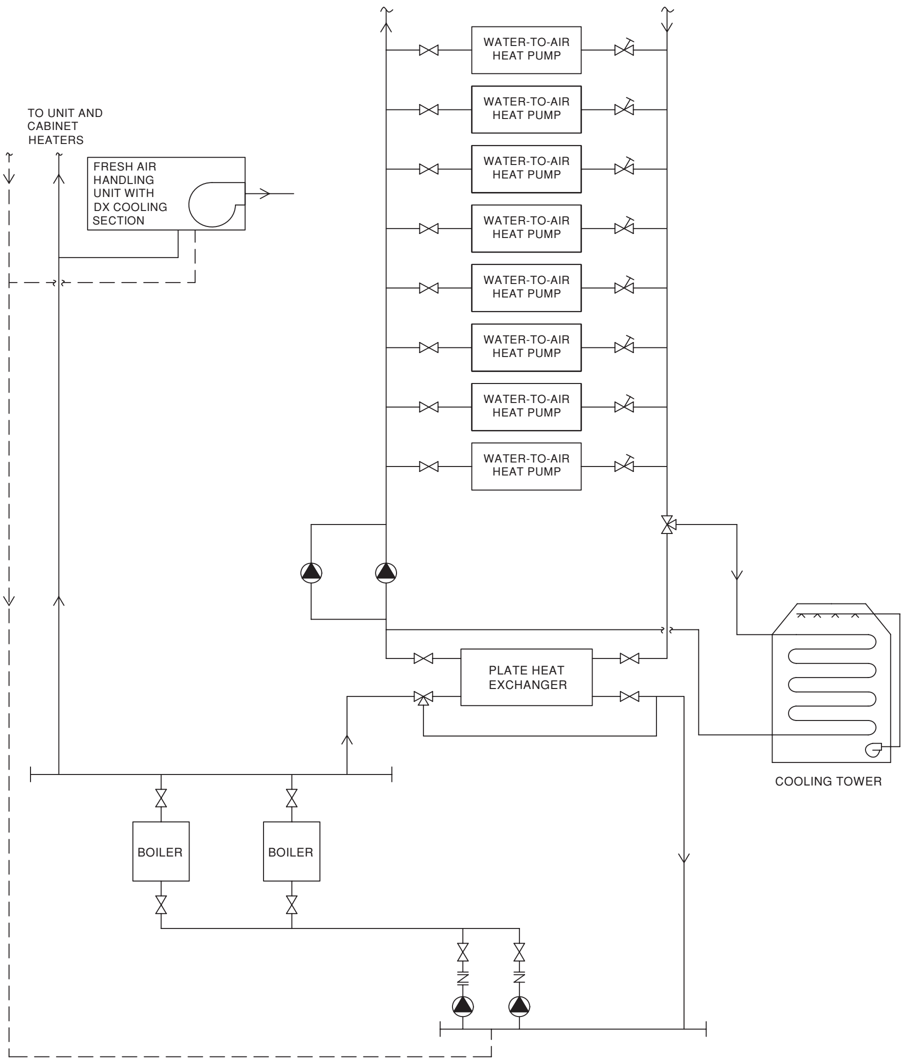

Figure 15. Conventional water loop heat pump system

and the water loop. The outside air-handling unit may have a DX cooling coil and a hot water coil or it may use a water-to-air heat pump for heating and cooling of outside air. See Figures 15 and 16.

The water-to-air heat pumps are available in the market from various manufacturers ranging in capacity from 1/2 tons and upward. Manufacturers include ClimateMaster, McQuay, Trane, Carrier, Waterfurnace and Florida Heat pump.

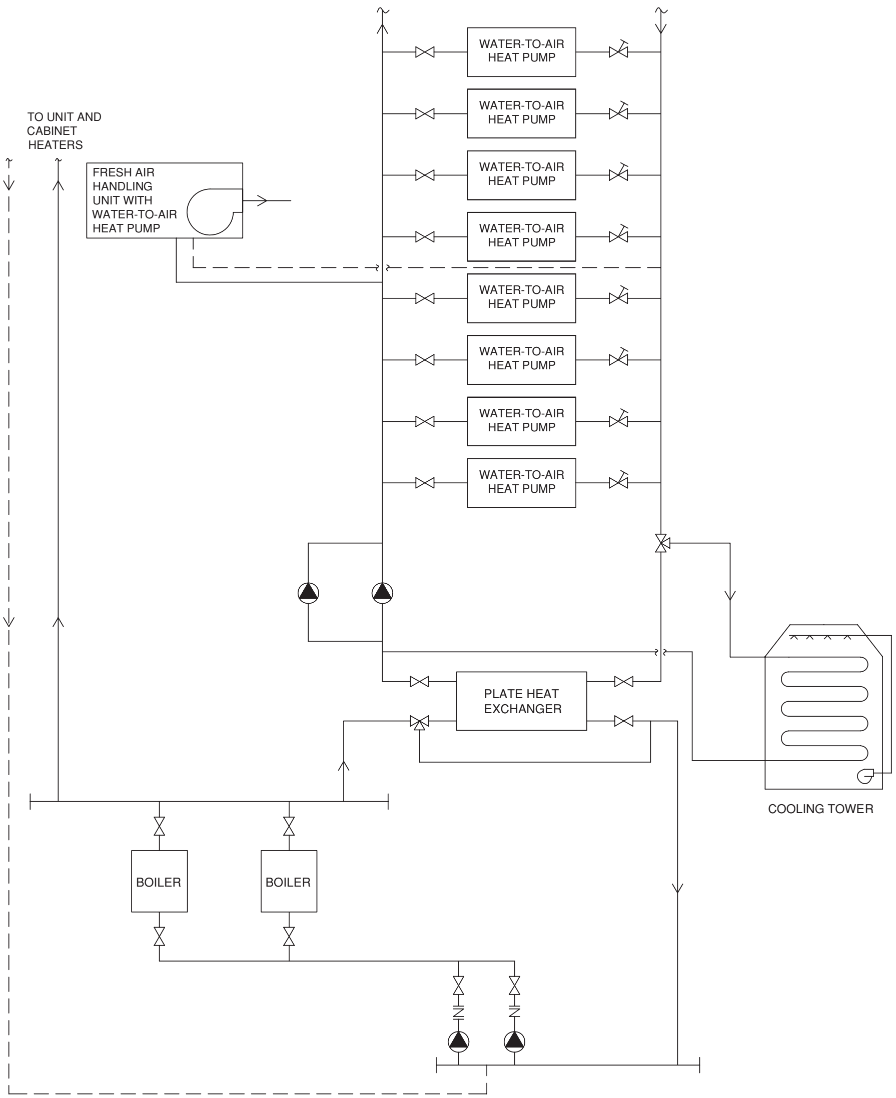

Figure 16. Conventional water loop heat pump system with fresh air heating by water-to-air heat pump

The system is very simple in configuration. The water-to-air heat pumps are installed in the individual spaces to be heated and cooled. The water loop connecting the heat pumps helps to transfer heat within the building. When the loop temperature rises above the design temperature, the water is directed through a cooling tower. The water loop extracts heat from the primary boiler heating circuit when the loop temperature falls below the design minimum temperature.

The simplicity of the system should result in lower first cost and lower maintenance cost when compared to the systems discussed in the following sections.

Advantages:

- Simultaneous heating and cooling available.
- Lower first cost than the central heat pump systems using ground loop.

Disadvantages:

- Noise in occupied spaces.
- Maintenance may become laborious and expensive.
- Not suitable for radiant cooling.
- Operating cost may be higher than the central heat pump systems using ground loop.

6.3.2.1. *WLHP with Air Source Heat Pump Water Heater*

This is a variation of the above system with a heat pump water heater replacing the boiler. The hot water is stored in a storage tank and released to the water loop to meet the peak loads during extreme weather conditions. The storage facility gives a certain degree of freedom for the air-to-water heat pump to choose the time of operation to suit the weather conditions. This system may not be suitable for regions with extreme cold weather. However, this is an environmentally cleaner system. See Figures 17 and 18.

Many of the commercially available air-to-water heat pumps have only published operating data in the entering air temperature range above 40 °F. The performance data below 40 °F is readily available for Toyo and Continental units. Capacity and performance data at ambient temperatures below 40 °F are required to be suitable for use in a water loop heat pump system. Commercially available air-to-water heat pumps in the North American market include Continental, Toyo Carrier, Mac Systems and Environmentally Engineered Equipment Inc.

6.3.2.2. *WLHP with Ground Loop Heat Exchanger*

Figure 19 depicts a water-loop heat pump system where the cooling tower and boiler have been replaced with a ground heat exchanger. This is the most common configuration of ground source heat pump used in commercial buildings.

6.3.3. CENTRAL HEAT PUMP (CHILLER) WITH GROUND LOOP

A central heat pump is used for providing heating water for space heating. The heat source is the ground loop. A stand-by boiler is provided to meet

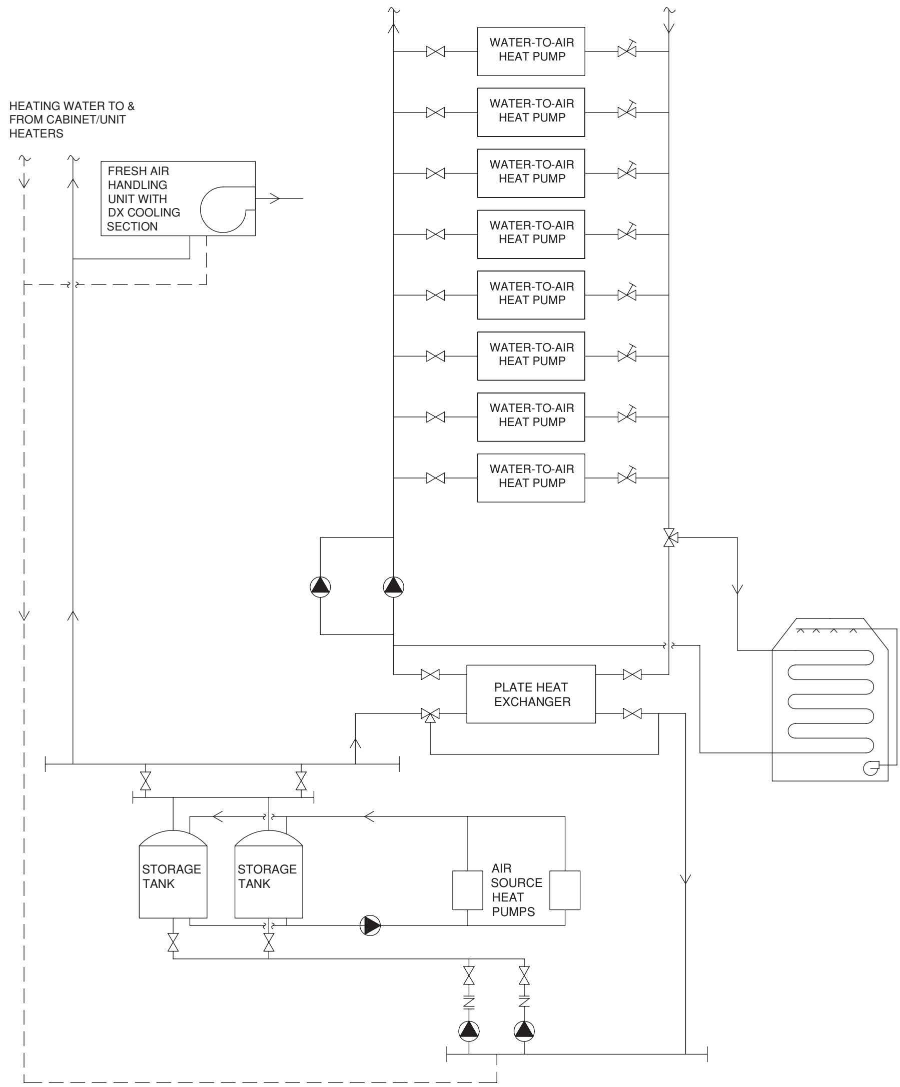

Figure 17. WLHP with air source heat pump water heater

peak loads during extreme weather. A 4-pipe fan coil unit is used at the zone level for cooling and heating. Outside air is supplied into the building by a central air-handling unit fitted with a heat recovery wheel and humidifier section.

The central heat pump is a reversing type. During summer, the central heat pump acts as a chiller supplying chilled water for cooling and rejecting heat to the ground loop. When there is a demand for simultaneous heating

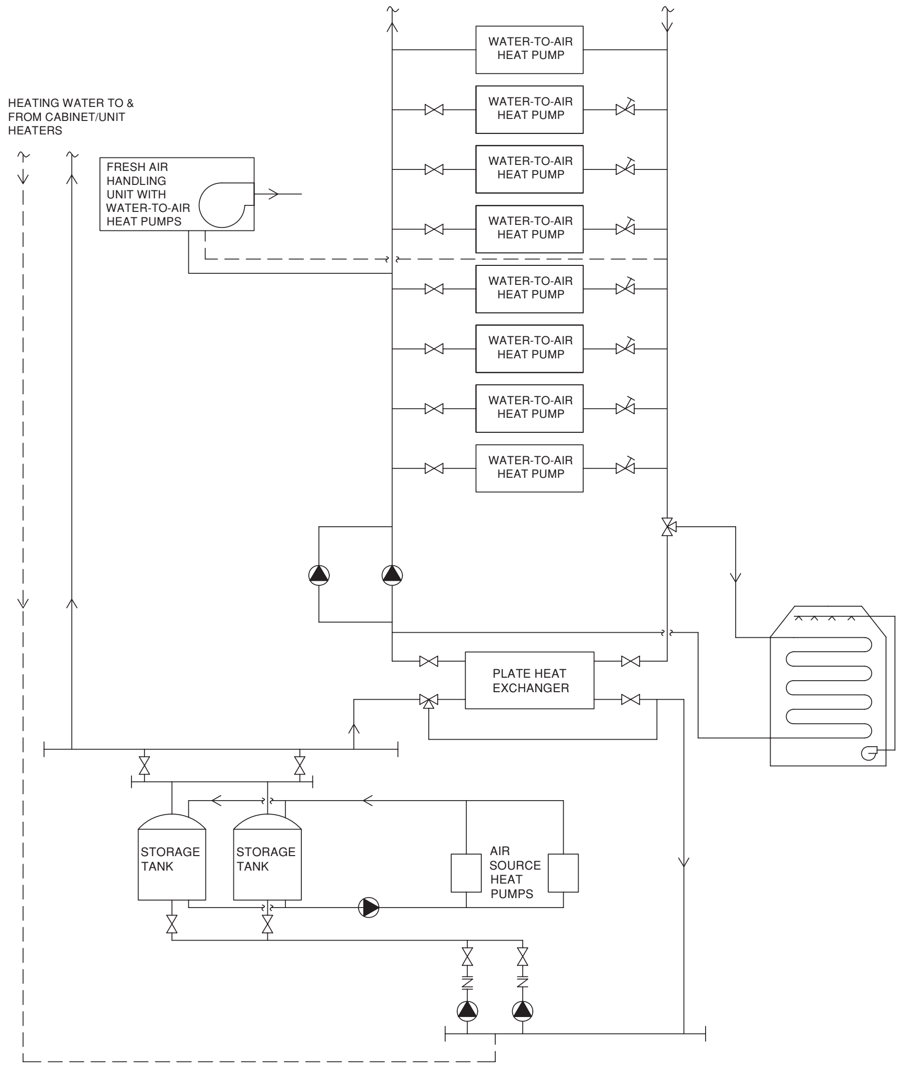

Figure 18. WLHP with air source heat pump water heater and fresh air heating by water-to-air heat pump

and cooling, the central heat pump resumes its heating mode of operation. The cooling circuit then bypasses the chiller and passes through the plate heat exchanger rejecting heat to the colder ground loop. It should be noted that the ground loop will be colder as it reaches the plate heat exchanger after passing through the evaporator of the central heat pump (see Figure 20).

A cooling tower is provided in series with the ground loop to assist during peak loads in extreme weather conditions. The cooling requirement of the

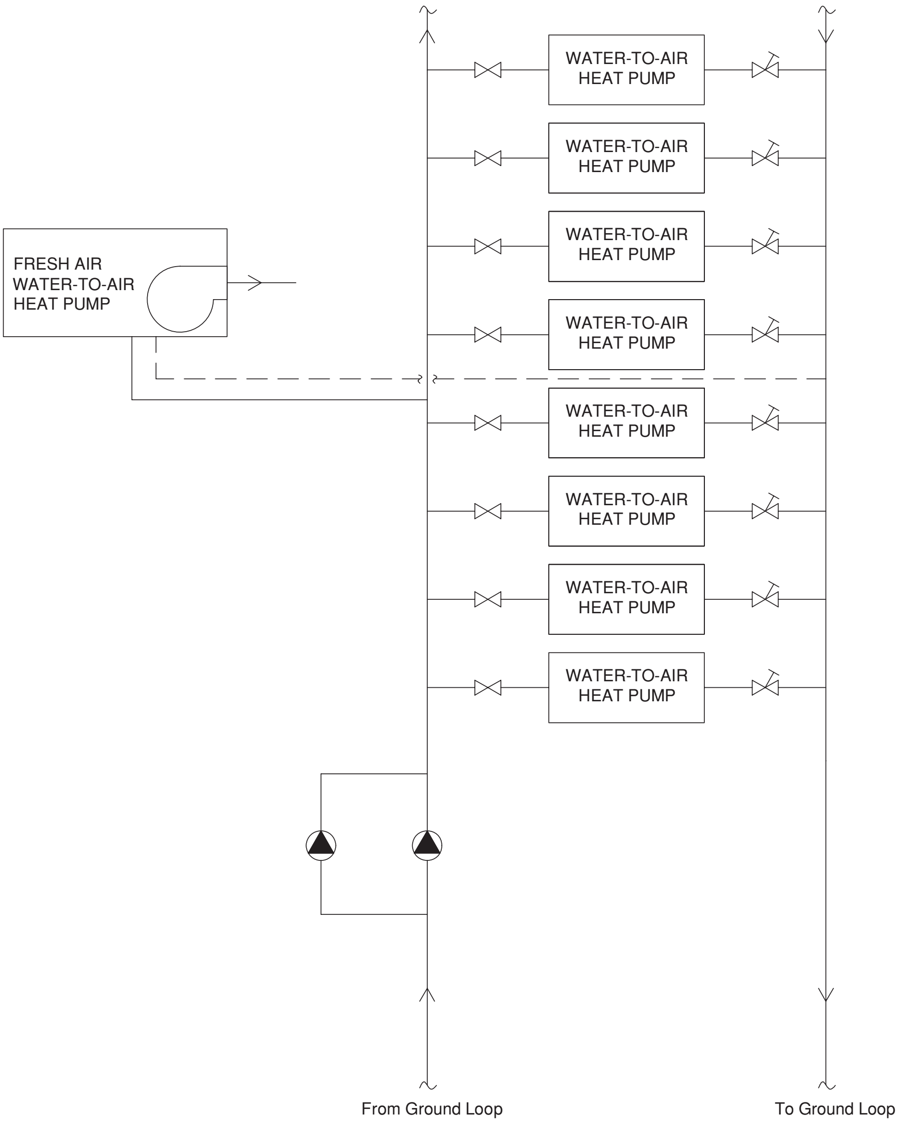

Figure 19. WLHP with ground loop

building is often far in excess of the heating requirement, which would lead to ground loop heat exchanger requirements that are too large and costly. The ground loop heat exchanger is then sized to the maximum amount of heat extracted during the design heat hour. The cooling tower is sized to meet the cooling heat rejection requirements in excess of the ground loop heat exchanger capability. A diverting valve directs the ground loop fluid through

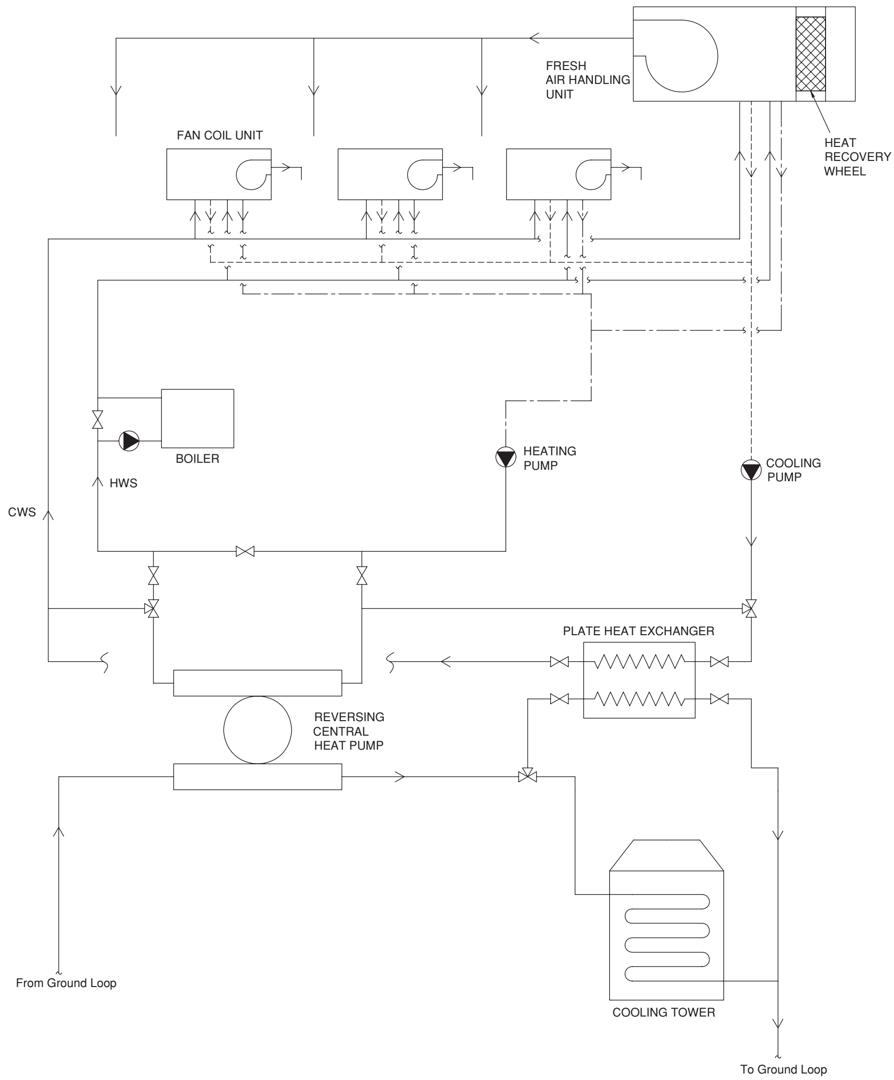

Figure 20. Central heat pump/chiller with ground loop

the cooling tower when the temperature of the fluid leaving the condenser of the chiller exceeds the design leaving temperature.

The parallel water flow circuits through the cooling tower and the plate heat exchanger helps to control the temperature of the cold water flowing through the heat exchanger during simultaneous cooling and heating demand. In such situations, the cooling tower circuit serves only as a return path for the diverted fluid to the ground heat exchanger.

If the temperature of the fluid from the ground loop falls below design during winter, the central heat pump may not be capable of maintaining the heating water supply temperature. In such a situation, the heating water will be directed through the stand-by boiler that will add the required heat to maintain the heating supply temperature. Use of evaporative pads in the air-handling unit for humidification will eliminate the need for natural gas for humidification and the system will be more energy efficient.

Commercially available large heat pumps (chillers) that can be employed include FHP manufacturing Co., Waterfurnace, ClimateMaster, Hydron Module LLC, Addison Products Co., Trane Heat Harvester Energy Efficient products and McQuay. More than one unit may have to be used in a building depending on the heating load and the model of equipment chosen. However this will not result in any significant change in the schematic shown for a single chiller/heat pump. The above manufacturers are located in North America.

Advantages:

- Simultaneous heating & cooling can be provided.
- Low noise levels in the occupied zone.
- Higher energy efficiency through the use of ground source and a central heat pump.
- Suitable for radiant floor heating and cooling.
- Lower maintenance cost.

Disadvantages:

- More plant space may be required.
- Higher first cost.
- Ground loop must be sized to meet the heating and cooling loads during extreme weather conditions.

6.3.4. CENTRAL HEAT PUMP (CHILLER) WITH GROUND LOOP AND HEAT PUMP WATER HEATER

This system is very similar to Central Reversing Heat Pump with Ground Loop but it uses a heat pump water heater instead of a boiler to meet peak loads during extreme weather conditions. The heat pump water heater is a water-to-water heat pump operating on the ground loop. The heated water is stored in a storage tank and is released to the hot water circuit when required. The overall system energy efficiency will improve by the use of the water-to-water heat pump in place of a boiler (see Figure 21).

The same models mentioned in Section 6.3.3 can be used to heat the water. However the heat pump can be a heating only type. The capacity of the unit is sized to meet the peak load in excess of the capacity of the central heat pump at

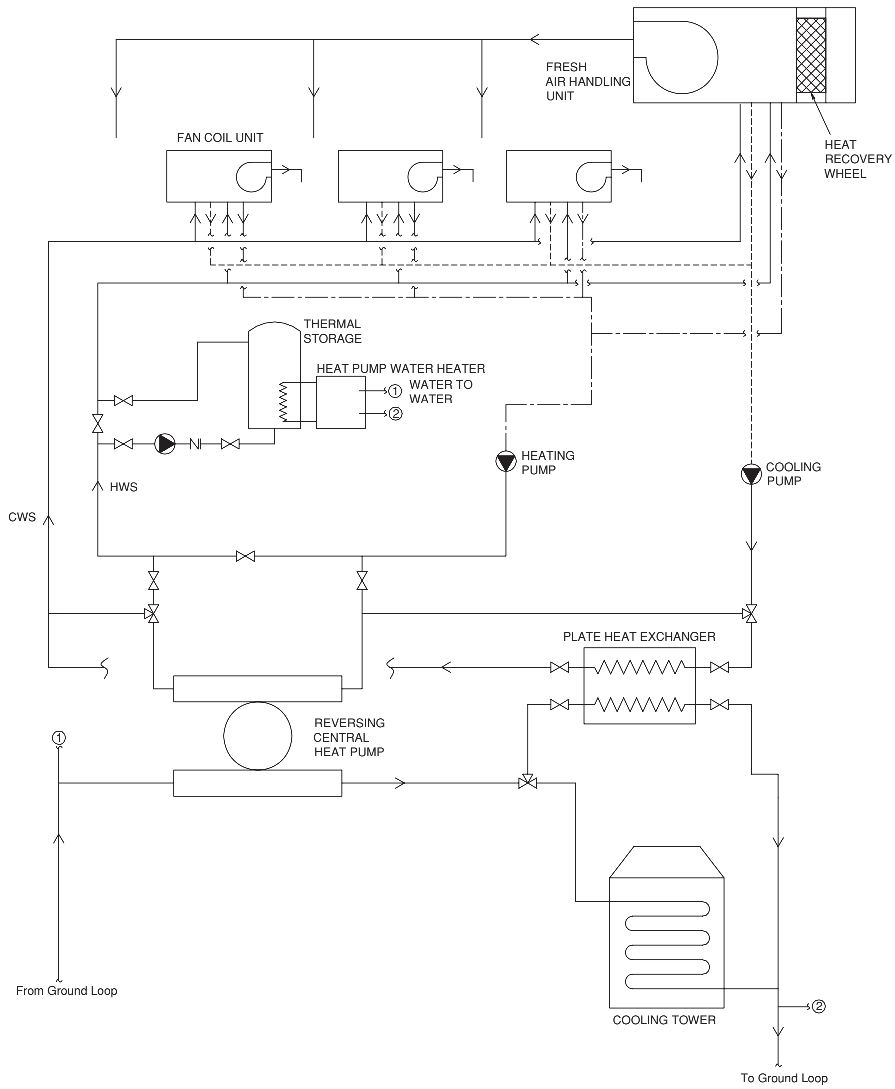

Figure 21. Central heat pump/chiller with ground loop and heat pump water heater

the extreme winter conditions. This means that the water-to-water heat pump can be much smaller in capacity than the central heat pump (chiller).

When the temperature of the heating water falls below the design temperature during the heating season, the water stored in the tank will be released to the heating circuit. The water-to-water heat pump will maintain the temperature of stored water in the tank. The storage tank allows the heat pump water heater to operate at a lower output than the heating demand of the building. Compared to a boiler-assisted system, depending on how electricity is generated, this system can have lower CO_2 emissions.

6.3.5. CENTRAL MULTIPLE HEAT PUMP SYSTEM WITH GROUND LOOP

This system comprises multiple water-to-water heat pumps working at the plant level providing simultaneous heating and cooling to the building using both the high and low temperature sides of each heat pump. It uses the ground loop during the cooling or heating season. A primary–secondary pumping system installed at both the low and high temperature sides circulate the cooling and heating water between the plant and the terminal units.

Small to medium size water-to-water heat pumps are considered in this system. Since the primary pumps are common to all the heat pumps, this system may not be ideal for projects where large size multiple heat pumps are used. It is better if dedicated primary pumps are used in the case of large multiple heat pumps. This is discussed in Section 6.4.1.

Commercially available water-to-water heat pumps under 10 tons capacity range that can be employed in this application include FHP manufacturing Co., Waterfurnace, ClimateMaster, Hydron Module LLC, Addison Products Co., and Trane.

During the summer, the heating side of the heat pump rejects heat to the ground loop. The water flow through the heat pump condensers is diverted from the secondary circuit to the ground loop by diverting valves as shown on the schematic diagram (Figure 22). When there is no cooling demand, the low temperature side of the heat pumps is connected to the ground loop to provide a heat source.

The combination of primary–secondary pumping and ground loop make the control system more complex. The secondary pumps have variable speed drives for better energy efficiency. The 3-way valve at the inlet of the heating secondary pumps is on–off type. This is because there is no mixing of supply and return water in the heating circuit. It should be noted that the heating supply temperature of the heat pumps will be in the range of 40–55 °C. It is economical to supply heating water at the maximum available temperature and this eliminates the need for mixing supply and return water in the heating circuit. The valve only isolates the secondary heating circuit from the primary when there is no heating demand. The primary circuit, on the other hand, will be connected to the ground loop when the condenser entering temperature exceeds the design value and the ground loop circulating pump is started. When heating demand rises, the secondary heating circuit will be connected to the primary again and the ground loop temperature will start dropping. A modulating 3-way valve at the inlet to the primary heating pumps will allow diverting only a portion of the fluid to the ground loop and result in better

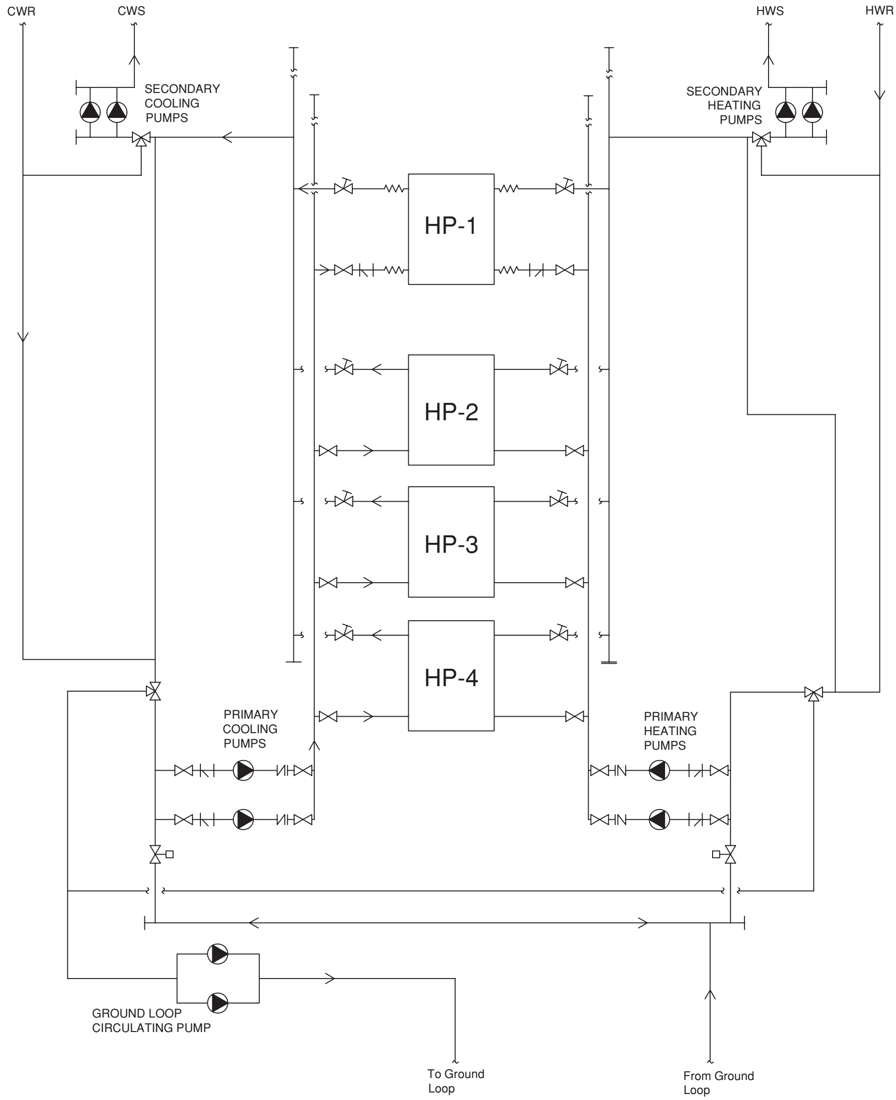

Figure 22. Central multiple heat pump system with ground loop and primary/secondary pumping

control of the temperature of the primary heating circuit. This requires variable flow capacity for the ground loop circulating pumps and will increase the first cost.

When the cooling load is smaller than the heating load, the 3-way valve at the inlet of secondary cooling pumps allows mixing of the supply and return water to maintain the design cooling supply temperature to the load.

The secondary cooling circuit is isolated from the primary when there is no cooling load.

The operation and control of this system will be complex and maintenance cost will be higher. As an alternative, secondary pumps are eliminated and primary only pumping is considered (Section 6.3.6).

6.3.6. CENTRAL MULTIPLE HEAT PUMP SYSTEM WITH PRIMARY PUMPING

In this case, each central water-to-water heat pump has a dedicated pump to circulate water. The water flow through the terminal devices is varied with the help of a bypass between supply and return header pipes. The advantage with this system is that the pumps can be shut off along with heat pumps with decreasing load during summer season. This scheme will be quite suitable for use with large multiple heat pumps as it will be less complicated and easy to operate than primary–secondary pumping discussed in the previous section. It is also suitable for use with small to medium size heat pumps as long as the number of units used is limited (see Figure 23).

The differential pressure controller shown in the schematic diagram helps to maintain the required differential pressure head and bypasses the flow from the supply to the return when the demand drops. This may lead to shutting off the heat pumps or connecting the primary circuit to the ground loop as discussed in the previous section. When the heat pump is shut off, the circulating pumps can be shut off simultaneously. This saves operating cost. The heat pumps in the Central System can be shut off only when both the cooling and heating demand is satisfied. In other words, the heat pump can be shut off when the heating supply temperature increases only if there is no cooling load to be met.

Advantages:

- Simultaneous heating & cooling can be provided.
- Low noise levels at the occupied zones.
- Higher energy efficiency through the use of ground source and central water-to-water heat pumps.
- Suitable for radiant floor heating and cooling.

Disadvantages:

- Higher first cost.
- More maintenance due to the presence of complex controls and rotating machinery.
- Ground loop shall be adequately sized to meet the heating & cooling loads during extreme weather conditions.

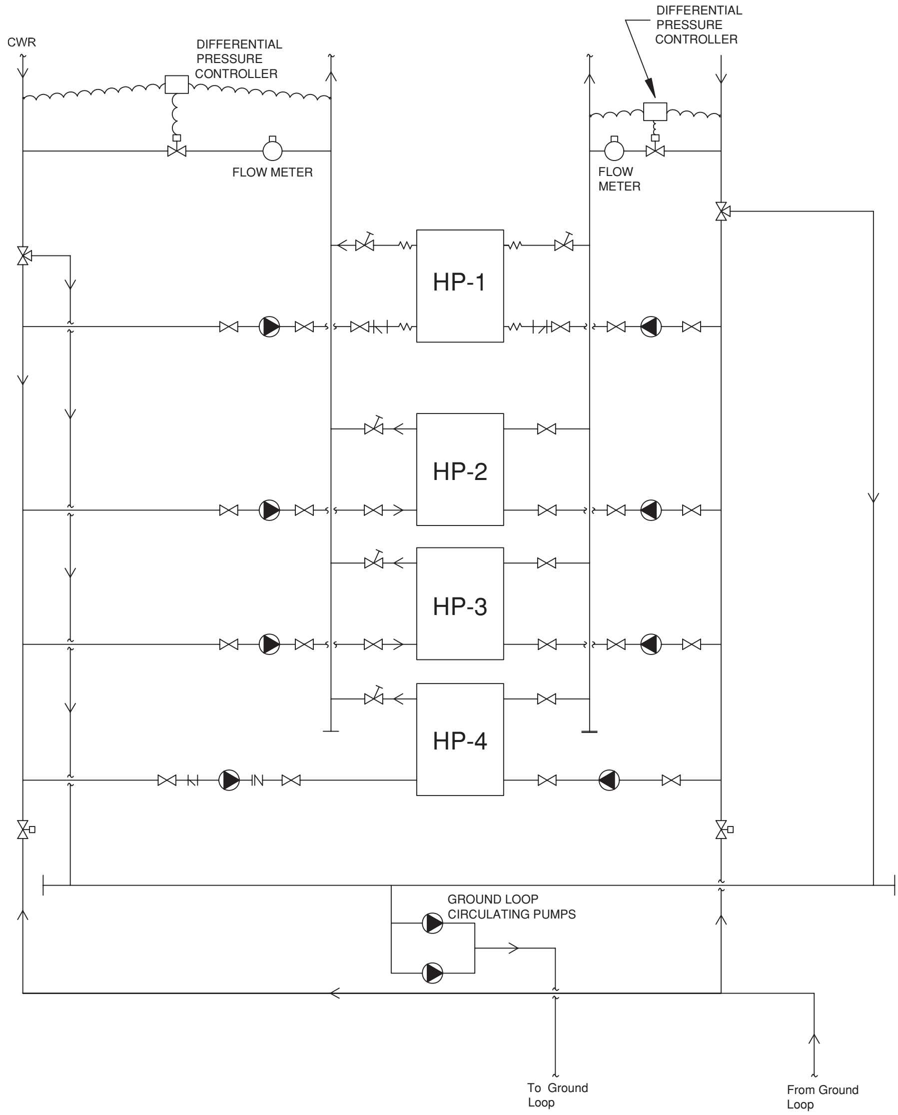

Figure 23. Central multiple heat pump system with ground loop and primary pumping

6.4. Some Recent Canadian Building Projects

6.4.1. PROJECTS

There are recent building projects in Canada that generally confirm the simulation results presented. Cost-effectiveness of some of these buildings is even greater than the simulated results due to savings in areas not modeled.

A brief summary of some recent projects is given, including both new and retrofit projects. Many projects have used the *Commercial Buildings Incentive Program (CBIP)*. Natural Resources Canada offers a financial incentive for the incorporation of energy efficiency features in new commercial and institutional building designs. The objective is to encourage energy-efficient design practices and to permanently Improve the Canadian building design and construction industry. A financial incentive of up to $60,000 is awarded to building owners whose designs meet CBIP requirements. The program requirements are based on the Model National Energy Code for Buildings. An eligible building design must demonstrate a reduction in energy use by at least 25% when compared to the requirements of the MNECB. The program runs to March 2007.

6.4.2. TELUS BUILDING

The Telus Building in Vancouver at 550 Robson Street is on a corner of a city block of related Telus structures built from 1917 through the 1950s. The Telus Building is well known for its double skin. The Telus has about 25% of the floor area of this block. The buildings needed updating due to changing seismic and other building requirements and the changing nature of the communications business post deregulation. Conventional retrofit would have resulted in a building with 60% of the original floor space. Demolition of this building was considered. A design competition sought for an innovative retrofit solution maintaining the floor area and meeting current code requirements.

The winning design proposed that the brick cladding (no insulation) be removed exposing the concrete shell. A glass skin (double-glazed, fritted and frameless) was added on the two street faces. This layer is hung 1 m outside the shell. It extends onto city property. It features operable panels and fritted glass (on the #2 surface) to control solar glare. The original single-glazed windows remain.

There are dampers bottom and top of this cladding cavity to promote natural ventilation. A few integrated photo-voltaic panels are sufficient to power the fans (twelve 1/2-HP motors) that assist ventilation through the double skin. There are tracks within the double skin that assist in the cleaning of surfaces #2 and #3. There is also a grid of fibre optic lights that blink periodically and change colour.

The building was strengthened by a bottom to top shear wall within the building and it has been left exposed. A commercial ground floor was also established. There was a mechanical room on the top floor. This has now been converted into a gym and immediately below are the executive offices. The original floor to ceiling height was 14–16 feet and a raised floor (18″) was introduced. It has wood core, metal-faced panels with a pressurized plenum below. Everything was raised 18″ including the stairwell (three risers were added) except for the washrooms, which are ramped. Heating is provided by abundant condenser water from an adjacent building. There are fan coil units on each floor.

The use of the building changed from mainly housing telephone exchange equipment to office space. The floor plate is 90 feet by 90 feet. Interior security railings were added to original double hung windows. We were told of frequent moves in the building. Costs were formerly $4 to $5 thousand per person per move and are now $500 per person per move.

The adjacent building is on the site of the main telephone central for downtown Victoria. Slab cores showed very poorly mixed concrete from 1917 construction. It will be taken down to two floors and a 8-storey atrium will take its place. Acoustics are a concern. The Telus Building will be opened up to this atrium changing dramatically the present closed in feeling.

Occupants like the operable windows in the original and the new facade, but it is noisy with the windows opened. Lighting is good but acoustics are a problem. Rolling and moveable partitions are commonly used—usually without a ceiling.

Value of the retrofitted building is double the construction cost and Telus is considering selling and leasing back. Building was pre-LEED and might qualify as Gold Star.

6.4.3. VANCOUVER ISLAND TECHNOLOGY PARK

Vancouver Island Technology Park was built in 1977 as a hospital for the mentally and physically handicapped. It was demolished and rebuilt as a 165,000 square foot building. It reduced energy usage 42.5% below MNECB and was the first Canadian LEED Gold Building. It uses a water loop heat pump system (see S15 in Table 1.) and carbon dioxide monitoring as a control parameter. The parking lot could have been used (horizontal system) as the heat pump heating and cooling source (S17 instead of S15). This would have significantly increased the energy reduction.

Operable windows were part of the design, but eliminated in the "value engineering" phase. Fifty-four % of the building materials were purchased within 500 miles of the site and 35% fly ash concrete was used. There were extensive savings on storm water runoff using ponding, weirs and plantings on the extensive property to avoid large diameter piping.

An "invisible structure" was used in the parking lot replacing the conventional ashphalt surface. Grass was planted in the parking area and the driving area was left bare. The shade of the cars apparently promotes grass without additional watering. Reduced runoff leads to savings with storm drainage.

The invisible structure is more expensive but there are maintenance savings and capital savings for drainage.

Dual flush toilets were used and waterless urinals (falconwaterfree.com). Two urinal types were used. One with a cartridge lasting several weeks. The other requires the liquid to be added periodically by staff. The specific gravity is chosen so that urine sinks beneath.

Estimated $15–$20 thousand fee for the LEED documentation. Separating deconstruction and reuse from construction allows more detailed design and more detailed specs.

6.4.4. INFORMATION AND COMMUNICATIONS TECHNOLOGY (ICT) BUILDING [7, 8]

A team including Stantec Architecture, HOK and Barry Johns Architecture created the new Information and Communications Technology Building a striking addition to the University of Calgary campus. Faculty members of both the Computer Science and Electrical & Mechanical Engineering groups insisted on having operable windows. This desire for direct access to fresh air meant a conventional mechanical approach could not be used. The resulting system exposes the building's concrete for use as a heat sink with an embedded network of tubing. The slab absorbs the heat generated by a heavy concentration of computers and people. Chilled water is run through plastic piping embedded in the lower portion of each above-grade floor's concrete slab. The slab then acts as a radiant cooling source, primarily affecting the spaces below the slab, which must be exposed for the system to work. Fortunately, the ICT Building's design already called for open ceilings.

The ICT Building may be the first use of structural slab radiant cooling in Canada. Unlike radiant heating, radiant cooling is not commonly used in North America. It is not well understood and has been misapplied in the past. Slab radiation will meet half its cooling needs and traditional air-based ventilation the rest. As a result, the ventilation ducts, air-moving fans and related equipment will be half their normal size, resulting in significant energy saving and allowing each floor height to be reduced by 375 millimeters. Other energy savings will come from the slab system's cooling water, which will be used twice. The cooling water will first be used in the building's air handling system's cooling coils and then reused in the slab cooling system before it is returned to the chiller plant.

The decision to expose the concrete led to a natural design strategy of leaving all the building's materials and systems unconcealed. The ICT building opened in September 2001 and houses Computer Engineering with labs and offices. The offices are mainly on the east-west perimeter with labs and graduate student offices on the interior. There is an interior walkway linking the University and the ICT gets 7,000–8,000 people daily walking through it. The budget was based on an area of 15,000 m^2 but turned out at 18,000 m^2. The construction budget was $25 million and the total budget was $40 million. It came in at $700,000 under budget. The design was a 35% reduction from the MNECB and received $80,000 from CBIP. It was a Design-Build under Ellis Don with many budget cuts. One of these cuts was all the outside sun screening. Another concept dropped was displacement ventilation. Maintenance department was against raised floors.

The original concept was operable windows with natural ventilation shutting down the conventional cooling during appropriate conditions. There are two solar towers at North and South ends of the building with fan assist since the towers are not high enough to ventilate under all conditions. The natural ventilation comes through the office and down the corridors. The Building Automation System has low temperature sensors to assist in locating freezing danger from open windows during winter. While we were touring with the facilities manager we spotted an open window with outside temperature about –15 °C. The office was unoccupied and the thermal plume was quite visible. Window cranks are taken away from the offending office occupants if the low temperature alarm goes off.

Control in the perimeter offices is conventional with about one thermostat per every three offices. The open windows probably could be correlated with the two offices without a thermostat. A central heating and cooling plant supply the entire campus.

6.4.5. C. K. CHOI BUILDING, UNIVERSITY OF BRITISH COLUMBIA [3]

Reused heavy timbers from an adjacent building demolition were structural elements. Reused Vancouver brick pavers were used for the building veneer. More than 50% of the total materials are reused or recycled. It was designed to use 50% of the energy based on ASHRAE 90.1 Standard as a reference. It was the first University of British Columbia green building). A $100,000 new sewer line was originally required but waterless urinals and composting toilets with rainwater capture have eliminated this need. Operable twist and turn windows employed with natural ventilation using roof vents with fan assist eliminate mechanical cooling. Trickle air leakage under window ventilation is also employed.

6.4.6. CONCLUSIONS OF BUILDING PROJECTS

Buildings that use dramatically less energy and display improved indoor environmental conditions are needed to achieve sustainable buildings as the standard. This is especially so when post-Kyoto era objectives are decided.

Energy and cost reference levels are available in every country. How much better performing can buildings be? At what price? It has been shown that a level of 65% less energy is attainable with existing technologies and at reasonable cost. Natural energy, especially underground thermal energy storage or earth energy, can play a useful role in achieving these reductions. Thermal energy storage becomes a requirement if reductions beyond 65% are to be made at reasonable cost. Design simulations have been confirmed in practice for both new and retrofit buildings. New design concepts and control concepts need to be developed for the ultra-low buildings of the future.

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7. HEAT STORAGE BY PHASE CHANGING MATERIALS AND THERMOECONOMICS

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Abstract. Heat storage systems by phase changing materials (PCM) need to identify the performance limits and optimize processes and cycles with thermodynamic analysis. Such an analysis consists of concepts like thermoeconomics, entropy generation minimization, and the extended exergy accounting, which calculates resource-based value of a commodity by establishing a comprehensive relation between exergy and economic values. Some other concepts are the cumulative exergy consumption method and exergy cost theory. Thermoeconomics connects mainly the second law of thermodynamics and economics to estimate the cost of exergy by using the cost 0accounting and structural theory. This study presents a thermodynamic analysis of latent heat storage by PCM. It also presents a case study for a seasonal solar heat storage system by paraffin wax by using a system of 18 solar air heaters with 27 m^2 of total absorber area in order to heat a 180 m^2 greenhouse.

Keywords: Latent heat storage; exergy analysis, thermoeconomics, economic analysis

7.1. Introduction

Effective solar energy utilization needs to store the heat in order to decrease the mismatches between the demand and availability. For storing and recovering heat, PCM can serve a suitable medium during melting and solidification. So that, the operation temperature depends on the composition of phase changing materials and the level of subcooling and sensitive heating. When a composite or a mixture of PCM is used the melting point may be multiple or an interval (Adebiyi et al., 1992; Demirel and Paksoy, 1993). As solar energy storage system is primarily a waste, clean energy management, it is environmentally friendly. When it is designed and operated by considering both the first and second laws of thermodynamics, engineers take into account the quality and the environmental aspects of energy usage (Krane, 1987). This practice is essential for efficient, environmentally friendly energy policies,

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 133–151.

and hence sustainable development even within the low temperature working range.

Combined principles of thermodynamics are widely utilized in assessing the performances of heat storage systems. Thermoeconomics further combines the thermodynamic principles with engineering economics to estimate the cost of exergy, and optimize the cost under various constraints. Although, Valero et al. (1989) tried to unify the thermoeconomic theories, the concepts and procedures may vary, and create ambiguity in practical applications (Szargut, 1990; Tsataronis, 1993; Erlach et al., 1999; Sciubba, 2003).

Still, scientists have shown growing interests in thermoeconomics and exergoeconomics within the last 20 years. This study reviews and summarizes the exergy analysis and thermoeconomics, and presents a case study for thermoeconomic analysis of a large scale, seasonal solar energy storage with paraffin wax as PCM for greenhouse heating (Demirel et al., 1993; Bascetincelik et al., 1999; Ozturk, 2005).

7.2. Latent Heat Storage (LHS)

Latent heat storage is a popular research area with industrial and domestic applications, such as energy recovery of air conditioning (Go et al., 2004), and under-floor electric heating by using PCM (Lina et al., 2005). Many studies have compared sensible heat storage and LHS systems; some suggest higher second law efficiencies for LHS systems (Adebiyi and Russell, 1987; Adebiyi, 1991; Rosen and Dincer, 2003), and consider that optimum phase-change temperature is the geometric average of the energy source and environment temperatures (Bjurstrom and Carlsson, 1985). However, as Van Den Branden et al. (1999) report that it may not be proper to conclude that one heat storage method is superior to another. Mainly operating conditions and the configuration of PCM identify a suitable heat storage method; also one has to consider the two-phase heat transfer phenomena within a PCM (De Lucia and Bejan, 1991; Cao and Faghri, 1991; Adebiyi, 1991). Some researchers suggest that multiple phase-change materials instead of single PCM may be more beneficial (Lim et al., 1992; Adebiyi et al., 1992). For example, Hidaka and Yamazaki (2004) used erythritol–polyalcohol mixtures with two melting points between 80 and 100 °C and a latent heat of 246 kJ/kg, while Tuncbilek et al. (2005) used 69.0 wt.% Lauric and 31 wt.% palmitic acids eutectic mixtures as PCM with a melting temperature of 35.2 °C and latent heat of 166.3 J/g.

Fluid flow temperature and the melting points of PCM may affect the contribution of sensible heat to the total heat stored (Farid and Kanzawa, 1989; Farid et al., 1990; Aceves-Saborio et al., 1994; Saman et al., 2005). Beside that, the effects of liquid superheating and solid subcooling should also be considered (De Lucia and Bejan, 1991; Ramayya and Ramesh, 1998; Saman et al., 2005). One of the main problems is how to assess the performance of a LHS system. Such an assessment may be performed with or without sensible heat effects before and after the phase change takes place. Ramayya and Ramesh (1998) suggest that overall second-law efficiency is higher for LHS with sensible heating and subcooling compared with that of the latent heat only. Saman et al. (2005) reported a numerical analysis with the sensible heat effects for LHS unit. Under mechanical equilibrium, exergy destruction in LHS is smaller compared with that of sensible heat storage due to finite differences between fluid temperature and heat storage unit (Rosen and Dincer, 2003). Improving the thermal conductivity of PCM may also reduce the phase changing time and increase the overall efficiency of charging and discharging operations (Shiina and Inagaki, 2005; Liu et al., 2005). When metal-PCM pairs are used in LHS, one has to consider the corrosion problems (Cabeza et al., 2005).

7.3. Thermodynamic Analysis

Thermodynamic analysis (TA) identifies the sources of exergy losses due to irreversibilities in each process in a system. This will not guarantee that economical process modifications would be generated. For that, relations between the energy efficiency and capital cost must be evaluated. Sometimes, improved energy efficiency will require more investment. TA is of considerable value where an efficient energy conversion is important. Optimization seeks the best solution under specific constraints, which usually determines the complexity of the problem. In every nonequilibrium system, an entropy effect leading to energy dissipation either within or through the boundary of system exists.

Currently, thermodynamic analysis has realized three main stages: (i) Second law analysis (Bejan, 1978, 1988; Krane, 1987; Adebiyi and Russell, 1987; Adebiyi, 1991) is mainly used in thermal engineering by combining the principles of thermodynamics with heat transfer and fluid mechanics to reduce entropy production. (ii) Later the exergy analysis combined the principles of thermodynamic with heat and mass transfer, fluid mechanics, chemical kinetics, and widely used in design and optimization in physical, chemical, and biological systems (Charach and Zemel, 1991; Duryamaz et al., 1998; Ramayya and Ramesh, 1998; Rosen and Dincer, 2003; Demirel, 2002). (iii) Lastly, exergy analysis is combined with economic analysis, and called as thermoeconomics or exergoeconomics (Hua et al., 1989, 1997; Tsataronis, 1993; Gonzales et al., 2003; Valero et al., 1994; Ayres, 1998; Valero et al., 2004a, 2004b; Demirel, 2002).

7.4. Exergy Analysis

Accessible work potential is called the exergy that is the maximum amount of work that may be performed theoretically by bringing a resource into equilibrium with its surrounding through a reversible process. Exergy analysis is essentially a TA, and utilizes the combined laws of thermodynamics to account the loss of available energy. Exergy is always destroyed by irreversibilities in a system, and expressed by

$$X = H - T_o S \tag{1}$$

where H is the enthalpy, T_o is the reference (dead state temperature) temperature, and S is the entropy. For an incompressible fluid initially at temperature T_i with constant heat capacity, the exergy is a simple function of temperatures when the pressure change is negligible

$$X = \dot{m} C_p \left[(T_i - T_o) - T_o \ln \left(\frac{T_i}{T_o} \right) \right] \tag{2}$$

where T_o is the dead state (environment) temperature, and C_p is the specific heat.

The exergy balance and the lost work is given by

$$\sum_{\substack{\text{into} \\ \text{system}}} \left[\dot{m} X + \dot{Q} \left(1 - \frac{T_o}{T_s} \right) + \dot{W}_s \right] - \sum_{\substack{\text{out of} \\ \text{system}}} \left[\dot{m} X + \dot{Q} \left(1 - \frac{T_o}{T_s} \right) + \dot{W}_s \right] = \dot{W}_{lost} \tag{3}$$

where W is the work, and superimposed dot shows the change of variable in time. The terms in square parenthesis show the exergy accompanying mass, heat, and work, respectively. W_{lost} represents the destruction of exergy. If a system undergoes a spontaneous change to the dead state without a device to perform work, then exergy is completely destroyed. Therefore, exergy is a function of both the physical properties of a resource and its environment.

Figure 24 shows the charging and discharging operations with appropriate valves, and temperature profiles for countercurrent LHS with subcooling and sensible heating. An optimum LHS system performs exergy storage and recovery operations by destroying as little as possible the supplied exergy (Krane, 1987; Domanski and Fellah, 1998). The first and second law efficiencies of LHS operations are summarized below.

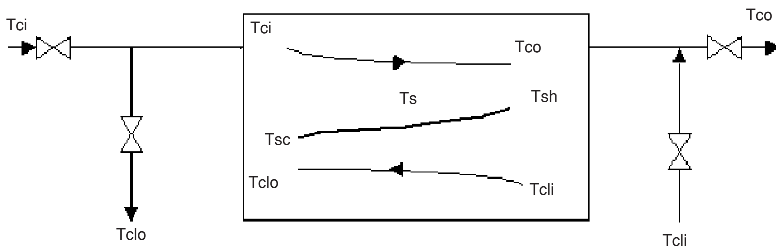

Figure 24. Typical temperature profiles of a LHS system for charging (c) and discharging (d) operations

7.4.1. CHARGING

A charging fluid heats PCM, which may initially be at a subcooled temperature T_{sc}, and may eventually reach to a temperature T_{sh} after sensible heating. Therefore, LHS undergoes a temperature difference of $T_{sh} - T_{sc}$, as shown in Figure 1. Heat available for storage would be

$$Q_c = UA(\Delta T_{lm})_c = \dot{m}_c C_{pc}(T_{ci} - T_{co}) \tag{4}$$

where U is the overall heat transfer coefficient, A is the heat transfer area, $\dot{m}_c$ is the charging fluid flow rate, and ΔT_{lm} is the logarithmic mean temperature difference expressed by

$$(\Delta T_{lm})_c = \frac{(T_{ci} - T_{sc}) - (T_{co} - T_{sh})}{\ln\left(\dfrac{T_{ci} - T_{sc}}{T_{co} - T_{sh}}\right)} = \frac{T_{ci} - T_{co}}{\mathrm{NTU}_c} \tag{5}$$

where $NTU = UA/(\dot{m}_c C_{pc}) = (T_{ci} - T_{co})/\Delta T_{lm}$ is the number of transfer units. Equation (5) relates the value of NTU with temperature. Heat lost by the charging fluid will be gained by the PCM

$$Q_c = Q_s = m_s[C_{ps}(T_l - T_{si}) + \Delta H_m + C_{p\mathrm{l}}(T_{sh} - T_h)] \tag{6}$$

where ΔH_m is the heat of melting, T_l and T_h are the lowest and highest melting points of phase change, and C_{ps} and C_{pl} denote the specific heats of solid and liquid states of PCM, respectively.

The net exergy X change of the charging fluid would be

$$\Delta X_c = (X_{co} - X_{ci}) = \dot{m}_c C_{\mathrm{pc}}\left[(T_{ci} - T_{co}) - T_o \ln\left(\frac{T_{ci}}{T_{co}}\right)\right]. \tag{7}$$

Exergy stored by the PCM is

$$X_s = Q_s\left(1 - \frac{T_o}{T_s}\right) \tag{8}$$

where T_s is an average temperature of storage, which may be approximated by $(T_{sc} + T_{sh})/2$.

The first and second law efficiencies are

$$\eta_I = \frac{\text{actual heat stored}}{\text{maximum energy gain}} = \frac{T_{ci} - T_{co}}{T_{ci} - T_s} \tag{9}$$

$$\eta_{II} = \frac{\text{exergy of PCM}}{\text{exergy of charge fluid}} = \frac{(T_{ci} - T_{co})\left(1 - \dfrac{T_o}{T_{sh}}\right)}{\left[(T_{ci} - T_{co}) - T_o \ln\left(\dfrac{T_{ci}}{T_{co}}\right)\right]}. \tag{10}$$

7.4.2. DISCHARGING

It is assumed that the PCM is totally melted and heated to a temperature T_{sh} when discharging fluid starts recovering heat estimated by

$$Q_d = UA\Delta(T_{lm})_d = \dot{m}_d C_{\mathrm{pd}}(T_{di} - T_{do}). \tag{11}$$

The heat gained by the discharging fluid will be lost by the PCM. Net exergy change of the charging fluid would be

$$\Delta X_d = (X_{di} - X_{do}) = m_d C_{\mathrm{pd}}\left[(T_{di} - T_{do}) - T_o \ln\left(\frac{T_{di}}{T_{do}}\right)\right]. \tag{12}$$

First and second law efficiencies are

$$\eta_I = \frac{T_{do} - T_{di}}{T_{di} - T_{sl}} \tag{13}$$

$$\eta_I = \frac{\text{exergy given to discharge fluid}}{\text{exergy of PCM}} = \frac{T_{do} - T_{di} - T_o \ln\left(\dfrac{T_{do}}{T_{di}}\right)}{(T_{do} - T_{di})\left(1 - \dfrac{T_o}{T_{sl}}\right)}. \tag{14}$$

Overall efficiencies for a LHS system become

$$\eta_{Io} = \eta_{Ic}\eta_{Id} \tag{15}$$

$$\eta_{IIo} = \eta_{IIc}\eta_{IId}. \tag{16}$$

All the temperatures are time dependent, and the charging and discharging cycles need to be monitored over the time of operation.

7.4.3. MODELING EQUATIONS

For charging mode, we have

$$\rho C_{\mathrm{pc}}\frac{\partial T}{\partial t} + V_o \rho C_{\mathrm{pc}} \nabla T = \nabla(k\nabla T) - \Delta H_m \frac{\mathrm{d}\beta}{\mathrm{d}t} \tag{17}$$

where β is the fraction of melted PCM, V_o is the average velocity, and k is the thermal conductivity of charging fluid. For one-dimensional heat transfer with axial conduction only, we have

$$\frac{\partial T}{\partial t} + V_o \frac{\partial T}{\partial x} = \alpha \frac{\partial^2 T}{\partial x^2} - \frac{\Delta H_m}{\rho C_{pc}} \frac{d\beta}{dt} \tag{18}$$

where, α is the thermal diffusivity. Initial and boundary conditions are

$$t = 0 \qquad T = T_i \tag{19}$$

$$x = 0 \qquad V_o T_o = -\alpha \frac{\partial T}{\partial x} + V_o T \tag{20}$$

$$x = L \qquad \frac{\partial T}{\partial x} = 0 \tag{21}$$

where L is the length of LHS. At steady state one-dimensional heat transfer for charging fluid is

$$\alpha \frac{\partial^2 T}{\partial x^2} - V_o \frac{\partial T}{\partial x} - \frac{\Delta H_m}{\rho C_{pc}} \frac{d\beta}{dt} = 0. \tag{22}$$

Equations (18) and (22) can be solved using finite difference technique (Cao and Faghri, 1991; Van Den Branden et al., 1999).

7.5. Thermoeconomics

Thermoeconomics combines exergy analysis with economic analysis, and calculates the efficiencies based on exergy; it assigns costs to exergy related variables by using the “exergy cost theory” (Szargut, 1990; Ayres, 1998) and “exergy cost balances”. Thermoeconomics can unify all balances that are mass, energy, exergy, and cost by a single formalism. “Extended exergy accounting” considers non-energetic costs, such as financial, labor and environmental remediation costs as functions of the technical and thermodynamic parameters of systems (Sciubba, 2003). There are two main groups of thermoeconomic methods: (a) cost accounting methods, such as exergy cost theory for a rational price assessment, and (b) optimization by minimizing the overall cost, under a proper set of financial, environmental and technical constraints to identify the optimum design and operating conditions (Erlach et al., 1999).

Structural theory facilitates the evaluation of exergy cost and incorporation of thermoeconomics functional analysis (Erlach et al., 1999). It is a common formulation for the various thermoeconomic methods providing the costing equations from a set of modeling equations for the components of a system. The structural theory needs a productive structure displaying how the resource consumptions are distributed among the components of a system. The flows entering a component in the productive structure are considered as fuels F and flows leaving a component are products P. The components are subsystems with control volumes as well as mixers and splitters. Therefore, the productive structure is a graphical representation of the fuel and product distribution (Erlach et al., 1999; Valero et al., 2004a, 2004b; Duryamaz et al., 2004). All the flows are extensive properties, such as exergy. For any component j, or a subsystem, the unit exergy consumption x_c is expressed on a fuel/product basis by

$$x_{cj} = F_j/P_j = F_j/X_j. \tag{23}$$

For linear modeling, the average costs of fuels and products are defined by (Erlach et al., 1999)

$$C^*_{jf} = \frac{\partial F_o}{\partial F_j}; \qquad C^*_{jP} = \frac{\partial F_o}{\partial P_j} \tag{24}$$

where F_o is the fuel to the overall system expressed as a function of the flow F_j or product P_j, respectively, and the other related parameters. Total annual production cost C_T in \$ is

$$C_T = \sum_{j=1}^{N} c_j X_j = \sum_{j=1}^{N} C_{jf} \tag{25}$$

where c_i is the specific cost of product in \$kW^{-1}, C_{fi} is the cost of fuel, and Xi is the rate of exergy as product of component j in kW and is expressed in terms of NTU using Equation (5)

$$X_j = \dot{m}_j C_p \left[NTU_j \Delta T_{\text{lm}j} - T_o \ln\left(\frac{T_{j1}}{T_o}\right)\right]. \tag{26}$$

Some optimization techniques minimize the cost of product of a system or a component. The optimum total production cost rate with respect to NTU is obtained from

$$\frac{\mathrm{d}C_T}{\mathrm{dNTU}} = 0. \tag{27}$$

Valero et al. (2004a, 2004b) extended the thermoeconomic approach to identify and locate anomalies causing reductions in efficiency and economic worth of energy utility systems. They suggested an ecovector to include the exergoeconomic costs or environmental impacts. Ecovector is a set of environmental burdens of an operation, and can be associated with input flows (Gonzales et al., 2003); it includes information about natural resources, exergy of these resources, and monetary costs. The external environmental costs associated with the environmental burdens may also be added into the ecovectors. The sum of internal and external costs results in the total social cost of energy and exergy flows (Gonzales et al., 2003).

Sciubba (2003) proposed an approach called the extended exergy accounting (EEA), which calculates the real, resource-based value of a commodity product. The time span of EEA is the whole life of a plant. The EEA includes the exergetics flow sheets for non-energetic costs of labor and environmental remediation expenditures, and hence uses "extended exergetic content." It also defines the criterion for an optimum process or operation (Sciubba, 2003).

Thermoeconomics of LHS systems involve the use of principles from thermodynamics and fluid mechanics and heat transfer. Therefore, thermoeconomics may be applied to both the use of those principles and materials, construction, and mechanical design, and a part of conventional economic analysis. The distinguished side of it comes from the ability to account the quality of energy and environmental impact of energy usage in economic considerations.

7.6. Case Study: Seasonal Solar Energy Storege by Paraffin as PCM

Turkey has extensive solar energy potential; the solar radiation period is 2624 h/year with an average intensity of 3.67 kWh/m^2 per day (Kaygusuz and Sari, 2003). Heat storage is an essential part of solar energy management. The case study involves a large-scale of solar LHS system using paraffin as PCM. Figure 25 shows the three basic components that are solar air heaters, LHS, and greenhouse:

1. *System of packed bed solar air heaters:* The system has a total solar heat collector area of 27 m^2 consisting of 18 packed bed solar air heaters (Demirel and Kunc, 1987). Each unit has 1.5 m^2 absorber area with a length of 1.9 m and width of 0.9 m. The Raschig ring type of packing made of polyvinyl chloride with the characteristic diameter of 0.05 m is used within the airflow

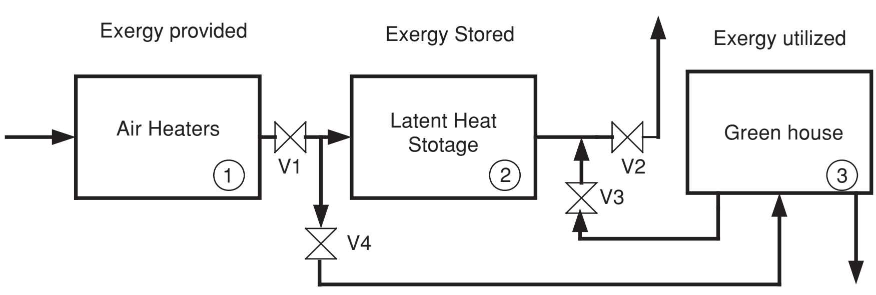

Figure 25. Productive structure with three components of the LHS system representing exergy transformation

passage. The packing enhances the wall-to-fluid heat transfer by increasing the radial and axial mixing as well as reducing the wall resistance (Demirel, 1989; Ozturk and Demirel, 2004). The volumetric flow rate of airflow is 600 m^3/h.

2. *Latent heat storage unit:* A horizontal steel tank, 1.7 m long and 5.2 m wide, contains 6,000 kg of technical grade of paraffin wax as PCM. Paraffin wax consists mainly straight-chain hydrocarbons and very little amount of branching. The *n*-alkane content exceeds 75%. Commercial waxes may have a range of about 8–15 carbon-number. Volume contraction is less than 12% during freezing. The tank is insulated with 0.05 m of glass wool. Inside the tank, there are two spiral coils made of perforated polyethylene pipes with a total length of 97 m and diameter of 0.1 m embedded into the PCM. The coils carry the warm airflow pumped from solar air heaters (Demirel et al., 1993; Bascetincelik et al., 1998; Ozturk, 2005). Differential scanning calorimeter measurements shows that the paraffin has a melting temperature range of 48–60 °C and approximately 190 kJ/kg of latent heat of melting. Paraffin wax freeze without super-cooling, and melt without segregation of components.
3. *Greenhouse:* The greenhouse with an area of 180 m^2 is covered by 0.35 mm thick polyethylene, and is aligned north to south. The floor area is 12 m × 15 m, and the height is 3 m. The LHS tank carries 33.3 kg of paraffin wax per square meter of the greenhouse ground surface area. Heat storage unit connects the solar air heater system to the greenhouse with appropriate fans, valves and piping (Ozturk, 2005). Whenever the temperature in the greenhouse drops below a set point, a fan circulates the air from greenhouse through the LHS unit until the temperature reaches the required level.

7.6.1. COST ANALYSIS

Costs are the amount of resources consumed to produce a flow or a product. When exergy added into a flow, the cost of flow leaving a component is equal to the cost of flow entering plus the fuel value of added exergy. When, on the other hand, exergy is removed, the fuel value of exergy is subtracted. The products and their average costs in the productive structure shown in Figure 2 are summarized below:

Component 1: Solar air heater system

Added exergy provided by the solar air heater system can be expressed in terms of NTU and ΔT_{lm} using Equation (26) as

$$\Delta X_1 = (X_{1p} - X_{1i}) = \dot{m}_1 C_p \left[\mathrm{NTU}_1 \Delta T_{lm1} - T_o \ln\left(\frac{T_{1\mathrm{o}}}{T_{1\mathrm{i}}}\right)\right]. \tag{28}$$

The airflow leaving the solar air heaters adds exergy, therefore the cost of it is

$$C_{1p} = C_{1i} + C_{1f} \tag{29}$$

where C_{1i} is the cost of the flow entering component 1, and C_{1f} is the fuel value of added exergy. Specific costs of warm air c_a and exergy c_{x1} are

$$c_a = \frac{C_{1p}}{X_{1p}} = \frac{C_{1p\mathrm{f}}}{X_{1p}}, \qquad c_{x1} = \frac{C_{1f}}{\Delta X_1} \tag{30}$$

where $C_{1p\mathrm{f}}$ is the fuel value of the product leaving solar air heating system.

Component 2: Latent heat storage system
Removed exergy by the LHS system during charging is

$$\Delta X_c = (X_{\mathrm{c}p} - X_{ci}) = \dot{m}_2 C_p \left[\mathrm{NTU}_c \Delta T_{\mathrm{lm}c} - T_o \ln\left(\frac{T_{co}}{T_{ci}}\right)\right]. \tag{31}$$

Cost of the product after charging is

$$C_{\mathrm{c}p} = C_{ci} - C_{cf}. \tag{32}$$

Specific costs of the product leaving LHS unit c_c and the removed exergy c_{xc} are

$$c_c = \frac{C_{\mathrm{c}p}}{X_{\mathrm{c}p}} = \frac{C_{\mathrm{c}p\mathrm{f}}}{X_{\mathrm{c}p}}, \qquad c_{xc} = \frac{C_{cf}}{\Delta X_c}. \tag{33}$$

Discharging flow adds exergy form the LHS, and is given by

$$\Delta X_d = (X_{\mathrm{d}p} - X_{di}) = \dot{m}_3 C_p \left[\mathrm{NTU}_d\, \Delta T_{lmd} - T_o \ln\left(\frac{T_{do}}{T_{di}}\right)\right]. \tag{34}$$

Cost of discharging flow is

$$C_{dp} = C_{di} + C_{df}. \tag{35}$$

Specific costs of discharging flow leaving LHS unit c_d and the added exergy c_{xd} are

$$c_d = \frac{C_{dp}}{X_{dp}} = \frac{C_{\mathrm{d}p\mathrm{f}}}{X_{dp}} \qquad c_{xd} = \frac{C_{df}}{\Delta X_d} \tag{36}$$

Component 3: Greenhouse
Exergy change within the greenhouse is

$$\Delta X_3 = (X_{3p} - X_{3i}) = \dot{m}_3 C_p \left[\mathrm{NTU}_3\, \Delta T_{lm3} - T_o \ln\left(\frac{T_{3o}}{T_{3i}}\right)\right]. \tag{37}$$

Exergy from the discharge flow is removed in the greenhouse, and the cost of flow leaving the greenhouse becomes

$$C_{3p} = C_{3i} - C_{3f}. \tag{38}$$

Specific costs of flow leaving greenhouse c_g and the exergy removed c_{xg} are

$$c_g = \frac{C_{3p}}{X_{3p}} = \frac{C_{3p\mathrm{f}}}{X_{3p}}, \qquad c_{xg} = \frac{C_{3f}}{\Delta X_3}. \tag{39}$$

Total cost of products of the three components would be

$$C_{p\mathrm{T}} = C_{1p} + C_{\mathrm{c}p} + C_{\mathrm{d}p} + C_{3p} = c_a X_{1p} + c_c X_{\mathrm{c}p} + c_d X_{\mathrm{d}p} + c_g X_{3p}. \tag{40}$$

Cost of a product for component j is based on a fuel/product basis $C_{jp} = C_{jp\mathrm{f}}$, so that the total cost of products is (Erlach et al., 1999)

$$C_{p\mathrm{T}} = C_{1p} + C_{\mathrm{c}p} + C_{\mathrm{d}p} + C_{3p} = C_{1p\mathrm{f}} + C_{\mathrm{c}p\mathrm{f}} + C_{\mathrm{d}p\mathrm{f}} + C_{3p\mathrm{f}}. \tag{41}$$

Equation (41) may be used in Equation (27) to find an optimum value of NTU to minimize the total cost of production. Cost optimization basically depends on the tradeoffs between the cost of energy (fuel) and capital investment as seen in Figure 26. Working with compatible design and operating conditions, and new technologies, it is possible to recover more and more exergy in energy conversion systems. Implementing pollution charges and incentives for environmentally friendly technologies may reduce the cost of exergy loss (Frangopoulos and Caralis, 1997).

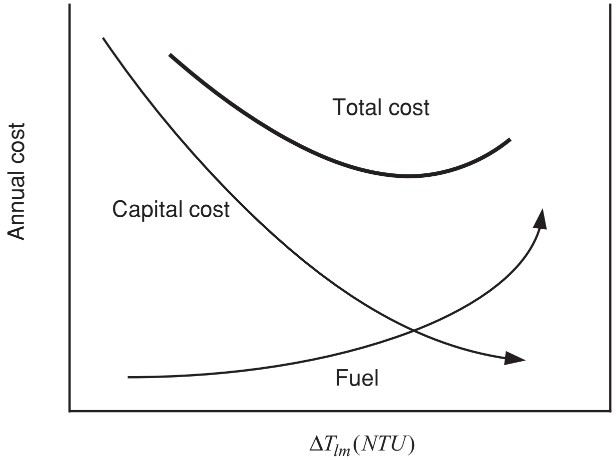

Figure 26. Annual cost optimization

Thermoeconomics of the LHS system involves fixed capital investment, operational and maintenance cost, and exergy costs (Domanski and Fellah, 1998). Total fixed capital investment consists of (i) direct expenses that are equipment cost, materials, and labor, (ii) indirect project expenses that are freight, insurance, taxes, construction, overhead, (iii) contingency and contractor fee, and (iv) auxiliary facilities, such as site development, auxiliary buildings.

7.6.2. A TYPICAL SEASONAL SOLAR HEAT STORAGE SYSTEM

This analysis considers the three basic components of a seasonal LHS system (Figure 25) constructed after one year. Table 7 shows the data used in thermoeconomic analysis.

Economic analysis can determine the discounted profitability criteria in terms of payback period (PBP), net present value (NPV), and rate of return (ROR) from discounted cash flow diagram, in which each of the annual cash flow is discounted to time zero for the LHS system. PBP is the time required, after the construction, to recover the fixed capital investment. NPV shows the cumulative discounted cash value at the end of useful life. Positive values of NPV and shorter PBP are preferred. ROR is the interest rate at which all the cash flows must be discounted to obtain zero NPV. If ROR is greater than the internal discount rate, then the LHS system is considered feasible (Turton et al., 2003).

Figure 27 shows the discounted cash flow diagram obtained from Table 8 using the data in Table 7. A net present value of \$102,462.21 is obtained at end of 15 years of useful life operation, which shows a profitable investment. Approximate discounted payback period is about eight years. Discounted

TABLE 7. A typical data used for thermoeconomic analysis of seasonal heat storage system

Fixed capital investments for the components:
$\mathrm{FCI}_1 + \mathrm{FCI}_2 + \mathrm{FCI}_3 = \$200{,}000 + \$200{,}000 + \$100{,}000 = \$500{,}000$
Cost of land: $L = \$50{,}000$
Working capital: $\mathrm{WC} = 0.2\ (\$500{,}000) = \$100{,}000$
Yearly revenues or savings: $R = \$160{,}000$
Total cost of production (COP) from equations (40) and (43):
$\mathrm{COP} = C_{P\mathrm{T}} = C_{1p} + C_{\mathrm{c}p} + C_{\mathrm{d}p} + C_{3p} = C_{1p\mathrm{f}} + C_{\mathrm{c}p\mathrm{f}} + C_{\mathrm{d}p\mathrm{f}} + C_{3p\mathrm{f}} = \$55{,}000$
$C_{1p\mathrm{f}} = \$20{,}000$, $C_{\mathrm{c}p\mathrm{f}} = \$15{,}000$, $C_{\mathrm{d}p\mathrm{f}} = \$10{,}000$, $C_{3p\mathrm{f}} = \$10{,}000$
Taxation rate: $t = 35\%$
Salvage value of the whole seasonal storage: $S = \$50{,}000$
Useful life of the system: $n = 15$ years; depreciation over 10 years
Discount rate $i = 8\%$

TABLE 8. Discounted cash flow estimations for a seasons LHS system

n	FCI	D[a]	A[b]	R	COP	B[c]	DCF	CCF
0	−50,000	0	500,000	0		−500,000	−50,000	−50,000
1	−500,000	0	500,000	0		−600,000	−555,555.6	−605,555.56
2	0	50,000	450,000	160,000	55,000	85,750	73,516.8	−532,038.75
3	0	50,000	400,000	160,000	55,000	85,750	68,071.11	−463,967.64
4	0	50,000	350,000	160,000	55,000	85,750	63,028.81	−400,938.83
5	0	50,000	300,000	160,000	55,000	85,750	58,360.01	−342,578.82
6	0	50,000	250,000	160,000	55,000	85,750	54,037.05	−288,541.77
7	0	50,000	200,000	160,000	55,000	85,750	50,034.3	−238,507.47
8	0	50,000	150,000	160,000	55,000	85,750	46,328.06	−192,179.41
9	0	50,000	100,000	160,000	55,000	85,750	42,896.35	−149,283.07
10	0	50,000	50,000	160,000	55,000	85,750	39,718.84	−109,564.22
11	0	50,000	50,000	160,000	55,000	85,750	36,776.71	−72,787.51
12	0	0	50,000	160,000	55,000	68,250	27,103.01	−45,684.50
13	0	0	50,000	160,000	55,000	68,250	25,095.38	−20,589.12
14	0	0	50,000	160,000	55,000	68,250	23,236.47	2,647.34
15	0	0	50,000	160,000	55,000	68,250	21,515.25	24,162.59
16	200,000	0	50,000	160,000	55,000	268,250	78,299.62	102,462.21

[a] Depreciation: straight line method: $D = (\text{FCI} - \text{S})/n$.
[b] Book value: $A = \text{FCI} - \Sigma(D_k)$.
[c] After tax cash flow: net profit + depreciation: $B = (R - \text{COP} - D_k)(1 - t) + D_k$.

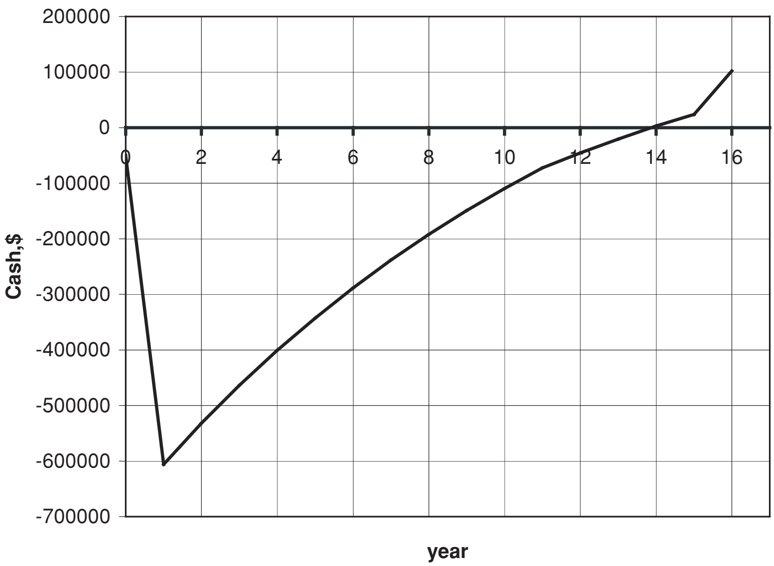

Figure 27. Discounted cash flow diagram for the productive structure of LHS system with three components

rate of return is around 10.485%, which is greater than the internal interest rate of 8%. By changing the values exergy costs in Equation (40), cash flow diagram can be modified easily. Similar cash flow diagrams can be produced for individual components.

7.7. Conclusions

Thermoeconomics combines the exergy analysis with economic analysis to obtain optimum design and operating conditions. Exergy analysis takes into account the quality of energy and impact of energy usage on the environment. An average cost balance based on fuel/product basis and structural theory might be helpful to estimate the cost of exergy flows for various components of a latent heat storage system. With a standardized exergy cost estimates and economic analysis, thermoeconomics could be a widely used tool in exergy management for a sustainable development.

Notation

A	area, m^{-2}
c	specific cost, \$ kW^{-1}
C	cost, \$
$C_{j\mathrm{pf}}$	fuel cost of product of component j \$
C_p	specific heat, kJ kg^{-1} K^{-1}
$C_{p\mathrm{s}}$	specific heat of solid PCM, kJ kg^{-1} K^{-1}
$C_{p\mathrm{l}}$	specific heat of liquid PCM, kJ kg^{-1} K^{-1}
d	depreciation
F_o	external resource, kJ
h	heat transfer coefficient, kJ m^{-2} K^{-1} s^{-1}
H	enthalpy, kJ kg^{-1}
i	rate of interest
k	thermal conductivity, kJ m^{-1} K^{-1} s^{-1}
L	cost of land, \$
m	mass flow rate kg s^{-1}
n	useful life, year
N	total number of components
NTU	number of transfer units
Q	heat flow, kJ s^{-1}
R	revenue, \$ $year^{-1}$
S	entropy, kJ K^{-1}
t	time, s; rate of taxation
T	temperature, K

T_l lowest melting point, K
T_h highest melting point, K
U over all heat transfer coefficient, kJ m^{-2} K^{-1} s^{-1}
V velocity, m s^{-1}
W work, kJ s^{-1}
x axial coordinate, m
X exergy, kJ s^{-1}

Greek Letters

α thermal diffusivity, m^{-2} s^{-1}
β fraction of melting efficiency
ρ density, kg m^{-3}

Subscripts

c charging
cc specific cost of charge
cd specific cost of discharge
ci charge in
co charge out
d discharging
di discharge in
do discharge out
f fuel
I, II first and second laws, respectively
k years for depreciation
lm log mean
m melting
o standard, reference (dead state conditions), out
s storage, shaft work
sc subcooling
sh sensible heating
T total

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PART IV. UNDERGROUND THERMAL ENERGY STORAGE

8. AQUIFER THERMAL ENERGY STORAGE (ATES)

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Abstract. Storage of renewable energy in the underground will reduce the usage of fossil fuels and electricity. Hence, these systems will benefit to CO_2 reduction as well as the reduction of other environmentally harmful gas emissions, like SO_X and NO_X. ATES, BTES and CTES are three options of Underground Thermal Energy Storage (UTES) systems. ATES and BTES are widely used in some countries. Relevant properties and different aspects of design and construction of ATES systems is discussed in this article.

Keywords: Underground Thermal Energy Storage, Aquifer, Borehole

8.1. Introduction to Underground Thermal Energy Storage (UTES)

There are several concepts as to how the underground can be used for underground thermal energy storage (UTES) depending on geological, hydrogeological and other site conditions. In Figure 28 some different options are schematically illustrated.

The two most promising options are storage in aquifers (ATES) and storage through borehole heat exchangers (BTES). These concepts have already been introduced as commercial systems on the energy market in several countries. Another option is to use underground cavities (CTES), but this concept is so far rarely applied commercially.

In ATES (Aquifer Thermal Energy Storage) systems groundwater is used to carry the thermal energy into and out of an aquifer. For the connection to the aquifer water wells are used. However, these wells are normally designed with double functions, both as production and infiltration wells, see Figure 29.

The energy is partly stored in the ground water itself but partly also in the grains (or rocks mass) that form the aquifer. The latter storage process takes place when the ground water is passing the grains and will result in the development of a thermal front with different temperatures. This front will move in a radial direction from the well during charging of the store and then turn back while discharging.

There are several hundreds of these systems in operation, with the Netherlands and Sweden as dominating countries of implementation. Practically all

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 155–176.

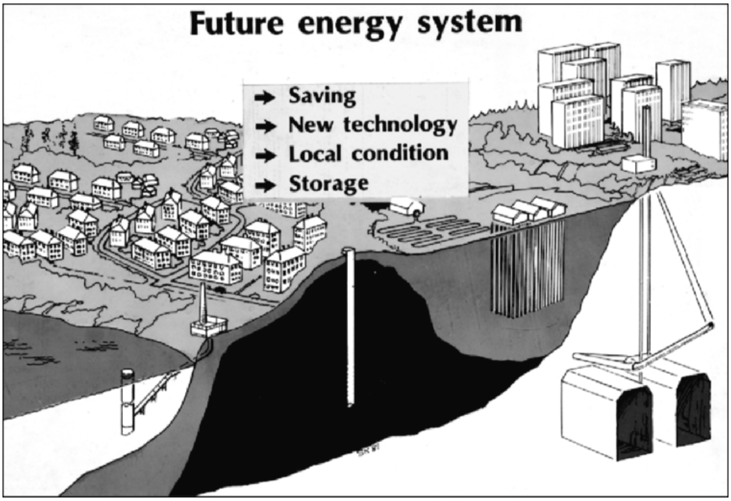

Figure 28. Different options using the underground for storage of thermal energy

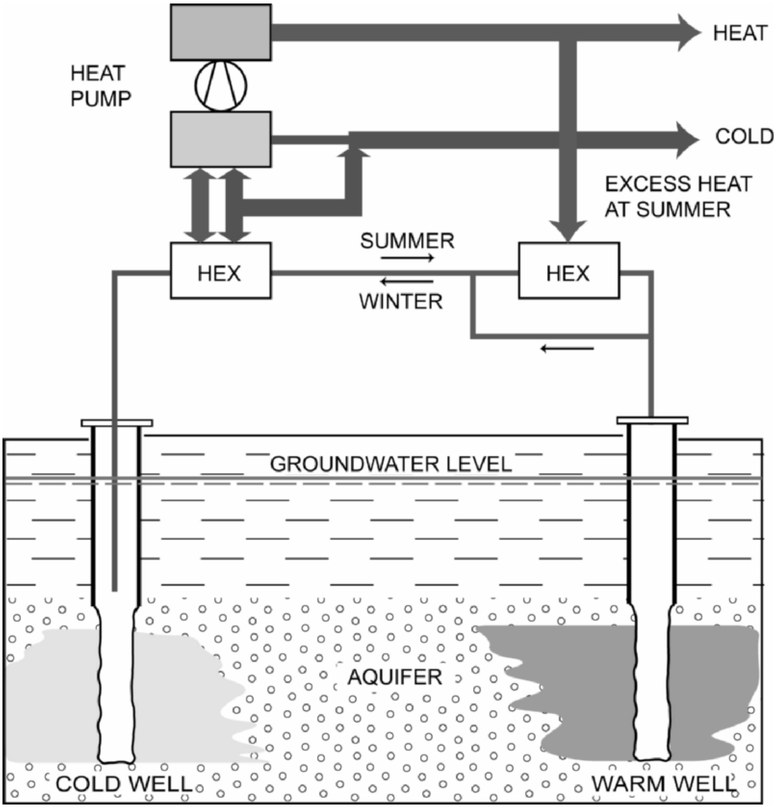

Figure 29. Principal ATES configuration

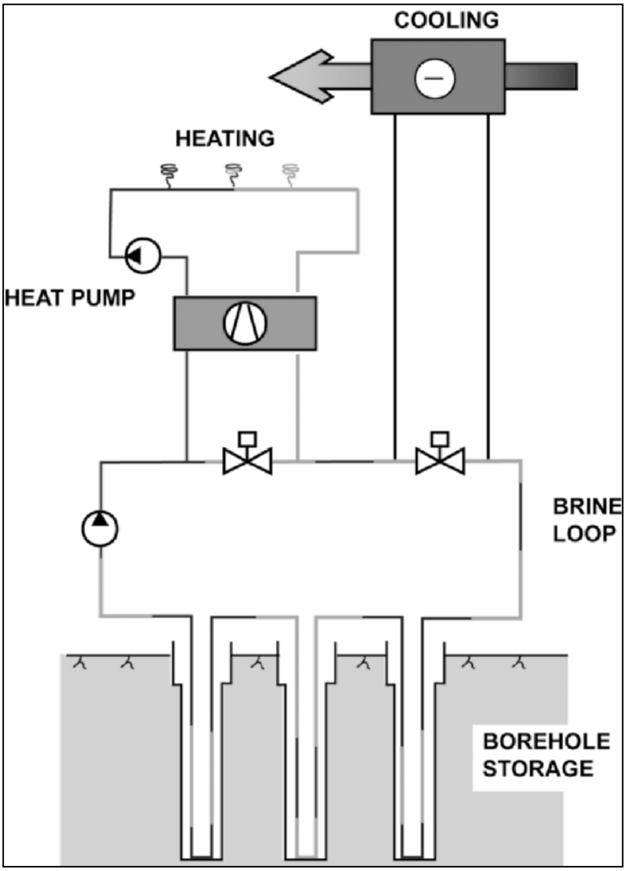

Figure 30. Principal BTES configuration

systems are designed for low temperature applications where both heat and cold are seasonally stored. However, the systems are some times also applied for short-term storage.

BTES (Borehole Thermal Energy Storage) systems consist of a number of closely spaced boreholes, normally 50–200 m deep. These are serving as heat exchangers to the underground. For this reason they are equipped with Borehole Heat Exchangers (BHE), typically a single U-pipe, see Figure 30.

In some countries the boreholes are grouted after the BHE installation, in others no backfill at all is being used. Instead the boreholes are naturally filled with groundwater. Using grout will normally decrease the thermal efficiency but on the other hand the groundwater will be protected.

In the U-pipe a heat (or cold) carrier is circulated to store or discharge thermal energy into or out of the underground. The storing process is mainly conductive and the temperature change of the rock will be restricted to only a few meters around each of the boreholes.

These systems have been implemented in many countries with thousands of systems in operation. The numbers of plants are steadily growing and new countries are gradually starting to use these systems. They are typically applied for combined heating and cooling, normally supported with heat pumps for a better usage of the low temperature heat from the storage.

8.2. Optional Configurations

Depending on type of application there are several different system configurations to consider. The four main systems are principally illustrated in Figure 31.

The simplest system (A) is based upon that ground water is directly used for preheating of ventilation air during the winter and for cooling during the summer season. In this case heat and cold from ambient air is seasonally stored

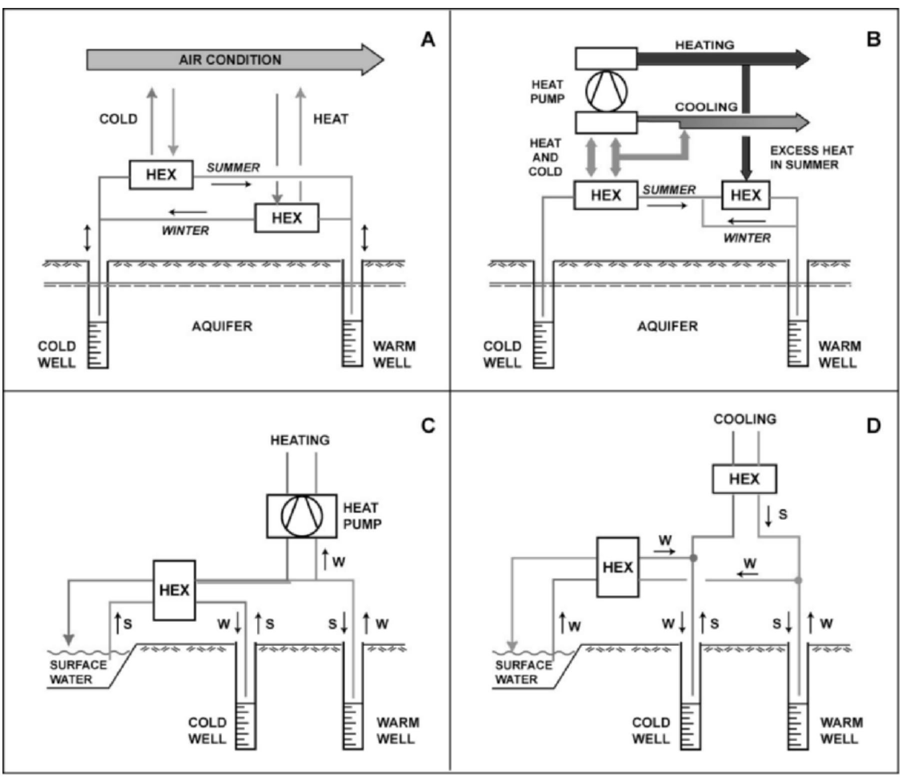

Figure 31. The main ATES configurations

in the aquifer at a temperature level of approx. +5 °C (winter) and +15 °C (summer). More commonly used is the heat pump supported system (B) that works the same way as system A. However, the production of heat is much larger and the temperature change is somewhat greater. System C represents an early type of ATES applications where surface water is used as a source of energy for the heat pump. This heat, at a temperature of 15–20 °C, is stored during the summer and used during the heating season. The fourth system (D) is similar, but in this case cold from the winter is stored to be used for district cooling.

Of these systems, heat pump supported combined heating and cooling applications (system B) are dominating. However, in recent years, there is a growing interest for storage of natural cold (system D), which is used for district cooling applications or for industrial process cooling.

8.3. Application Statistics and Experiences

In Table 9 the recent statistics of ATES utilisations in Sweden are presented. As can be seen the technology is so far preferably used for commercial and institutional buildings with small or medium sized applications. Large-scale plants are applied for some district heating and cooling systems while the industry sector only has a couple of systems applied for manufacturing industries. The rest represents cooling in the telecom sector.

In the Netherlands the number of applications is much larger (approx. 200 plants in 2004). In this country the use for industrial process cooling has a dominant portion of applications as well as for green house heating and cooling.

Some 20 years of experiences has reviled that a significant number of ATES plants have had or have operational problems or failures. The major part of these has been solved by fairly simple measures. However, a few plants have continued difficulties with the well capacities.

The dominating reason behind the problems is clogging of the wells mainly caused by iron precipitation. For this reason the research on ATES is focused on how to collect geological, hydrogeological and hydrochemical data in a proper way in order to design functional systems and wells.

Accurate site investigation data with test drillings and pumping tests are also of importance for modelling and simulations to be used for permit applications. The simulations are used to predict the thermal and hydraulic influences and are used for environmental assessment issues as well as for prediction of potential physical damages caused by the pumping of ground water.

TABLE 9. Statistics of ATES plants in Sweden

UTES system (for illustration, see Figure 31)	Number of plants	Average storage capacity (MW)		Utilisation sector (number of plants)				
		Heat	Cold	Com/Ins building	District heating	Combined DC/DH	District cooling	Industry[a]
Direct heating and cooling	1	0.3	0.3	1	—	—	—	—
Heat pump supported heating and cooling	25	1.30	1.45	15		4		6
Heat pump supported heating only	5	1.9	—	1	4	—	—	—
D. Cooling only	7	—	6.9	—	—	—	4	3
Total	38	—	—	17	4	4	4	9

[a] Process cooling of telecommunication stations included.

TABLE 10. Economics and potential energy savings by ATES

System application (see Figure 31)	PF	Energy saving (%)	Payback (years)
A. Direct heating and cooling	20–40	90–95	0–2
B. HP supported heating and cooling	5–7	80–87	1–3
C. HP supported heating only	3–4	60–75	4–8
D. Direct cooling only	20–60	90–97	0–2

8.4. Economics and Environmental Benefits

In general, the investment in an ATES system will not bee significantly higher than the investment for a conventional system. In some cases it may even be less. Since the operational cost is much less due to the energy savings, the ATES system is normally quickly paid. However, the experienced pay back time will differ with type of system. This is clearly shown in Table 10 which illustrate the performance factor (PF), the resulting energy savings compared to conventional systems and the calculated pay back time. The figures are derived from the same Swedish applications that are shown in former Table 9. Conventional systems in these cases consist of fossil fuels or electricity for heating and chillers or district cooling for air conditioning.

The environmental benefits are related to energy savings and will in most cases support the usage of ATES in any country. The obvious benefit is the reduction of CO_2 by using a large portion of natural renewable heat and cold in the systems. Besides the reduction of CO_2, there are also fewer emissions of NO_X and acidity (acid rain) to the atmosphere.

In the coming chapters fundamental issues for understanding the underground requirements for developing ATES systems are described.

8.5. Types of Aquifers

To be able to construct an ATES systems a proper aquifer has to at hand at or close to the site where the ATES user is located.

By definition groundwater can be found almost anywhere. The groundwater table is defined as the level under which all pores or fractures are water saturated, see Figure 32.

An aquifer is in practice defined to be a limited geological formation from which ground water can be pumped by using water wells.

There are two kinds of aquifers. If the groundwater stands in direct contact with the atmosphere as in the figure above, the aquifer is regarded as unconfined. If, on the other hand a permeable formation below the groundwater table is covered by a less permeable layer, the aquifer is regarded as confined, see Figure 33.

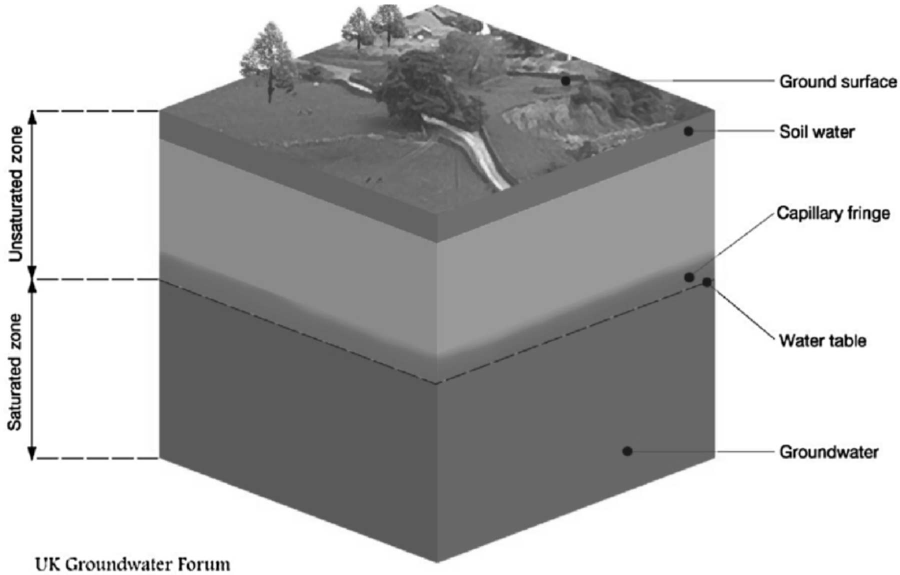

Figure 32. The saturated zone defines the groundwater table

The confined type of aquifer has a hydraulic pressure (static head) that is on a higher level than the top of the aquifer. This artesian pressure can sometimes reach above the surface level resulting in self flowing wells (artesian wells).

In nature, the groundwater is a part of the hydrological cycle. Hence, groundwater is naturally recharged and drained. Sometimes the draining is shown up as springs, but more common it flows out to a lake or a river as shown in Figure 34.

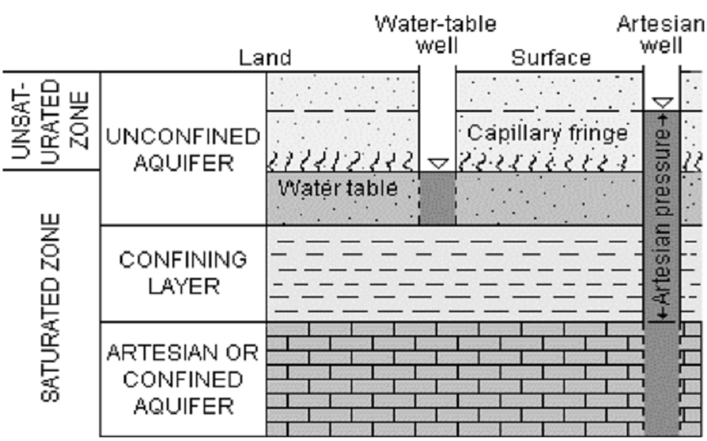

Figure 33. Confined and unconfined aquifers

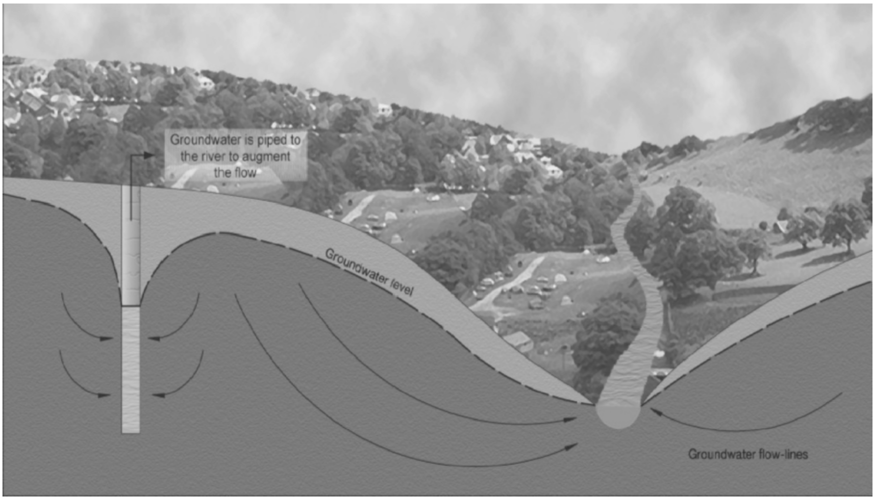

Figure 34. Flow of groundwater from an unconfined aquifer, drained by a river and by pumping from a water well

The figure also show how pumping of groundwater will create a cone of depression around the well. The size and shape of this cone is mainly related to the pumping rate and the hydraulic conductivity of the aquifer.

In the case of confined aquifers, the natural recharge and flow is more restrained, and as illustrated in Figure 35, the flow distance will normally be significantly longer and more time consuming.

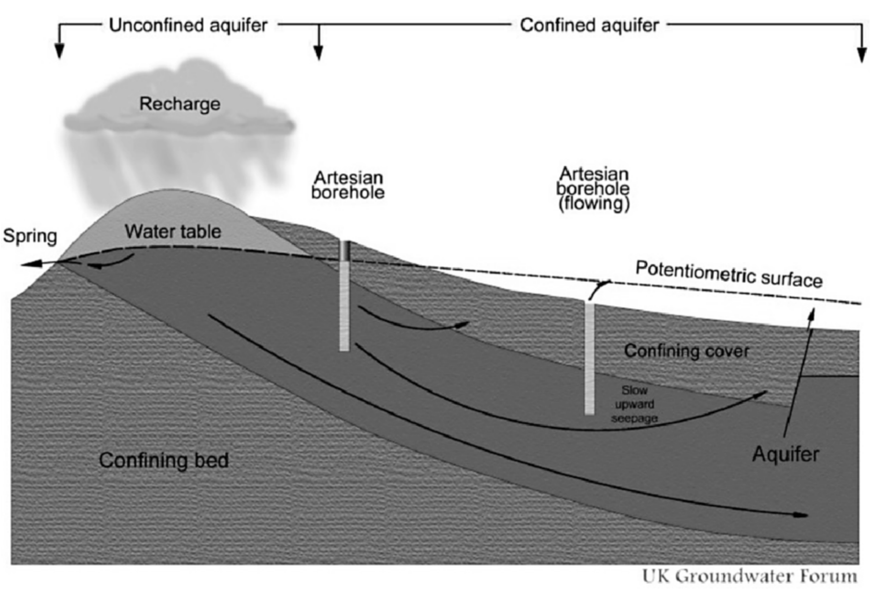

Figure 35. Flow of groundwater in an confined aquifer with potential artesian wells

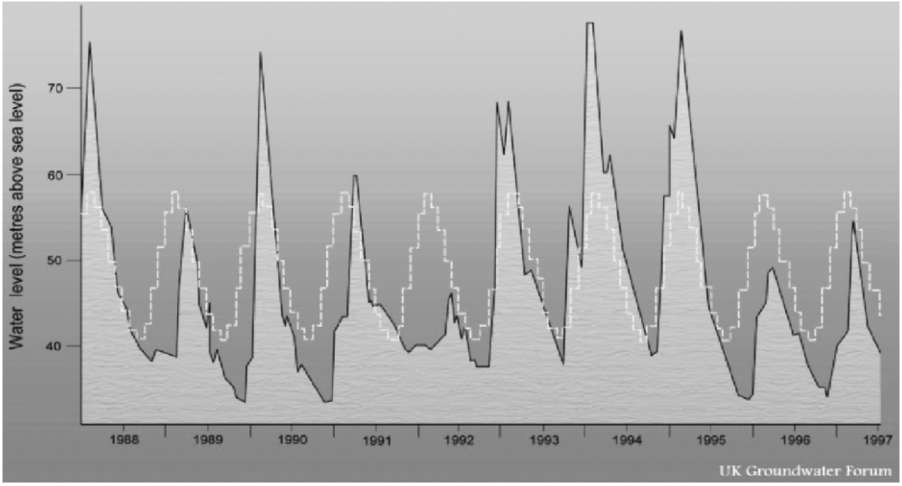

Figure 36. Variations of the groundwater table in UK, compared to “normal” values

Another difference is that the cone of depression will be larger since it only reflects a drop of hydraulic pressure around the well and the recharge area of the groundwater is restricted to the unconfined part of the aquifer. Still, if the wells in the figure should be pumped it is important to recognize that some recharge will take place also through the confining bed. This may create consolidation of this bed followed by potential damages to buildings.

The principal variation of the undisturbed groundwater table over a series of years is shown in Figure 36. This natural variation is mainly climate dependent. Most important is the precipitation and under what season the precipitation occurs. Normally, most of the recharge take place under non growing seasons in west European climate zones. In arid climate zones the recharge take place more occasionally and then combined with temporary “cold and rainy” conditions.

8.6. Aquifer Properties

Any ATES application will require a good knowledge of the aquifer being the target to use.

The most important properties are

- Geometry (surface area and thickness).
- Stratigraphy (different layers of strata).

- Static head (groundwater or pressure level).
- Groundwater table gradient (natural flow direction).
- Hydraulic conductivity (permeability).
- Transmissivity (hydraulic conductivity × thickness).
- Storage coefficient (yield as a function of volume).
- Leakage factor (vertical leakage to the aquifer).
- Boundary conditions (surrounding limits, positive or negative).

The first four items are studied by using topographical, geological and hydrogeological maps and descriptions, data from existing wells and older site investigations. The latter ones may contain geophysical data as well as older pumping tests and so forth. Any information on groundwater chemistry is of importance as well as information of the natural groundwater temperature.

This material will give the first picture of the aquifer and will be the basis for complementary site investigations. These may include test drillings, geophysical logs and pumping tests with water chemistry in order to fully understand the aquifer conditions.

The latter five bullets are obtained by using different forms of pumping tests. Such a test is normally done with a pumping well and a number of observation wells or pipes placed at certain distances from the well and in different directions. The duration varies but is commonly 1–2 weeks.

During the test the groundwater table is monitored and the drawdown cone around the pumping well is established. From these drawdown data the hydraulic properties of the aquifer can be analyzed as shown in Figure 37.

The most important parameter is the *permeability* (or *hydraulic conductivity*) is defined as the resistance for water to flow in the aquifer material and it is expressed as a flow rate (m/s) when it is affected by the gravity (gradient 1.0).

In an unconfined aquifer the permeability relates to the specific yield. This term express the ability for the aquifer to be drained and is clearly related to the grain size. The specific retention reflects the capillary forces that tend to retain water close to the grains. How these parameters relate to each other is shown in Figure 38.

From a pumping test the permeability is evaluated as *Transmissivity* (m^2/s). This parameter express the added permeability's meter by meter. By dividing the transmissivity with the thickness of the aquifer an average permeability is given.

If the test has proceeded till a steady state (cone of depression fully developed), the leakage factor as well as the storage coefficient can be determined. These factors are less important but may be critical when it comes to modelling

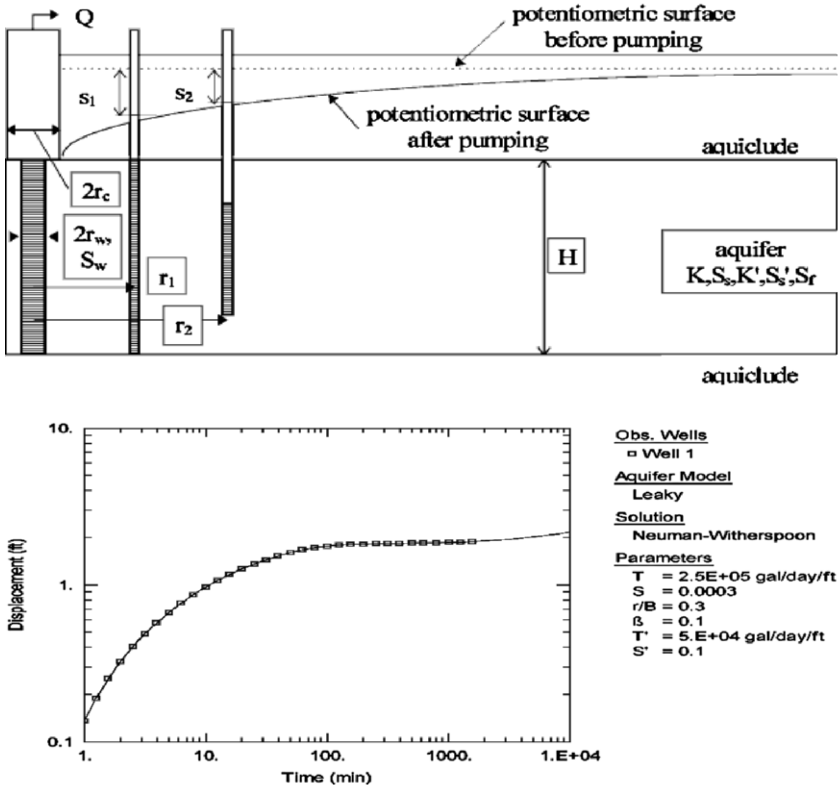

Figure 37. A pumping test is used to evaluate the hydraulic properties of the aquifer

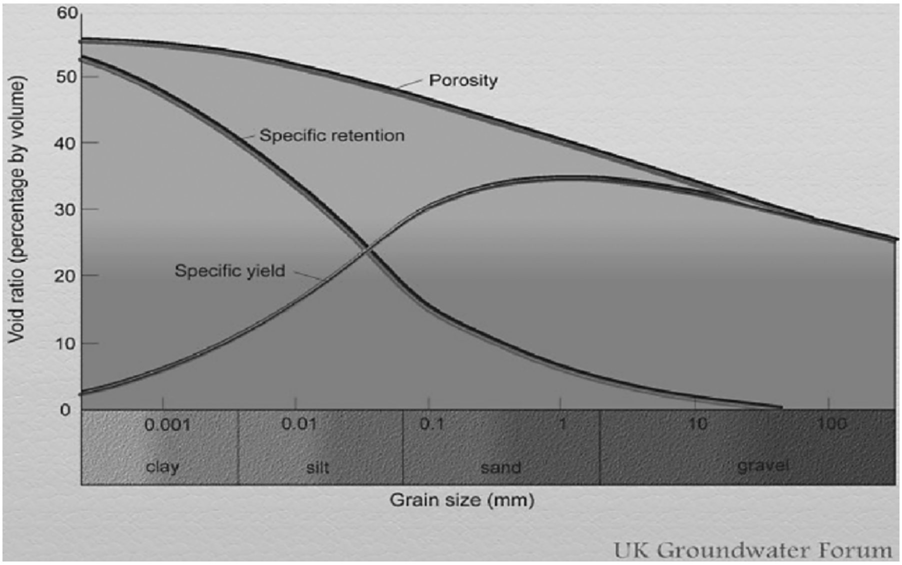

Figure 38. The relation of porosity, specific yield, specific retention and grain size in an unconfined aquifer

and simulations. Also positive and negative the boundary conditions may be essential for modelling, especially if they occur close to the well sites.

8.7. Ground Water Chemistry

8.7.1. PROBLEMS RELATED TO WATER CHEMISTRY

As stated earlier, the chemical composition of the groundwater is of uttermost importance when it comes to the design of any ATES system. The reason is the potential risks for functional problems with wells and other components in the system.

Potential problems with an ATES loop are illustrated in Figure 39 and encounter a number of events that are related to the chemical behaviour of the system. However, the figure also illustrates problems connected to a general system design, such as aeration and sand production. These types of problems may also have secondary damaging impacts on surroundings buildings and the environment.

8.7.2. HOW TO TRACE WELL PROBLEMS

The most common technical problem is clogging of wells. Clogging is defined as an increased flow resistance for water to enter the well (or be disposed

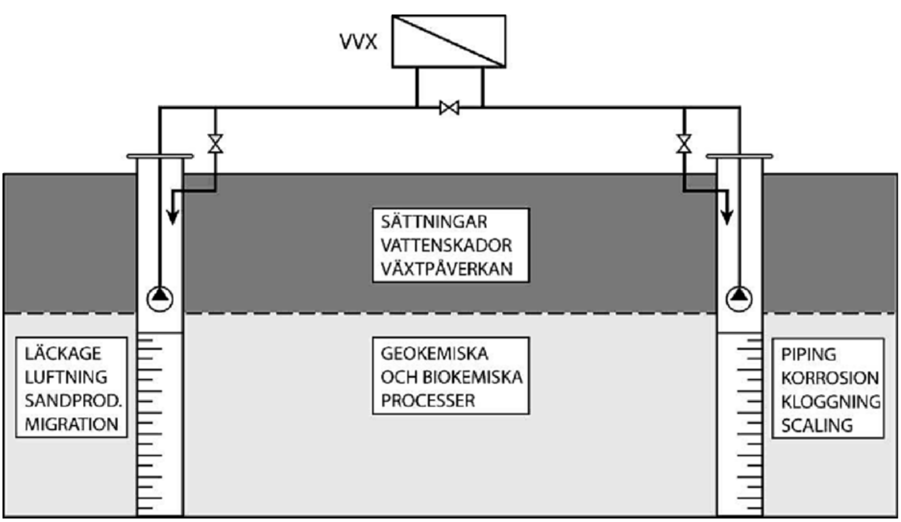

Figure 39. Potential technical problems in an ATES loop. Secondary impacts on the surrounding environment may occur

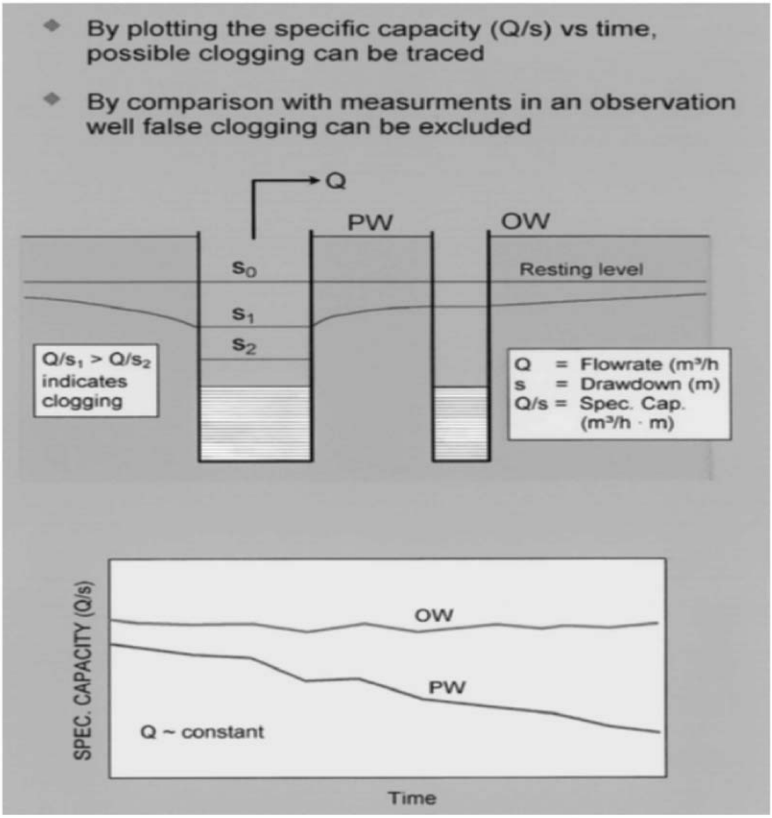

Figure 40. By using a monitoring program clogging can easily be traced in an early stage

through the well). The clogging process normally gets more and more evident with time and will result in a lower and lower well capacity.

Clogging can easily be traced and dealt with in an early stage by monitoring flow rates and drawdown as shown in Figure 40.

The figure illustration shows occurrence of clogging by plotting data from an observation well (OW) and compare that to the production well (PW). In this case the production well shows a decreased specific capacity while the observation well shows a steady level versus time. The only explanation is then that the resistance for water to enter the production well is increasing. The increased resistance will lower the drawdown inside the well, while the groundwater table outside the well is kept constant. This will increase the hydraulic gradient (the driving force) between the well and the aquifer and hence maintain a constant flow rate.

"False clogging" sometimes occurs. Such events are either explained by a general lowering of the groundwater table or by failure of the submersible pumps cutting down the flow rate. However, by monitoring both the production wells and the aquifer in observation wells or pipes such events can be excluded as a result of clogging.

8.7.3. CLOGGING PROCESSES

There are three main clogging processes

- Clogging by fines.
- Hydro-chemical clogging.
- Biochemical clogging.

Already during drilling and well construction there are certain risks for clogging of the aquifer porosity. As illustrated in Figure 41, fines may invade into permeable beds decrease the yield capacity. A well known such clogging additive to the drilling mud is bentonite, which ones invaded is very difficult to clean out. For this reason it is much better to use polymers in the mud.

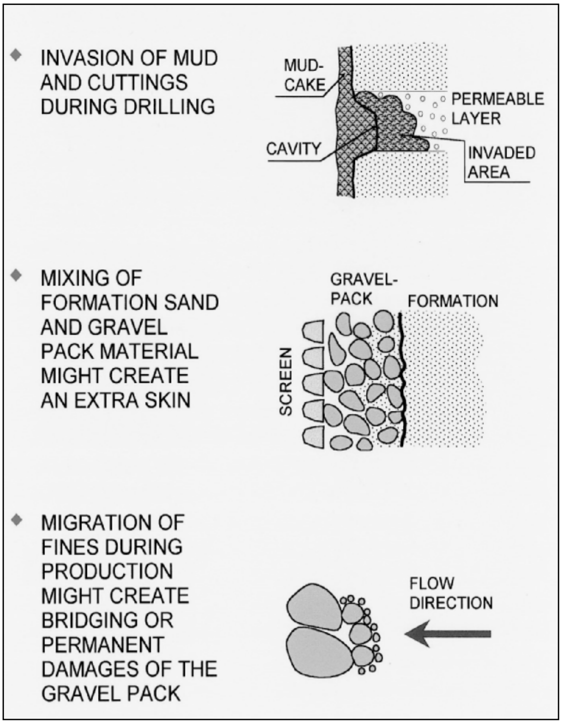

Figure 41. Different forms of clogging by fine particles during drilling, well construction and later on, production

During the construction of gravel packed wells a critical stage is the placement of the gravel pack. As shown in the figure this procedure should be carried out in a way that the gravel is not mixed with formation sand. For this reason its better to pump down the gravel through a tremie pipe rather than to poor it down and let it sink by gravity forces.

Even if a well has been properly designed and constructed, there is always a certain risk for a gradually clogging process cased by migration of fine grained formation particles. Typically, these fine particles will form bridges in the well vicinity which will increase the flow resistance. However, these bridges can be broken down by a reversed flow and the fines may be flushed to the surface by a further well development. For this reason wells with potential bridging should be constructed so they easily can be flushed.

8.7.4. HYDRO CHEMICAL CLOGGING

Under certain conditions the wells may be clogged by solid chemical precipitates. Most common of these are iron and calcium compounds.

The main processes behind these types of clogging are illustrated in Figure 42.

In general the processes are initiated by changes of pH and the redox potential (Eh) of the water.

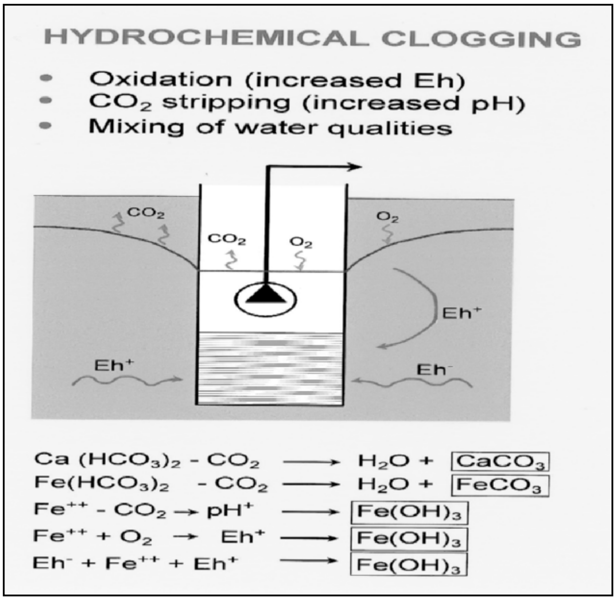

Figure 42. Hydro-chemical well clogging processes

Precipitation of carbonates (often referred to as scaling) may take place if carbon dioxide is allowed to be stripped out from the water. This happens if the draw down in the well exceeds the bubble point for CO_2. For this reason large draw downs should be avoided for scaling sensitive waters.

Stripping of CO_2 may also cause iron to precipitate, normally as iron hydroxide. That type of precipitation will also occur if an reduced type of water is entering the well from one side and an oxidized type from the other side. A third and obvious iron oxidation process will take place if reduced water gets in contact with oxygen.

To prevent from hydro-chemical clogging the systems should be designed so there is no entrance of air to the ground water loop. Hence, the loop should be perfectly air tight and constantly under pressure.

8.8. Design and Construction

8.8.1. DESIGNING STEPS AND PERMIT PROCEDURE

Any ATES realization is a quite complex procedure and has to follow a certain pattern to be proper developed. Typical designing steps are as follows:

- Pre feasibility studies (describes the principal issues).
- Feasibility study (tells the technical and economical feasibility and environmental impact compared to one or several reference systems).
- The first permit applications (local authorities).
- Definition of hydro-geological conditions by means of complementary site investigations and measurements of loads and temperatures, etc on the user side.
- Evaluation of results and modeling (used for both technical, legal and environmental purposes).
- Final design (used for tender documents).
- Final permit application (for court procedures).

The technical issues are general, but the permit procedure may vary from country to country. However, in most countries the use of ground water for energy purposes will be restricted and will be an issue for application according to different kind of acts.

8.8.2. FIELD INVESTIGATIONS

One essential part in developing an ATES project is to perform site investigations. The more knowledge that is obtained of the underground properties, the better basis for design is achieved.

The site investigations will most commonly cover the following procedure:

1. Geological mapping.
2. Geophysical investigations.
3. Test drillings.
4. Pumping tests.

The test drillings will define the stratigraphical units in the area while the geophysics and geological mapping are used for extrapolation of the layers and for definition of geometry, see Figure 43.

Test drillings may like in the figure be a part of the final system and can be looked upon as an early investment in system. However, more commonly they are drilled in a small dimension and do not fit into the final system after design. In these cases they still can serve as observation wells.

For shallow aquifers in the overburden it is common to drive slim steel pipes that are perforated in the lower meter or so. This method has proven to be an excellent way of taking samples for the design of screened production wells.

Based on the results a conceptual model is created and the hydraulic properties of the aquifer and its surrounding layers are derived from the pumping test.

The final outcome will be a geological model that is more or less accurate and that can be used for the final design using simulation models.

To be able to make model simulations the load of heat and cold have to known. For this reason it is common to perform measurements on how the loads are varied at different out door temperatures.

Such investigations that also covers supply and return temperatures in the distribution systems are often done prior to or in parallel with the underground site investigations. The results are key factors as basis for design in order to calculate flow rates and size of the ATES storage. In Figure 44 an example of heating and cooling load for an ATES application in workshop factory is given.

8.8.3. MODEL SIMULATIONS

Simulations are used for several reasons, but preferably to study how different flow rates and different number and distances between the wells are functioning. The results will then guide the decision where to place the wells and with what flow rate they should be operated.

The outcomes of such simulations are of two kinds, namely

1. The hydraulic impact shown as cones of depression and uplift around the wells.
2. Configuration of the thermal front around the wells.

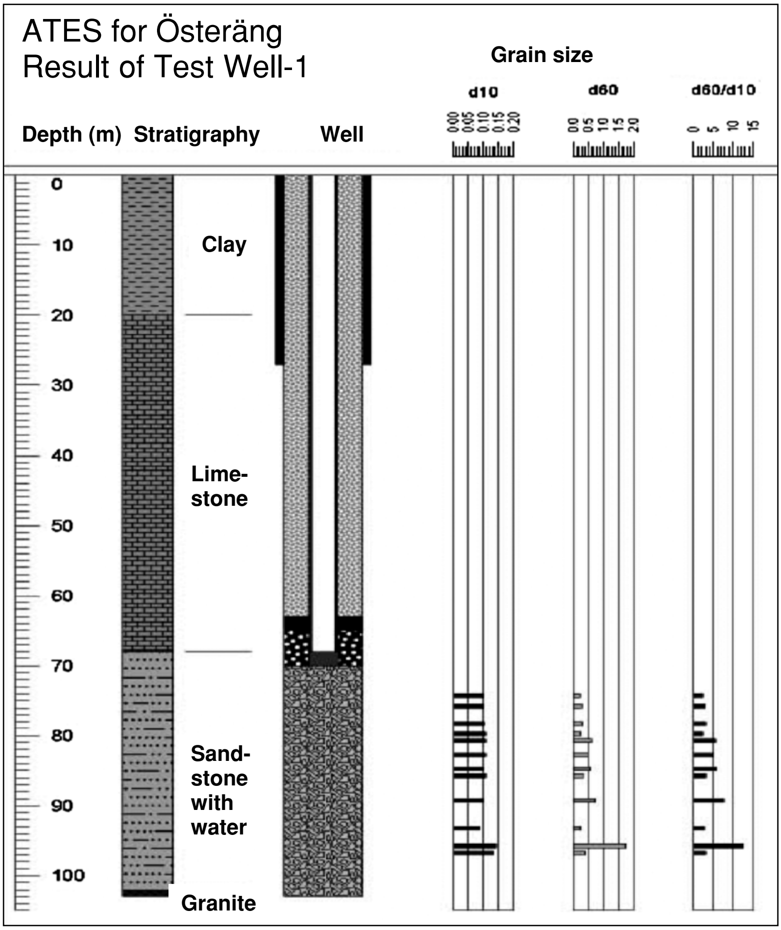

Figure 43. Example of results from a test drilling

The hydraulic impact will be more or less extensive depending on the distance between the groups of wells. With a long distance the impacted area will be larger than for a shorter distance. In Figure 45 an example from a simulation at Bo 01 in Malmö, Sweden is shown.

As can be seen, the impact area, defined as a change of static head with 0, 3 m or more, reach quite a distance.

In Figure 46 the corresponding thermal front is illustrated. As can be seen the thermally impacted area is much less. As a matter of fact, in this case the warm and cold fronts are somewhat overlapping one each other. This will to some extent be a disadvantage for the production temperature, but due to restricted surface availability this was accepted.

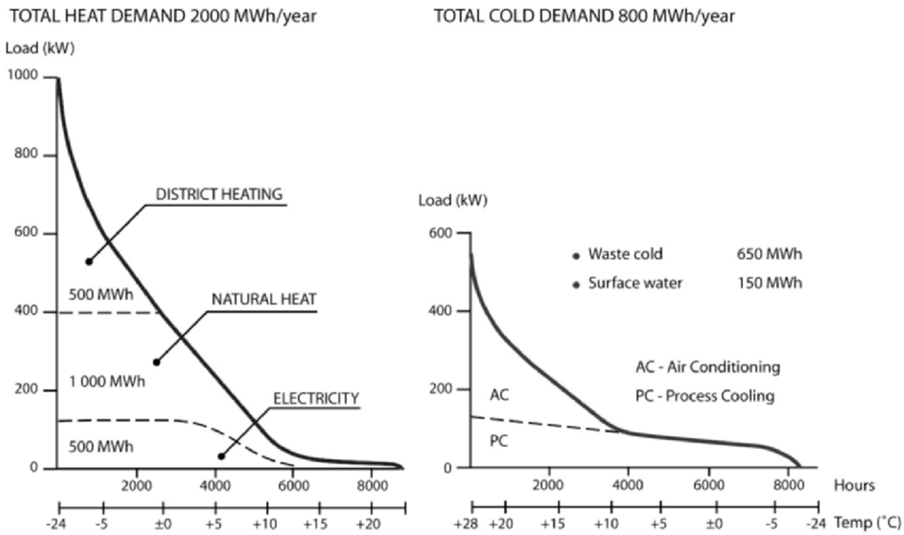

Figure 44. Example of heat and cold load duration diagrams

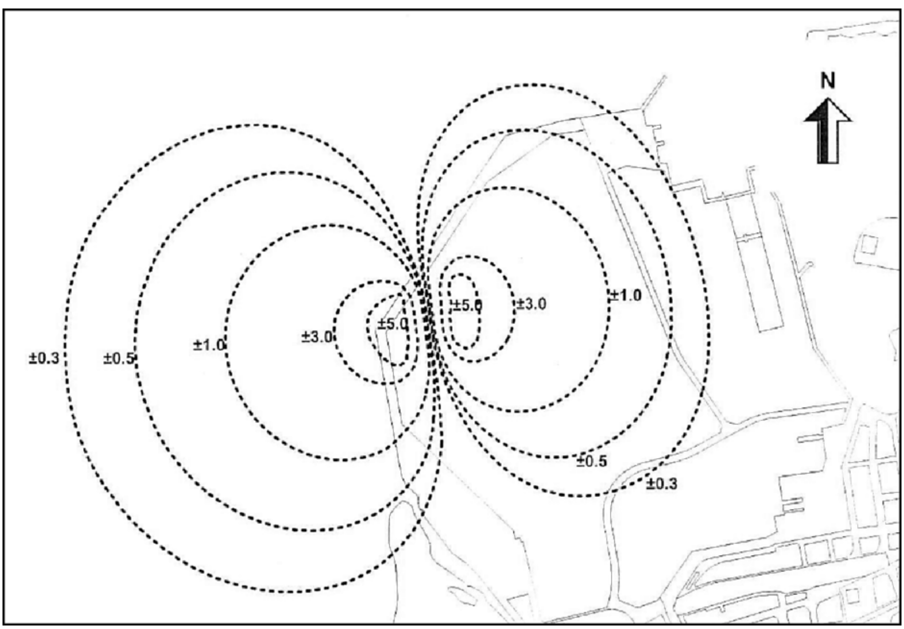

Figure 45. Example of a hydraulic simulation with 2 groups of wells, 7 warm and 7 cold, run with a flow rate of 32 l/s. The simulation is done with MODFLOW. Figures in meter. The distance between the well groups are 150 m (see also Figure 46)

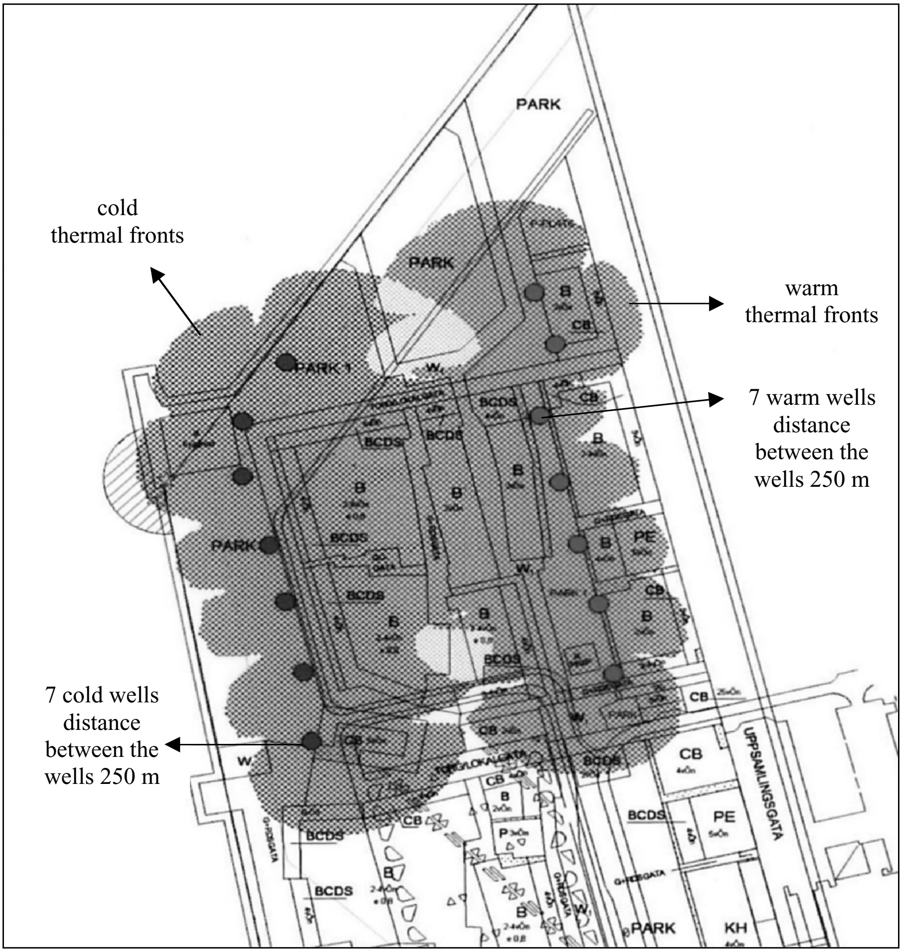

Figure 46. Warm and cold thermal fronts as they have been simulated with CONFLOW for the Bo 01 plant

There are a number of simulation models available on the market. Some are user friendly and quite easy to use, while others are more advanced. In Figure 47 some of the models are listed.

There are a lot of aspects that can be addressed when it comes to model simulations. However, most of these can be summarized with these bullets:

- Model simulations are necessary for design and for ATES impact studies.
- Some user friendly models are available on the market at a fairly low cost.

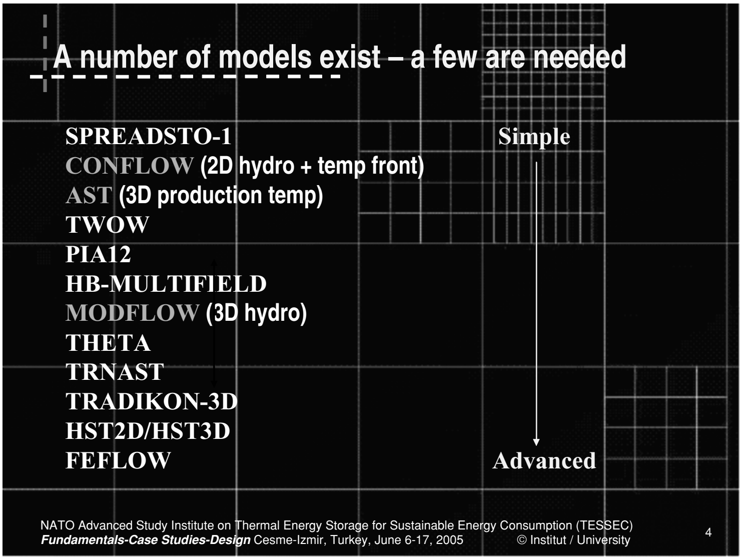

Figure 47. Models for hydraulic and thermal simulations. Market ones is commonly used in commercial ATES projects

- The accuracy of the model and hence the simulation results will increase with the number of data input and the quality of these.
- Still, models are always simplifications of real conditions. Simulated results may differ significantly from operational results.

9. ADVANCES IN GEOTHERMAL RESPONSE TESTING

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Abstract. A Geothermal Response Test measures the temperature response to a thermal energy forcing of a borehole heat exchanger. The temperature response is related to the ground and borehole thermal parameters such as thermal conductivity, heat capacity and the conductivity of the borehole material and is therefore used to obtain estimates on these important parameters. Generally such data are analysed using a line source method. Although quick and, when certain conditions are met, accurate, the line source method has several drawbacks. First of all it is only valid for the constant heat flux case, secondly it only allows estimates of ground thermal conductivity and borehole resistance, other parameters, like heat capacity, cannot be estimated. Thirdly effects of ground water flow cannot be quantified. Moreover, selecting test parameters a-priory is not always easy and it is difficult to adjust test conditions due to the necessity of constant heat flux. Parameter estimation techniques have been developed to allow analysis of test results under varying heat flux conditions and to allow simultaneous estimates of different parameters to be obtained. In this paper we develop such a method based on the TRNSYS simulation package with the DST borehole model, using a generic optimisation package—GenOpt—to perform the calibration. Moreover we extend the GRT test protocol to use different heat extraction and injection energy levels within the same experiment. Apart from improving estimates of ground and borehole thermal parameters, it is our final goal to allow the characterisation of ground water flow using the in situ thermal response test.

Keywords: Geothermal response test, ground thermal conductivity, line source, GENOPT, TRNSYS, borehole heat exchanger

9.1. Introduction

In the design of borehole heat exchangers accurate information on the soil thermal parameters, such as thermal conductivity, heat capacity and temperature, is essential for the design of an economically sized and well-functioning thermal energy store. Especially the soil thermal conductivity is critical, as

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 177–192.

it is affects both total length of heat exchanger needed as well as optimum inter-borehole distances. Moreover, it is very difficult to estimate indirectly, e.g. from data on the geological profile (Austin, 1998; Ghelin, 1998; Witte et al., 2002). Due to the importance of ground thermal conductivity, several Geothermal Response Tests (GRT) methods (Eklöf and Gehlin, 1996; Austin, 1998; van Gelder et al., 1999) have been developed to measure the effective thermal conductivity of the ground and the local thermal resistance of the heat exchanger—borehole installation. These tests all operate under the assumption that the principal heat transport mechanism is conduction and therefore that there is a relation between the thermal power applied to a heat exchanger, the temperature development with time and the thermal conductivity of the material. Other mechanisms of heat transport, such as radiation from the surface, effects of geothermal gradients, thermally induced convection or advection of ground water, are not taken into account in the analysis. When these processes play an important role in the heat transfer in the ground, the analysis methods include the effects at best lumped in the effective ground thermal conductivity estimated. However, in many cases the basic assumptions made by the GRT are invalidated. For instance, with ground water flow the effective thermal conductivity depends on the difference between local and global temperature and therefore is not constant in time during the experiment (Witte, 2002). Moreover, the parameters measured or estimated with the method are limited to the background temperature, ground thermal conductivity and borehole resistance. Other parameters, like the heat capacity of the borehole and ground, shank spacing, etc are not considered. In this paper we present an alternative approach to the data analysis of GRT data, based on parameter estimation. Also we extend the test procedure to include non-constant energy pulses applied to the borehole heat exchanger. In a following paper we will apply the method to data from a controlled groundwater flow experiment and investigate the possibilities of separating the different mechanisms of heat transport in the ground.

9.2. Geothermal Response Test Field Methods

The basis of the line source GRT test methodology is to apply a constant energy flux to the ground and measure the temperature development. From the relation between time and temperature the thermal conductivity from the material, in this case the ground, can be estimated using a relatively simple line-source approach (Ingersoll and Plass, 1948; Mogeson, 1983). Moreover, such a test yields important information on drilling conditions, geological profile, background temperature and the borehole response (borehole resistance).

Two approaches have been developed to carry out such a test in field conditions. The type I test, developed concurrently in Sweden (Eklöf and Gehlin,

1996) and the United States (Austin, 1998), employs resistance heating to provide thermal energy to force the ground. This technique, although successful, has several drawbacks. First of all, only heat rejection to the ground is possible. Especially at higher temperatures advection or thermally induced convection of ground water may affect results. Moreover, the method depends on a constant electrical power source to provide a constant thermal energy flux to the ground, in reality often considerable variations in the power grid are observed. We therefore developed (van Gelder et al., 1999; Witte et al., 2002) a test methodology using a reversible heat pump to provide a heat extraction or heat rejection to the ground. In this type II test the energy flux to the ground is actively controlled, using temperature measurements and a three-way regulating valve. An inertia tank is used for thermal storage, so that electrical power fluctuations do not affect results. The system is very robust, housed in a rugged container, and supports full telemetry. To date we have performed about 40 odd tests in the Netherlands, Belgium and the United Kingdom under varying conditions and the method has proved very reliable and yields accurate and reproducible results.

9.3. Data Analysis

9.3.1. DATA ANALYSIS USING THE LINE SOURCE APPROACH

With the line source approach it is possible to calculate from the experimental data the ground thermal conductivity and borehole resistance. At the start of the experiment, before an energy pulse has been applied, the undisturbed ground temperature is measured.

The data analysis procedure for the case of constant heat flux is based on the theory describing the response of an infinite line source model (Ingersoll and Plass, 1948; Mogeson, 1983). Although this model is a simplification of the actual experiment, it can successfully be used to derive the geothermal properties (e.g. Kavenaugh, 1984; Austin,. 1998; Gehlin, 1998).

The model approximates the transient process of heat injection or extraction by

$$T_f = \frac{-\left(q_v \rho c \left(T_{out} - T_{in}\right)\right)}{4\pi\lambda H}\left[\ln\left[\frac{4at}{r_0^2}\right] - \gamma\right] + \frac{q_v \rho c \left(T_{out} - T_{\text{in}}\right) R_b}{H} + T_g \tag{1}$$

where q_v is the volume flow circulation medium (m^3/s), ρ is the density circulation medium (kg/m^3), c is the heat capacity circulation medium (J/(kg K)), T_{out} is the return temperature circulation medium (°C), T_{in} is the injection temperature circulation medium (°C), T_f is the average temperature of circulation medium (°C), T_g is the far field (ground) temperature (°C), λ is the ground thermal conductivity (W/m K), H is the ground loop length (m), R_b is the borehole resistance (K/(W/m)), γ is the Eulers constant (−), t is the time (s), r_0 is the borehole diameter (m), k is the coefficient of the regression T_f with ln(t) (−), a is the thermal diffusivity (λ/C, where C is the thermal capacity) (m^2/s).

This formula can be used as an approximation of the transient process under the condition that

$$t \geq \frac{5r_0^2}{a}. \tag{2}$$

The thermal conductivity can be estimated from the data by

$$\lambda = \frac{-\left(q_v \rho c \left(T_{out} - T_{in}\right)\right)}{4\pi Hk} \tag{3}$$

where the parameter [k] equals the slope of a linear regression of temperature with logarithmic time. When λ has been estimated, the borehole resistance R_b can be calculated using (1).

As an example we show the results from the reference experiment (Table 11) described in Witte et al. (2002) in Figure 48. From the data an average conductivity value of 2.1 ± 0.02 can be inferred when the complete data range is included in the analysis. As the first hours of the experiment results are mainly due to the borehole response, these are often discarded. The estimate of the soil thermal conductivity excluding the first five hours of data was 2.14 ± 0.03 W/m K. After the ground thermal conductivity has been estimated the borehole resistance can be calculated and fitted in a straightforward graphical way (Figure 49). The borehole resistance obtained in this way was 0.13 K/(W/m).

TABLE 11. Experiment parameters reference experiment (Witte et al., 2002)

Experiment data	Energy extraction pulse (ground cooling)
Date	November 1999
Experiment duration (h)	265
Loop type	HDPE PN16 U-loop 25/21 mm
Length (m)	30
Borehole radius (m)	0.15
Circulation medium	Ethylene glycol
Flow rate (m^3/h)	0.78 ±
ΔT (°C)	-1.3 ± 0.0697
Energy flux (W)	$-1.122.23 \pm 60.04$
Energy flux per meter (W/m)	-37.41 ± 2.0
Logging interval (s)	60

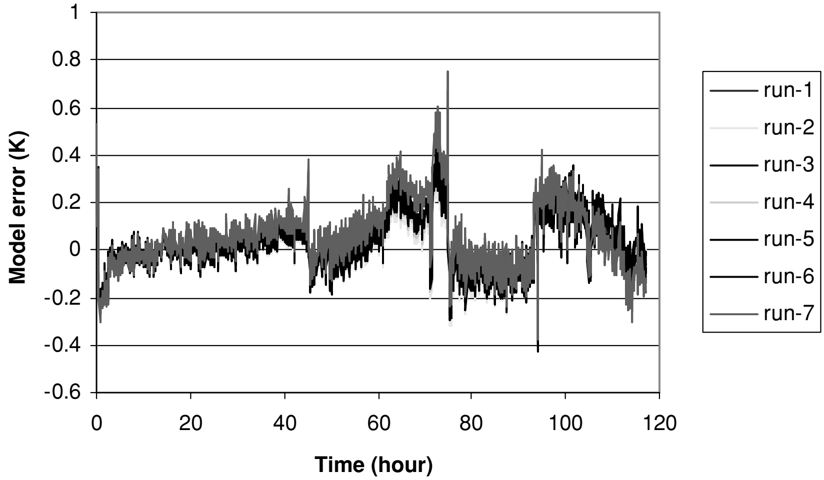

Figure 48. Temperature response reference experiment (Witte et al., 2002) and linear regression line. Relevant experiment data are summarized in Table 1

9.3.2. STABILITY AND CONVERGENCE

It is known that the line-source model employed can be rather sensitive to perturbations caused by outside influences such as the daily atmospheric temperature cycle or passing weather fronts (e.g. Austin, 1998; Gehlin, 1998;

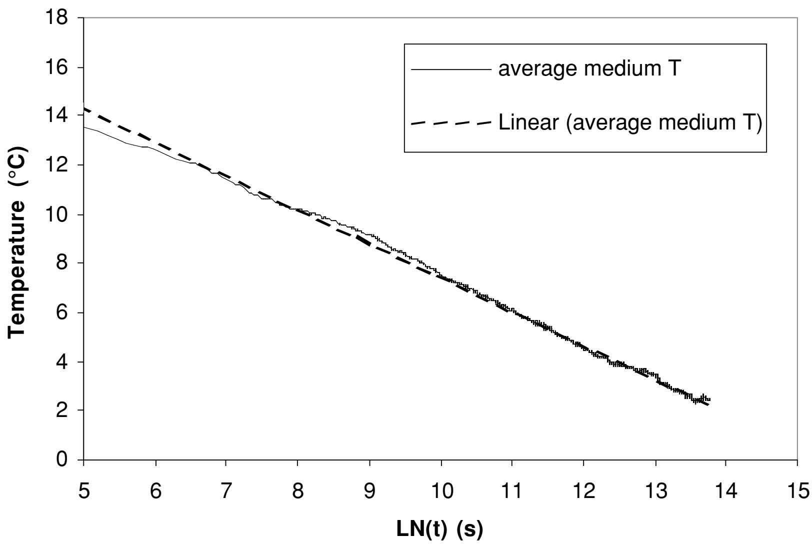

Figure 49. Estimated borehole resistance reference experiment (Witte et al., 2002)

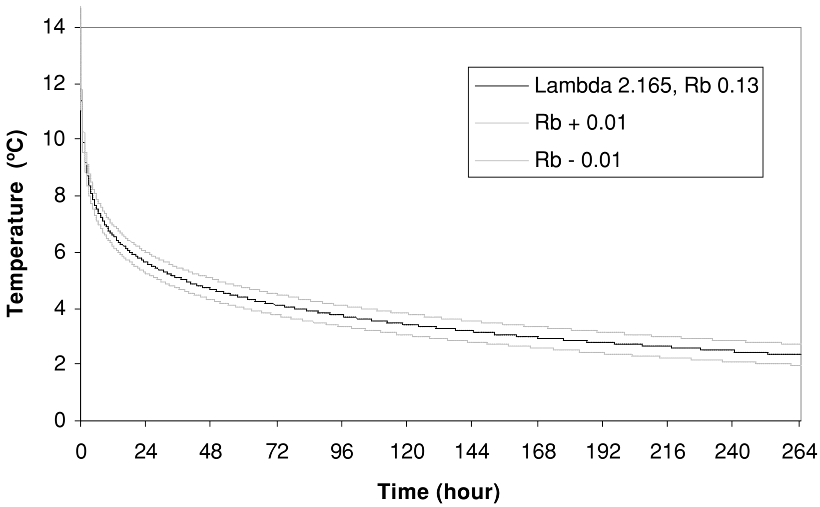

Figure 50. CUSUM test, stability and convergence of the estimates of ground thermal conductivity as a function of starting time and amount of data included, obtained in the reference experiment

Van Gelder et al., 1999). The stability and speed of convergence of the estimated soil thermal conductivity, as a function of starting time selected and amount of data points included, can be evaluated using a graphical method based on the CUSUM test (Brown et al., 1975) (Figure 50). These graphs are constructed by calculating estimates of soil thermal conductivity with the data points added in a stepwise fashion, each step adding a certain amount (e.g. 4 h) of data. The sensitivity to the starting time selected can be evaluated by constructing several of these series, each with a different starting time.

We can see that the series converge faster to a common value of estimated ground thermal conductivity when the first three hours of data are discarded. During the first period, between 50 and 100 h, the values converge to a common estimate of ground thermal conductivity. After 100 h the estimated value for the ground thermal conductivity shows a significant drift. This drift can be correlated with transient ambient temperature trends affecting the stability of the estimate. Subsequently the test apparatus was improved by fitting the temperature sensors directly in the borehole, greatly dampening any effect of ambient temperature.

During the many tests we performed we found that that ground water flow will significantly affect results of the In Situ Geothermal Response Test. This is especially clear in the CUSUM graphs, that show an increasing estimate of ground thermal conductivity with time. The reason for this is that the effect

TABLE 12. Description of the experiment parameters for the reference experiment and ground water extraction experiment

Parameter	Reference experiment	Ground water extraction experiment
circulation medium flow rate (m^3/h)	0.85 ± 0.012	0.87 ± 0.0072
ΔT (°C)	2.18 ± 0.084	2.11 ± 0.068
Energy rate (W/m)	69.23 ± 2.13	68.78 ± 2.23
Groundwater extraction rate (m^3/h)	0	2.89

of ground water flow depends on the difference between the groundwater temperature and fluid temperature. As the experiment progresses the fluid temperature decreases (or increases, depending on the experiment) and the difference with the ground water increases as well, enhancing the effect. To better investigate this behaviour we conducted two controlled experiments on a reference heat exchanger (Table 12).

In the reference experiment ground water flow was absent or very low, in the second experiment ground water flow was enhanced by pumping in a nearby well, placed at about 3 m from the borehole heat exchanger (Witte, 2001). The results of the convergence graphs are shown in Figure 51. The estimate for ground thermal conductivity converges quite well to a value between 2.3 and 2.4 while in the experiment with ground water flow the

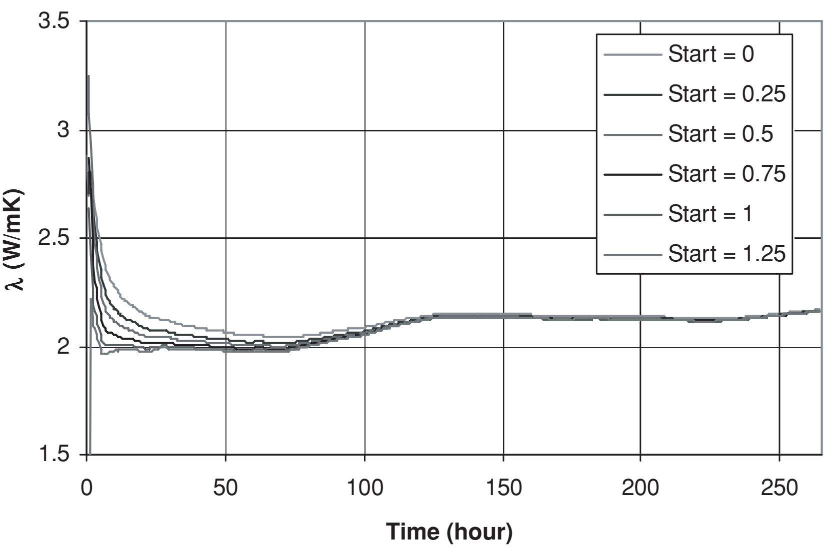

Figure 51. Conductivity convergence (CUSUM graph) curves for the reference (no groundwater) and ground water flow experiment

estimate does not converge but increases to a value at the end of the experiment of 3.1.

9.3.3. DATA ANALYSIS USING A NUMERICAL MODEL AND PARAMETER ESTIMATION TECHNIQUES

Originally parameter estimation techniques have been developed for use with GRT data mainly to overcome problems using the line source analysis method when the power source is not constant. Shonder et al. (1999) developed a 1D numerical model based on the cylindrical source representation. Yavuzturk et al. (1999) developed a numerical short time step two-dimensional finite volume model (Austin, 1998; Austin et al., 2000; Yavuzturk et al., 1999) that uses a Nelder-Mead Simplex parameter-estimation method (Nelder and Mead, 1965; Austin et al., 2000) to obtain estimates of several parameters of the ground loop heat exchanger. Gehlin and Hellström (2003) present an explicit one-dimensional finite difference model with a short time step. In this model a U-tube is approximated by a concentric heat exchanger. Using this model they compare two line source approximations, a cylindrical source approximation and the 1D numerical model. They conclude that the line source method is fastest and that the numerical models tend to overestimate ground thermal conductivity. Still, they conclude that using a numerical method with parameter estimation is the only solution for situations with varying heat flux. However, Witte et al. (2002) showed that when heat flux is constant the line source and numerical approximations give very similar results.

A few years ago we started using Yavutzturk's method with GRT data to obtain an independent estimate of ground thermal conductivity, and allow the evaluation of other parameters, such as ground heat capacity and U pipe shank spacing. The concurrent estimates of conductivity, heat capacity and shank spacing proved instable in Yavuzturks method, we therefore now only use the method to estimate ground and borehole conductivity. The main drawbacks of Yavuzturks approach are the long running times required to obtain an estimate and the fact that only conductivity can be estimated robustly.

More recently Wagner and Clauser (submitted to Journal of Geophysics and Engineering) developed a parameter estimation technique based on the SHEMAT code (Clauser, 2003) to concurrently estimate ground thermal conductivity and ground heat capacity. According to Wagner and Clauser the expected effect of uncertainties in heat capacity (on the order of 20% for a single rock type) may affect the response temperature of a borehole heat exchanger by 2%. In this method the borehole—heat exchanger is not modelled explicitly and therefore the exact borehole—heat exchanger configuration cannot be investigated in detail.

9.4. Parameter Estimation Based on the Trnsys Model

To offer more flexibility we adopt an approach, based on the transient simulation model TRNSYS (Klein et al., 1976), making use of the Lund DST borehole model (Hellström, 1989). The parameter estimation procedure is carried out using the GenOPT (Wetter, 2004) package with the Nelder and Mead Simplex minimization algorithm (Nelder and Mead, 1965) or Hooke and Jeeves minimization algorithm (Hooke and Jeeves, 1961).

We use a fairly straightforward TRNSYS model (Figure 52). The model consists of a data reader reading the experiment data from an excel spreadsheet that also calculates the basic error (difference between modelled and measured temperature at a certain time step), the DST component that calculates the borehole response with the experiment description (ground typology, borehole and heat exchanger configuration, etc) and an integrator to integrate the errors. An inertia tank was added to the model to take into account the inertia of the water volume in the ground loop, yielding a better fit for the first few time steps when calibrating on heat flux in stead of directly on temperature. Inputs to the DST model are the borehole water flow temperature and flow rate, output is the borehole return temperature. The latter is compared with the measured return temperature to obtain the error of the current estimate.

An advantage over the line source method and other parameter estimation techniques is that the estimate can be made directly on the measured return temperature. Using the derived average heat extraction or injection rate may

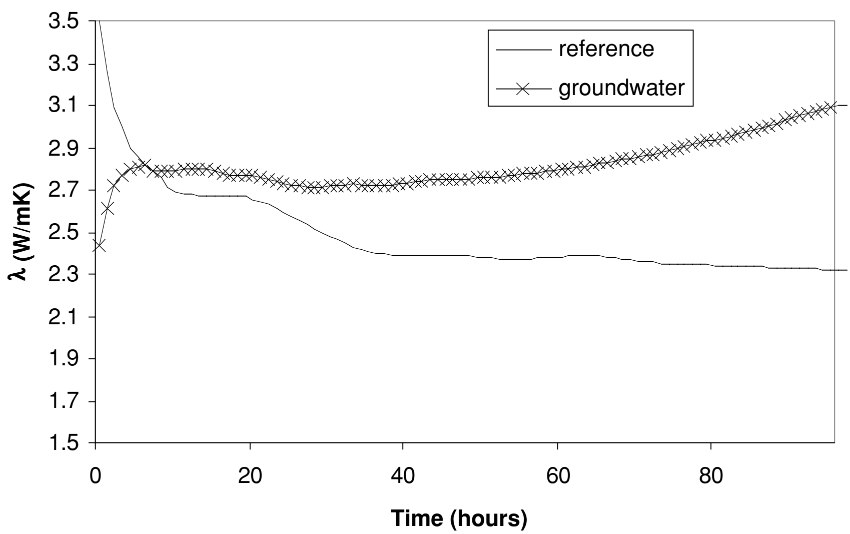

Figure 52. TRNSYS model with Excel data reader, simple tank (pipe volume) DST borehole heat exchanger model, and error function

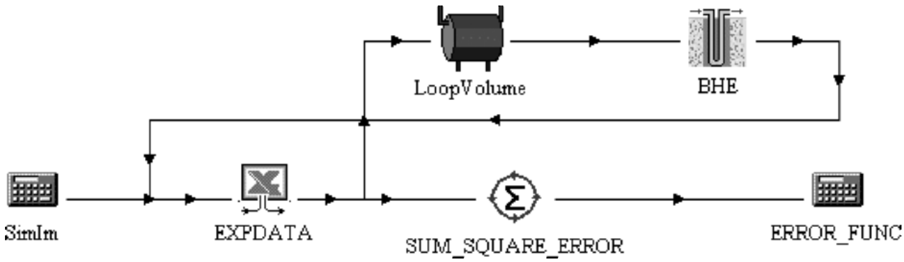

Figure 53. Example of GenOpt calibration run, using a Nelder Mead Simplex algorithm

introduce some error as, apart from flow rate and ΔT, the density and heat capacity of the fluid need to be taken into account.

For longer boreholes the data may need to be synchronized by comparing the return temperature not with the current time step but with the time step—n, where n is the travel time. The error minimized is the sum square error of the difference between the calculated and measured borehole heat exchanger return temperature. We have set up the analyses procedure in such a way that it is easy to select discrete data-windows for the calibration.

The complete data series is used to calculate the temperature response, but only certain parts of the experimental data are used to calculate the error. An example of a calibration run is given in Figure 53, the final calibrated TRNSYS model run is shown in Figure 54. Using the first part of the data (with constant heat flux) an estimate of ground thermal conductivity of 2.15 was obtained. Yavatzturk's method yielded an estimate of 2.18, while the estimate obtained with the TRNSYS parameter estimation method was 2.10.

In principle all parameters of the model can be entered in the parameter estimation procedure. For the time being we limit the parameters to be calibrated to the ground thermal conductivity, ground heat capacity and borehole filling conductivity.

9.4.1. TEST PROTOCOL USING NON-CONSTANT ENERGY FLUX

In recent experiments we have started to apply a second energy pulse (either injection or extraction) after the first pulse yielded a good estimate of ground thermal conductivity using the classical line source approach (Figure 54). This second energy pulse is analysed exclusively using a numerical model and parameter estimation technique. The first idea behind this extension of the classical line source GRT is that the second pulse provides additional information on the response of the borehole. In a normal test only the first ten hours or so provide information on the borehole response. With a second pulse however, the borehole response is again important during the first hours

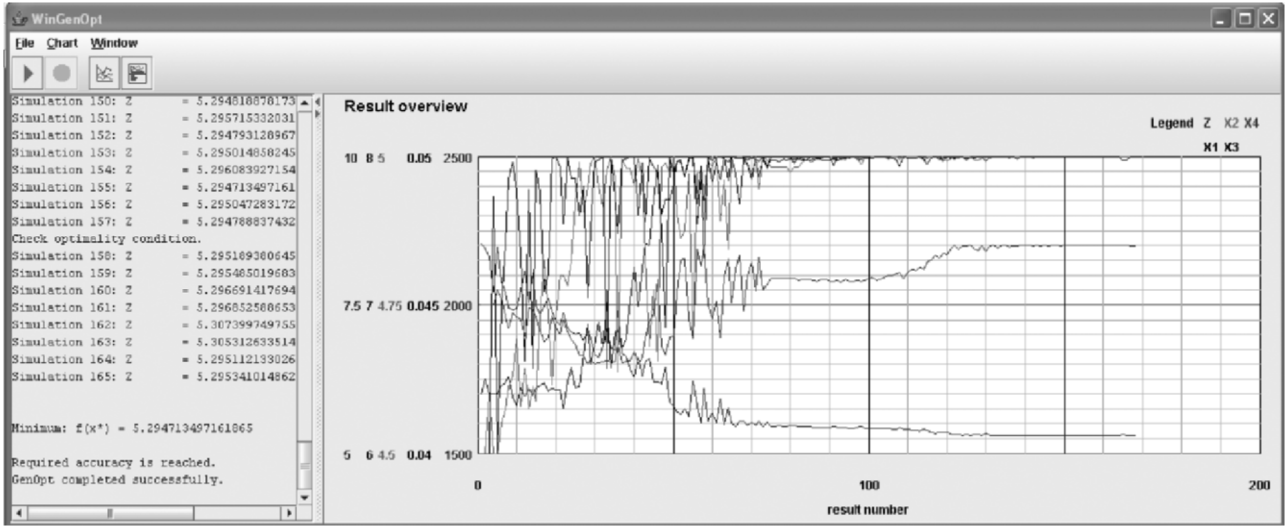

Figure 54. TRNSYS simulation, measured temperatures (red: borehole-heat exchanger T-in, blue: borehole-heat exchanger T-out) and purple: modelled borehole-heat exchanger T-out), green line is the cumulative error

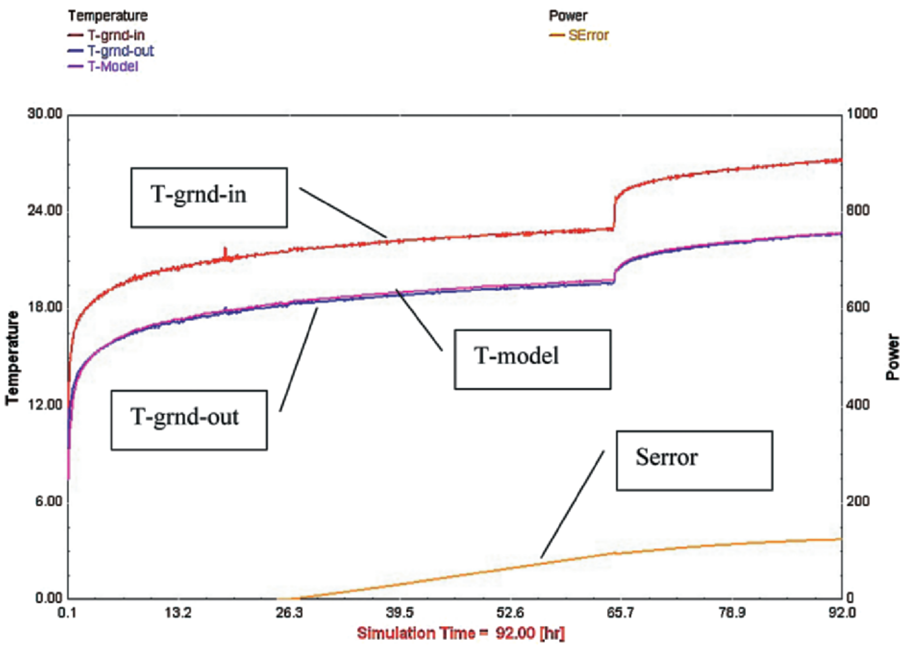

Figure 55. Geothermal response test with combined heat extraction/heat injection and different energy levels, shown are the borehole heat exchanger flow and return temperatures and the calculated heat flux. Borehole flow rate is not shown in the graph

and therefore a second estimate of borehole properties can be made. Another advantage is that the test is carried out at more than one energy level. When choosing the test parameters we in principle need to select a proper energy flux rate: too low and the error in the measurement is relatively large, too high and the ground may saturate too quickly and test length be too short. Using different energy levels means that the first test period can be more conservative, while the second pulse can be selected on basis of the results of the first pulse and therefore provides more accurate data.

Such a second energy level can be applied to either a heating or cooling GRT. However, it is also possible to combine heating and cooling tests in one test and at the same time use different energy levels (Figure 55).

The test commences with a heat extraction of about 1 kW (duration 45 h), subsequently this extraction rate was increased to ± 1.35 kW. After a total time of 60 h, a short recovery period (10 h) was allowed. During this period the heat pump was reversed and started storing heat in the buffer tank. Subsequently a heat injection of about 1 kW was started, after which there is a transient power injection with a maximum power injection rate of about 3.4 kW. The experiment settings have to be adjusted manually, therefore some control problems are obvious in the graph. For instance, when changing to heat injection initial power selected was too high and a very quick response of the borehole temperature to almost 30 °C is evident.

The first hours were run with a constant heat flux, allowing an estimate with the line source model. Results of the line source model are an estimated ground thermal conductivity of 2.05 W/m K and a borehole resistance of 0.12 K/(W/m), taking into account fluid properties, flow conditions and average shank spacing this borehole resistance equates to a conductivity of the borehole material of about 2.0 W/m K.

Using different data periods of this response data and the parameter estimation procedure with TRNSYS, we first calibrated the ground thermal conductivity, borehole conductivity and heat capacity concurrently. However, we noted that the best fit heat capacity always converged to the maximum allowed value (run 1, run 2), while at the same time the ground thermal conductivity decreased. Subsequently we therefore only calibrated on ground thermal conductivity and borehole conductivity, here the borehole thermal conductivity tended to the maximum value allowed. As a third approach we estimated ground thermal conductivity, borehole conductivity and heat capacity separately, keeping the other parameters fixed. The search algorithm was changed to Hooke Jeeves as the GenOpt Nelder Mead algorithm does not allow calibration on one parameter only. The results are summarized in Table 13.

Table 3 gives the results of the GenOpt parameter estimation (Nelder Mead Simplex parameter estimation algorithm, run 7 Hooke Jeeves) procedure. Parameters estimated are: ground thermal conductivity, borehole thermal conductivity and heat capacity ground. Shown are also the summed calibration errors (Z) for the calibration period and the error when calculating the error for the complete experiment period of 117 h. Method was: AP all parameters (conduction and capacity), λ only thermal conduction, S parameters estimated separately (order: ground thermal conductivity, borehole conductivity, ground heat capacity). Heat capacity used when not calibrating this parameter shown between brackets.

TABLE 13. Results of the GenOpt parameter estimation

Run	Type	Cal. period (h)	Z cal. period	Z total	λ ground (W/m K)	Heat capacity ground (kJ/m^3 K)	λ borehole (W/m K)
1	AP	1–40	3.25	37.853	1.48	2,399	2.22
2	AP	1–40	3.118	35.683	1.34	2,799	2.22
3	λ	1–40	3.326	38.72	1.52	(2,290)	2.22
4	λ	1–60	4.725	38.712	1.52	(2,290)	2.22
5	λ	1–117	44.529	44.529	1.55	(2,290)	2.16
6	λ	80–117	33.49	38.79	1.59	(2,290)	2.22
7	S	1–60	7.488	59.501	1.80	2,600	2.00

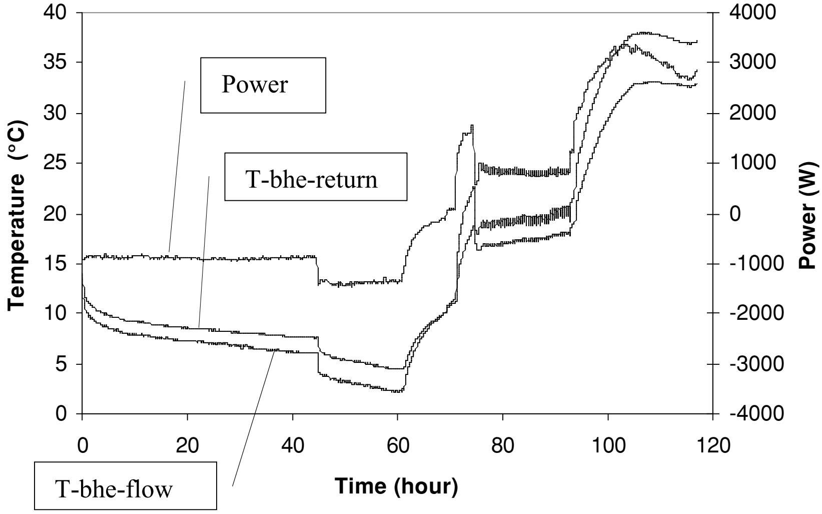

Figure 56. Temperature error between measured and calculated temperature for each timestep using the parameter values from the different calibration runs (see Table 13)

With the exception of the last calibration run (calibrating the parameters in series) all ground thermal conductivity and borehole conductivity values are very comparable. The ground thermal conductivity values estimated are significantly lower than the estimates obtained with the line source method (on the first 40 h of data). When the parameters are estimated separately, estimated ground thermal conductivity is higher.

Plotting the errors for all individual data points (Figure 56) for the different calibration runs it is clear that largest errors are associated with periods of transient heat flux. Moreover, errors during the period with heat extraction (1–40 h and 1–80 h), increase with time.

9.5. Conclusions

The Geothermal Response Test as developed by us and others has proven important to obtain accurate information on ground thermal properties for Borehole Heat Exchanger design. In addition to the classical line source approach used for the analysis of the response data, parameter estimation techniques employing a numerical model to calculate the temperature response of the borehole have been developed. The main use of these models has been to obtain estimates in the case of non-constant heat flux. Also, the parameter estimation approach allows the inclusion of additional parameters such as heat capacity or shank spacing, to be estimated as well.

We developed a parameter estimation procedure using the TRNSYS model with the DST borehole model. This model runs fairly quick compared to other numerical models, allowing rapid estimation of parameters even for larger datasets. The generic optimisation package GenOpt is used for the parameter estimation procedure. Results show that concurrent estimates of ground thermal conductivity, borehole conductivity and heat capacity is not possible, apparently due to coupling between these parameters Estimates of ground thermal conductivity are also lower than expected based on line source theory. The error of the calculation is largest for periods of transient heat flux. Although it is possible to analyse GRT data with different levels heat flux, the heat flux during any period should be maintained constant.

There are several improvements that can be made to the present calibration procedure. First of all the numerical grid in the DST model can be adjusted to yield more accurate results for short time step data. Secondly the accuracy of the calibration can be increased. At present error for any time step of the calibrated temperatures is ±0.2–0.4 K, this should probably be decreased to about ±0.1–0.2 K.

During the past several years we have applied a second heat injection or extraction pulse to the GRT test, after a sufficiently long response for the line source method had been obtained. This second energy pulse allowed us to improve characterisation of the borehole response and ground thermal conductivity. When test conditions were unknown we were able to first run a test at a low energy level and, when a first estimate had been obtained, switch to a higher level. This prevents a rapid temperature saturation of the borehole and nearby ground volume, leading to too short test times, when initial thermal properties of the ground are over-estimated. Finally the second pulse sometimes provided information on ground water flow, as the thermal conductivity estimated with the second data window was sometimes significantly higher than the estimate based on the line source method or first data window.

In this paper we present for the first time a test that combines heat extraction and heat injection pulses in one experiment. It is expected that differences in the ground thermal conductivity, when different data windows are used to obtain an estimate, can be related to advection and convection of ground water. The "real" ground conductivity should be derived from the experimental data where the response is close to or lower than the natural ground temperature, minimizing effects of advection and convection. First results, for a case of no ground water flow, show that estimates of ground thermal conductivity are very comparable for the different data windows.

After the calibration procedure and the control over the experiment have been improved, a similar experiment with enhanced ground water flow will be done to develop the method further.

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10. FREEZING PROBLEMS IN BOREHOLE HEAT EXCHANGERS

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Abstract. There are approximately 300,000 borehole systems in Sweden with an annual growth rate of 30,000 systems. Such borehole systems are mainly used for heating of single family houses but in recent years several hundred larger systems, for both heating and cooling, have been constructed. The systems delivered 16% of all space heating of Sweden in 2000 and are estimated to cover 27% in 2010. This strong development indicates that borehole systems are reliable and supply heat at a lower cost than more conventional alternatives. This paper concerns a rare freezing problem that occurs in 1 of 10,000 systems. In such cases freezing that occurs in the boreholes as a result of heat extraction creates an over pressure that flattens the pipe system, thereby stopping the circulating of the heat carrier fluid. Even if this is a rare problem it means big problems for the unlucky individuals and also for the industry since one such problem reduces the market in that region.

Keywords: borehole heat exchangers, freezing problem, principal solution

10.1. Introduction

There are presently almost 300,000 borehole systems in Sweden with an annual growth rate of 30,000 systems. The systems are mainly used for heating of single family houses but in recent years a several hundred larger systems, for both heating and cooling, have been constructed. These systems delivered 16% of all space heating in year 2000 and are estimated to cover 27% in year 2010. This strong development indicates that borehole systems are reliable and supply heat at a lower cost than more conventional alternatives. However, we have studied a rare problem where such systems do not work.

10.1.1. BEDROCK HEAT

Heat extraction from the bedrock by a heat pump system is an environmental friendly heating method where 70% of the heat is taken from the ground. This

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 193–203.

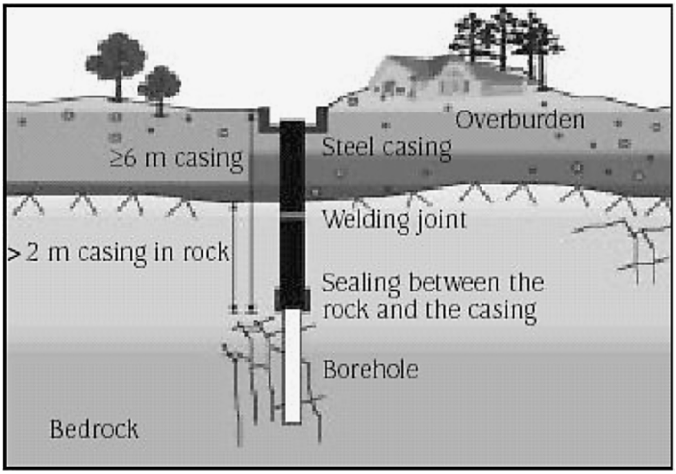

Figure 57. Outline of borehole system for heat extraction

part is renewable energy while the rest (30 percent) needs to be provided as driving energy to the heat pump.

The systems (Figure 57), which extract solar heat from the ground, consist of one or more boreholes drilled to about 150 m depth, a pipe system and a heat pump. The pipe system is installed in the borehole. By circulating brine through the pipe system, at a lower temperature than the ground, heat is flowing to the borehole from the warmer surroundings. The energy of the warmed brine is extracted by the heat pump, which also raises the temperature to the level required by the heating system.

When the brine has given its heat to the heat pump it is a few degrees colder on the way back to the rock. The heat extraction from the borehole lowers the ground temperature, but after a few years this temperature drop will reach to a point, a steady state, where it will be restored during the summers.

10.1.2. PROBLEM

In systems where the borehole temperature is below freezing (0 °C) the borehole groundwater will freeze to ice. This freezing occurs from the top and down into the borehole. This is normally not a problem and is even an advantage because ice conducts heat more easily than water.

In certain unusual cases, about 1 borehole of 10,000, the freezing will cause problems. In these cases the freezing causes a high pressure that deforms (flattens) the pipes in the borehole. In such cases some water is confined between two frozen parts (plugs) and when that water finally freezes the occurring expansion pressure becomes high enough to deform the pipes.

10.1.3. OBJECTIVE

There are a lot of different reasons how water can be confined by the ice. This article presents principal explanations of the problem and principal solutions to solve the problem.

10.2. Frozen Boreholes

When the water in the borehole starts freezing and the water turns to ice it will expand with almost 10%, which is about 11 per meter borehole. To have enough space in the borehole the water that has not yet been frozen needs to leave the borehole, which it can do through the fissures in the rock.

The normal case is that the water freezes from the surface and down into the ground, but it can also start freezing deeper down in the hole. As long as the freezing goes gradually and the water has somewhere to go it will not be any pressure increase in the borehole. The ice that already has been frozen does not cause any pressure on the surroundings and cannot damage the pipe lines.

The pressure from the freezing will only appear if water gets shut in the borehole. This can for instance occur between two plugs of ice if the borehole between these two plugs is watertight, see example in Figure 58. When water between these two plugs of ice starts freezing the expansion of the ice will

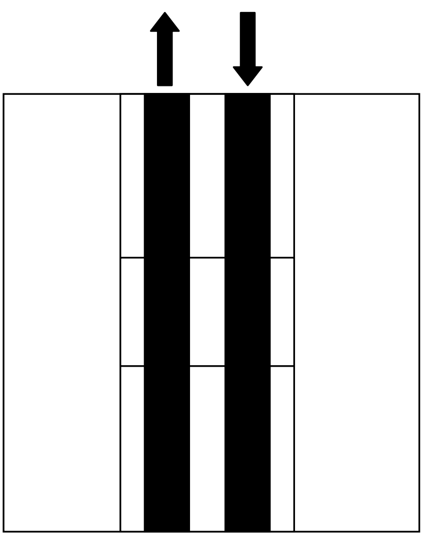

Figure 58. Section of borehole and pipe lines. A volume of water has been enclosed by two plugs of ice. The two tubes indicates cold (entrance) respective warm (outlet) pipe

cause an overpressure in the water. The pressure will be taken up by the most brittle thing, namely the pipes, and deform them. This will reduce the flow of brine in the pipes and therefore reduce the function of the bedrock heat system. Since it is unusual with totally watertight bedrock, this type of problem in most cases will appear in the casing, i.e., in the steel pipe protecting the borehole through the overburden.

There are several reasons why the freezing occurs unevenly along the borehole, for example a larger flow of ground-water in a section of the borehole will slow down the cooling of the rock. Inside the casing the water normally freezes from the top and down. If ice is also formed below the casing, water is trapped inside the casing.

It is normal that fissures in the rock drains the pressurized water and thus reduces the pressure and there is less chance for the problem to occur in the borehole. However, in the cased part of the hole there are no fissures and the water has nowhere to go.

10.2.1. SOURCE OF ERROR THAT RISE THE RISK OF FREEZING

In many cases it is taken into account already when dimensioning the borehole that it may freeze. In other cases the freezing occurs due to incorrect dimensioning, as exemplified below:

1. Miscalculation of the power and energy demand of the building.
2. Wrong assumption about thermal properties of the rock, i.e., thermal conduction.
3. Deeper soil overburden than assumed in the design.
4. Lower groundwater table than assumed in the design.

1: Miscalculation is very common, especially for older properties. It is especially difficult to estimate the energy required for cooling (heat injection into the borehole). The problem can also arise after the construction when expanding the house, which will increase the demand of energy.

2: Usually the designer of the system assumes typical bedrock for the area, which sometimes shows to be wrong. The driller will notice if it is an unusual kind of rock, but the depth of the hole is already decided and in the most cases there will not be any redesign made.

3: Since the soil normally means poorer heat conduction than the rock an unexpected large depth of soil results in a too short borehole. This will lead to under dimensioning and a risk of freezing in the casing.

4: The groundwater level is important in the design of borehole systems since the depth of the borehole below the groundwater level is the so-called active borehole depth, i.e., the part of the hole that provides the heat. If the groundwater level is lower than estimated the system will be under dimensioned. It can sometimes be solved by filling up the borehole with water or to fill it up with sand or bentonite.

If only one of these problems occurs it will not be a big problem, but if more than one problem occurs it will lead to under dimensioning and freezing will take place. The easiest and most common ways to solve the problem is either to drill the borehole deeper, drill a new borehole or to reduce the heat extraction. To reduce the heat extraction an additional heater must be used.

Except for these four problems, there can also be other reasons for freezing and often they occur together, like:

- *Varying flow of groundwater:* If the horizontal groundwater flow varies along the vertical borehole, it will supply correspondingly more or less heat to the borehole, resulting frozen and unfrozen parts along the borehole and in the casing. This would cause freezing problems in the casing but not in the rock since areas with greater groundwater flow indicates fissures in the rock.
- *Water tight borehole wall:* A water tight borehole would mean the same problem as in the casing, but overpressure is more likely to occur in the casing.
- *Low flow in the brine filled pipe system:* A low flow rate though the borehole pipes indicates a lower fluid temperature and a greater risk of freezing.
- *Varying rock thermal conduction:* If there is one type of rock that is conducting more heat than the surrounding rock, the rock with greater heat conduction will stay warmer than the surrounding one. In such cases water would be trapped and overpressure would occur when freezing.
- *Neighboring systems:* If a neighboring borehole system is located closer than 15–20 m the system will influence each other, which shows in a lower ground temperature than in a single borehole case. The lower ground temperature increases the risk of freezing.

10.2.2. METHOD TO COUNTERACT FREEZING PROBLEMS

The easiest way to prevent problem with freezing is to avoid temperatures below freezing. If freezing is allowed there are many ways to control, reduce or to drain the pressure. Figure 3 shows some examples of what can be done.

The examples below and Figure 59 shows how to avoid problem with freezing in borehole, but in particular what to do about the problem that already exists. In the principal drawings (Figures 60–67), which outlines different ways to solve freezing problems, the scales are incorrect.

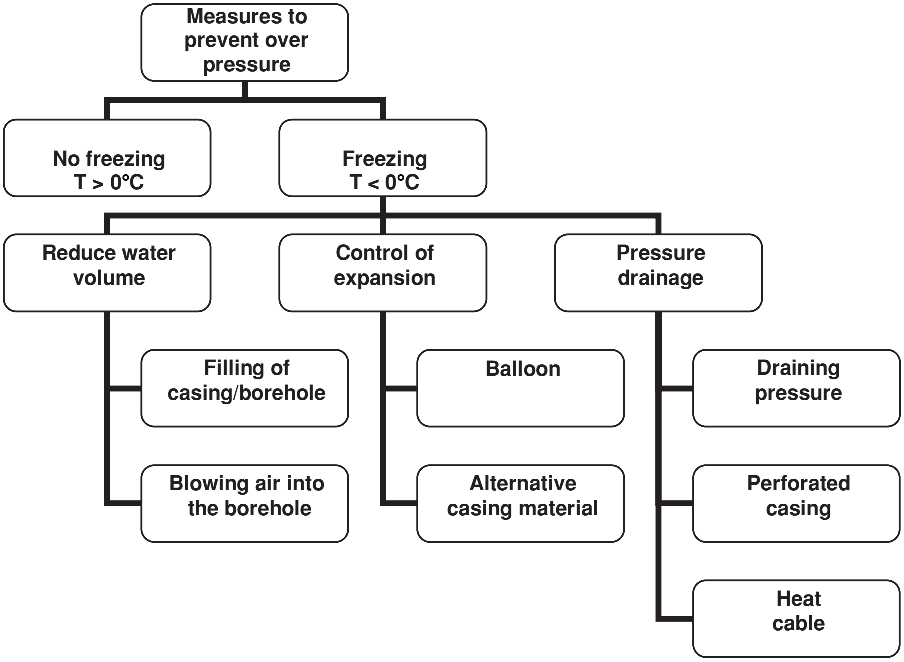

Figure 59. Overview over measures that can be done when freezing in borehole damage the tubes of brine

10.2.2.1. *Filling of Casing/Borehole*

By filling the casing with a material that replace the water it will not be possible for the freezing pressure to occur. It would be possible to use a material like Styrofoam, Figure 60. Even if not all water is replaced the freezing pressure will be substantially reduced.

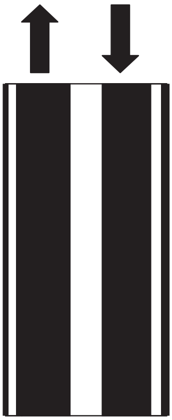

Figure 60. Borehole top where the casing through the soil is thermally insulated

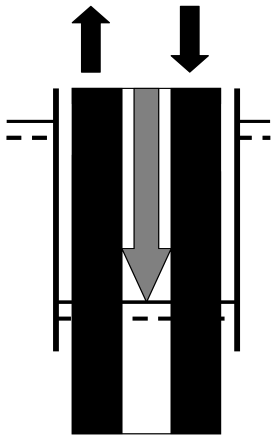

Figure 61. Lowering of the groundwater level. By pumping in air at top of the borehole

This kind of filling will reduce the heat transfer from the ground through the casing. This means that it will be necessary to drill a deeper borehole.

One alternative would be to replace water by filling the borehole with cement. Cement has almost the same thermal conductivity as ice, which is 3–4 times greater than water. This will increase the capacity of the hole. It is also possible to fill the hole with sand instead of cement.

10.2.2.2. *Blowing Air into the Borehole*

With an airproof lid on top of the borehole and an air valve it would be possible to pump air into the casing. This way the groundwater level will be lowered, to preferably a bit above the lower edge on the casing as shown in Figure 61.

It is not possible to have freezing problem when the tubes are surrounded by air, so the lower level in the casing will minimize the risk of a freezing problem in the casing. This solution means that almost no heat will be extracted from the soil (through the casing) and that a deeper borehole in the rock is required.

10.2.2.3. *"Balloon"*

It is possible to install an air filled soft tube ("balloon") in the borehole between the two other tubes as shown in Figure 62. This elastic balloon is compressed more easily than the brine filled pipe and will therefore take the pressure from the formation of ice. The amount of water will also decrease because of the space of the balloon, which decreases the risk of freezing.

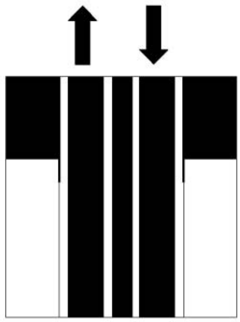

Figure 62. A "balloon" between the tubes that take the pressure when freezing occurs

10.2.2.4. *Alternative Casing Material*
If the casing is manufactured in another material with more elastic properties, the casing will be pressed out instead of having the tubes with brine squeezed together, see Figure 63.

The ordinary casing will be outside the elastic one with air between and welded together to prevent water leakage. When the ice melts, the material will go back to its original shape again.

10.2.2.5. *Draining Through a Pipe*
The pressure in the borehole can be reduced by having a tube of elastic material filled with silicone oil. When a slight overpressure due to the freezing occurs the silicone oil will be pressed away to an expansion tank, see Figure 64.

10.2.2.6. *Perforated Casing*
By perforating the casing with millimeter holes a small over-pressure will force the water to flow out, as shown in Figure 65. With this solution over-pressure can never occur. This very attractive solution might not be all that legal (for now).

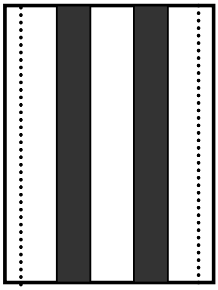

Figure 63. The elastic casing material will take up the pressure

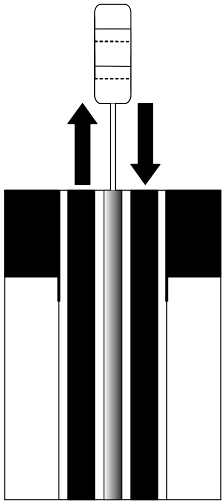

Figure 64. The pressure from the freezing will be taken care of by the silicone oil and regulated by an expansion tank

10.2.2.7. *Heat Cable in the Borehole Connected to an Expansion Tank*

By inserting a heat cable in the borehole it is possible to prevent the overpressure, see Figure 66. The heat cable should be connected to the brine expansion tank. When the expansion of freezing water increases the volume in the expansion tank the heating starts, which makes it possible to always have a drainage hole though the ice.

The disadvantage with this solution is that you must supply heat to extract heat. This is only an emergency solution for constructions that already have this problem and should not be considered as a standard solution.

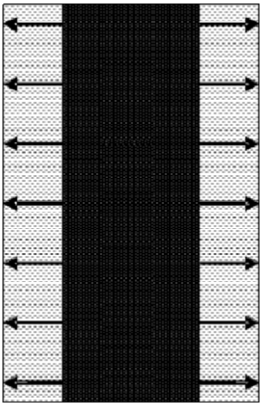

Figure 65. Perforated casing that allows the water to trickle out when overpressure occurs

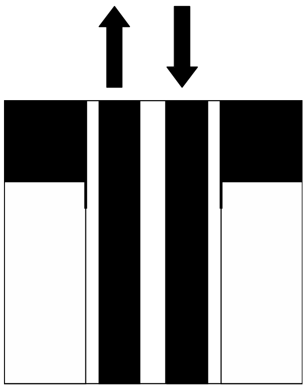

Figure 66. Heat cable in the casing on the cold side of the tube

10.2.2.8. *Temperature Indicator in the Borehole*

To get a better understanding of the problem with freezing a temperature indicator could be placed along the collectors to monitor the temperature during heat extraction, as shown in Figure 67.

This way it is possible to understand where the problem occurs and makes it easier to find the best solution to solve the problem.

10.2.3. CONCLUSIONS

The studied freezing problem in borehole systems is rare, but it causes big problems for those individual who are affected. It is also a problem for the

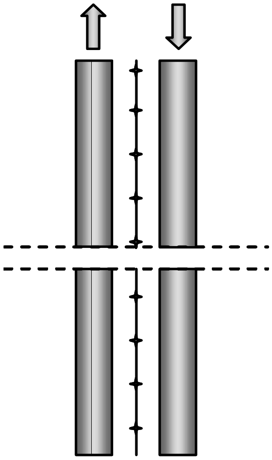

Figure 67. Temperature giver placed between the tubes in the borehole

companies that are delivering the systems when bad reputation reduces the market in the neighborhood.

So far there has not been any solution for this problem. In this study however, the problem is analyzed and explained. The risk increases with the soil depth, i.e., long casing. Some of the suggested principal solutions will solve the freezing problem by controlling, reducing or draining the pressure.

In new systems when there is a high risk of freezing problems to occur some of the suggested measures should be used.

Reference

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11. THREE YEARS MONITORING OF A BOREHOLE THERMAL ENERGY STORE OF A UK OFFICE BUILDING

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Abstract. In the autumn of 2000, the British Engineering Council awarded an Environmental Engineering award to the groundsource heatpump project at Commerce way, Croydon Surrey. This, one of the larger UK groundsource projects, is a speculative built industrial building of about 3,000 m^2 with both offices and warehouse facilities. The building, that is leased by Ascom Hassler Ltd. (a Swiss based IT company), is expected to have an annual cooling load of 100–125 MWh and a heating load of 90–100 MWh. Peak loads under hot summer conditions are anticipated to reach up to 130 kW. During the normal life span of a building (25 years) the surplus of heat rejection would lead to increasing ground temperatures. This results in a less efficient heat pump operation and may even result in insufficient capacity during cooling peak demands. As a solution a hybrid system, incorporating a dry-cooler, was developed. The principal idea is to use the dry-cooler to store cold in the wellfield during early spring, when the required summer peak load cool can be generated very efficiently and cheaply. The operation and efficiency of the wellfield, the installed heat pump system and dry-cooler is controlled and monitored under a Building Management System (BMS). The results of the first three years of operation of the system are presented. Using the monitoring data an evaluation of the original design will be made.

Keywords: heat pump, geo exchange, ground source, borehole heat exchanger, monitoring, croydon

11.1. Introduction

In the Croydon project, one of the larger UK ground source projects to date, the client (AXA Sunlife) opted for a low temperature geothermal energy system as the capital cost were only slightly higher than for traditional HVAC systems (VRV, four pipe fan coils). Using the Building Management System

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 205–219.

several key variables have been monitored, including borehole circulation medium temperatures and ground temperatures. In this paper we present an overview of these monitoring results for the period 2000–2003. Using these measurements an attempt will be made to evaluate the original design.

At the core of any geo-energy system is a heat pump. A heat pump is a machine that can transport heat (thermal energy) from a low to a high temperature level, hence the term "pump". Heat pumps are very attractive as a technology for space conditioning (heating and/or cooling). In fact, in a study conducted in 1993 the US EPA concluded: "GeoExchange systems are the most energy-efficient, environmentally clean, and cost-effective space conditioning systems available". One of the most important benefits of heat pump technology is that the transfer of energy occurs with a very high efficiency. For every kilowatt of electrical energy used to drive the compressor (and secondary side circulation pumps) five or six kilowatts of heat or cool is generated. The efficiency of a heat pump is expressed as it's COP, the ratio between the electrical power consumption and thermal energy generated. Modern heat pumps have typical COP's between 4 and 5 in heating mode and between 3.5 and 4 in cooling mode. This makes a heat pump an economically interesting machine, as the thermal energy taken from the environment comes at no additional cost. Moreover, savings on primary energy and greenhouse gas emissions can be realised. Studies carried out by the IEA for instance have demonstrated that savings of up to 50% of primary energy are possible. With respect to CO_2 reduction the global emissions reduction potential has been estimated at 6% in 1997. In addition to the savings on primary energy consumption and greenhouse gas emissions heat pump technology offers other advantages. The attractiveness of the ground source system lies in the low running cost, low maintenance, emission reductions, small plant room and lack of external plant, absence of sound emissions and of course the marketable sustainable "green image". This potential of ground source is recognised by high-end real estate developers and more progressive architects and consultants.

The heat pump exchanges thermal energy with the building on one side, and with the environment on the other side. In a geo-energy system the environment is the ground (other possibilities are e.g. air or surface water). The ground is particularly suited for low temperature energy exchange: the usual operating temperature bandwidth is between -5 °C and 30 °C (not taking into account high temperature energy stores). The ground can be used either through a closed loop borehole heat exchanger, through direct use of ground water or even through direct expansion systems. Advantages of closed loop borehole heat exchangers are their very long life span (50 years or more) and the fact that they are virtually maintenance free.

The goal of a design of a geothermal heat exchanger is to maintain a specified bandwidth of temperatures in the ground loop heat exchanger at which the heat pump may operate efficiently. The main issue with such a design is that we need to consider both the local process, occurring in and around the borehole, and the global process of the whole ground volume that is thermally influenced and its interaction with the boundaries. Especially for the local process, the earth is usually not capable of transporting the thermal energy rapidly enough to the heat exchanger. This means that the ground temperature around the heat exchanger tends too increase or decrease as long as the heat pump is in operation. For the global process, in a system utilising the ground for both cooling and heating there may either be a balance or imbalance in the total energy. In the first case the ground temperature will tend to show a yearly fluctuation around a mean value, in phase with the energy demand. In the latter case these fluctuations are also present, but the average ground temperature will tend to increase or decrease with time. The interactions with the boundaries include energy exchange due to conduction, groundwater flow, geothermal gradient and interactions at the surface.

In such a system there is a direct and dynamic coupling between the energy supplier (the ground) and the energy user (the building). Not only are the loads on the ground system determining the thermal response of the ground, the actual power generated will in turn depend on the source temperature. The design therefore needs to specify accurately the total seasonal loads that determine the global response, as well as the peak load demands and duration. These peak loads are superposed on the seasonal response of the system.

For a successful design detailed knowledge of the building loads as well as detailed geological and geo-hydrological knowledge are necessary. Moreover, the actual engineering of the boreholes and ground loop heat exchangers, material choice, etc all has to be considered carefully.

We will not consider the building load design in any detail. For the ground source design the principal input parameters are the seasonal loads and peak loads and duration. Several methods and models exist to calculate the energy requirements of a building (e.g. Ashrae, 1998; Blast, 1986). Main uncertainties are the climatic circumstances and the variations therein and the actual building use during its lifespan.

With respect to heat conduction in the ground the most important parameter is the thermal diffusivity of the ground, the ratio between the ground thermal conductivity and the volumetric heat capacity. Especially the thermal conductivity is difficult to establish with sufficient accuracy (Austin, 2000; van Gelder et al., 1999). Although each soil type has a specific conductivity, the conductivity depends not only on the material itself but also to a large extent on factors such as packing, pore volume and water content. Moreover, the sequence of soil types and presence of ground water or ground water flow in the different formations in the soil profile will affect the overall conductivity and relative contribution to the thermal forcing of the different depth intervals. As the soil conductivity is so important, several methods have been developed to directly measure the overall soil conductivity (Eklöf and Gehlin, 1996; Austin, 1998; Witte et al., 2002).

For the local process especially the borehole resistance is important. The borehole resistance depends mainly on the loop type and material, loop dimensions, circulation fluid properties, temperature of the process, borehole engineering (Hellström, 1991). Furthermore the far field temperature in the ground and geothermal gradient needs to be measured.

11.1.1. THE CROYDON PROJECT

The Croydon building (Figure 68) is a three-story office building located at the Commerce way, Croydon Surrey (UK). Total surface area is about 3,000 m^2, with both offices and warehouse facilities. In the offices 85 Geothermic water-to-air heat pumps have been installed. The warehouse, of approximately 690 m^2 is heated or cooled using a low temperature under floor heating, with to a water-to-water heat pump (26 kW). Total installed heating capacity is 225 kW, maximum cooling capacity installed is 285 kW.

As all Geothermic heat pumps are connected in parallel to the pipe work supplying the source and return water, therefore simultaneous cooling and heating loads are balanced in the building. During periods with a net cooling or net heating demand the ground heat exchanger (Figure 69) supplies the

Figure 68. The Croydon building, Commerce way, Croydon Surrey (UK)

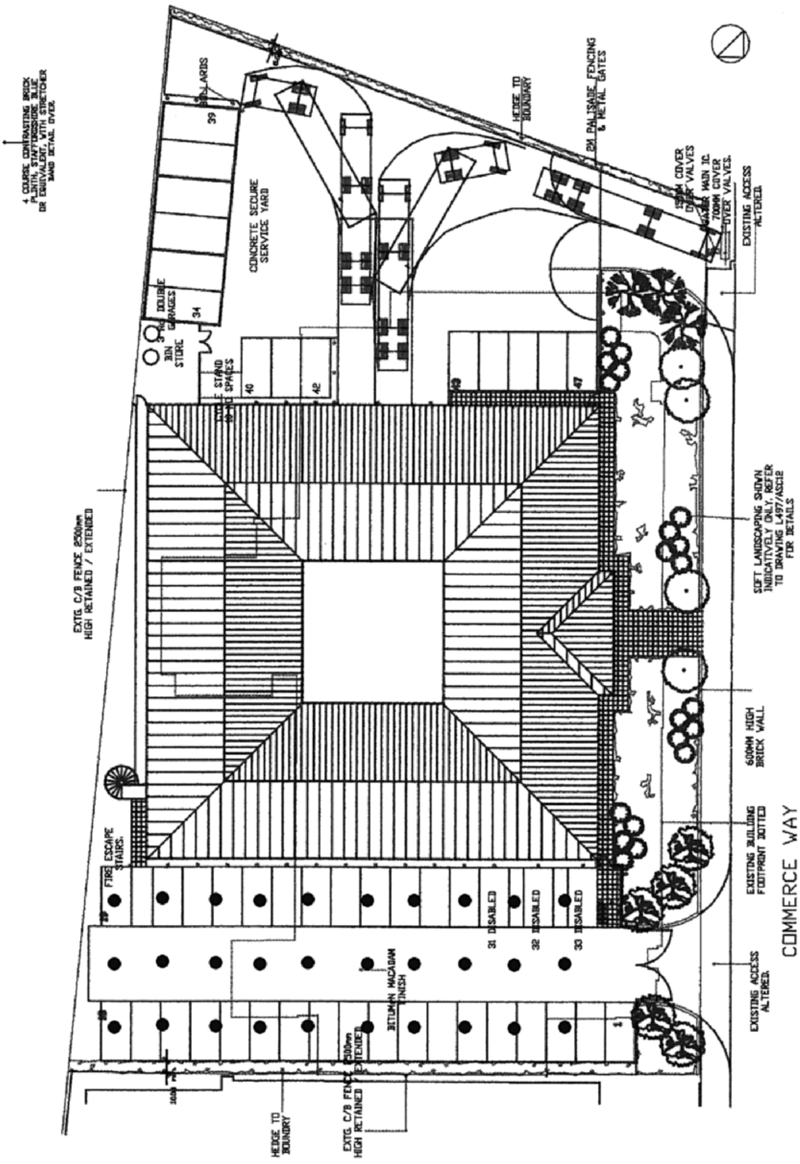

Figure 69. Location map showing the building and parking lot with positions of the installed ground loop heat exchangers and observation wells

additional heat or cool. The ground loop heat exchanger consists of thirty U-loops (40 mm, PE PN16) installed in 100 m deep boreholes, with a distance between the boreholes of about 5 m. Boreholes were fully grouted with a bentonite/cement mixture.

11.1.2. BOREHOLE HEAT EXCHANGER DESIGN

Geology of the site has been described on the basis of three borelogs of the Geological Survey (British Geological Survey, 1997) in the vicinity and on the basis of the borelogs made during the drilling of a test borehole. Main geological sequence is: a surface layer (1.5 m), Thanet Sands (1.5–13 m), Upper Chalk (13–80 m) and Middle Chalk formation (80 m). Groundwater levels are at about −2 m with respect to the surface level. Groundwater flow is in a north to northwesterly direction with an estimated Darcy flow of 20–25 m/year in chalk formations and of between 75 and 100 m/year in Thanet sands.

Critical design parameters are the ground thermal conductivity, far field temperature and energy loads. The thermal ground parameters were measured on-site with an In Situ Thermal Response Test [4]. Results of this test (Figure 70) showed a thermal conductivity of 2.2 W/m K, an average far field temperature of 11.6 °C and a geothermal gradient below 40 m of approximately 0.009 °C/m.

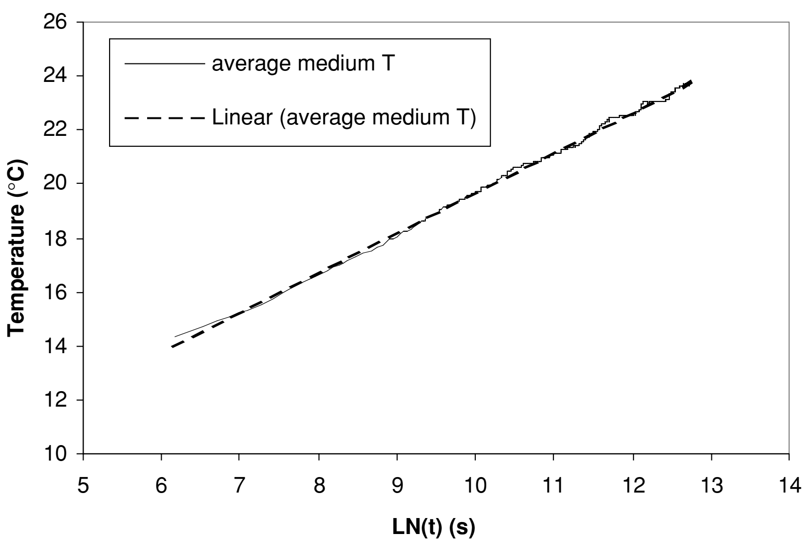

Figure 70. Result (average circulation medium temperature with LN(time)) of the In Situ Thermal Response Test for the Croydon test site

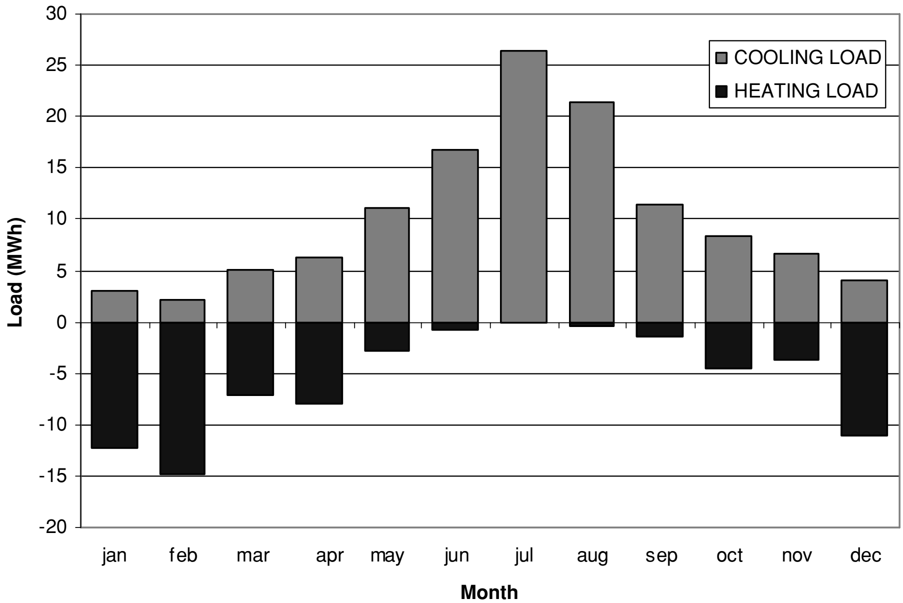

Figure 71. Design loads building (net loads to the ground), monthly heating and cooling loads

Building loads were calculated by EDSL (Environmental Design Solutions, Milton Keynes, UK) using a three-dimensional dynamic model. Total yearly loads have been calculated as 100 MWh/year (average summer) up to 120 MWh/year (warm summer) cooling and about 95 MWh/year heating. Taking into account the average efficiency of the heat pumps, this translates to a load on the ground of 65 MWh heating and between 120 and 145 MWh cooling. Seasonal distribution of the average loads is depicted in Figure 71.

Evident from Figure 4 is that even in winter appreciable cooling occurs and that only in July no heating demand is present. The total net load on the ground during an average year is 55 MWh annual heat rejection.

Average monthly fluid temperatures (Figure 72) in the ground loop heat exchanger were modelled using Earth Energy Designer (EED, Eskilson et al., 2000) and GhlePro (Spitler, 2000). Clearly average temperatures in the ground tend to increase, from about 13.5 °C in the first year to about 17 °C in the twenty-fifth year. The installed Geothermic heat pumps can operate efficiently within a temperature bandwidth of 0–30 °C, and design temperatures were selected accordingly. The limiting design temperature is the temperature during cooling peak loads, with a 30 °C limit.

Superposing a 136 kW cooling peak load on the modelled average medium temperatures a 7 h peak load can be accommodated in year 5, in year 10 only 5 h and in year 25 just 2 h can be accommodated.

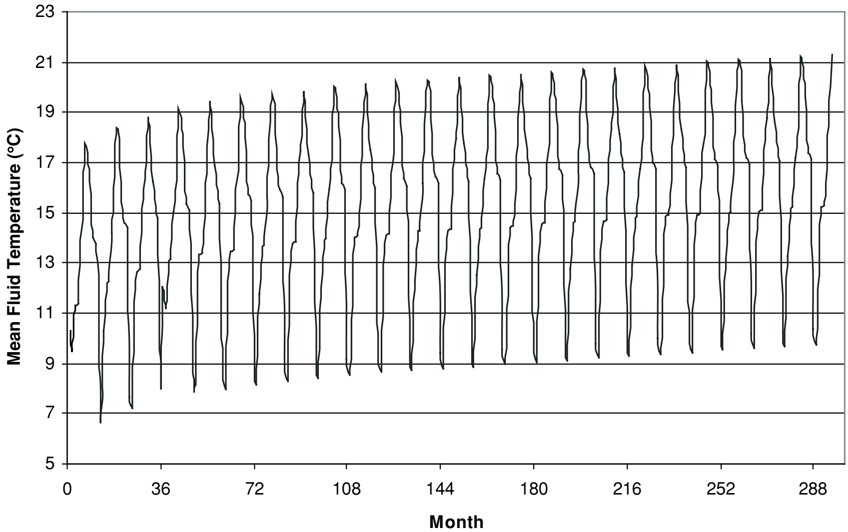

Figure 72. Average modelled monthly fluid temperatures in the ground loop heat exchanger, for 25 years

Instead of increasing the size of the ground loop-heat exchanger a dry cooler was incorporated in the system. Depending on the operating temperatures (source and sink) a dry cooler can very efficiently reject heat. The principal idea is therefore to use the dry cooler to store cold in the ground during times when this can be done at high efficiency (spring and at night). Such a hybrid system allows more flexibility in building use (adding or removing heat pumps), and makes an active management of the ground temperatures possible. Also, a dry cooler is a relatively cost-effective way of rejecting heat.

11.2. Monitoring Results

A number of variables have been monitored with a relatively high frequency from September 2000 until July 2003 (20 min interval). In this paper we will focus on the borehole heat exchanger temperatures, thermal loads on the ground and ambient temperatures. Using the borehole flow and return temperatures the load on the ground at each time step was calculated using the known fluid properties and (fixed) pumping rate. The building was commissioned in September 2000. The monitoring data from September 2000 up to July 2003 is now available. In July 2003 the tenant suspended UK operations in response to market developments. As the lease still runs and no new tenant is using

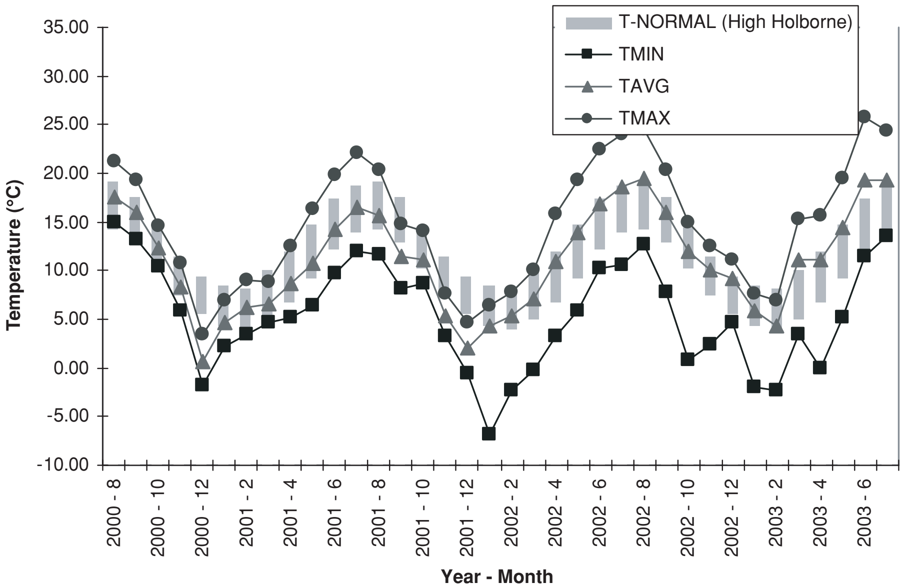

Figure 73. Average monthly day-minimum, day-maximum and day-average temperatures at Croydon and average typical temperature range of the climate station High Holborne London Weather Centre—(design temperatures)

the building, no new data is being gathered at the moment as all systems are on standby. During the operational period the dry cooler has not been used, energy demands presented therefore only refer to the building.

The main questions we would like to address are:

1. How well do the actual measured building loads compare with the design building loads, taking into account possible differences in climate.
2. How well does the borehole heat exchanger design match with the measured temperatures, and what impact do the measured loads have on the design?

Ambient temperatures (Figure 73) measured at Croydon are by and large comparable to the design temperatures at High Holborn. It can be noted that the winters of 2000 and 2001 have been somewhat colder than normal, while summers (and especially maximum temperatures reached) have been quite a bit warmer in 2002 and 2003. The difference between minimum and maximum temperatures is much larger, as in the monitored data the temperatures are not averaged over a period of more years.

Daily ambient and borehole temperatures monitored at the Croydon facility (Figure 74) also show a clear trend. Design limits of the borehole heat exchanger of 0 °C during heating and 30 °C during cooling have not been exceeded, but average winter temperatures have increased during the season

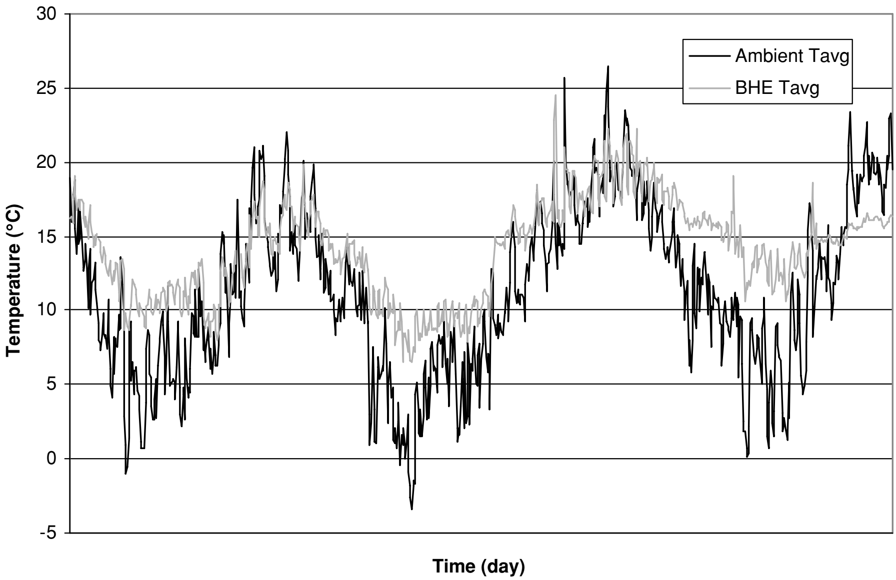

Figure 74. Average daily ambient and borehole heat exchanger temperatures at Croydon (period 2001–2003)

2002–2003 with respect to previously recorded values. Minimum heat pump source temperature (ground loop heat exchanger return temperature) recorded throughout the period is 4.2 °C, maximum heat pump source temperature was 26.8 °C. The loads that have been recorded are shown in Figure 75. Clearly the net cooling loads to the ground greatly exceed the heating loads. During the monitoring period total heat extraction was 140 MWh (200 MWh design), total heat injection was 441 MWh (368 MWH design). Heating loads are 30% lower than anticipated while cooling loads are 20% higher. The original imbalance factor (cooling/heating) was 1.84, in reality the imbalance is 3.15.

A direct comparison of the measured and design monthly loads is depicted in Figure 76. Here also, the trend of heating loads being smaller than anticipated while cooling loads are higher than anticipated is very clear. One of the main reasons for the difference in loads is the fact that occupancy level (number of people per m^2) was higher than anticipated.

The question of how well the borehole heat exchanger system holds up under these conditions is an interesting one. We compare the measured temperatures in the borehole heat exchanger system with the predicted temperatures using the design and measured loads. The program Earth Energy Designer (EED) was used to calculate temperatures for the first year, the analysis of the complete monitoring period was done using TRNSYS with the DST (Hellström, 1989) model. Figure 77 shows the simulated temperatures using the design and measured building heating and cooling loads.

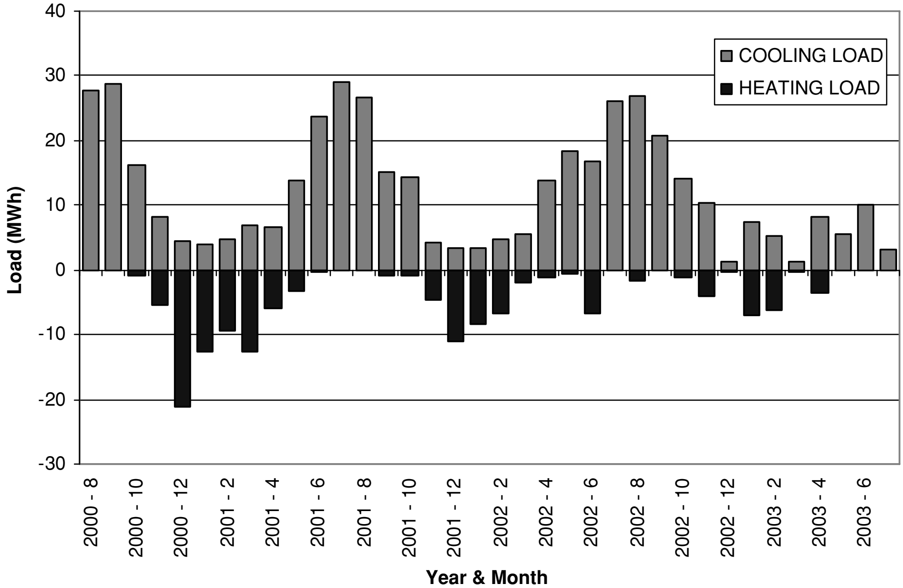

Figure 75. Monthly heating and cooling loads (measured loads to the ground) at Croydon (period 2001–2003)

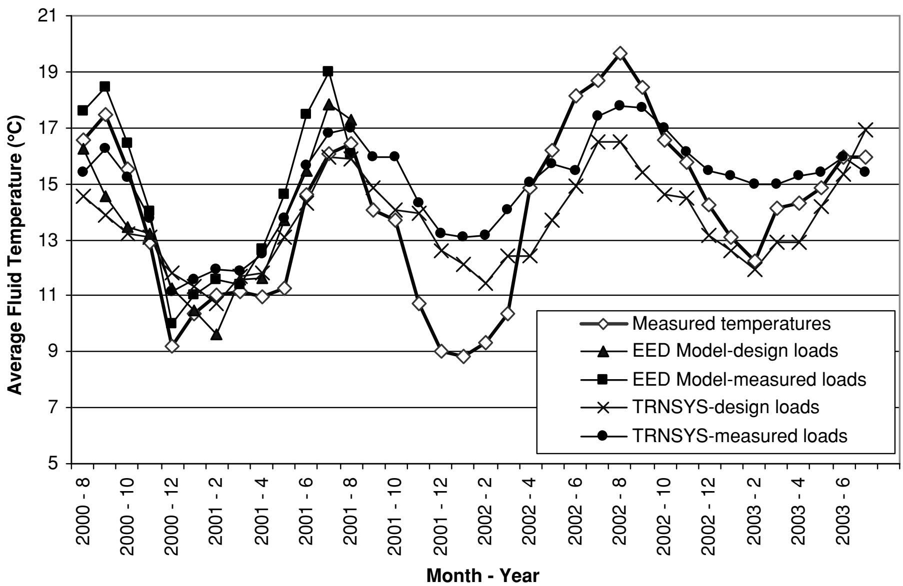

Figure 76. Comparison between design and measured monthly loads (period 2001–2003)

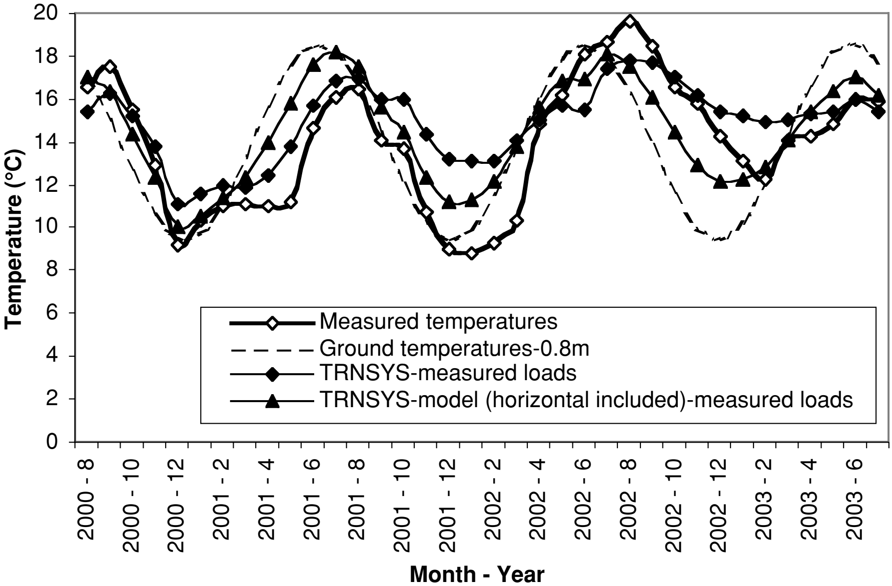

Figure 77. Comparison of measured and modelled borehole heat exchanger temperatures (period 2001–2003)

Some interesting observations can be made from these graphs. First of all, the measured temperatures and the calculated temperatures for the first year are all in fair agreement. EED tends to overestimate temperatures during cooling when using the measured loads. The TRNSYS model, that has been used to predict temperatures for the complete monitoring series, predicts the increasing average store temperature relatively well. The measured fluid temperatures however, show a larger temperature amplitude than the model calculations. Also, fluid temperatures in the 2001–2002 winter season are much lower than expected considering the measured loads.

As the larger temperature amplitude seems to be a temperature effect (and not a thermal load effect, which would mean larger temperature differences as well), and the low temperatures in the 2001–2002 winter season correlate with the low ambient temperatures during this period, it is thought that the horizontal connecting pipes may experience a larger than expected seasonal temperature effect. In the Croydon Borehole Heat Exchanger well field a total of about 1,200 m of horizontal pipework has been laid, beneath a black asphalt surface. We extended the TRNSYS model to include the effects of the horizontal pipes, using the average ambient temperature and average temperature amplitude to calculate the ground temperature at the depth of the horizontal pipes (±0.8 m below surface). The results are show in Figure 78.

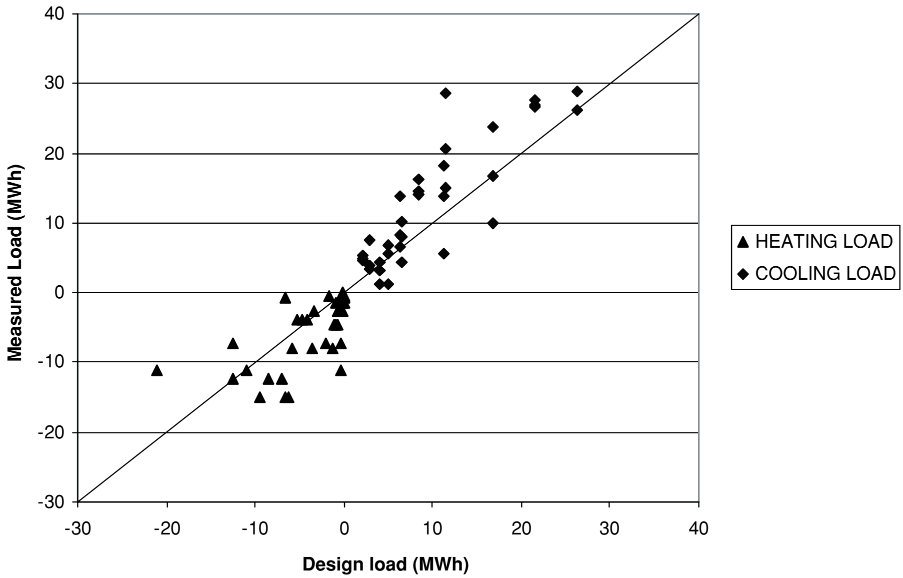

Figure 78. Comparison between TRNSYS model with and without horizontal connecting pipes, measured and modelled borehole heat exchanger temperatures (period 2001–2003). Also shown is the undisturbed ground temperature at depth of horizontal pipes

From this figure it is clear that a much better fit is obtained. The fluid temperatures during the summer season 2001 and in the winter season 2001–2002 are still too high. This can be explained by taking into account that the real ambient temperatures during those years have not been used in the model, only average atmospheric temperatures. Fit in the winter season 2002–2003 is excellent.

11.3. Conclusions

The borehole heat exchanger at Commerce Way (Croydon, UK) has, during its 36 month of operation, rejected 440 MWh of heat into the ground and extracted 140 MWh. In spite of the large difference between anticipated and actual cooling and heating loads, cooling loads being 20% higher while heating loads were 30% lower, the design temperature limits have not been exceeded. The average store temperature appears to be increasing due to the imbalance in heating and cooling loads. This was anticipated during the design, and a dry-cooler was incorporated to reject excess heat in the winter/early spring period when conditions are favorable. It had been expected to start operation of the dry cooling in 2004/2005. However, due to changes in the market the activities of the tenant at the Croydon office were suspended, at present the offices remain closed and no heat is rejected or extracted. Unfortunately, further monitoring of data had to be suspended as well.

A first comparison between measured and modelled temperatures has been made, using standard software such as EED and TRNSYS. During the first year of operation measured and modelled temperatures agree fairly well. After the first seasonal cycle results are less comparable, the measured fluid temperatures showing a much wider amplitude. Including a model of the horizontal connecting pipes in the model greatly improved the fit. Given the type of surface (asphalt), the influence of the seasonal temperature changes in the shallow ground may be much bigger than anticipated. Normally the influence of horizontal pipes is not considered to be very great, as the conductivity in the usually dry zone where the pipes are buried limits the downward migration of the temperature pulse at the surface. However, the thermal properties of the surface as well as the shallow ground will influence the temperature amplitude experienced there. In these cases (e.g. horizontal pipes under asphalt) additional insulation may be required. Although the energy exchanged is not affected, the coefficient of performance will be affected as the heat pumps will operate less economically with the higher temperatures in summer, or lower temperatures in winter.

In this paper we have presented a general overview of the monitoring data. The dataset provides more detail, both in several additional systems components that have been monitored as well as in the frequency of logging. In the future a more detailed statistical analysis of the high-resolution dataset (20 minutes logging interval) will provide more insight in the behaviour of the system during daily cycles.

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12. A UNIQUE BOREHOLE THERMAL STORAGE SYSTEM AT UNIVERSITY OF ONTARIO INSTITUTE OF TECHNOLOGY

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Abstract. The borehole thermal energy storage system (BTES) at University of Ontario Institute of Technology (UOIT) is described and its technical details are presented from the energy conservation point of view. An illustrative example is given to demonstrate performance aspects of the of the heat pump system integrated with it. The BTES system forms an important component of the drive for an efficient campus, and is also used in UOIT's energy-related engineering programs and research.

Keywords: case study, application, university, thermal energy storage, data analysis

12.1. Introduction

Energy storage technologies are usually a strategic and necessary component for the efficient utilization of renewable energy sources and energy conservation. Thermal energy storage (TES) serves at least three different purposes [1]: (i) energy conservation and substitution (by using natural energy sources and waste energy), (ii) energy peak shifting (from more expensive daytime to less expensive nighttime rates), and (iii) electricity conservation (by operating efficient devices at full load instead of part load to reduce peak power demands and increase efficiency of electricity use).

The benefits of TES include [2]: (i) reduced energy costs, (ii) reduced energy consumption, (iii) improved indoor air quality, (iv) increased flexibility of operation, (v) reduced initial and maintenance costs, (vi) reduced equipment size, (vii) more efficient and effective utilization of equipment, (viii) conservation of fossil fuels, and (viii) reduced pollutant emissions.

Various TES systems have been used in practice. Underground thermal energy storage systems may be divided into two groups [3]: (i) closed storage systems in which a heat transport fluid (water in most cases) is pumped through heat exchangers in the ground, and (ii) open systems where groundwater is

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 221–228.

pumped out of the ground and then injected into the ground using wells (aquifer TES) or in underground caverns.

The University of Ontario Institute of Technology (UOIT) is striving to becoming an innovator in engineering, driven by the strength of its programs and research. A major thrust has been established at UOIT in the area of energy engineering, and is aimed at addressing many of the present and future energy challenges facing society. One particular area of energy research is on thermal energy storage and this work is being greatly facilitated by the availability of a leading-edge, on-site borehole thermal energy storage facility. This is of course a critical component of the university's heating and cooling system, and helps keep costs down and efficiency up. In addition, the thermal storage system is used for research and to educate students in thermal energy storage in various courses, e. g., thermodynamics, as well as in graduate courses and thesis work in the future.

Although there are some underground thermal energy storage applications in Canada, such as those at Scarborough Centre in Toronto, Carleton University in Ottawa, the Sussex Hospital in New Brunswick and Pacific Agricultural Centre in Agassiz, B.C. [4], the UOIT borehole thermal energy storage project is unique in Canada in terms of the number of holes, capacity, surface area, technology, etc. Borehole thermal energy storage, which is similar to a borehole geothermal system, involves storage and provides for both heating and cooling on a seasonal basis. Large-scale storage systems, comparable to the UOIT one, have been implemented at Stockton College in New Jersey, USA and in Sweden [4].

In this study, a closed BTES using boreholes at UOIT is described and its technical details are presented, along with an illustrative example providing a performance analysis of the heat pumps.

12.2. System Description

The underground borehole thermal energy storage system considered in this article is installed at the University of Ontario Institute of Technology (UOIT) in Oshawa, Ontario, Canada. The UOIT campus includes four new buildings that are designed to be heated and cooled, sometimes with renewable energy in order to minimize greenhouse gas emissions. Test drilling programs were carried out to determine the feasibility of thermal storage in the overburden and bedrock formations at the UOIT site. In-situ tests were conducted to determine the groundwater and thermal characteristics. An almost impermeable limestone formation was encountered from 55 m to 200 m below surface, as shown in Figure 79. The homogeneous, non-fractured rock proved to be ideally suited to thermal energy storage since there is virtually no groundwater

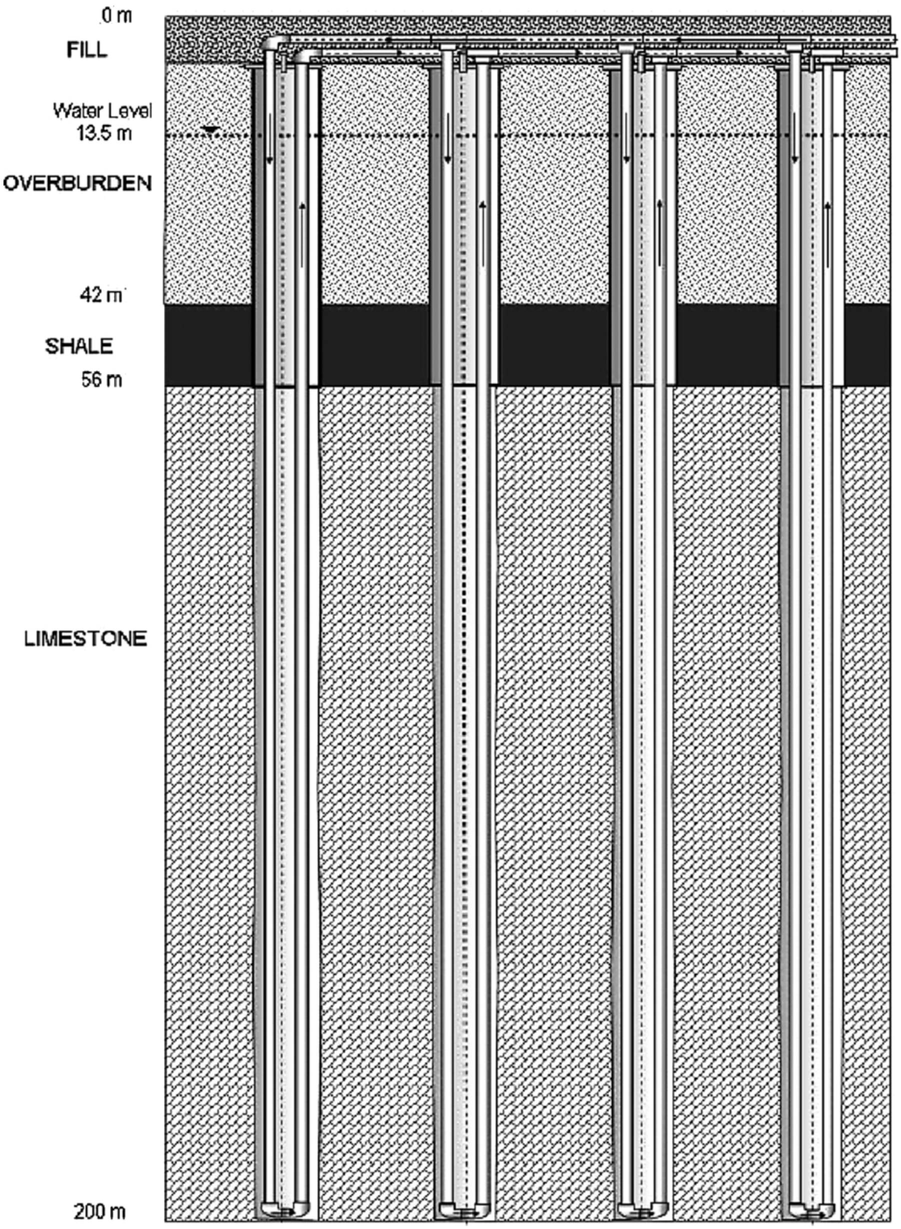

Figure 79. Illustration and details of the UOIT BTES system

flux to transport thermal energy away from the site. The total cooling load of the campus buildings is about 7,000 kW. Using the thermal conductivity test results, it was determined that a field of 370 boreholes, each 200 m in depth, would be required to meet the energy demand. In addition, five temperature monitoring boreholes were installed, increasing the total drilling for the project to 75 km. The borehole drilling was carried out using three drilling rigs, operating 24 h per day. Steel casing was installed in the upper 58 m of each borehole to seal out groundwater in the shallow formations. Design changes were made to the borehole heat exchangers (BHE) as a result of the lack of groundwater flow in the rock. The Swedish practice of water-filled BHEs was utilized instead of the North American practice of grouted BHEs. Waterfilled BHEs improve the efficiency of the U-tube installation and extend the life of the boreholes indefinitely [5].

The BTES field occupies the central courtyard of the new UOIT campus. The field is divided into four quadrants in order to optimize seasonal energy storage. The BHEs are located on a 4.5 m grid and the total field is about 7,000 m^2 in area; thus the system has a total of 1.4 million m^3, or 1.9 million tons of rock, and 0.7 million tons of overburden. A string of temperature probes in each of the five monitoring wells monitors the temperature of the thermal store within and outside the BHE field. The fluid and energy flows are being monitored in the first few years of operation to optimize the long-term performance of the BTES system.

UOIT's central plant provides a cooling and heating system for the entire campus, utilizing the BTES. Chilled water is supplied from two multistack chillers, each having seven 90-ton modules, and two sets of heat pumps each with seven 50-ton modules. The 90-ton modules are variable displacement centrifugal units with magnetic bearings that allow for excellent part-load performance. The condenser water goes in to a large borehole field that has 370 holes each 198 m deep. The field retains the heat from the condensers for use in the winter (when the heat pumps reverse) and provides low-temperature hot water for the campus. All but a few services use this low-temperature (52.8 °C supply) hydronic heat; energy provides trim heat for the low-temperature system, heat for the small medium-temperature (82.0 °C supply) system (used for vestibules, radiant panels, etc) and backup heat for the campus. With relatively low return temperatures of the ground source system, the boilers operate at high efficiency. Via a tunnel system that surrounds the BTES field, services are distributed through the campus. Each building is hydronically isolated with a heat exchanger, and has an internal distribution system [5].

A schematic flow chart of the BTES is presented in Figure 80, and the technical specifications of the chiller and two heat pumps, each having seven modules, are listed in Table 14. The BTES consists of a field of borehole heat exchangers (BHEs), a system of horizontal headers and tie-piping, and pumping and flow control equipment in the central system. As mentioned above, the BHE field is equipped with a single U-tube heat exchanger. A glycol solution, encased in polyethylene tubing, circulates through an interconnected, underground network. During the winter, fluid circulating through tubing

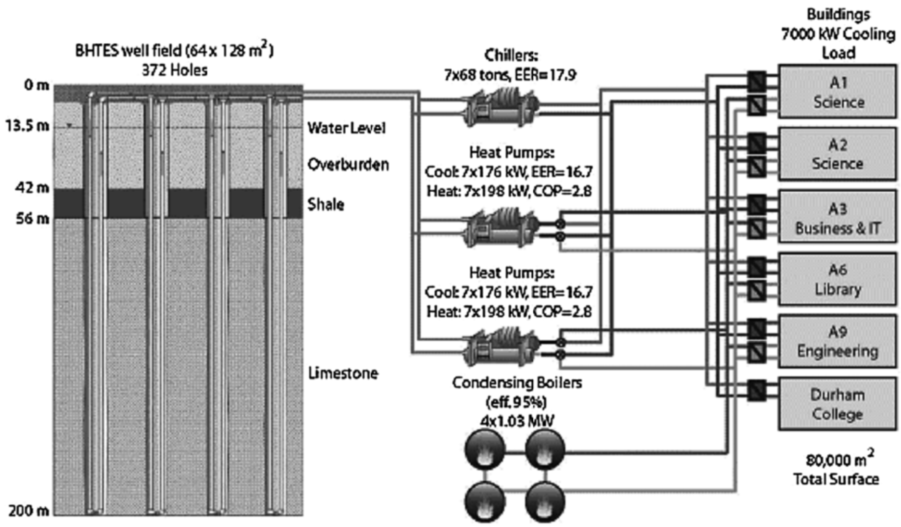

Figure 80. Schematic flow diagram of the BTES at UOIT

extended into the wells collects heat from the earth and carries it into the buildings. In summer, the system reverses to take heat from the building and place it in the ground. Supplemental heating is also provided by condensing boilers [6].

A site view during construction of the BTES system at UOIT well field, showing the grids of boreholes and piping that interconnect them is shown in Figure 81. Here, uncapped well heads show up as black dots. Four-inch piping runs from the wells into the mechanical corridors that circle the field.

TABLE 14. Design values for heat pumps

Total heating/cooling loads	1,386/1,236 kW
For heating	
Load water	
Entering/leaving water temperatures	41.3/52 °C
Source water	
Entering/leaving water temperatures	9.3/5.6 °C
For cooling	
Load water	
Entering/leaving water temperatures	14.4/5.5 °C
Source water	
Entering/leaving water temperatures	29.4/35 °C
COP_{design} for heating/cooling	2.8/4.9

Figure 81. A site view during construction of the BTES at UOIT, showing the grids of boreholes and piping that interconnect them

12.3. Analysis and Illustrative Example

We examine the performance aspects of the system heat pumps by considering three types of coefficient of performance: Carnot, actual and exergetic.

The energy (or first law) efficiency is simply a ratio of useful output energy to input energy and is referred to as a coefficient of performance (COP) for refrigeration systems. The energy efficiency of the heat pump unit can thus be defined as follows:

$$\mathrm{COP}_{actual} = \frac{\dot{Q}_h}{\dot{W}_{comp}} \tag{1}$$

where $\dot{Q}_h$ is the heating load and $\dot{W}_{comp}$ is the work input rate to the compressor.

Different ways of formulating exergy efficiency (second law efficiency) are considered. The exergetic COP (i.e., efficiency ratio) used here is as follows:

$$\mathrm{COP}_{exergetic} = \frac{\mathrm{COP}_{actual}}{\mathrm{COP}_{Carnot}} \tag{2}$$

where

$$\mathrm{COP}_{Carnot} = \frac{T_H}{T_H - T_L} \tag{3}$$

Here, COP_{Carnot} is the maximum heating coefficient of performance, based on a Carnot (ideal) heat pump system operating between low- and high-temperature reservoirs at T_L and T_H, respectively.

A parametric study was conducted using the Engineering Equation Solver (EES) software. Figure 4 illustrates the variation of the exit temperature of the heat pump (or supply temperature of the heat distribution system) in the heating mode versus COP. Normally, in heating systems, the supply temperature of the heat distribution network plays a key role in terms of exergy loss. This temperature is determined via an optimization procedure.

In this procedure, increasing the supply temperature is considered to reduce the investment cost for the distribution system and electrical energy required for pumping stations. However, this change increases heat losses in the distribution network.

Unless there is a specific reason, the supply temperature should be higher in order to increase the exergy efficiency of the heat pumps and hence the overall system, as shown in Figure 82. Other points to be considered in the design include the effect of outdoor conditions on the return temperature of the heat distribution network, the type of users connected to the system, and the characteristics of the heating apparatus. Also, in the heat exchanger

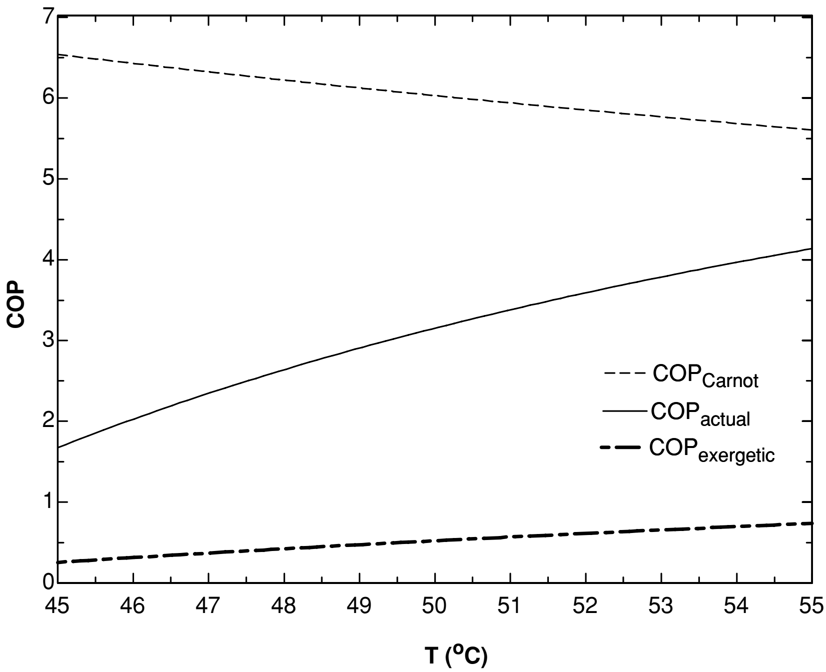

Figure 82. Variation of COP vs. heat pump supply temperature

design, a certain temperature difference is desired depending on the type of heat exchanger to be used. But, decreasing the supply temperature increases the size of building heating equipment. Oversizing does not only lead to increased cost, but also to greater exergy destruction due to unnecessarily increased irreversibilities in pumping, pipe friction, etc.

12.4. Closing Remarks

In this study we have described the innovative borehole thermal energy storage system at UOIT and presented a brief performance analysis focusing on coefficient of performance. An illustration is presented. Further research is anticipated to improve the overall BTES system at UOIT, and to produce generalized recommendations for the design of similar systems.

Acknowledgement

The authors thank the technical facilities staff of UOIT for providing data and information and Dr. Arif Hepbasli and Mr. Kurtulus Bakan for the illustrative example calculations. The authors also acknowledge the support provided by Science and Engineering Research Canada.

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13. BTES FOR HEATING AND COOLING OF THE ASTRONOMY HOUSE IN LUND

Olof Andersson
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Abstract. Borehole Thermal Energy Storage (BTES) system is used for heating and cooling of the new Institution for Astronomy at the University of Lund in Sweden. The system has been in operation since autumn 2001. 20 boreholes each with a depth of 200 m are placed under the parking area. The system delivers free cooling of 150 MWh with a approximate COP of 50 in summer. The building is heated by the heat pump using the storage as a source of heat and providing 300 MWh of heat.

Keywords: Borehole Thermal Energy Storage, heating, cooling

13.1. Type of Application

The storage is applied to a new Institution for Astronomy at the University of Lund (Figure 83). The building was finalised in the autumn 2000 with a total heating and cooling floor area of 4,200 m^2.

The building has a separate minor energy central with a heat pump designed for 300 kW output of heat (Figure 84). The heat pump is connected to 20 boreholes, each one 200 m deep. The space between the boreholes is approx. 4 m.

In the boreholes, which are placed at a parking lot, single U-pipes are installed. There is no back filling, but the holes are filled with ground water. The boreholes are drilled through 65 m of clayey soil and 135 m of shale. The thermal conductivity has been measured with a Thermal Response Test (TRT) to be approx. 2.8 W/m K.

The purpose with the BTES system (Figure 85) is to deliver basic free cooling to the building at a temperature level below approx. +10 °C. During the winter season the building is heated by the heat pump using the storage as a source of heat. The storage is then chilled down to approx. +2 °C, giving the source of free cooling. The designed heating and cooling demands are shown in Figure 86.

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 229–233.

Figure 83. The Astronomy House with the Energy Central in front. The boreholes are placed on the asphalt parking lot in front (lids)

Figure 84. The heat pump inside the Energy Central

13.2. Operational Experiences

The plant was taken into operation in the autumn 2001 and so far it has been proved to function even better than expected.

The monitoring system has shown that for the year 2002 the system delivered approx. 150 MWh of cold (36 kWh/m^2) at a COP of approx. 50. Even

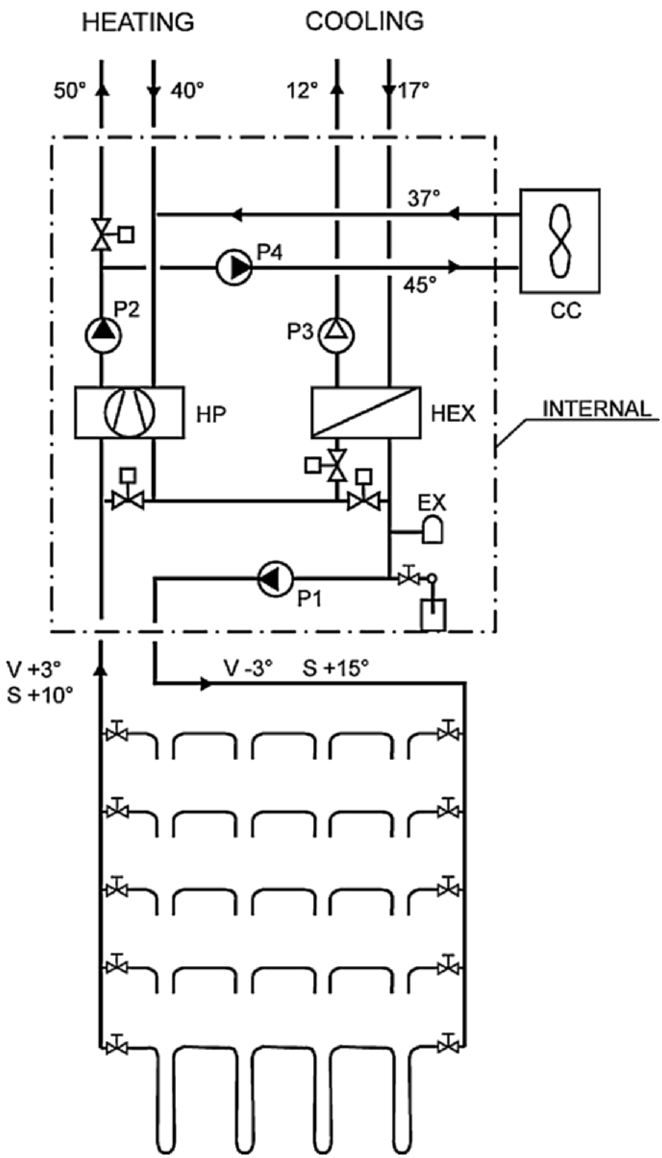

Figure 85. The system configuration diagram

if the summer temperatures peaked on some 30 °C, the storage covered the total load without using the heat pump evaporator for peak shaving. This was better than expected.

The heat production included (approx. 300 MWh), the COP for the total system was 4.8 (approx. 95 MWh of electricity).

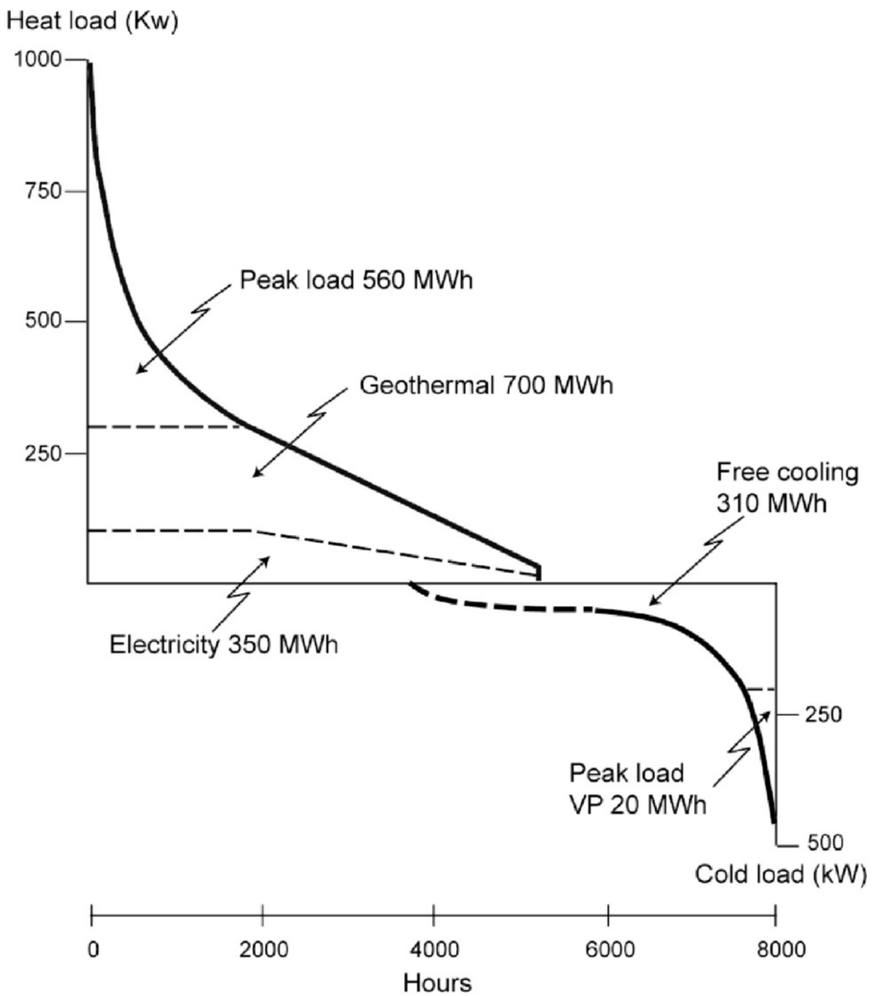

Figure 86. The designed heating and cooling demands

13.3. Economics

The total investment cost for the BTES system (boreholes, heat pump, condenser cooler, tubing, control equipment, electrical cables, and heat exchangers) was approx. 160,000 Euro.

The maintenance cost for the system has been budgeted to 3,200 Euro/year, and the energy cost for running the pumps in the system approx. 6,800 Euro/year. The calculated savings based on these figures are 15,000 Euro/year. This will result in a straight payback time of some 10 years.

13.4. Project Team

Project management: Akademiska Hus i Lund AB (Birgitta Wickman).
BTES system design: SWECO VIAK AB, Malmö (Olof Andersson).
HVAC design: Scandiakonsult AB, Malmö (Ronny Lundqvist).

13.5. Financing

Mainly financed as a commercial project by Akademiska Hus but with some support from the local environmental authority.

13.6. Contacts

Current operator: Akademiska Hus (jonny.ask@akademiskahus.se) Contact person: As above, Contact person SWECO VIAK: olof.andersson@sweco.se.

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SWECO VIAK AB, 1999. BTES System for the New Astronomy House in Lund, A Feasibility Study, Consultancy Report in Swedish, SWECO VIAK AB Malmö, 1999-09-23, Commission Number 1240.161.120.

14. BO 01 ATES SYSTEM FOR HEATING AND COOLING IN MALMÖ

Olof Andersson
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Abstract. Newly established residential area in Västra Hamnen (West Harbour) in the city of Malmö uses Aquifer Thermal Energy Storage (ATES) as part of the district heating and cooling system. ATES system has 5 warm and 5 cold wells that are 70–80 m deep. Cold from the nearby sea (Öresund) and water cold from heat pump is stored from winter to summer. The purpose of the system is to deliver free cooling to district cooling system at a temperature level below +6–8 °C. Operational experiences and economic aspects are discussed.

Keywords: Aquifer Thermal Energy Storage, district cooling

14.1. Type of Application

The storage is linked to a local district heating and cooling system that serves a newly established residential area in Västra Hamnen (West Harbour) in the city of Malmö (Figure 87). The storage is an essential part of the system that also contains a centralised heat pump and chiller system. Electricity to run the system is obtained from a windmill.

In the aquifer a combination of cold from the nearby sea (Öresund) and waste cold from the heat pump is stored from winter to summer. During the cooling season cold is directly delivered to the local DC network. Peak loads are covered heat pump and/or a chiller. The designed cold storage load capacity is 1,300 kW with a turnover of 3,900 MWh/year.

The purpose with the Aquifer Storage is to deliver basic free cooling to the DC system at a temperature level below +6–8 °C. This is achieved by storage of evaporator cold at +4.5 °C and cold from the seawater at temperatures below +4 °C. The warm side of the system works with temperatures between +13 –and 15 °C. The maximum flow rate is for legal reasons limited to 32.5 l/s (117 m^3/h).

The storage system contains (Figure 88) two groups of wells with a set of heat exchangers in between. There are 5 warm and 5 cold, 8″ wells with

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 235–238.

Figure 87. The Bo 01exhibition area seen from the south at the final stage of construction. The in-take of seawater is placed opposite the yellow building

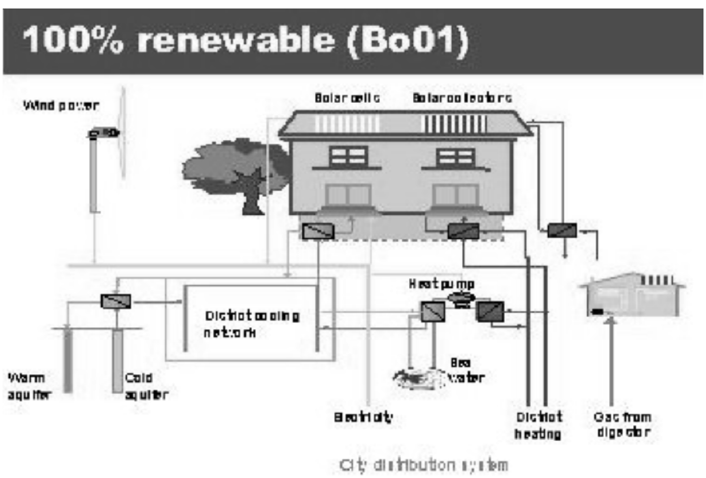

Figure 88. The system concept in overview, based on local renewable resources

depths that are 70–80 m deep. The distance between the wells in the same group is some 50 m and between the groups approx 250 m.

The soil is approx 10 m and consists of clay till landfill on top. The bedrock consists of different limestone sections (Figure 89). The main aquifer is related to porous layers 30–60 m below surface. The transmissivity is approx. $2 \times 10^{-3}\,\mathrm{m}^2/\mathrm{s}$.

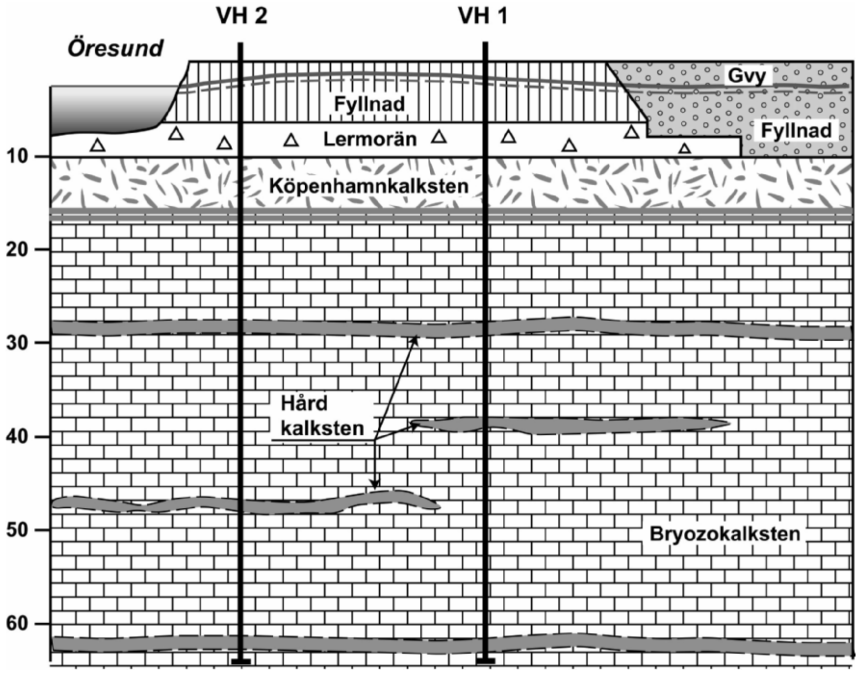

Figure 89. The hydro geological conditions at Bo 01

14.2. Operational Experiences

The plant was constructed in 2000 and it was functionally tested in the spring 2001. The function of the storage and the sea water system worked as planned. However the distribution of district cold did not functioned due to a hidden short cut between the supply and return line (unclosed buried valve). For this reason the system was not fully in operation this season.

The summer season 2002 the storage was delivering cold approx. half of the designed amount. The reason for the less free cooling production was that the heat pump, from which waste cold should be stored during the winter, was shut down because of technical problems. Hence, the only cold stored in the aquifer came from the surface water.

14.3. Economics

The total investment cost for the aquifer system (wells, submersible pumps, tubing, control equipment, electrical cables, well chambers and heat ex- changers) was approx. 350,000 Euro. The specific investment cost is then roughly 370 Euro/kW.

The maintenance cost for the aquifer has been budgeted to 10,000 Euro, and the energy cost for running the pumps in the system approx. 5,000 Euro/year. The specific variable cost is then some 4 Euro/MWh of produced cold.

14.4. Project Team

Project management: Sydkraft Värme Syd AB, Malmö (Olle Göransson).
Aquifer system design: SWECO VIAK AB, Malmö (Olof Andersson).
Surface system design: Sycon Energikonsult, Malmö (mats Egard).

14.5. Financing

Mainly financed as a commercial project by Sydkraft but with some support from the European Commission.

14.6. Contacts

Current operator: Sydkraft Värme Syd AB.
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Contact person SWECO VIAK: olof.andersson@sweco.se.

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15. ATES FOR DISTRICT COOLING IN STOCKHOLM

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Abstract. Aquifer Thermal Energy Storage (ATES) system is used to serve the district cooling of the inner city of Stockholm in Brunkebergs Torg with natural cold from a lake, which is stored during night and recovered during peak hours at daytime. The wells are situated at two narrow streets, six wells each street. The system is designed for a capacity of approx. 25 MW cooling power at a storage working temperature of +4 to +14 °C and at a flow rate of 600 l/s. The operational problems, solutions and economic aspects are discussed.

Keywords: Aquifer Thermal Energy Storage, district cooling

15.1. Type of Application

The storage is linked to a district cooling system that serves the inner city of Stockholm in Brunkebergs Torg (Figure 90) with natural cold from a lake (Värtan). The surface water is produced from a depth of 35 m and has a temperature ranging from +4 to +6 °C. A heat pump system can lower the supply temperature to +3 °C if required.

The designed capacity with the surface water is 60 MW. The purpose with the Aquifer Storage is to increase the capacity in order to connect more customers to the system. This is achieved by short-term storage where cold from the lake is stored during night and recovered during peak hours at daytime.

The storage system contains two groups of wells with a set of heat exchangers in between (Figure 91). The system is designed for a capacity of approx. 25 MW cooling power at a storage working temperature of +4 to +14 °C and at a flow rate of 600 l/s. The wells are situated at two narrow streets, six wells each street. The distance between the streets is approx. 60 m and the distance between wells approx. 10 m (Figure 91). All the wells are completed with Ø 360 mm continues slotted filter screens 10–14 m in length and a Ø 400 mm production casing, 14–20 m in length.

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 239–243.

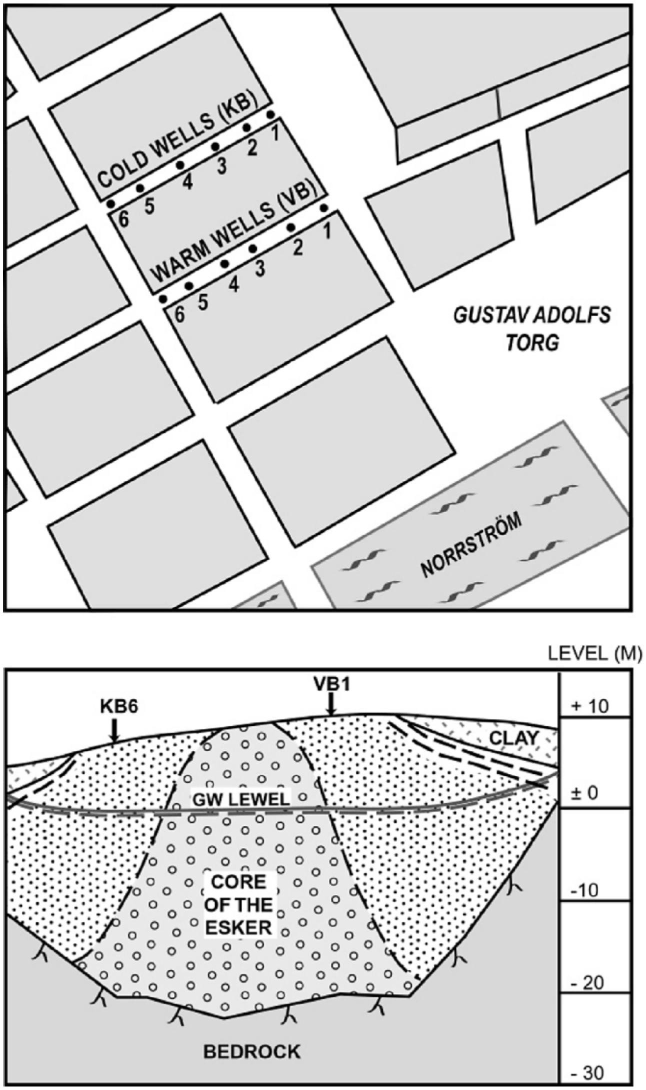

Figure 90. Map of the urban environment for the ATES system and a schematic section across the esker. The scale of the map is approx. 1:3,000

15.2. Operational Experiences

The plant was constructed in late 1997 and early 1998 and it was functionally tested in the summer 1998. It was then reviled that a major part of the wells produced sand. It was established that the wells were not sufficiently developed and that this was the reason for the problem. During the winter season

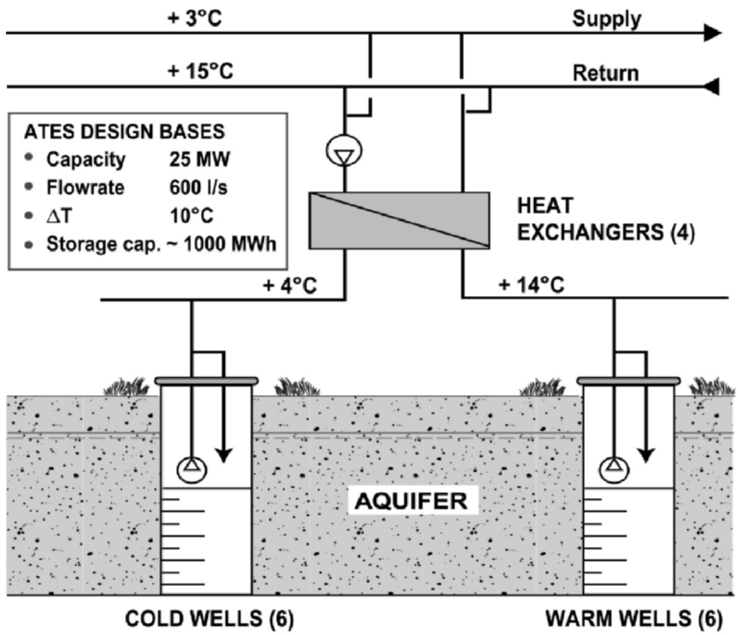

Figure 91. Principal ATES design and connection to the DC net-work

1998–1999 a program for further well development was executed. Due to a high risk of settlement damages of the buildings close to the wells a smooth development method was necessary and the amount of sand coming up from the wells had to be limited. For this reasons the on–off pumping method was chosen. This treatment was performed with the existing submersible pumps at a flow rate of roughly 120 l/s. As shown in the Table 15, the method proved to be efficient for the major part of wells. However, two wells, KB5 and KB6, were reconstructed with inner screens. This attempt was carried out in 2000 and was not successful. Hence, these wells were shut down and the plant is now operating with a reduced flow capacity of approx. 360 l/s (1,300 m^3/h) and some 15 MW of cooling power.

Due to a resisting minor sand production in most of the wells and some problems with precipitation of iron, a well maintenance programme has been developed. In this program the wells are back flushed twice a year and in addition to that, some wells are redeveloped with air lifting once a year.

The monitored thermal behavior of the storage indicate storage losses that are less then 5% over a period of 4 months. This is in accordance with the predicted losses.

The first year of commercial operation (1999) some 1,500 MWh of cold was stored and some 900 MWh was recovered in 33 storing cycles. The storage

TABLE 15. Initial well capacities, results of on–off pumping treatment and well capacity changes after the operational season 1999

Well number	Screen length (m)/slot size (mm)	Initial capacity (l/s × m)	Sand production at on–off pumping (ml/m^3 H_2O) At start	At end	Number of on–offs	Well capacity change (% of initial)
KB1	10/2.0	114	−1500	103	330	−95%
KB2	10/2.0	123	180	0.2	40	−35%
KB3	10/2.0	120	30	2.5	80	Approx. −20%
KB4	10/1.0–2.0	130	800	7.5	310	−150%
KB5[a]	12/0.75–2.0	32	Too much	Not ended	2	Not tested
KB6[a]	12/1.0–2.0	30	Too much	Not ended	2	Not tested
VB1	10/1.5–2.0	26	300	10.0	230	Approx. −150%
VB2	14/0.75–2.0	91	260	2.0	60	−50%
VB3	10/1.5–2.0	50	100	10.0	90	Approx. −100%
VB4	10/2.0–2.5	59	800	10.0	160	+25%
VB5	14/0.75–1.5	103	450	10.0	160	−50%
VB6	14/0.75–1.5	77	150	7.0	20	+30%

[a] Reconstruction attempt in spring 2000.

efficiency of approx. 60% this year was due to an overloading of cold to keep the store continuously chilled over the whole cooling season. The three latest years (2000–2002) the use and the efficiency of the storage have gradually increased due to a better operational strategy.

15.3. Economics

The total investment cost was approx. 3 milj. Euro. The additional cost to solve initial problems (1998–1999) was approx. 1 milj. Euro. This means a specific investment cost of approx. 250 Euro/kW calculated with the reduced capacity of 15 MW.

The maintenance cost the last three years (2000–2002) has been in the order of 50,000 Euro/year. In average some 2,500 MWh/year has been produced from the aquifer, which means a cost of approx. 20 Euro/MWh.

The energy needed to run the pumps in the system is approx. 4% of the output. This means a consumption of approx. 100 MWh/year at a cost of approx. 5,000 Euro (2 Euro/MWh).

In summary, and with capital and labor costs excluded, cold are produced for 20–25 Euro/MWh, while the market value for cold is about 100 Euro/MWh.

15.4. Project Team

Project management: Birka Energi, Stockholm (Hans Rudling).
Aquifer system design: SWECO VIAK AB, Stockholm (Olof Andersson).
Surface system design: Birka Teknik o Miljö, Stockholm (Agne Gustavsson).
Operational management: Birka Teknik o Miljö, Stockholm (Peter Fridén).
Maintenance program: SWECO VIAK AB, Stockholm (Håkan Djurberg).

15.5. Financing

Totally commercial financed by Birka Energi. Risk supported by STEM with 10% of the investment cost.

15.6. Contacts

Current operator: Fortum Service, Stockholm.
Contact person Fortum Service: jan.henriksson@fortumsevice.com.
Contact person SWECO VIAK: olof.andersson@sweco.se.

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16. ENERGY PILE SYSTEM IN NEW BUILDING OF SAPPORO CITY UNIVERSITY

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Abstract. Energy Pile System uses building foundation piles as ground heat exchangers. For the new building at Sapporo City University steel foundation pile is decided to be used with the HVAC system. The construction work started in January 2005. Total of 51 steel pipes have been screwed. The paper gives the results from performance analysis and construction phases of the project.

Keywords: Energy Pile System; Ground Heat Exchangers; GSHP

16.1. Use of Building Foundation Piles as Ground Heat Exchangers, "Energy Pile System"

Use of building foundation piles as ground heat exchangers allows great possibility of cost reduction in the construction of ground heat exchangers and attracts a lot of attention in Japan now. This technique has been called as "Energy Pile System" in the European countries. At the same time, it was found the first idea has been already presented in 1964 by Takashi in the journal of Japanese Association of Refrigeration.

Types of foundation piles are classified broadly into three categories. First is the cast-in-place concrete pile. Second is the pre-casting concrete pile, which has a hole in the center. The last one is the steel foundation pile with a blade on the tip of the pile, which is screwed into the ground by a rotating burying machine. The steel foundation pile is able to easily utilize as the ground heat exchanger just after filling water and inserting several sets of U-tubes in the pile. There are two typical methods that enable the steel foundation pile to provide ground heat exchanging. One is direct water circulation method and the other one is indirect method using U-tubes soused in filled water. The latter one can take closed circulating system, which is better in terms of maintenance for many years.

The advantages of the steel foundation pile to be the ground heat exchanger are its high water-tightness and low thermal resistance due to high thermal

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 245–253.

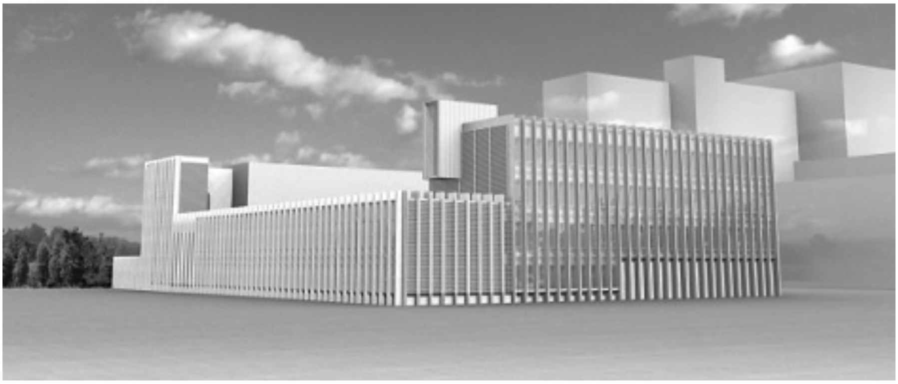

Figure 92. New building of the school of nursing, Sapporo City University (Tentative Name, inaugurated in spring 2006)

conductivity of the steel pile and effect of heat convection of filled water in the pile. In addition to above, huge amount of heat capacity of water in piles expects to play a role of a buffer tank of the heat source system.

16.2. Planning, Designing and Construction of Energy Pile System in Sapporo City University

16.2.1. SAPPORO CITY UNIVERSITY AND ENERGY PILE SYSTEM

Sapporo City University (Tentative Name), which consists from two faculties—the school of design and the school of nursing, will be inaugurated in spring 2006. A new building of the school of nursing shown in Figure 92 is located about 2 km west of the Sapporo central railway station. The mayor claimed to build the environmentally friendly public buildings. After many systems have been examined in this building, Sapporo city council decided to adopt the GSHP system using foundation piles into HVAC system of a new building of the school of nursing in November, 2004. Construction work has started form January 2005. This Energy Pile System will be the world first one which utilizes the steel fundamental piles of the building as ground heat exchangers.

16.2.2. PLANNING OF ENERGY PILE SYSTEM USING BUILDING STEEL FOUNDATION PILES

A new building of the school of nursing consists from two parts. One is a high-rise building for professor's rooms which has 2,800 m^2 of floor area and the other one is a low-rise one for practical training rooms which is 2,000 m^2. Only high-rise one provides steel foundation piles. The layout of steel foundation piles can be realized in Figure 93. 51 piles in all were screwed into the ground under the baseplate at −4.0 m deep from the ground level. The diameter of used steel piles ranges from 600 mmϕ to 800 mmϕ. As shown in Figure 94 hard gravel and pebble layer appears from about 10 m deep in this area. Consequently piles lengths are not so long and their average is 6.2 m. The average effective length for the ground heat exchanger will be 4.7 m long after deducting needed head space of 1.0 m long for footing and the bottom space of 0.5 m long. Total effective heat exchanging length can be 240 m. Total filled water volume is 115 m^3 and its large heat capacity has to be considered in the calculation of dynamic response of this system. The authors had confirmed that heat extracting performance for the indirect heat exchanging method was almost comparable in magnitude to that of direct method when two sets of U-tubes were inserted in a pile in the full scale field experiments. From these reasons, the indirect closed circulating system was adopted and two sets of U-tubes were inserted in to each steel pile.

Hourly heating loads including ventilation load were calculated by using the computational simulation program and then allowable heat supply from Energy Pile System was evaluated by using a GSHP designing tool which the authors have developed under the constrained condition that the minimum thermal medium temperature to the U-tubes in steel foundation piles from the heat pump unit did not drop under −2 °C. Calculation results showed that Energy Pile System can supply daily base heating load of 40 kW. However, as at least heating output of 50 kW is required to claim the subsidy from the Japanese Ministry of the Environment, additional boreholes were planed to be drilled to support remaining 10 kW. Reasonable length for drilling was 75 m deep and it was estimated that three boreholes were needed to satisfy heat output of 10 kW under the same minimum brine temperature condition. Anyhow heating output of 50 kW supplied from the GSHP was only a fraction of building heating load. Eventually, heat from the GSHP was decided to apply to heating and cooling the outside fresh air in the ventilation unit because it requires relative low temperature for heating and natural cooling is possible during summer. Recovered heat by natural cooling is released into the ground for recharging heat capacity of the soil.

16.2.3. DESIGNING TOOL

The layout of building foundation piles always depends on the building structure, geological condition and the type of foundation piles. It is not in regular lattice pattern in most cases. Many existing designing soft wares utilize so-called “G-function” and they cannot support the irregular layout. The author’s

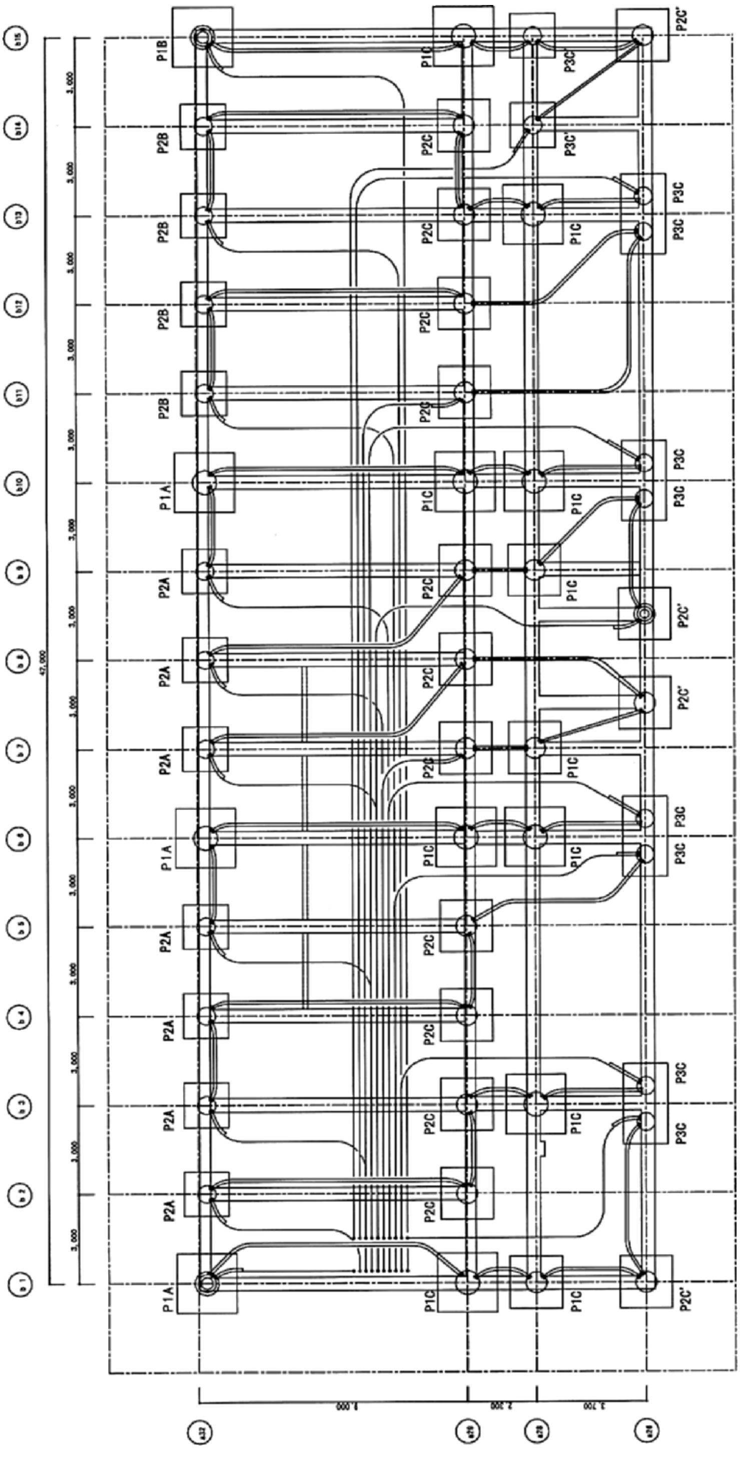

Figure 93. Layout of installed steel foundation piles and piping (51 piles with 600 ~ 800 mm ϕ in diameter)

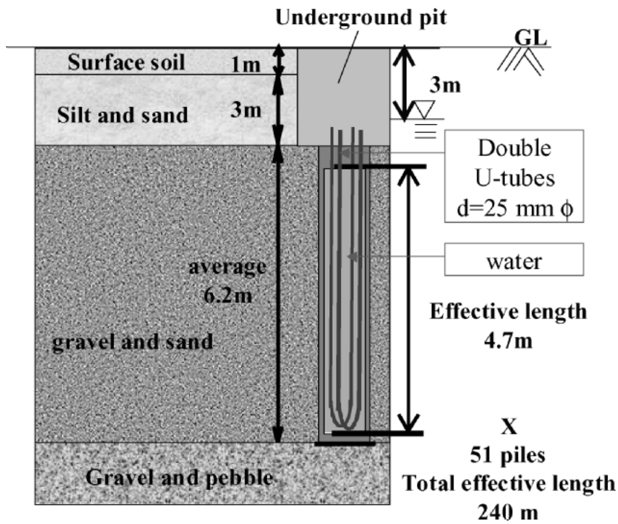

Figure 94. Geological condition and details of underground construction

group has developed a novel GSHP designing and performance prediction tool which is able to treat the random layout of ground heat exchangers with high speed calculation algorithm. Other features of this tool are that it can treat the effect of huge heat capacity in the ground heat exchanger and the thermal response of a short ground heat exchanger with a big diameter is calculated using cylindrical heat source theory modified by the similar method which Eskilson presented. Convolution integral is carried out according to the hourly heating and cooling loads. In addition, this tool includes database of heat pump performance curves according to both outlet temperature of the primary side and inlet temperature of the secondary side, energy prices and specific CO_2 emissions. Consequently, this tool can calculate hourly energy consumption and energy cost. Then life cycle energy and life cycle CO_2 emissions are evaluated. Graphical input screens and also visually output screens provide the user-friendly interface as shown in Figure 95. Particularly, it takes only for a few minutes to get results of multiple ground heat exchangers for two years.

16.2.4. PREDICTION OF PERFORMANCE

Before the calculation of performances, heating and cooling loads which can be covered by the Energy Pipe System are evaluated. Total amount of heating and cooling is 230 GJ and 75.6 GJ, respectively, and peak loads are 50 kW for heating. On the other hands, average effective thermal conductivity was measured by the on-site thermal response test and it was 2.2 W/(mK).

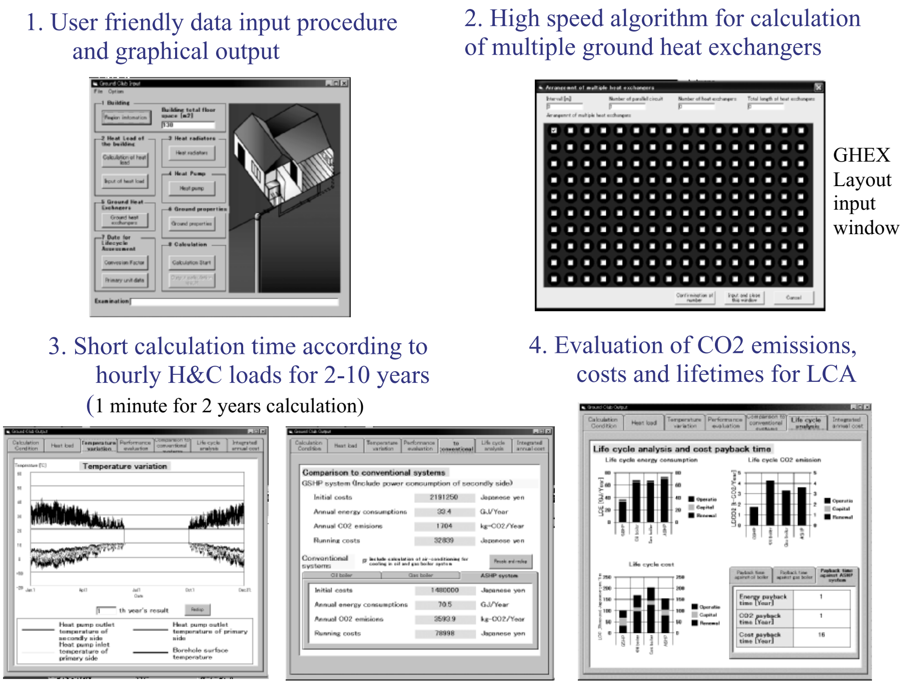

Figure 95. User-friendly interfaces for input and output of a developed design tool

The minimum brine outlet temperature is −0.8 °C during the fifth year's operation. During summer the outside air for the ventilation is cooled by the circulated brine between ventilation units and U-tubes in foundation piles and then removed heat is released into the ground. Finally it is realized that brine temperature after the summer season completely recovered to the initial temperature at the start of the fifth year's operation. This means that sustainable operation can be kept in this Energy Pile System.

Table 16 describes the predicted performance of the GSHP system. Total amount of extracted heat from the ground reaches 180 GJ and electricity consumption is 13.9 MWh. In this time COP of the heat pump unit is 4.4. When this system will adopt a constant—speed pump which can cover the maximum heat output, SCOP is 2.7. This result suggests that a variable speed pump according to the heat loads is effective to improve SCOP. On the other hand, released heat into the ground during summer is 56 GJ and the electricity of 3.6 MWh is consumed to circulate the brine between U-tubes in the foundation piles and ventilation units. SCOP during summer is estimated to be 5.7. This can be also improved.

TABLE 16. Predicted annual performance of the GSHP system

Amount of heat extraction	Electric power consumption	Electric power consumption of the circulation pump	Average COP	Average SCOP
Heating period				
180.2	13,886	7,155	4.4	2.7
Amount of heat injection to the ground (GJ)	Electric power consumption	Electric power consumption of the circulation pump	Average COP	Average SCOP
Cooling period				
75.6	0	3,689	—	5.7

Comparisons of annual CO_2 emission and annual operating cost of the GSHP system with those of gas systems which are a gas boiler without a chiller and a gas cooling and heating machine are shown in Figure 96. Annual operating cost of GSHP is 3,500 USD and it is only a half of a gas boiler system and 42% of a gas cooling and heating system. Annual CO_2 emission of GSHP is 12 tons. This is 3.8 tons and 7.4 tons smaller than gas systems, respectively.

16.3. Process of Construction

Pit excavation work of this building has started in January 2005. Totally 51 steel piles were screwed by using rotating burying machine from February 2005 indicated in Figure 97. After all piles were installed, internal spaces were filled up by tap water and two sets of U-tubes were inserted as shown in Figure 98. Figure 99 shows how U-tubes come outside through the reinforced frame of footings. U-tubes were protected by sheathed plastic tubes.

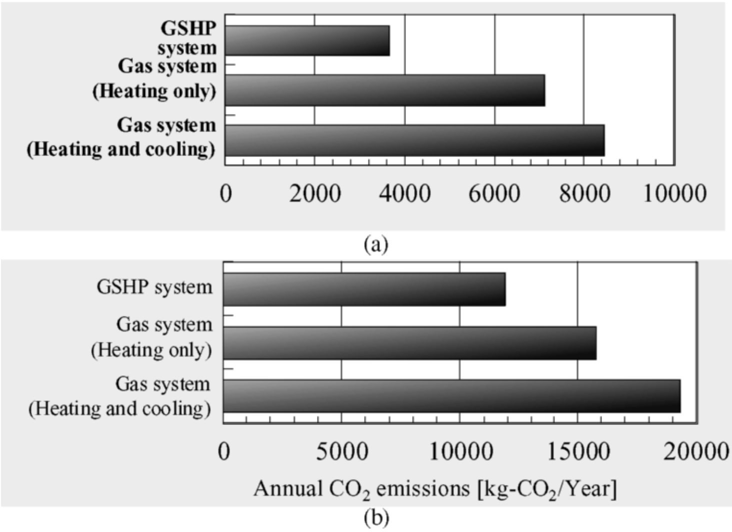

Figure 96. Comparisons of annual CO_2 emission and annual operating cost of the GSHP (a) Comparisons of annual operating cost of the GSHP (b) Comparisons of annual CO_2 emission

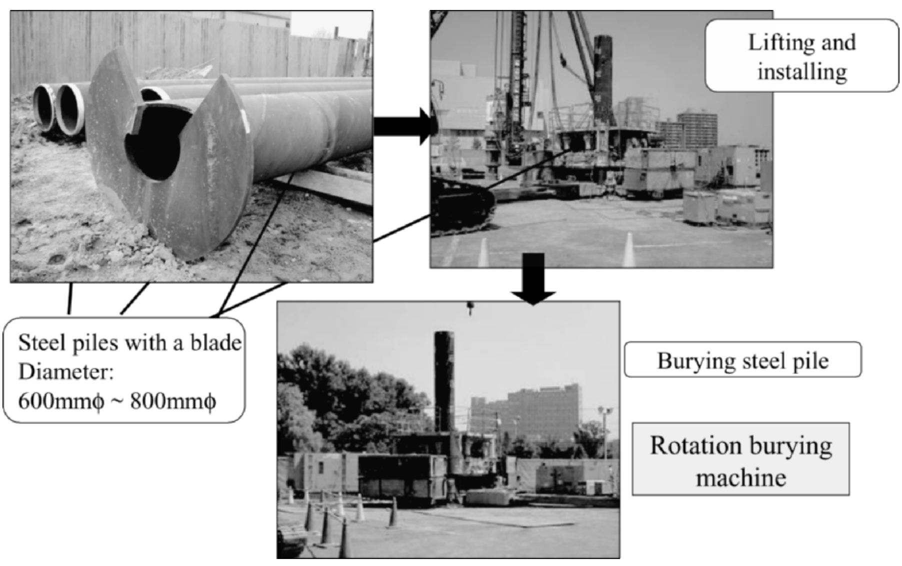

Figure 97. Steel piles with a blade on the tip and a rotation burying machine

[SPEC of used U-tube]

Nominal diameter	ISO 25A
Material	PE100
Wall thickness	2.9 mm

Figure 98. Layout of installed 51 steel foundation piles and inserting of two sets of U-tubes

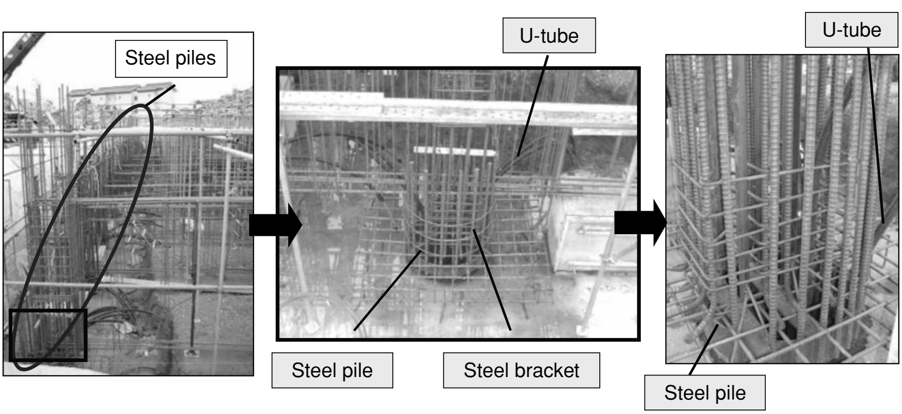

Figure 99. U-tubes come outside through the reinforced frame of footings

PART V. PHASE CHANGE MATERIALS

17. PHASE CHANGE MATERIALS AND THEIR BASIC PROPERTIES

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Abstract. This section is an introduction into materials that can be used as Phase Change Materials (PCM) for heat and cold storage and their basic properties. At the beginning, the basic thermodynamics of the use of PCM and general physical and technical requirements on perspective materials are presented. Following that, the most important classes of materials that have been investigated and typical examples of materials to be used as PCM are discussed. These materials usually do not fulfill all requirements. Therefore, solution strategies and ways to improve certain material properties have been developed. The section closes with an up to date market review of commercial PCM, PCM composites and encapsulation methods.

Keywords: PCM; phase change; latent heat; melting; heat storage; cold storage; corrosion; phase separation; incongruent melting; subcooling; nucleator; products.

17.1. Basic Thermodynamics

Heat and cold can be stored using different physical and chemical processes. These processes have different, distinct advantages and disadvantages.

17.1.1. HEAT STORAGE AS SENSIBLE HEAT

By far the most common way of heat storage is as sensible heat. As Figure 100 shows, heat transferred to the storage medium leads to a temperature increase of the storage medium. The ratio of stored heat to temperature rise is the heat capacity of the storage medium.

This temperature increase can be detected by a sensor and the heat stored is thus called sensible heat. Sensible heat storage in most cases uses as storage materials solids (stone, brick, . . .) or liquids (water, . . .). Gasses have

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 257–277.

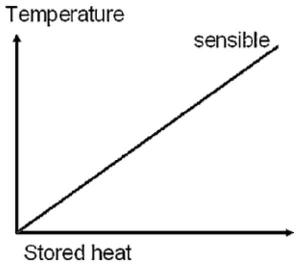

Figure 100. Heat storage as sensible heat leads to a temperature increase when heat is stored

very low heat capacity and are therefore usually not used for heat or cold storage.

17.1.2. HEAT STORAGE AS LATENT HEAT

If heat is stored as latent heat, a phase change of the storage material is used. Different options are:

1. *Evaporation of the storage material.* Evaporation is a phase change with usually large phase change enthalpy; however the process of evaporation strongly depends on the boundary conditions:
   - *Constant volume:* evaporation leads to a temperature and large pressure change within that volume and is therefore technically not applied.
   - *Constant pressure in closed systems:* this leads to a large volume change, which is therefore also technically not applied.
   - *Constant pressure in open systems:* upon loading the storage with heat, the storage material is evaporated and lost to the environment (open system). To unload the storage, the storage material has to be retrieved from the environment. The only technically used material is therefore water.
2. Solid–liquid phase changes (melting). Melting is a phase change with large phase change enthalpy, if a suitable material is selected. Melting is characterized by a small volume change, usually less than 10%. If a container can fit the phase with the larger volume, usually the liquid, the pressure is not changed significantly. Then melting and solidification of the storage material proceeds at a constant temperature. solid–liquid phase changes are therefore suitable for many technical applications.
3. Solid–solid phase changes. Solid–solid phase changes have the same characteristics as solid–liquid phase changes, but usually do not posses a large phase change enthalpy. However, there are exceptions.

The melting of the storage material is shown in Figure 101.

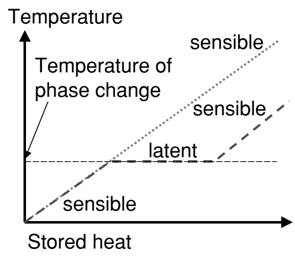

Figure 101. Heat storage as latent heat for the case of melting (solid–liquid phase change)

Upon melting heat is transferred to the storage material while the material keeps its temperature constant at the melting temperature. If the melting enthalpy has been transferred to the storage material the melting is completed and further transfer of heat results in sensible heat storage. The storage of the heat of melting cannot be detected from the temperature and the heat stored (melting enthalpy) is called latent heat. Materials with a solid–liquid (or solid–solid) phase change, which are suitable for heat or cold storage, are commonly referred to as "latent heat storage material" or simply "phase change material" (PCM).

17.1.3. POTENTIAL APPLICATIONS

Potential fields of application for PCM can be found directly from the basic difference between sensible and latent heat storage as explained in Figure 102.

17.1.3.1. *Stabilization of Temperature*

As Figure 102 shows, heat can be supplied or extracted from a latent heat storage material without significant temperature change. PCM can therefore be applied to stabilize the temperature in an application, for example the indoor temperature in a building or the temperature of the interior of transport boxes.

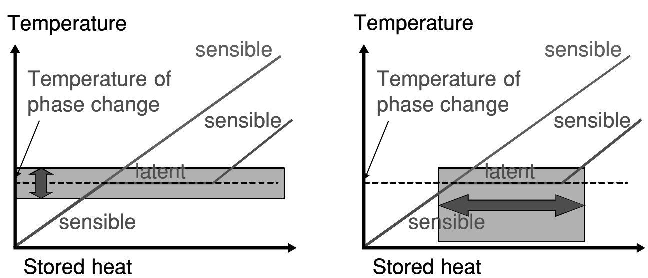

Figure 102. Potential fields of application of PCM: temperature stabilization (left) and storage of heat or cold with small temperature change (right)

TABLE 17. Comparison of typical storage densities of diffrent energy storage methods

	kJ/l	kJ/kg	Comment
Sensible heat			
Granite	50	17	$\Delta T = 20°C$
Water	84	84	$\Delta T = 20°C$
Latent heat of melting			
Water	330	330	0 °C
Paraffin	180	200	5–130 °C
Salthydrate	300	200	5–130 °C
Salt	600–1,500	300–700	300–800 °C
Latent heat of evaporation			
Water	2,452	2,450	Ambient conditions
Chemical energy			
H gas	11	120,000	300 K, 1 bar
H gas	2,160	120,000	300 K, 200 bar
H liquid	8,400	120,000	20 K, 1 bar
Gas (petroleum)	33,000	44,600	
Electrical energy			
Battery		200	Zinc/manganese oxide

17.1.3.2. *Storage of Heat or Cold with High Storage Density*
As Figure 102 shows, PCM are also able to store large amounts of heat or cold at comparatively small temperature change. PCM can therefore be applied to design heat or cold storages with high storage density, for example in domestic heating. A comparison of energy storage densities achieved with different methods is shown in Table 17. PCM can store about 3–4 times more energy per volume as is stored as sensible heat storage in solids or liquids in a temperature interval of 20 °C. This can be a significant advantage in many applications. Chemical energy storage in petroleum however shows a storage density about 100 times larger than that of PCM.

17.2. Physical, Technical and Economical Requirements on Phase Change Materials

A suitable phase change temperature and a large melting enthalpy are the requirements that have always to be met by a PCM. However, there are more requirements that have to be met for most, but not all applications. These are:

Physical requirements:

- Suitable phase change temperature ⇒ to assure storage and extraction of heat in an application with a fixed temperature range.
- Large phase change enthalpy ΔH ⇒ to achieve high storage density compared to sensible storage.

- Large thermal conductivity ⇒ to be able to extract the stored heat or cold with sufficiently large heat flux.
- Reproducible phase change ⇒ to use the storage material many times (also called cycling stability).
- Little subcooling ⇒ to assure that melting and solidification proceed at the same temperature.

Technical requirements:

- Low vapor pressure ⇒ to reduce requirements of mechanical stability on a vessel containing the PCM.
- Small volume change ⇒ to reduce requirements of mechanical stability on a vessel containing the PCM.
- Chemical and physical stability ⇒ to assure long lifetime of the PCM.
- Compatibility with other materials ⇒ to assure long lifetime of the vessel containing the PCM and surrounding materials in case of leakage.

Economic requirements:

- Low price ⇒ to be competitive with other options for heat and cold storage.
- Non toxicity ⇒ for environmental and safety reasons.
- Recyclability ⇒ for environmental and economic reasons.

A first selection of materials is usually done with respect to phase change temperature, enthalpy and reproducible phase change. The state of the art with respect to that selection is discussed in the following section "Classes of materials". Usually a material is not able to fulfill all the above mentioned requirements. For example the thermal conductivity is generally small and an encapsulation is always needed. Therefore strategies and approaches have been developed to cope with these problems. These are discussed after the section "Classes of materials" in "Approaches to solve material problems".

17.3. Classes of Materials

17.3.1. OVERVIEW

By far the best known PCM is water. It occurs naturally and has been used for cold storage for more than 2000 years. Today, cold storage with ice is state of the art and even cooling with natural ice and snow is used again.

For applications where the melting point of water at 0 °C is not useful, different material classes have been investigated in the past. Figure 103 shows the typical range of melting enthalpy over melting temperature for the most promising material classes.

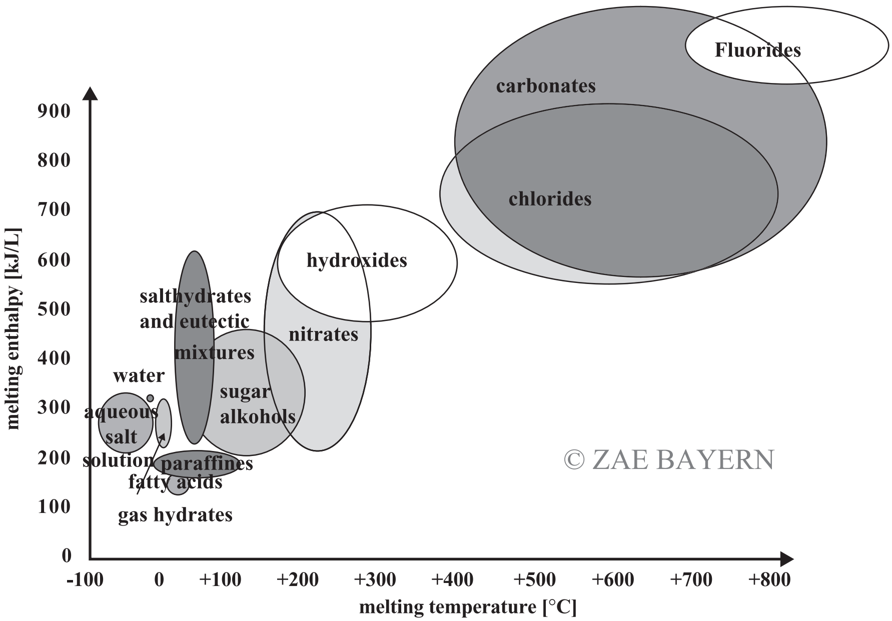

Figure 103. Classes of materials that can be use as PCM with regard to their typical range of melting temperature and melting enthalpy (graph: ZAE Bayern)

17.3.2. EXAMPLES OF MATERIALS INVESTIGATED AS PCM

A complete treatment of materials investigated as PCM is out of the scope of this book. The following text focuses on important and typical examples of materials. A more detailed treatment is found in Zalba et al. (2003) and Lane (1983).

17.3.2.1. *Inorganic Materials*

Inorganic materials, as Figure 103 shows, cover a wide temperature range. They include water at 0 °C, aqueous salt solutions at temperatures below 0 °C, salt hydrates between about 5 °C and 130 °C and finally different salts at temperatures above about 150 °C. Thanks to their density, usually larger than 1 g/cm^3, they have larger melting enthalpies per volume than organic materials. Material compatibility with metals can be problematic as some PCM-metal combinations show severe corrosion. Table 18 shows a selection of a few typical examples which are the basis for many commercial PCM. Where several data are given, they reflect the typical variation of literature date.

In order to get new PCM, mixtures of 2 or more inorganic materials have also been investigated. Table 19 shows some examples of mixtures that

TABLE 18. Selection of inorganic materials that have been investigated for use as PCM

Material	Melting temperature (°C)	Heat of fusion (kJ/kg)	Thermal conductivity (W/m K)	Density (kg/m^3)
H_2O	0	333 334	0.612 (liquid, 20 °C)	998 (liquid, 20 °C) 917 (solid, 0 °C)
$LiClO_3 \cdot 3H_2O$	8	253	Not available	1,720
$CaCl_2 \cdot 6H_2O$	29 30	171, 192	0.540 (liquid, 39 °C) 1.088 (solid, 23 °C)	1,562 (liquid, 32 °C), 1,496 (liquid), 1,802 (solid, 24 °C) 1,710 (solid, 25 °C)
$LiNO_3 \cdot 3H_2O$	30	296	Not available	Not available
$Na_2HPO_4 \cdot 12H_2O$	35–44	265 280	0.476 (liquid) 0.514 (solid)	1,522
$Na_2S_2O_3 \cdot 5H_2O$	48–55	187, 209	Not available	1,670 (liquid) 1,750 (solid)
$Na(CH_3COO) \cdot 3H_2O$	58	226, 264	Not available	1,280 (liquid) 1,450 (solid)
$Ba(OH)_2 \cdot 8H_2O$	78	265, 280	0.653 (liquid, 86 °C) 1.255 (solid, 23 °C)	1,937 (liquid, 84 °C) 2,180 (solid)
$Mg(NO_3)_2 \cdot 6\ H_2O$	89 90	149, 163	0.490 (liquid, 95 °C) 0.669 (solid, 55.6 °C)	1,550 (liquid, 94 °C) 1,636 (solid, 25 °C)
$MgCl_2 \cdot 6H_2O$	117	165, 169	0.570 (liquid, 120 °C) 0.704 (solid, 110 °C)	1,450 (liquid, 120 °C) 1,569 (solid, 20 °C)
$NaNO_3$	307	172	0.5	2,260
KNO_3	333	266	0.5	2,110
KOH	380	145	0.5	2,044
$MgCl_2$	714	452	Not available	2,140
NaCl	800	492	5	2,160
Na_2CO_3	854	276	2	2,533
KF	857	452	not available	2,370
K_2CO_3	897	236	2	2,290

are base on materials from Table 18. In the case of $CaCl_2 \cdot 6H_2O$, small amounts of NaCl and KCl are added to achieve a better melting behavior without significant change of the melting temperature. The combination of $Mg(NO_3)_2 \cdot 6H_2O$ and $MgCl_2 \cdot 6H_2O$ results in a much lower melting point.

17.3.2.2. *Organic Materials*

Organic materials, as Figure 103 shows, cover a smaller temperature range from about 0 °C to 150 °C. They include mainly paraffins, fatty acids and sugar alcohols. In most cases, their density is smaller than 1 g/cm^3. Thus paraffins and fatty acids usually have smaller melting enthalpies per volume than inorganic materials. They tend to be more expensive, but usually do

TABLE 19. Examples of inorganic mixture that have been investigated for use as PCM

Material	Melting temperature (°C)	Heat of fusion (kJ/kg)	Thermal conductivity (W/m K)	Density (kg/m^3)
48% $CaCl_2$ + 4.3% NaCl + 0.4% KCl + 47.3% H_2O	27	188	Not available	1,640
58.7% $Mg(NO_3) \cdot 6H_2O$ + 41.3% $MgCl_2 \cdot 6H_2O$	58, 59	132	0.510 (liquid, 65 °C) 0.678 (solid, 53 °C)	1,550 (liquid, 50 °C) 1,630 (solid, 24 °C)
66.9% NaF + 33.1% MgF_2	832	Not available	Not available	2,190 (liquid), 2,940 (solid, 25 °C)

not subcool. Table 20 shows examples of paraffins and sugar alcohols that have been investigated for use as PCM. Examples of fatty acids are shown in Table 21.

Organic materials can also be mixed to modify the melting point.

17.3.2.3. *Mixtures of Organic and Inorganic Materials*

In the last few years, also mixtures of organic and inorganic materials have been investigated. However, at present not many results have been published.

TABLE 20. Examples of paraffins and sugar alcohols that have been investigated for use as PCM

Material	Melting temperature (°C)	Heat of fusion (kJ/kg)	Thermal conductivity (W/m K)	Density kg/m^3)
Paraffin C14	4	165	Not available	Not available
Paraffin C15–C16	8	153	Not available	Not available
Paraffin C16–C18	20–22	152	Not available	Not available
Paraffin C18	28	244	0.148 (liquid, 40 °C), 0.358 (solid, 25 °C)	774 (liquid, 70 °C) 814 (solid, 20 °C)
Erythritol	118	340	0.326 (liquid, 140 °C), 0.733 (solid, 20 °C)	1,300 (liquid, 140 °C), 1,480 (solid, 20 °C)
High density polyethylene (HDPE)	100–150	200	Not available	Not available

TABLE 21. Examples of fatty acids that have been investigated for use as PCM

Material	Melting temperature (°C)	Heat of fusion (kJ/kg)	Thermal conductivity (W/m K)	Density (kg/m^3)
Caprylic acid	16	149	0.149 (liquid, 38 °C)	901 (liquid, 30 °C) 981 (solid, 13 °C)
Butyl stearate	19	140 123–200	Not available	Not available
Capric acid	32	153	0.153 (liquid, 38 °C), 0.149 (liquid, 40 °C)	886 (liquid, 40 °C), 1,004 (solid, 24 °C)
Lauric acid	42–44	178	0.147 (liquid, 50 °C)	870 (liquid, 50 °C), 1,007 (solid, 24 °C)
Myristic acid	49–58	186, 204	Not available	861 (liquid, 55 °C), 990 (solid, 24 °C)
Palmitic acid	61, 64	185, 203	0.162 (liquid, 68 °C), 0.159 (liquid, 80 °C),	850 (liquid, 65 °C) 989 (solid, 24 °C)

17.4. Approaches to Solve Material Problems

Usually, a material selected to be used as PCM does not fulfill all of the above requirements. Therefore, different strategies have been developed to solve or avoid potential problems. Some of these strategies are now discussed. A more detailed discussion of this subject can be found in Lane (1983) and Lane (1986).

17.4.1. PHASE SEPARATION

The effect of phase separation, also called semicongruent or incongruent melting, is a potential problem with PCM consisting of several components. Phase separation is explained in Figure 104 with a salt hydrate as example.

A salt hydrate consists of two components, the salt (e.g. $CaCl_2$) and water (e.g. $6H_2O$). The single phase of the salt hydrate is first heated up from point 1 (solid) to point 2. At point 3 the liquidus line is crossed and the material would be completely liquid. Upon heating or cooling, between point 2 and 3, 2 phases are formed, the liquid and a small amount of a phase with less water (point 4). If these phases differ in density, this can lead to macroscopic separation of the phases and therefore concentration differences of the chemicals forming the PCM material (points 5 and Figure 104 right).

When the temperature of the sample is reduced to below the melting point, the latent heat of solidification can usually not be released. This would require

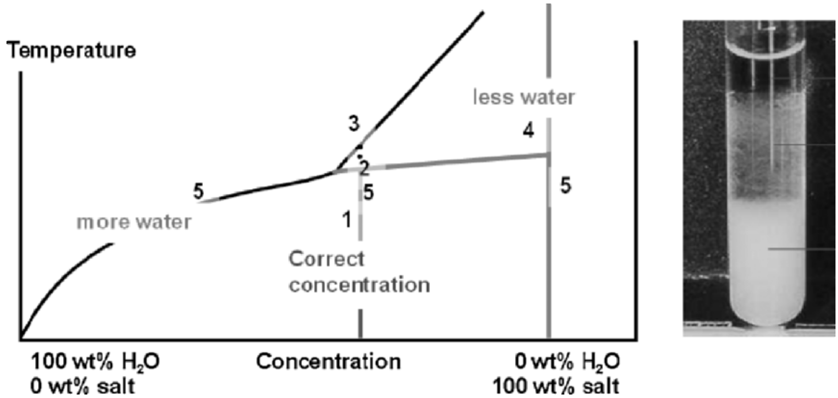

Figure 104. Phase separation of a salt hydrate (e.g. $CaCl_2 \cdot 6H_2O$) into three distinct phases with different water concentration and density (right) and corresponding phase diagram (left)

the correct concentration of the chemical components throughout the whole sample to form the solid PCM again. When the sample is heated up to a temperature where the phase point of the whole sample is in the liquid region (point 3) the different phases should mix again by molecular diffusion. If the sample is not mixed artificially, this can however take many hours or even days.

In most cases phase separation can be overcome using a gelling additive. A gelling additive forms a fine network within the PCM and thereby builds small compartments which restrict phases with different density to separate on a macroscopic level (Figure 105). If the sample is then heated to a temperature

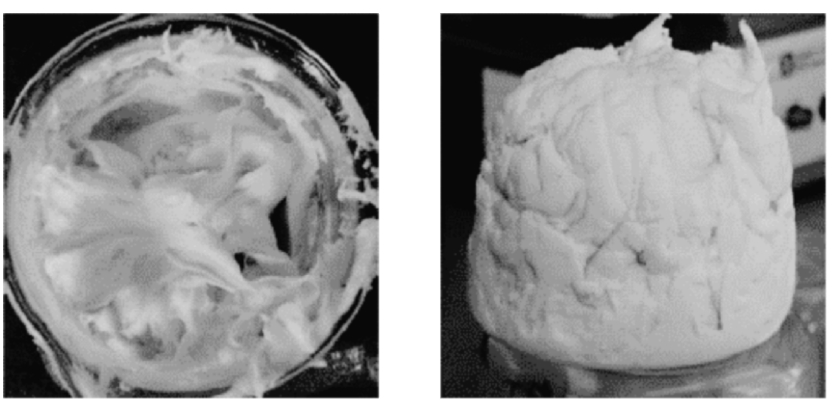

Figure 105. Gelling of a salthydrate to prevent phase separation: $CaCl_2 \cdot 6H_2O$ gelled with cellulose

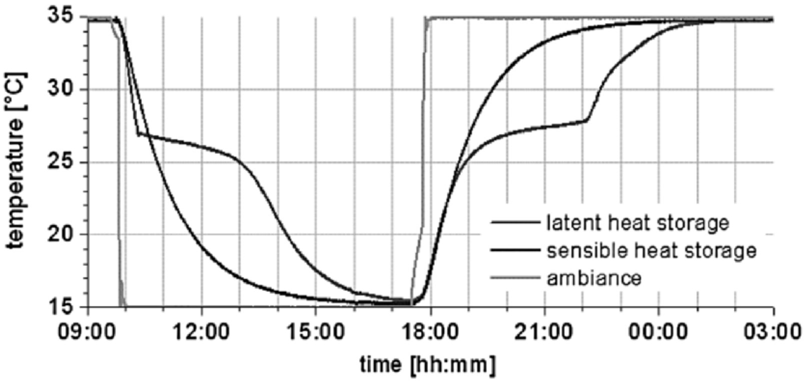

Figure 106. Solidification and melting of a PCM that is nearly ideal

somewhat above the melting point molecular diffusion can homogenize the PCM material again.

In some cases, phase separation can also be overcome by adding other chemicals to the original PCM and thus changing the phase diagram in a way that phase separation is prevented completely.

17.4.2. SUBCOOLING

An ideal PCM would solidify and melt at the same temperature as shown in Figure 106.

Many PCM however do not get solid right away if the temperature of the PCM is below the melting temperature (Figure 107).

This effect is called subcooling or supercooling. During subcooling, the PCM is in a metastable state, which means it is not in thermodynamic equilibrium. Subcooling is typical for many inorganic PCM. To reduce or

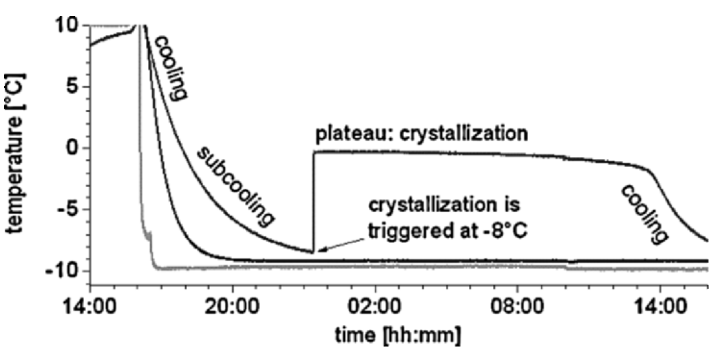

Figure 107. Subcooling of water

suppress subcooling, a nucleator has to be added to the PCM to ensure that the solid phase is formed with little subcooling. Potential nucleators are:

- *Intrinsic nucleators:* particles of solid PCM. They have to be kept separately from the PCM as they would otherwise melt with the PCM and thereby become inactive.
- *Extrinsic nucleators:* often chemicals that show very similar crystal structure as the solid PCM. This usually means that they have similar melting temperatures as the PCM itself and thus become deactivated at temperatures very close to the melting point of the PCM itself.

Nucleators have been developed for many, but not all, well investigated PCM. For a new PCM however, the search for a nucleator is usually time consuming and often not successful, as there is still no reliable theoretical approach for the search for a nucleator.

17.4.3. LOW THERMAL CONDUCTIVITY

The low thermal conductivity of PCM is an intrinsic property of nonmetallic liquids in general (Table 18–21). It poses a problem, because PCM store a large amount of heat in a small volume and this heat has to be transferred through the surface of this volume to the outside to be used in a system.

There are generally two ways to improve heat transfer:

- Improvement of heat transfer using mass transfer, which is convection. Convection only occurs in the liquid phase and therefore only acts when heat is transferred to the PCM. When heat is extracted, the solid phase forms at the heat exchanging surface.
- Improvement of heat transfer through increasing the thermal conductivity. This can be achieved by the addition of objects with larger thermal conductivity to the PCM. A special case are fins which are attached directly to the heat exchanger. Figure 108 shows two examples.

17.4.4. ENCAPSULATION AND COMPOSITE MATERIALS

Encapsulation and composite materials are a key issue in PCM technology. Because some of their positive effects can result from similar microscopic properties, they are discussed together (Figure 109). In almost all cases a PCM has to be encapsulated for technical use, as otherwise the liquid phase would be able to flow away from the location where it is applied. Macro

Figure 108. Improvement of heat transfer in a storage model through increasing the thermal conductivity with copper pieces (left) and graphite (right)

encapsulation, which is encapsulation in containments usually larger than 1 cm in diameter, is the most common form of encapsulation.

Besides holding the liquid PCM and preventing changes of its composition through contact with the environment, macro encapsulation also

- improves material compatibility with the surrounding, through building a barrier.
- improves handling of the PCM in a production.
- reduces external volume changes, which is usually also a positive effect for an application.

Micro encapsulation, which is encapsulation in containments smaller than 1 mm in diameter, is a recently developed new form of encapsulation for PCM. It can currently only be applied to water repelling PCM. Micro encapsulation serves the same purpose as mentioned above for macro encapsulation, but additionally

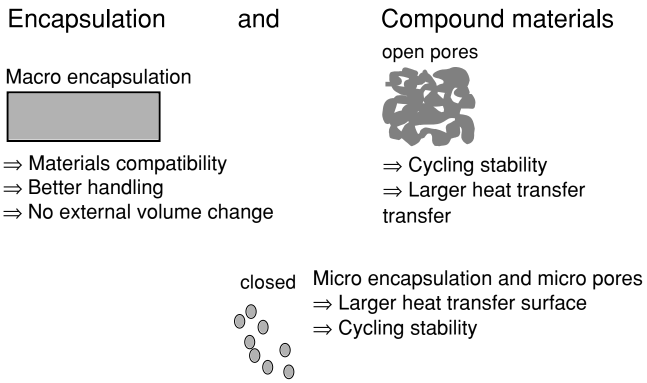

Figure 109. Encapsulation and composite materials and some of their positive effects

- improves heat transfer to the surrounding through its large surface to volume ratio.
- improves cycling stability since phase separation is restricted to microscopic distances.

Composite materials are materials consisting of a PCM and at least one other material. The other material serves to improve at least one of the PCM properties. In most cases this is handling of the PCM, but compounds can also

- improve the cycling stability, again by microscopic structures that reduce phase separation.
- improve heat transfer, through the addition of materials with large thermal conductivity as for example graphite.

17.4.5. COMPATIBILITY WITH OTHER MATERIALS

The compatibility of PCM with other materials is important with respect to lifetime of the encapsulation (or vessel) that contains the PCM, and the potential damage to the close environment of the encapsulation within the system, in case of leakage of the encapsulation.

Common problems in materials compatibility with PCM are:

- corrosion of metals in contact with inorganic PCM.
- stability loss of plastics in contact with organic PCM.
- migration of liquid or gas through plastics that affect the performance of a contained organic or inorganic PCM and outside environment.

To avoid compatibility problems, compatibility tests under conditions typical for the planned application are performed. From their results suitable material combinations are selected. Figure 110 shows a compatibility test for metals in contact with inorganic PCM. Test tubes containing both materials (center) are kept in a controlled environment for a fixed time (left) and later effects on the metal are analyzed (right).

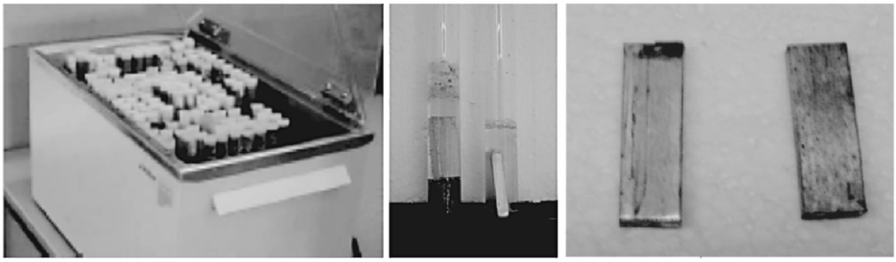

Figure 110. Compatibility test for metal-inorganic PCM combinations

Figure 111. Compatibility test for plastic-inorganic PCM and plastic-organic PCM combinations

Figure 111 shows a similar setup to test plastics in contact with organic and inorganic PCM materials.

17.5. State of the Art

17.5.1. PCM ON THE MARKET

At the moment more than 50 PCM are commercially available from the following companies:

- RUBITHERM GmbH in Germany (http://www.rubitherm.de/).
- Dörken GmbH & Co. KG in Germany (http://www.doerken.de/bvf/de/produkte/pcm/produkte/cool25.php).
- Climator AB in Sweden (http://www.climator.com/).
- TEAP in Australia (http://www.teappcm.com/).
- CRISTOPIA Energy Systems in France (http://www.cristopia.com/).
- Mitsubishi Chemical in Japan.

Their price varies in the 0.5–10 € /kg range. This means for being competitive in an energy system, daily loading and unloading should be targeted. Figure 112 shows an overview of commercial PCM with respect to melting temperature and melting enthalpy per volume and mass.

17.5.2. ENCAPSULATIONS ON THE MARKET

Encapsulations can be divided into two groups (page 219), depending on their size: macro and micro encapsulation.

17.5.2.1. *Examples of Macro Encapsulation*

Macro encapsulation in plastic containers is widely used, usually for inorganic PCM. Examples of macro encapsulation are shown in Figure 113 (plastic

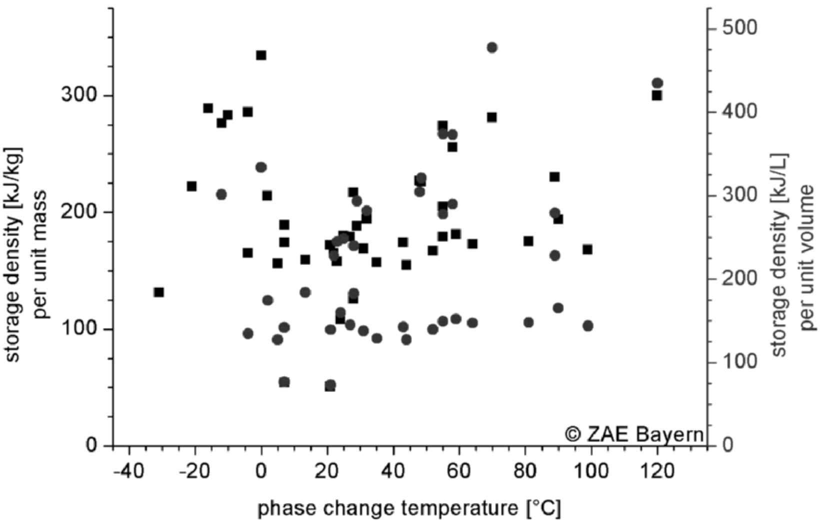

Figure 112. Overview of melting points and storage densities of commercial PCM. (●) in kJ/l, (■) in kJ/kg

containers), Figure 114 (bags), Figure 115 (capsule stripes) and Figure 116 metal containers.

17.5.2.2. *Examples of Micro Encapsulation*

Micro encapsulated PCM are available from BASF (http://www.basf.com/corporate/080204_micronal.htm) (Figure 117).

BASF encapsulates different paraffins with a special process and sells the micro capsules under the brand name Micronal® as fluid dispersion or as dried powder (Figure 118).

17.5.2.3. *PCM Compounds on the Market*

The market for PCM compounds is currently dominated by the companies Rubitherm and SGL, both from Germany.

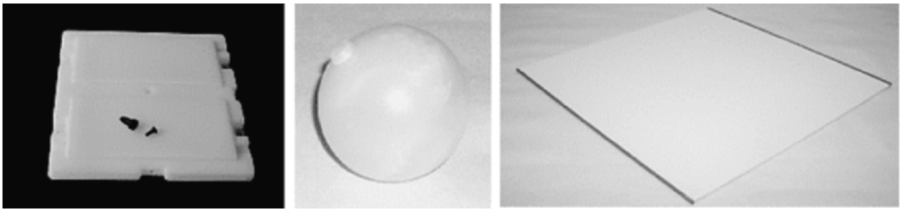

Figure 113. From left to right: Flat container (Kissmann/Germany), spheres and bar double plates (Dörken/Germany)

Figure 114. Macro encapsulation in bags (left from Climator/Sweden; right from Dörken/Germany)

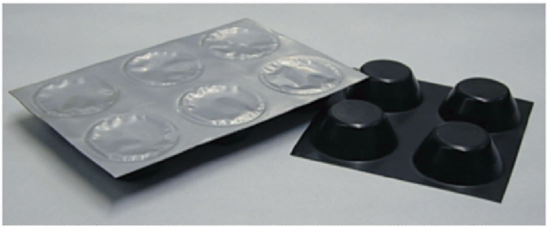

Figure 115. Macro encapsulation in capsule stripes as produced by TEAP/Australia and Dörken/Germany for inorganic PCM.

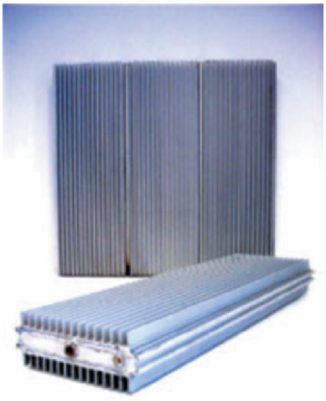

Figure 116. Macro encapsulation in aluminum profiles with fins for improved heat transfer (Climator/Sweden)

17.5.2.4. *PCM Composite Materials to Improve Handling and Applicability*

Rubitherm produces a set of different composite materials, mainly to improve handling and applicability. Some composites, which are based on different granulates and fiber boards are shown in Figure 119.

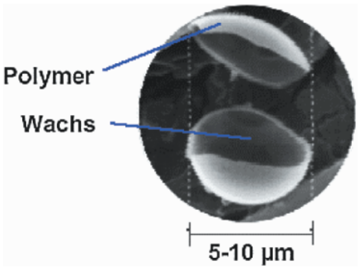

Figure 117. Microscope image of an opened micro capsule (picture: BASF/Germany)

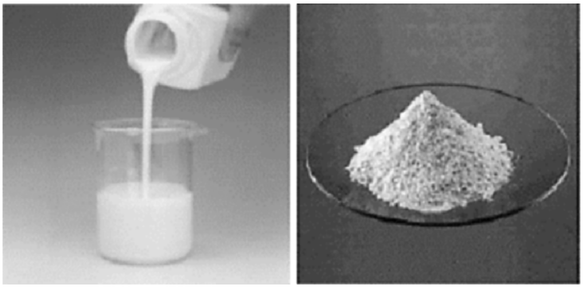

Figure 118. Micro encapsulation of paraffin produced by BASF/Germany as fluid dispersion (left) and dry powder (right) (pictures: BASF/Germany)

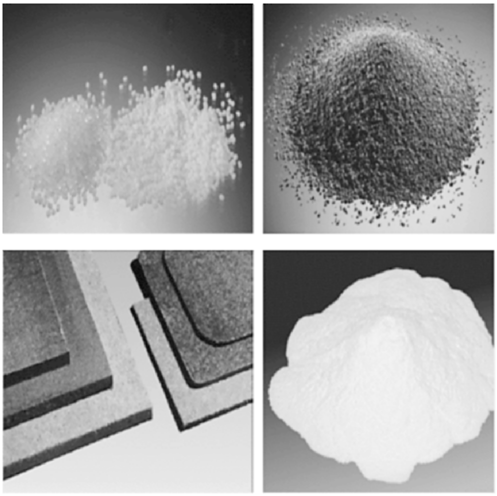

Figure 119. PCM composite materials produced by Rubitherm: compound (PK), granulate (GR), fiber board (FB) and powder (PX)

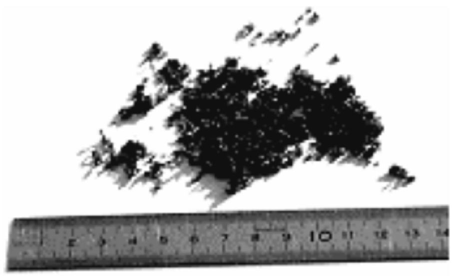

Figure 120. Expanded graphite which is the basic heat transfer structure for PCM-graphite composites

17.5.2.5. *PCM-Graphite Composites to Increase the Thermal Conductivity*

The use of graphite particles and fibers to enhance the thermal conductivity of PCM has been proposed about 10 years ago. Since then, numerous publications have experimentally proved the concept. Besides the high thermal conductivity of graphite its stability to high temperatures and corrosive environments is a big advantage to other materials. SGL in Germany sells different PCM composites with graphite (http://www.sglcarbon.com/sgl_t/expanded/markets/energy/heat_storage_d.html). SGL uses expanded graphite (Figure 120) on a large scale for producing graphite sheets which are then processed e.g. for high temperature seals. To form a PCM-graphite composite, the expanded graphite is used in two different ways.

PCM-Graphite Matrix

The PCM-graphite matrix is produced in two steps:

- In a first step the expanded graphite is pressed in a continuous process to form about 1 cm thick plates (Figure 121 left). These plates form a graphite

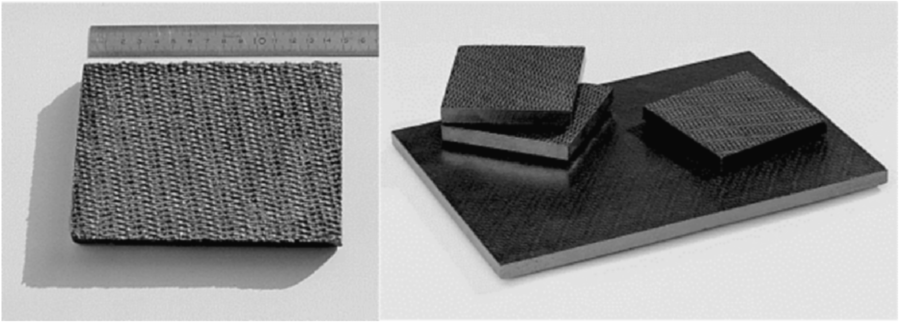

Figure 121. Left: prepressed expanded graphite forms a graphite matrix with thermal conductivity of about 25 W/m K and porosity of 90 vol.%. Right: PCM-graphite matrix after infiltration of PCM with about 85 vol.%

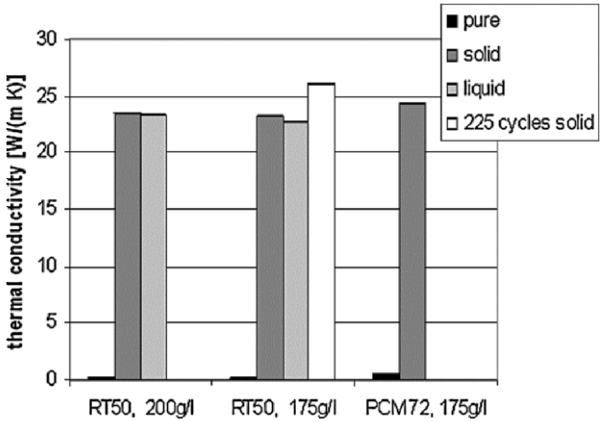

Figure 122. Thermal conductivity of pure PCM, PCM-graphite matrix in the solid and liquid state after production and in the solid state after cycling for different PCM and graphite densities

matrix with about 90 vol.% porosity, good mechanical stability and thermal conductivity of about 20–25 W/m K in plate and 5–8 W/m K perpendicular to the plate surface.

- In a second step, the PCM is infiltrated into this graphite matrix until about 80–85 vol.% PCM are reached.

Figure 122 shows the thermal conductivity of pure PCM, PCM-graphite matrix in the solid and liquid state after production and in the solid state after cycling for different PCM and graphite densities. Compared to the pure PCM with a thermal conductivity of 0.2–0.5 W/m K, the thermal conductivity is enhanced by a factor of 50–100.

The PCM-graphite matrix can be produced with many organic and inorganic PCM, but with some important exceptions. Furtheron, limitations in producing the graphite-matrix with respect to shape and size exist. SGL has therefore developed a second method to produce PCM-graphite composites.

Figure 123. PCM-graphite compound produced from mixing PCM with expanded graphite

PCM-Graphite Compound

In this method the PCM is mixed with expanded graphite in a compounding process. The result is a compound in granular form (Figure 123).

The compound can be produced with any arbitrary PCM and can also be brought into any arbitrary form e.g. by injection molding. The final result has a similar volumetric composition as the PCM-graphite matrix, which is 10 vol. % graphite, 80 vol. % PCM and 10 vol. % air. The thermal conductivity however is only about 4–5 W/m K due to the loose contact between the graphite particles in the compound. Nevertheless, compared to the pure PCM, the thermal conductivity is still enhanced by a factor of 5–20.

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Zalba B., J.M. Marın, L.F. Cabeza, and H. Mehling, 2003. Review on thermal energy storage with phase change: Materials, heat transfer analysis and applications, Appl. Thermal Eng., 23, 251–283.

18. PHASE CHANGE MATERIALS: APPLICATION FUNDAMENTALS

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Abstract. This chapter covers fundamentals for the application of PCM for different systems and products. The chapter starts with an introduction into heat transfer mechanisms by analytical and numerical models. Then different designs for storages are discussed including their advantages and disadvantages with respect to liquids and gases as heat transfer medium. The chapter ends with a presentation of the different PCM measurement technologies. These technologies are DSC for small samples, and T-history method for bigger samples. In situ measurement will also be commented.

Keywords: PCM; storage design; slurry; heat exchanger; products

18.1. Fundamental Aspects of the Application of PCM

18.1.1. POTENTIAL APPLICATIONS

Applications of PCM cover many diverse fields. As mentioned before, the most important selection criterion is the phase change temperature. Only an appropriate selection ensures repeated melting and solidification. Connected to the melting and solidification process is the heat flux. The range of heat flux in different applications covers a wide range from several kW for space heating with water or air, domestic hot water and power plants to the order of several W for temperature protection and transport boxes (Figure 124).

Because PCM generally have low thermal conductivity, the problem of heating and cooling power was not discussed in the materials section. The usual approach to reach sufficient heat flux is by designing the system.

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 279–313.

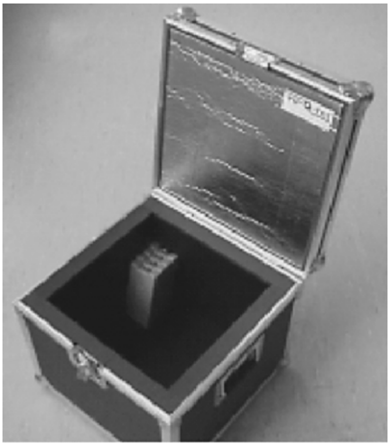

Figure 124. Transport box from Va-Q-tec. Due to the vacuum super insulation a cooling power of less than 10 W by a PCM is sufficient to sustain −20 °C for 4 days in the box in a +30 °C environment (picture: Va-Q-tec GmbH)

18.1.2. THINGS TO CONSIDER

From a thermodynamic point of view, the things to consider are the:

- Loading and unloading temperatures.
- Loading and unloading power.
- Heat transfer medium.
- Storage density (kWh or kJ per kg or m^3).

From a commercial point of view the following points have to be considered:

- Volume and weight of the storage.
- Design flexibility.
- Cost.

In this section, we will describe and explain the fundamentals for both, thermodynamic and commercial considerations. We will start looking at the basics of heat transfer in PCM, how the heat flux and time to complete a phase change are calculated. Later we will look at the design of complete storages to supply a warm or cold heat transfer fluid (liquid or gas) and discuss general design strategies.

18.2. Heat Transfer Basics

18.2.1. ANALYTICAL MODELS

The first and easiest example to look at when discussing heat transfer in PCM is a one-dimensional semi-infinite layer as shown in Figure 125.

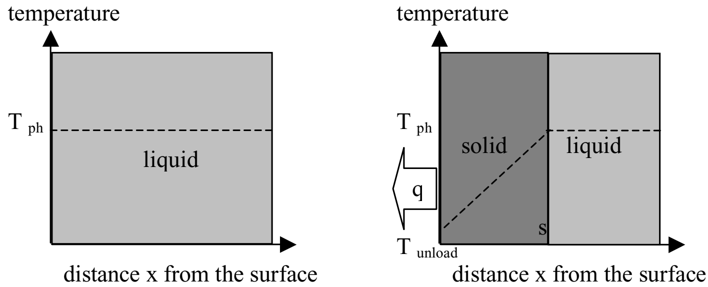

Figure 125. Cooling of a semi-infinite PCM layer. Left: initial situation, right: situation at a later time

These are severe geometrical restrictions, but to come to an analytical solution we also have to apply severe thermal restrictions.

18.2.1.1. *Analytical Model for the Heating and Cooling of a Semi-Infinite PCM Layer*

For the cooling of the semi-infinite PCM layer, we apply the following thermal restrictions:

- The heat capacity c_p is negligible compared to the phase change enthalpy ($c_p \cdot \Delta T \ll \Delta H$), therefore only latent heat at the phase change temperature has to be considered.
- There is no convection, only heat conduction.
- At $t = 0$ the PCM is completely liquid and exactly at the phase change temperature throughout.
- The temperature at $x = 0$ is then changed to T_{unload} and kept at that level for all times.

Figure 125 shows the cooling of the semi-infinite PCM layer. Because the heat capacity c_p is negligible, the temperature change from the location of the phase front at distance s to the surface is linear. The heat flux at the surface as a function of the location of the phase front is then

$$q(s) = \lambda \cdot \frac{T_{ph} - T_{unload}}{s}. \tag{1}$$

For the heating, the situation is reversed (Figure 126).

The dependence of q on the location of the phase front s is useful in the case of a PCM layer of finite thickness. If heat is extracted or supplied from one side, and if s is equal to the thickness of the PCM layer, then q(s) is the heat flux at the end of the phase change and therefore the lower limit of the heat flux. The lower limit can be helpful, but usually the heat flux as a function of time $q(t)$ is more useful.

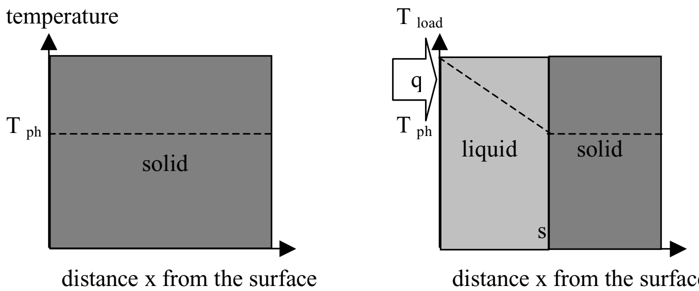

Figure 126. Heating of a semi-infinite PCM layer; initial situation left and later at the right

From neglecting the sensible heat everywhere

$$c_p \cdot (T_{ph} - T_{unload}) \ll \Delta H \tag{2}$$

follows, that the heat flux at the surface is equal to the heat released when the phase front is moving, that is

$$q = \Delta H \cdot \frac{\mathrm{d}s}{\mathrm{d}t}. \tag{3}$$

On the other hand, q is also given by

$$q(s) = \lambda \cdot \frac{T_{ph} - T_{load}}{s} \tag{4}$$

therefore

$$\lambda \cdot \frac{T_{ph} - T_{load}}{s} = \Delta H \cdot \frac{\mathrm{d}s}{\mathrm{d}t}. \tag{5}$$

Separating the variables s and t to different sides and integrating from $t' = 0$ to $t' = t$

$$\int_{t'=0}^{t'=t} \frac{\lambda \cdot (T_{ph} - T_{load})}{\Delta H} \cdot \mathrm{d}t' = \int_{s(t'=0)}^{s(t'=t)} s \cdot \mathrm{d}s \tag{6}$$

gives

$$\frac{\lambda \cdot (T_{ph} - T_{load})}{\Delta H} \cdot t = \frac{1}{2} \cdot s(t)^2. \tag{7}$$

The location of the phase boundary as a function of time is therefore given by

$$s(t) = \sqrt{2 \cdot \frac{\lambda \cdot (T_{ph} - T_{load})}{\Delta H} \cdot t}. \tag{8}$$

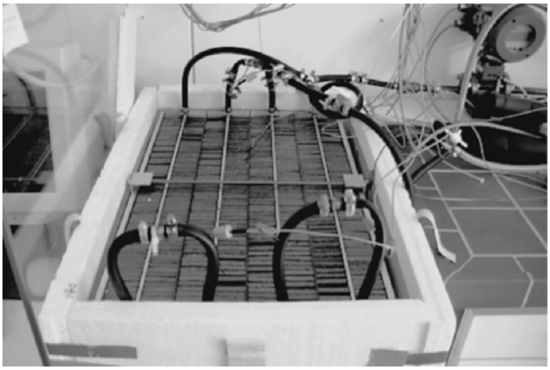

Figure 127. Example of a real, three-dimensional heat storage. 4 flat plate heat exchangers connected exchange heat between the storage medium and the heat transfer fluid (picture: ZAE Bayern)

Substituting in Equation (4), the heat flux as a function of time is finally given by

$$q(t) = \sqrt{\frac{(T_{ph} - T_{load}) \cdot \Delta H \cdot \lambda}{2 \cdot t}} \tag{9}$$

Many people are surprised that ΔH is part of $q(t)$, but this can be understood easily. The larger ΔH, the longer it takes for the phase front to move a certain distance away from the surface. As this distance enters into the heat flux via the temperature gradient, also ΔH influences $q(t)$.

18.2.1.2. *Analytical Model for a Simple Heat Storage*

The solution for the semi-infinite layer can give quite some insight into heat transfer within PCM. However, it is also clear that for a real heat storage as shown in Figure 127, the one-dimensional approach is insufficient.

In a two- or three-dimensional storage we have to take into account that the heat transfer medium will exchange heat with the storage medium along the heat exchanger. Thereby the heat transfer medium will change its temperature, and the temperature gradient entering the equation of the heat flux changes along the heat exchanger. Instead of having the one-dimensional problem discussed already, we now have at least a two-dimensional problem.

It might be surprising, but a simple analytical model, that can be quite useful, can be derived easily from the solution of the heat exchange in a simple heat exchanger (Figure 128).

Figure 128 shows a simple heat exchanger that exchanges heat supplied by fluid 2 to heat up fluid 1 while flowing along the heat exchanger surface. For such heat exchangers, it is known that the total heat flux from fluid 2 to

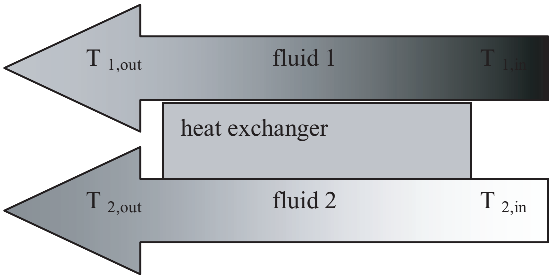

Figure 128. Heat exchanger exchanging heat from fluid 2 to fluid 1 (dark = low temperature)

fluid 1 is given by

$$\dot{Q} = A \cdot k \cdot \Delta T_{lm} \tag{10}$$

where A is the heat exchanging area, k is the total heat transfer coefficient and ΔT_{lm} is the so-called logarithmic mean temperature difference between fluid 1 and fluid 2 given by

$$\Delta T_{lm} = \frac{\Delta T_{in} - \Delta T_{out}}{\ln \frac{\Delta T_{in}}{\Delta T_{out}}}. \tag{11}$$

The heat transferred to fluid 1 will raise its temperature, and therefore

$$\dot{Q} = A \cdot k \cdot \frac{\Delta T_{in} - \Delta T_{out}}{\ln \frac{\Delta T_{in}}{\Delta T_{out}}} = c_p \cdot \frac{\mathrm{d}V}{\mathrm{d}t} \cdot (T_{1,\mathrm{out}} - T_{1,\mathrm{in}}) \tag{12}$$

where c_p is the heat capacity and $\mathrm{d}V/\mathrm{d}t$ the volume flow of fluid 1.

If we now replace fluid 2 by a PCM, the heat exchanger from Figure 128 becomes a heat storage as shown in Figure 129. To simplify the calculations we first assume that the PCM is everywhere at the phase change temperature

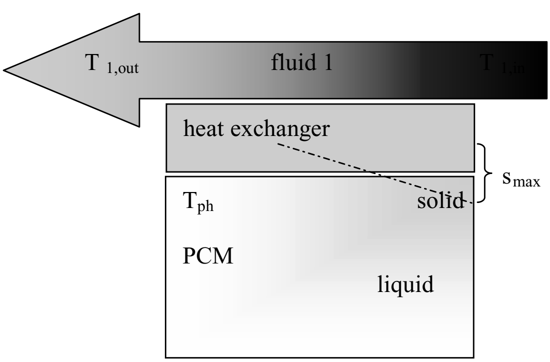

Figure 129. Approximation of a heat storage as a heat exchanger

and eliminate the subscript 1 for fluid 1 as we only have one fluid left. Then

$$c_p \cdot \frac{\mathrm{d}V}{\mathrm{d}t} \cdot (T_{out} - T_{in}) = A \cdot k \cdot \frac{(T_{ph} - T_{in}) - (T_{ph} - T_{out})}{\ln \frac{T_{ph} - T_{in}}{T_{ph} - T_{out}}}$$
$$= A \cdot k \cdot \frac{T_{out} - T_{in}}{\ln \frac{T_{ph} - T_{in}}{T_{ph} - T_{out}}}. \tag{13}$$

We now have to solve this equation for T_{out}:

$$\ln \frac{T_{ph} - T_{in}}{T_{ph} - T_{out}} = \frac{A \cdot k}{c_p \cdot \frac{\mathrm{d}V}{\mathrm{d}t}} \tag{14}$$

$$T_{out} - T_{in} = (T_{ph} - T_{in}) \cdot \left[1 - \exp\left(-\frac{A \cdot k}{c_p \cdot \frac{\mathrm{d}V}{\mathrm{d}t}}\right)\right]. \tag{15}$$

And then the heating power of the storage is

$$\dot{Q} = c_p \cdot \frac{\mathrm{d}V}{\mathrm{d}t} \cdot (T_{ph} - T_{in}) \cdot \left[1 - \exp\left(-\frac{A \cdot k}{c_p \cdot \frac{\mathrm{d}V}{\mathrm{d}t}}\right)\right]. \tag{16}$$

We now have a very nice formula for the outlet temperature and the heating power. The heating power is the maximum power possible if the full temperature rise can be reached multiplied by a factor that incorporates the ratio of the heat exchanger quality to the fluid parameters.

One important question however remains: when is the assumption that the temperature on the PCM-side of the heat exchanger is the phase change temperature valid. In other words, when does this simplification lead to significant errors in the results?

The error due to the simplification becomes significant when the thermal resistance in the solid PCM-layer (Figure 129) leads to a significant reduction below T_{ph} at the heat exchanger surface. If the thermal resistance of the heat exchanger itself is neglected, then the driving temperature difference for the heat transfer is reduced by the ratio of the thermal resistances

$$\frac{s/\lambda}{1/\alpha + s/\lambda} < 10\%. \tag{17}$$

If an error of 10% due to the simplification is accepted the maximum distance of the phase front to the heat exchanger surface s_{max} is given in Table 22. For a typical heat transfer coefficient if water is taken as heat transfer fluid, two different cases can be observed. For the pure PCM, the maximum thickness allowed before the simplification leads to serious errors in the result is only 0.5 mm. In that case the simplification is of no practical use. If the

TABLE 22. Maximum layer thickness of solid PCM S_{max} to have an error smaller 10% due to the simplification

	Paraffin ($\lambda = 0.25$ W/m K)	Paraffin–graphite matrix ($\lambda = 25$ W/m K)
$\alpha_{water} = 250$ W/(m^2 K)	$s_{max} = 0.5$ mm	$s_{max} = 50$ mm
$\alpha_{air} = 25$ W/(m^2 K)	$s_{max} = 5$ mm	$s_{max} = 500$ mm

PCM-graphite matrix is used, a layer thickness of 50 mm still gives reasonable results. In this case the simplification can therefore be used for practical calculations. For air as heat transfer medium the simplification can already be used for the pure PCM as a layer thickness of 5 mm is already typical for many applications.

18.2.1.3. *Summary for Analytical Models*

The analytical models shown result in simple formulas for the heat transfer. These formulas give general insight into the basic relations between different parameters and how important they are for the final result.

However, strong limitations exist with respect to

- Geometry.
- Incorporation of sensible heat.

Only due to severe restrictions in these points the differential equations can be solved. For a detailed and more realistic analysis of real problems therefore a different approach has to be used. We have to use numerical models because there we only have to define the differential equations in the form of difference equations. These are then solved by a computer.

18.2.2. NUMERICAL MODELS

18.2.2.1. *Numerical Models: One Dimensional*

To show how numerical models work we start with the one-dimensional heat transfer problem.

At first, the medium is divided into volume elements as shown in Figure 130. Each volume element is represented by a knot and these knots store

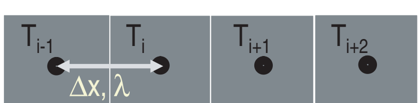

Figure 130. Creation of volume elements denoted by the variable *i*

the energy and exchange energy between them. The thermal properties of a knot with volume ΔV are then its temperature T and heat capacity c_p. Heat is transferred between two knots according to

$$q = A \cdot \lambda \cdot \frac{\Delta T}{\Delta x} \tag{18}$$

where A is the heat exchanging area between two knots, λ the thermal conductivity, ΔT the temperature difference between two knots and Δx the distance between two knots. If the temperature within the problem is given at some time t, an amount of heat given by

$$Q = q \cdot \Delta t = \Delta t \cdot A \cdot \lambda \cdot \frac{\Delta T}{\Delta x} \tag{19}$$

is exchanged between two knots in a time interval Δt. From the heat exchanged, the initial temperature and the heat capacity, the new temperature of each knot can be calculated.

These calculations have to be done by the computer successively for all volume elements. To do this, the volume elements are numbered by a two subscripts:

1. First subscript: for the volume element using the variable i and distance Δx.
2. Second subscript: for the time element using the variable k and the time interval Δt.

Then the heat transferred into volume element i from the neighboring elements $i + 1$ and $i - 1$ during time step k is

$$Q_{i,k} = \Delta t \cdot A \cdot \lambda \cdot \frac{T_{i+1,k} - T_{i,k}}{\Delta x} + \Delta t \cdot A \cdot \lambda \cdot \frac{T_{i-1,k} - T_{i,k}}{\Delta x}. \tag{20}$$

The heat transferred leads to a temperature change

$$Q_{i,k} = \Delta V \cdot c_p \cdot (T_{i,k+1} - T_{i,k}). \tag{21}$$

Therefore

$$\begin{aligned} \Delta V \cdot c_p \cdot (T_{i,k+1} - T_{i,k}) = \Delta t \cdot A \cdot \lambda \cdot \frac{T_{i+1,k} - T_{i,k}}{\Delta x} \\ + \Delta t \cdot A \cdot \lambda \cdot \frac{T_{i-1,k} - T_{i,k}}{\Delta x}. \end{aligned} \tag{22}$$

Using $\Delta V = A \cdot \Delta x$ the new temperature at time step $k + 1$ for volume element i is

$$T_{i,k+1} = T_{i,k} + \frac{\Delta t \cdot A \cdot \lambda}{c_p \cdot \Delta x^2} \cdot (T_{i+1,k} - 2 \cdot T_{i,k} + T_{i-1,k}). \tag{23}$$

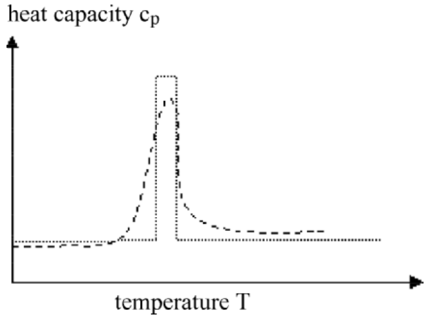

Figure 131. Example curves for temperature dependent c_p to incorporate phase change into numerical heat transfer

To include the thermal effect of a phase change, usually c_p as a box function or measured values as shown in Figure 131.

In Equation (23), all input data are known from the earlier time step. This way of calculating the values for the next time step is called explicit method. More information on numerical modeling, and for example how to integrate convection, can be found in Farlow (1983) and Özisik (1968).

18.2.2.2. *Numerical Model: Three Dimensional*

To model a real heat storage as in Figure 127, the net of knots from Figure 130 has to be at least two dimensional. Further on, it has to include knots for the heat transfer fluid and the environment. Such a net of knots is shown in Figure 132.

This model was used to simulate the unloading of the storage shown in Figure 127. The storage was equipped with PCM-graphite matrix as storage

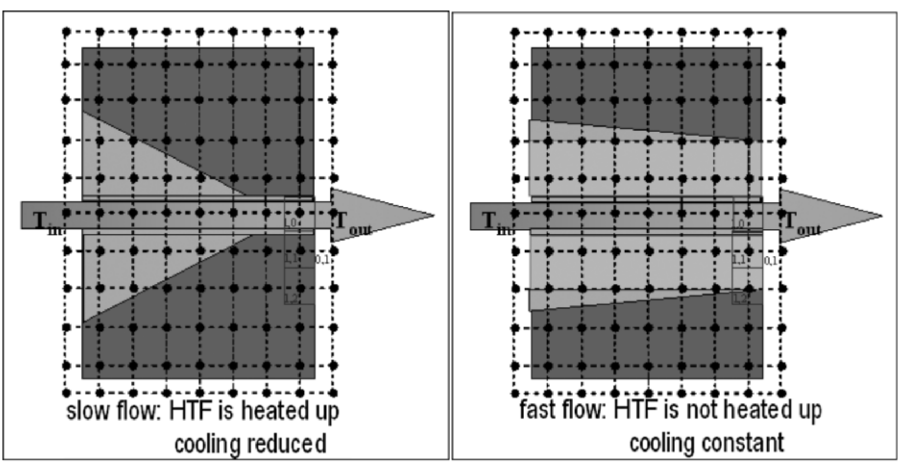

Figure 132. Two-dimensional net of knots to simulate a heat storage including knots for the heat transfer fluid and also the environment. For slow flow (left) and fast flow (right) of the heat transfer fluid, the phase front will move differently

material. The matrix was oriented in a way that heat transfer was preferably towards the heat exchanger and limited by air gaps perpendicular to the heat exchanger. The melting point of the PCM was 55 °C, the starting temperature 65 °C. The storage was unloaded with water at 10 °C at a rate of 1 l/min. The result for one storage module is shown in Figure 133 with a time sequence.

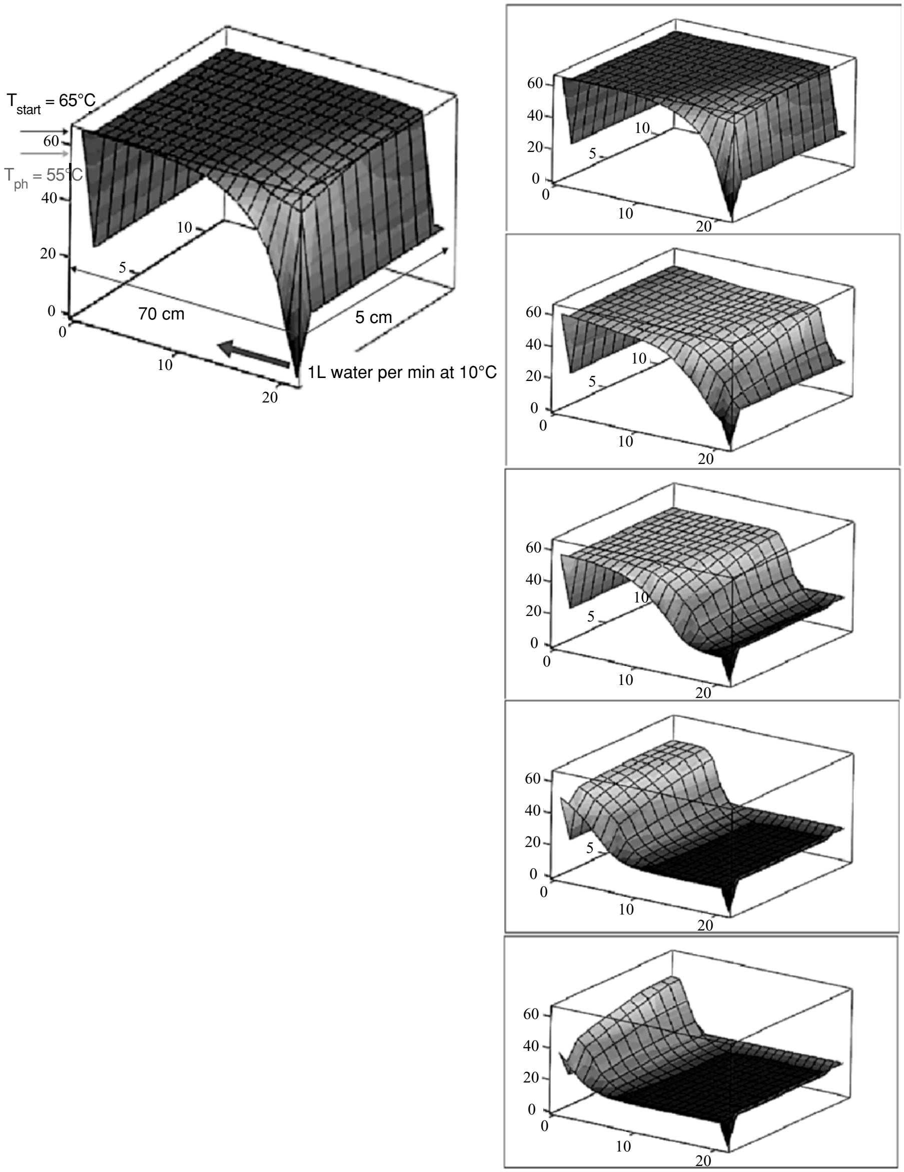

Figure 133. Unloading of one module of the heat storage from *Figure 127*. The temperature profiles in the picture series show the cooling down of the storage material with time (picture: ZAE Bayern)

The 1st part of the time sequence in Figure 133 shows the early cool down at the inlet. The 2nd and 3rd picture show the cool down of most parts in the storage to the phase change temperature. The 4th and 5th part show the cool down of the whole storage close to the inlet temperature.

18.2.3. MODELIZATION

The simulation of heat transfer in a PCM in simple geometry as well as in a whole storage as described above can be done with many mathematical and engineering software tools like MathCad and EES. Better results can be achieved with commercial CFC solutions like FLUENT that only require very basic knowledge on numerical methods for heat transfer.

The modelization of whole systems like buildings with a storage integrated or with building components that include PCM can be done using Esp-r or TRNSYS.

18.2.4. MODEL VERSUS EXPERIMENT

The analytic solutions and the numerical simulations look very nice, but are they correct, that means do they agree with real measurements? To test that we have to make measurements on real models or heat stores and compare them with calculations based on the thermodynamic properties of the PCM used. The determination of these properties is discussed later.

18.2.4.1. *One-Dimensional Heating and Cooling of a Semi-Infinite PCM Layer*

For the heating and cooling of a semi-infinite PCM layer the model shown in Figure 134 has been developed. It consists of a heat exchanger made from a copper block and an acrylic glass container for the PCM.

Within the container thermocouples are located on the surface of the heat exchanger and in distances of 1 and 2 cm.

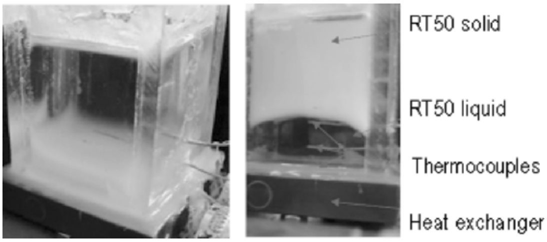

Figure 134. Experimental model for the heating (right) and cooling (left) of a semi-infinite PCM layer (pictures: ZAE Baayern)

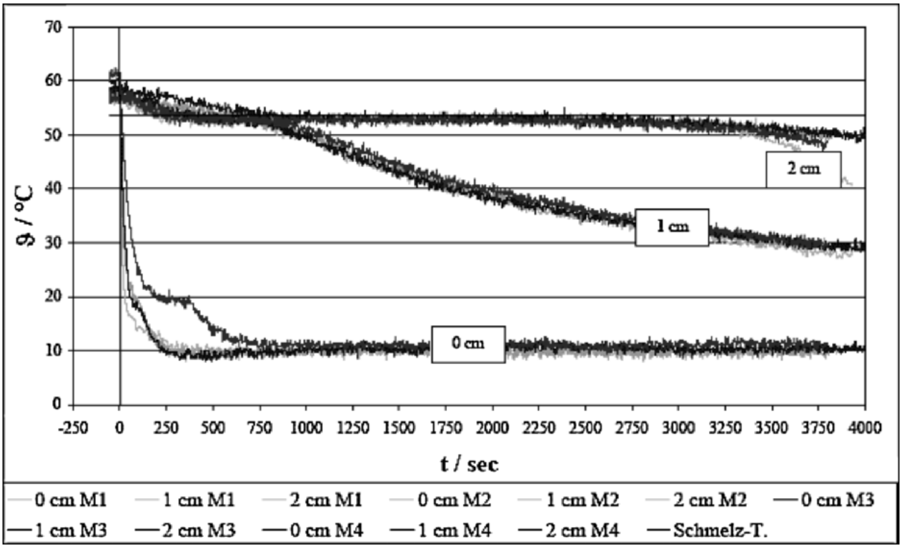

Figure 135. Temperature history in a cooling experiment

Figure 135 shows a temperature history in a cooling experiment. In Figure 136 the time when the temperature at the different thermocouples in the experiment dropped below the phase change temperature is compared with predictions from different models.

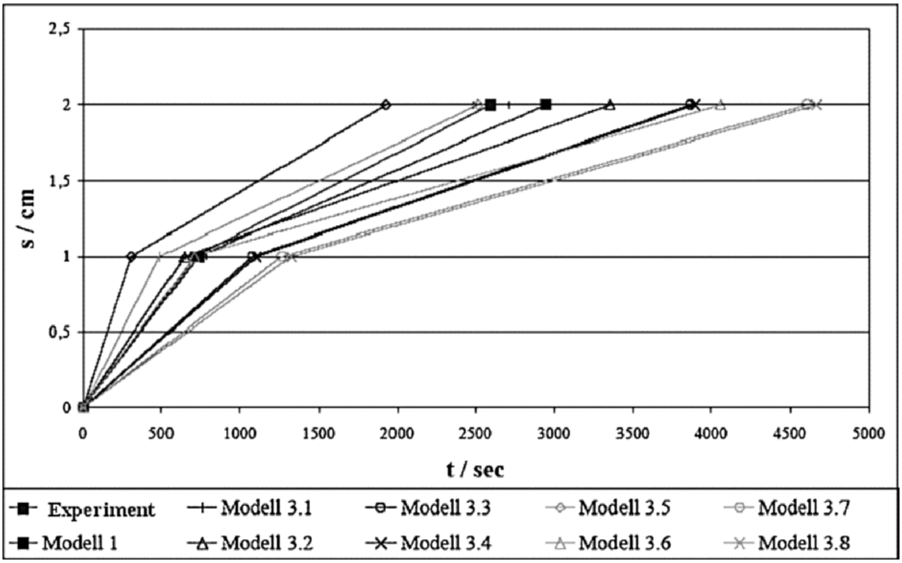

Figure 136. Time when the temperature at the different thermocouples in the experiment dropped below the phase change temperature, compared with predictions from different models

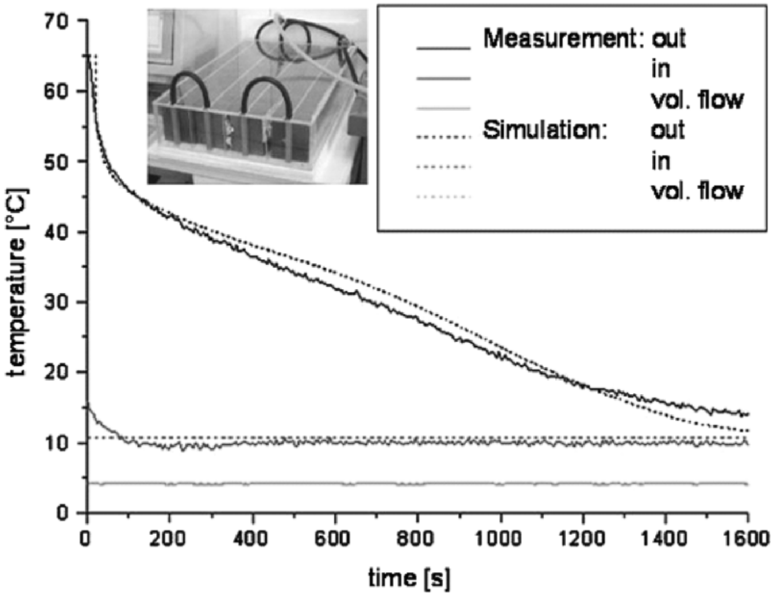

Figure 137. Comparison of the outlet temperature history recorded in an experiment and from simulations for a real heat storage.

Depending on the model, the predictions can agree well with experimental observations or be completely wrong.

18.2.4.2. *Real Heat Storage*

For a real heat storage, a comparison of the outlet temperature history recorded in an experiment and from simulations is shown in Figure 137.

The agreement between measurement and simulation is very good.

18.2.4.3. *Subcooling*

Currently there is still one important effect that cannot be treated analytically or numerically; this is the heat transfer in a subcooled PCM. The subcooled state is not a state of thermodynamic equilibrium and can therefore not be described by any of the models described above. If salthydrates find widespread application for building applications in the future, this problem will have to be solved as even 1 or 2 °C of subcooling can have a mayor effect on system performance.

18.2.5. METHODS TO IMPROVE THE HEAT TRANSFER

18.2.5.1. *Optimizing the Geometry of the Encapsulation*

Besides predicting heat transfer it has to be improved in many cases. If the PCM is encapsulated, this can be done by optimizing the encapsulation. The optimization is done with respect to

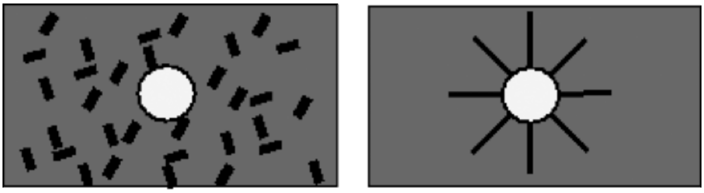

Figure 138. Methods to optimize the heat transfer within the storage material

1. Surface to volume ratio, that is heat exchanging surface to volume of the storage material.
2. Largest geometric dimension that influences heat transfer to the heat exchanging surface via the temperature gradient.

Regarding the geometry we can choose between balls (spherical geometry), pipes (cylindrical geometry) or slabs (plate like). The best surface to volume ratio has a slab. It also has good heat transfer characteristics as the largest geometric dimension that influences the heat transfer is the thickness. On the other hand, if micro encapsulation is used the heat transfer characteristic is even better as the largest geometric dimension is only a fraction of a millimeter.

18.2.5.2. *Optimizing the Heat Transfer within the Storage Material*
There are also two methods to optimize the heat transfer within the storage material as shown in Figure 138. The first method is to add particles with a higher thermal conductivity than the PCM. This is commercial at the moment only with graphite. The second method is to add larger metallic plates and attach them to the heat exchanger for better heat transfer. These plates are commonly called fins and are widely applied.

18.3. Storage Design and Examples

After discussing the heat transfer within a PCM, its mathematical description and how it can be improved, it is now time to discus the most important concepts for storage design.

18.3.1. BASICS

Figure 139 shows the basic working scheme of a storage. Heat or cold from a source is transferred to the storage, stored, and later used to cope with a demand. Besides the shift in time from supply to demand, the storage can also be used to increase the available peak power or smooth out fluctuations of the supply.

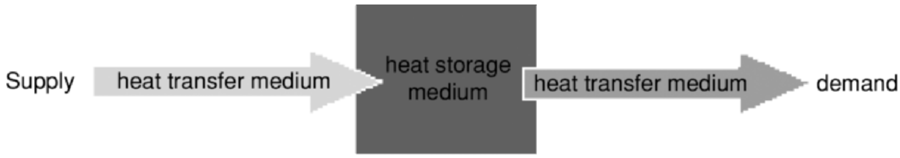

Figure 139. Basic working scheme of a storage. Heat or cold from a source is transferred to the storage, stored, and later used to cope with a demand

Two of the most important boundary conditions for storage design are

- the kind of heat transfer between heat storage medium and heat transfer medium, and
- the heat transfer medium.

Others, like energetic and exergetic efficiency are discussed in Dincer et al. (2002). Basically we can distinguish between two kinds of heat transfer fluids with different ranges of heat capacities and heat transfer coefficients:

- *Gases:* heat capacity $c_p \cong 1$ J/LK, heat transfer coefficient $\alpha = 3, \ldots, 30$ W/m^2 K.
- *Liquid:* heat capacity $c_p \cong 1$ kJ/LK, heat transfer coefficient $\alpha = 100, \ldots,$ 1,000 W/m^2 K.

18.3.2. STORAGE TYPES

The following discussion of different storage types focuses on two of the most important selection criteria:

- Storage density (volume and mass).
- Loading and unloading power.

18.3.2.1. *Direct Discharge Type*

The most common storage type is the direct discharge type (Figure 140). In this type the heat transfer fluid for the demand side is stored preferably at

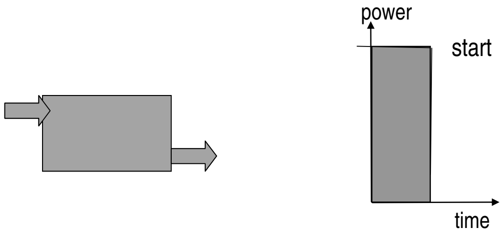

Figure 140. Direct discharge type

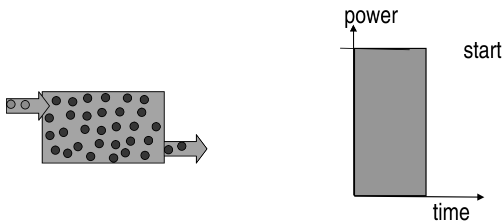

Figure 141. Slurry type

the right temperature in a vessel and discharged directly to the demand when needed. This storage type only exists for liquids as heat transfer fluid. Gasses do not have sufficiently large storage capacities. Direct discharge storages have a simple storage design and there is no heat transfer from storage medium to transfer medium necessary for unloading.

Direct discharge types have:

- low storage density, as only sensible heat storage is used.
- high power, restricted only by the discharge volume flow.

18.3.2.2. *Slurry Type*

Quite similar to direct discharge storages are slurry types (Figure 141). They differ from direct discharge types only because the heat transfer fluid also contains microencapsulated PCM. The micro encapsulated PCM increases the stored heat per volume of heat transfer fluid, which again has to be a fluid and not a gas.

Slurry types have:

- increased storage density.
- high power, even higher than direct discharge types as more heat is transported per volume of heat transfer fluid.
- improved heat transfer on the demand side, as the improved storage density also raises the heat transfer coefficient α.

One good example are ice-water slurry storages, which are widespread in cooling applications. Water is cheap and easy to handle. Today, it is state of the art to produce ice-water slurries that can be stored and pumped to a demand when needed.

From an energetic point of view, it would be desirable to produce cold at somewhat higher temperatures. Then other PCM with higher melting points in the 5–10 °C range are preferred. Such slurries can be produced using microencapsulated PCM. Figure 142 shows such a slurry produced by BASF.

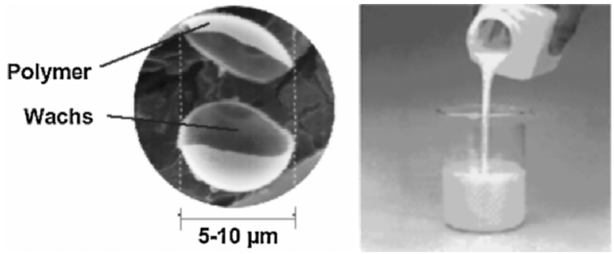

Figure 142. Slurry from micro encapsulated PCM (picture: BASF AG)

Slurries from microencapsulated PCM can of course also be produced for heating applications.

18.3.2.3. *Modular Type*

Modular heat storages use macro encapsulated PCM instead of microencapsulated PCM to increase the storage capacity. During discharge the macro encapsulated PCM however remains in the storage.

Modular types (Figure 143) have:

- Increased storage density compared to direct discharge types, due to 50–70 vol.% PCM.
- High power at the start, due to the amount of heat transfer fluid within the storage during stand still.
- Low power later due to bad heat transfer between PCM-modules and heat transfer fluid.

Modular systems mostly use flat plate bags for air has heat transfer medium and spheres (balls) for liquids as heat transfer medium. In cooling applications the system used by Cristopia (France) using PCM filled balls is the state of the art. Systems with bags for cooling air in air-conditioning systems are currently developed.

The systems described until now have no stratification. In heat storages with stratification the lower layer is subject to large temperature changes while

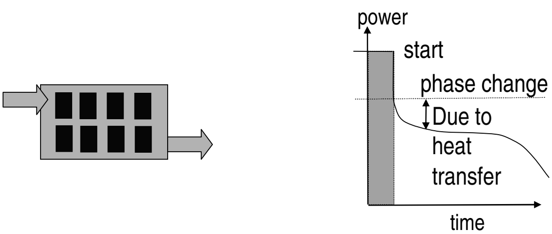

Figure 143. Modular type

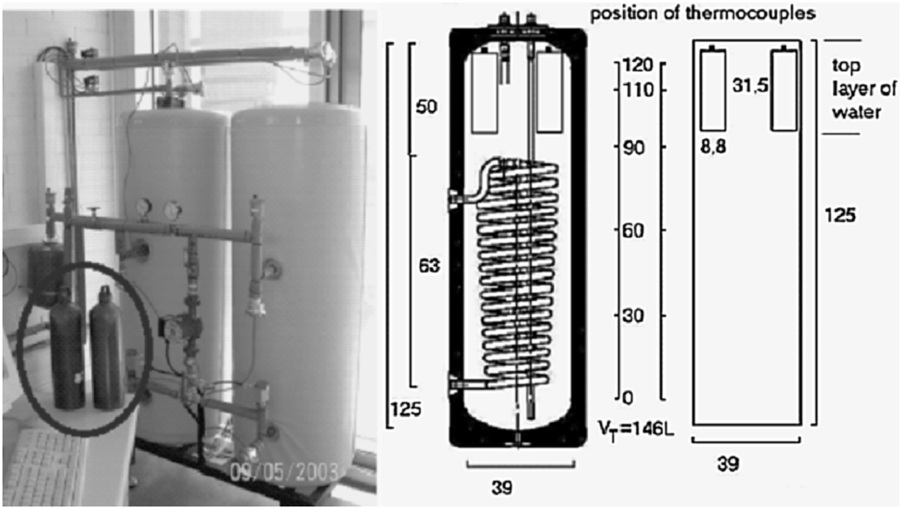

Figure 144. Hot water heat storage with stratification and PCM-modules to increase the amount of stored heat (picture: Universitat de Lleida

the top layer is always close to the demand temperature. Therefore PCM modules only improve the thermal performance of the top layer significantly; the lower layer already stores enough sensible heat.

This concept was developed first presented by Mehling et al. (2002, 2003). Figure 144 shows an experimental setup with PCM modules. It was shown that with a comparatively small amount of PCM the amount of stored heat is increased drastically.

Example 1. One-dimensional simulation of a pcm module in a water tank with MathCad

A numerical method to simulate the performance of the storage with the PCM module was implemented using an explicit finite-difference method. The discretization of the model can be seen in Figure 145.

The heat store of 120 cm height and 20 cm diameter is divided into 12 layers, each one 10 cm thick. Each layer is represented by one knot with one temperature. The top three layers contain the PCM module which is represented similar to the water layers.

The thermal effects taken into consideration in the model were:

- Heat conduction from one water layer to the next water layer,
- Heat loss from the water to the surroundings (ambient),
- Heat transfer PCM-water, and
- Heat loss of PCM.

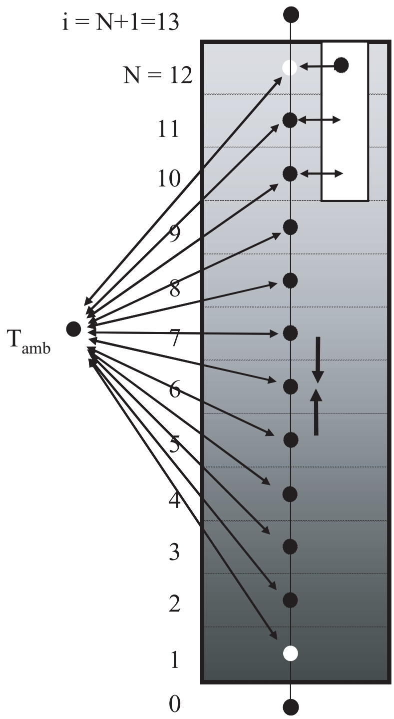

Figure 145. Model of the Cylindrical vertical tank used in experiments

The temperatures were discreticized as $T_{j,i}$, where j corresponds to the time knot and i to the space knot. Black dots in Figure 145 are knots where thermocouples recorded the temperatures in the experiments.

The basic formulas for heat transfer calculations are for thermal conductivity between the water layers:

$$q = \Delta A \cdot \lambda \cdot \frac{\Delta T}{\Delta x} = \Delta A \cdot k \cdot \Delta T \tag{24}$$

and

$$k = \frac{\lambda}{\Delta x} \tag{25}$$

with λ being the thermal conductivity in (W/m K), and $\Delta T/\Delta x$ the temperature gradient.

For heat losses, this can generally be abbreviated to:

$$q = \Delta A \cdot \alpha \cdot \Delta T \tag{26}$$

with α being the heat transfer coefficient in (W/m^2 K).

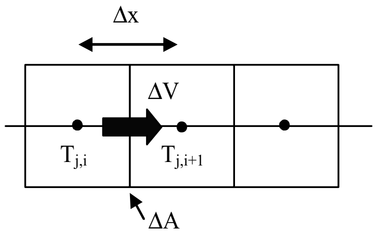

Figure 146. Energy balance of a volume element

From energy balance (Figure 146), heat gain and heat loss in a volume element with one dimensional heat transfer gives

$$\Delta A \cdot \lambda \cdot \frac{T_{j,i} - T_{j,i+1}}{\Delta x} \cdot \Delta t = c_p \cdot \Delta V \cdot \left(T_{j+1,i+1} - T_{j,i+1}\right) \tag{27}$$

where c_p is the volume specific heat capacity. Because

$$\Delta V = \Delta A \cdot \Delta x \tag{28}$$

follows:

$$\Delta A \cdot \lambda \cdot \frac{T_{j,i} - T_{j,i+1}}{\Delta x} \cdot \Delta t = c_p \cdot \Delta A \cdot \Delta x \cdot \left(T_{j+1,i+1} - T_{j,i+1}\right) \tag{29}$$

$$\lambda \cdot \frac{T_{j,i} - T_{j,i+1}}{\Delta x} \cdot \Delta t = c_p \cdot \Delta x \cdot \left(T_{j+1,i+1} - T_{j,i+1}\right) \tag{30}$$

and the temperature change in volume element $i + 1$ from time step j to $j + 1$ is

$$T_{j+1,i+1} - T_{j,i+1} = \frac{\lambda}{c_p} \cdot \frac{\Delta t}{\Delta x \cdot \Delta x} \cdot \left(T_{j,i} - T_{j,i+1}\right). \tag{31}$$

Heat transfer by conduction between the water layers, gives a temperature change based on the temperatures at time step $j - 1$:

$$T^w_{j,i} \Leftarrow T^w_{j-1,i} + \frac{\lambda^w}{c^w_p(T^w_{j-1,i})} \cdot \frac{\Delta t}{\Delta x \cdot \Delta x} \cdot \left(T^w_{j-1,i+1} - 2 \cdot T^w_{j-1,i} + T^w_{j-1,i-1}\right) \tag{32}$$

where $\Leftarrow$ means that the value of the parameter on the left side is calculated by the right side.

Heat loss to the surrounding from each water layer gives additionally:

$$T^w_{j,i} \Leftarrow T^w_{j,i} + \frac{\alpha^{amb} \cdot \Delta A^w}{c^w_p(T^w_{j-1,i}) \cdot \Delta V^w} \cdot \Delta t \cdot \left(T^w_{j-1,i} - T^{amb}\right) \tag{33}$$

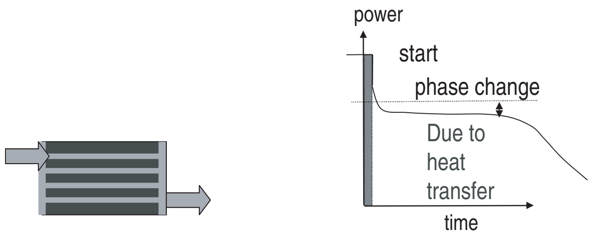

Figure 147. Heat exchanger type and unloading profile

and in those layers with PCM with the reduced water volume:

$$T^w_{j,i} \Leftarrow T^w_{j,i} + \frac{\alpha^{amb} \cdot \Delta A^w}{c^w_p(T^w_{j-1,i}) \cdot \left(\Delta V^w - \Delta V^{PCM}\right)} \cdot \Delta t \cdot \left(T^w_{j-1,i} - T^{amb}\right) \tag{34}$$

With

$$C_{j,i} = \alpha^{PCM} \cdot \Delta A^{PCM} \cdot \Delta t \cdot \left(T^w_{j-1,i} - T^{PCM}_{j-1,i}\right) \tag{35}$$

the heat transfer PCM-water can be calculated to give the temperature rise in the top layers due to the PCM module:

$$T^w_{j,i} \Leftarrow T^w_{j,i} + \frac{1}{c^w_p(T^w_{j-1,i}) \cdot \left(\Delta V^w - \Delta V^{PCM}\right)} \cdot C_{j,i}. \tag{36}$$

Finally, the PCM module lost heat and therefore must be at a lower temperature which is given by its value at time step $j-1$ and the heat transfer PCM-water:

$$T^{PCM}_{j,i} \Leftarrow T^{PCM}_{j-1,i} + \frac{1}{c^{PCM}_p\left(T^{PCM}_{j-1,i}\right) \cdot \Delta V^{PCM}} \cdot C_{j,i} \tag{37}$$

In a last step, if water is moved in the heat store by a pump, the temperatures are shifted from one space knot to the neighboring one by

$$T^w_{j,i} \Leftarrow T^w_{j,i-1} \tag{38}$$

The implementation of such a system in MathCad can be seen in Figure 148.

18.3.2.4. *Heat Exchanger Type*

From the modular type it is not far to the heat exchanger type. Instead of using PCM modules there is a storage container with an internal heat exchanger (Figure 147).

The heat exchanger can be a pipe system as shown in Figure 149 or with flat plate heat exchangers as in Figure 127.

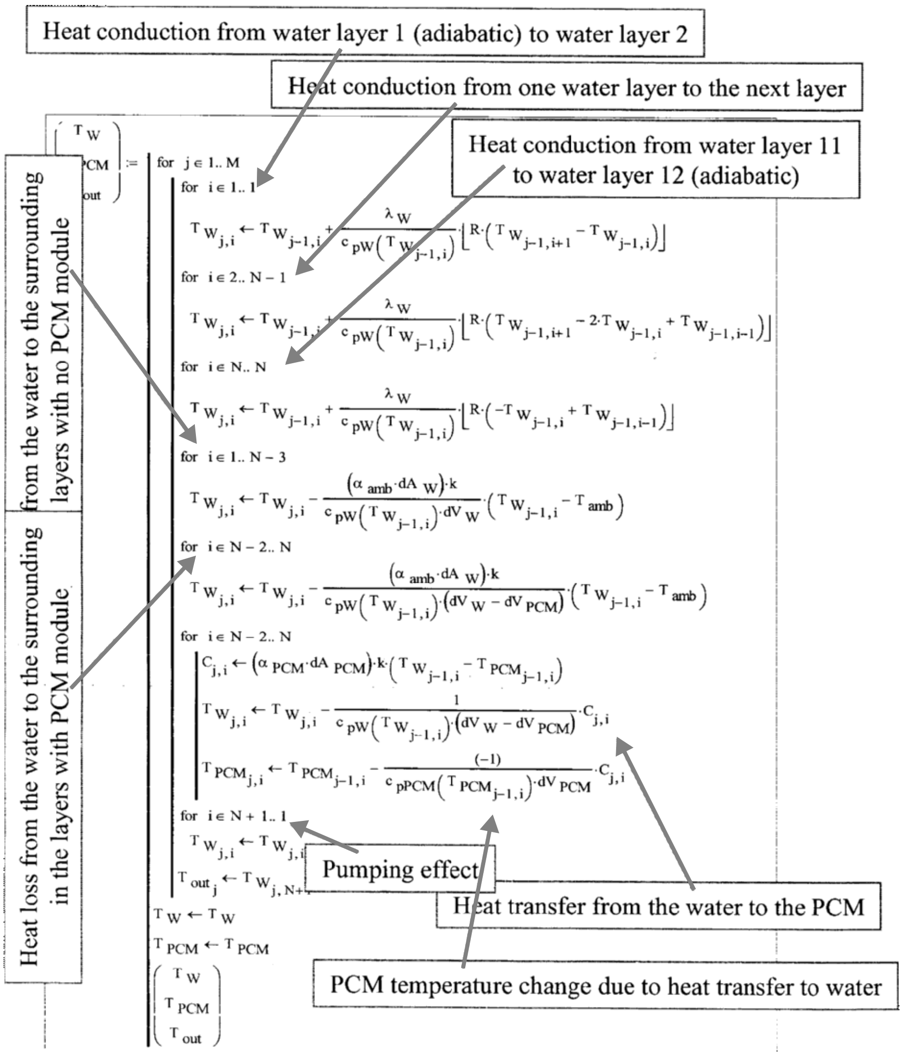

Figure 148. 1D modelisation of a PCM mdule in a water tank

The construction with heat exchangers offers less flexibility in the design of the storage, but drastically increases the volume fraction of the PCM in the storage.

Heat exchanger types have:

- Maximum storage density, as up to 95 vol.% are PCM.
- High power only for the first seconds, as there is only a small amount of heat transfer fluid in the storage.

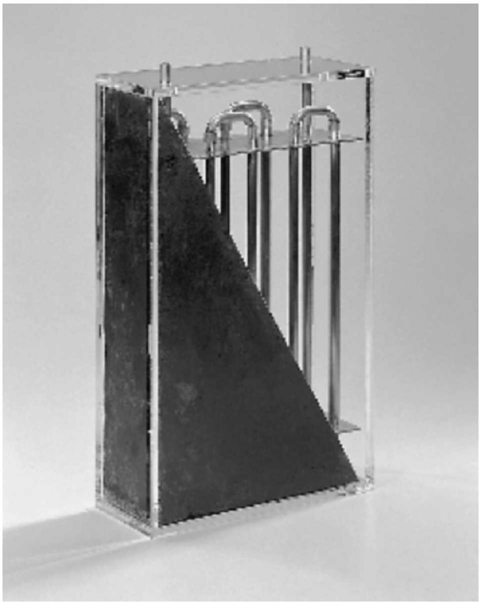

Figure 149. Heat exchanger type with pipes aas heaat exchanger (picture: SGL TECHNOLOGIES)

- A power very much dependent on the proper design of the heat exchanger later.

Gasses due to their low heat capacity do not store and transport as much heat as similar volumes of a liquid. Therefore their low heat transfer coefficients are often not a special problem. From analytical model for a simple heat storage given previously, we know that the temperature of the heat transfer fluid on the outlet of the storage model was

$$T_{out} - T_{in} = (T_{ph} - T_{in}) \cdot \left[1 - \exp\left(- \frac{A \cdot k}{c_p \cdot \frac{dV}{dt}} \right) \right]. \tag{39}$$

If the heat transfer coefficient on the side of the heat exchanging fluid α dominates the value of k than k/c_p is 10–100 times higher for gasses than for liquids. Latent heat storages for heating or cooling gasses therefore tend to have smaller ratios of heat exchanging area A to volume flow $\mathrm{d}V/\mathrm{d}t$ for similar outlet temperatures.

18.3.2.5. *Direct Contact Types*

A rare type of storage is the direct contact storage (Figure 150). Direct contact storages have similar performance than heat exchanger type storages; however

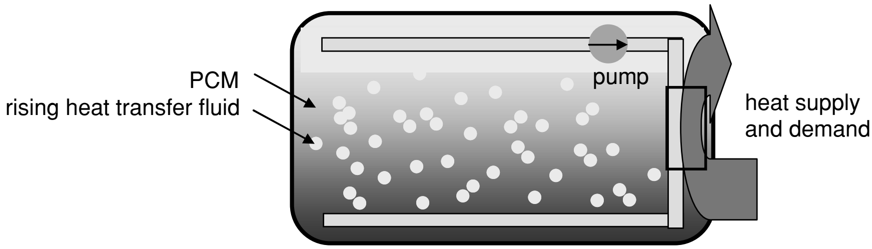

Figure 150. Direct contact heat storage. During loading the heat transfer fluid is heated by the supply from outside. It is then pumped to the bottom of the storage and released in small droplets. These rise, melt the PCM and are collected at the top to go back to the inner circuit

do not need a heat exchanger to separate the storage medium form the heat transfer medium.

They use specially selected storage and heat transfer media that do not mix with each other and separate by themselves due to density differences.

During loading the heat transfer fluid is heated by the supply from outside. It is then pumped to the bottom of the storage and released in small droplets. These droplets rise from the bottom due to their lower density, exchange heat with the PCM by direct contact, melt the PCM and are collected at the top to go back to the inner circuit. Unloading works with cold heat transfer fluid pumped in at the bottom, but otherwise the same.

Direct contact types have:

- Maximum storage density, because up to 95 vol.% are PCM.
- High power all the time, due to the direct contact between storage medium and heat transfer medium.
- They need an extra pump for the heat transfer medium. They are therefore only applied to large storages.

18.3.3. CONCEPT AND EVALUATION OF STORAGE

18.3.3.1. *Concept*

A thermal energy system should be designed considering condition of a building and its load characteristics. As shown in Figure 151, the volume of tank is designed have capacity able to equalize heat required in daytime.

First of all, the hourly heat load of building is calculated. This step is crucial for deciding the volume of storage tank. The heat load can be calculated by manual calculation or computer program, such as TRNSYS or EnergyPlus.

After the hourly heat load is calculated, the capacity of heat source machine would be decided. As total heat produced by heat source machine should be identical to total heat load, the capacity can be calculated dividing head load

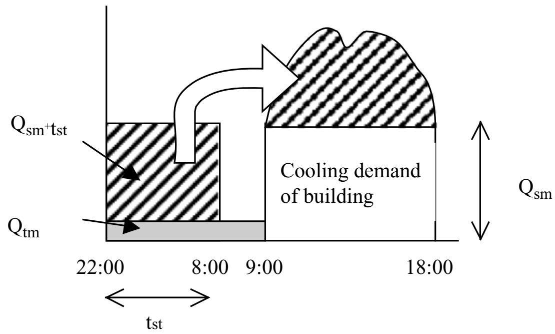

Figure 151. Concept of cool storage

by operating hours. The operating hours is sum of daytime operation and nighttime operation, in which discount tariff is usually applied.

Capacity of heat source machine is calculated by Equation (40).

$$Q_{sm} = \frac{Q_{cd}}{T_p} \tag{40}$$

where Q_{sm} is the capacity of heat source machine, Q_{cd} is the total head load of the peak day, T_p is the operation hours of heat source machine.

The volume of storage tank is decided considering heat to be stored and the efficiency of tank.

$$V_s = \frac{Q_{sm} \times t_{st} - Q_{tm}}{\eta_v \times \Delta t \times \rho \times c} \tag{41}$$

where V_s is the volume of tank, Q_{sm} is the capacity of heat source machine, t_{st} is the operation hours for storage, Q_{tm} is the heat load needed during nighttime, η_v is the volumetric efficiency of the tank, ρ is the density of water, c is the specific heat of water.

18.3.3.2. *Definition of Efficiency*

The efficiency of tank, which is used in Equation (41), can be defined by temperature profile as shown in the figure. Although latent heat of ice cannot be represented by temperature profile, it is represented by imaginary temperature difference, which is a quotient of latent heat by heat of fusion of water. The area made by $\Delta\theta_i$ is equal to stored latent heat. The hatched area in Figure 152 represents sensible heat defined by temperature difference of air handling units for design. The ratio of latent heat and sensible heat to hatched area may be defined as storage efficiency of ice storage tank. Although the value of efficiency can exceed 100%, excess value represents.

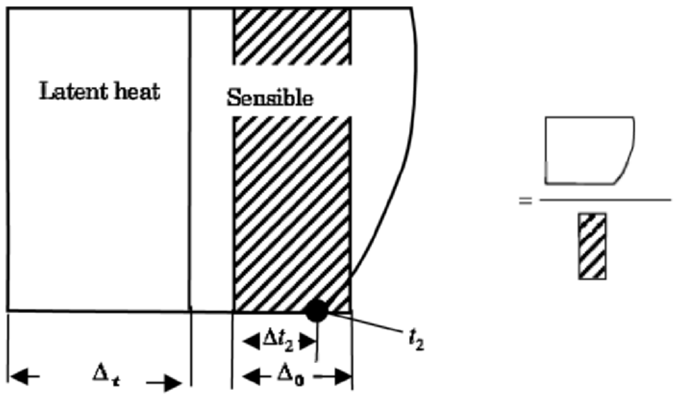

Figure 152. Definition of the efficiency of ice storage tank

18.3.3.3. *Factors Having Effect on the Efficiency*

Since water has highest density at 4 °C, thermal characteristics of ice storage tank are complicated for melting mode. Usually, warm water flows top of the tank, and then water is cooled down going downward. In a way to bottom, water is mixed and reaches to the bottom when its temperature became 4 °C. Usually, without mixing or other measures, it is difficult to get output temperature under 4 °C for ice on coil type storage tank. The outlet temperature gradually went up to 4 °C and was kept until ice was melted (Figure 153).

Ice making

For ice on coil type ice storage tank, water inside the tank is sometimes agitated to enhance heat transfer on coil surface. Small water or air pumps are used for this purpose. This agitation is effective until ice is formed on the coil, because temperature during ice forming is uniformly freezing point. Degree

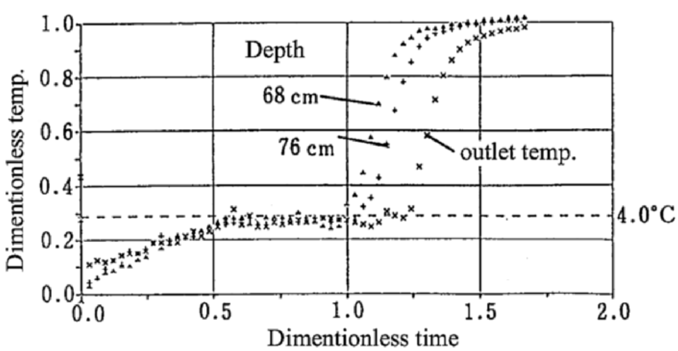

Figure 153. Temperature response of ice storage tank for melting without agitation

of agitation, which means flow rate of water or air, does not have much affect on ice-making rate especially for small ice packing factor.

18.3.3.4. *Ice Melting*
Mixing status in the tank has large affection on outlet response for the ice storage tank with out agitation. If the ice packing factor is small, thermal characteristics of the tank resembles to a stratified water tank. Since the ratio of sensible heat to latent heat is relatively large, conditions should be set to maintain stratification as much as possible. Therefore, large temperature difference in the inlet is preferable.

18.3.3.5. *System Performance*
Since ice storage tanks are used in HVAC system, the performance should be evaluated from system point of view. If the water distribution system had poor performance, performance of tank would be worse than its design value. Furthermore, heat load characteristics of the building also have large relation to system performance. System performance will be different for the building that has peaky load comparing to the building with flat load.

18.4. Determination of Thermophysical Properties

Before any simulation or basic calculation of a heat storage can be performed, the thermophysical properties of the PCM have to be determined. These properties are the most important input parameters for any analytical or numerical model.

18.4.1. REPRODUCIBLE PHASE CHANGE

The technical use of PCM in any application is based on the assumption that the PCM will perform as expected for the life time of the application. Therefore, reproducibility in the phase change properties of the PCM has to be tested. This is performed using any equipment that allows to cycle the PCM around its melting temperature (usually between ±10 and 15 °C), but ensuring that the material is totally melted (Figure 154).

18.4.2. STORED HEAT

The stored heat as a function of temperature is determined using calorimeters. According to the calorimetric formula

$$\Delta Q = \frac{\delta Q}{\delta T} \cdot \Delta T = C_p \cdot \Delta T \tag{42}$$

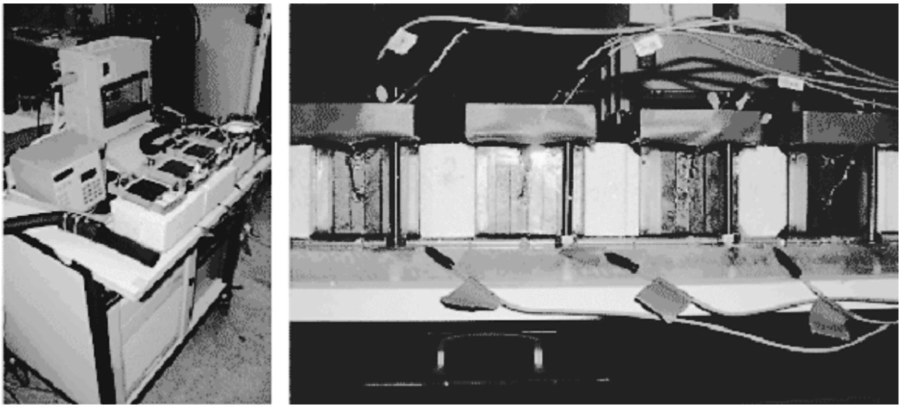

Figure 154. Experimental setup for automatic heating and cooling of four samples and recording of the temperature over time(picture :ZAE Bayern)

a calorimeter supplies or extracts heat ΔQ from a sample and measures the temperature change ΔT to calculate C_p or changes the temperature of the ambient and measures the heat flux to the sample until its temperature is equal to the temperature of the ambient. The definition of C_p above includes all thermal effects, even phase changes. When we talk about measuring the stored heat in a PCM, we actually are referring to three properties: the melting point, the melting enthalpy, and the specific heat. At least the definition for the determination of the melting point and melting enthalpy is not well defined as many PCM have a melting range. It is therefore recommended to give the stored heat in fixed temperature intervals as shown in Figure 155.

T-Interval [°C]	stored heat [J/g]	stored heat [J/ml]
19,5 – 20,5	2,1	3,0
20,5 – 21,5	2,2	3,2
21,5 – 22,5	2,5	3,6
22,5 – 23,5	3,2	4,6
23,5 – 24,5	4,3	6,2
24,5 – 25,5	10,5	15,2
25,5 – 26,5	25,6	37,1
26,5 – 27,5	45,3	65,7
27,5 – 28,5	64,3	93,2
28,5 – 29,5	7,8	11,3
29,5 – 30,5	3,1	4,5
30,5 – 31,5	3,0	4,4
31,5 – 32,5	3,1	4,5
32,5 – 33,5	3,0	4,4
33,5 – 34,5	3,1	4,5
density at 34°C [g/ml]	1,45	

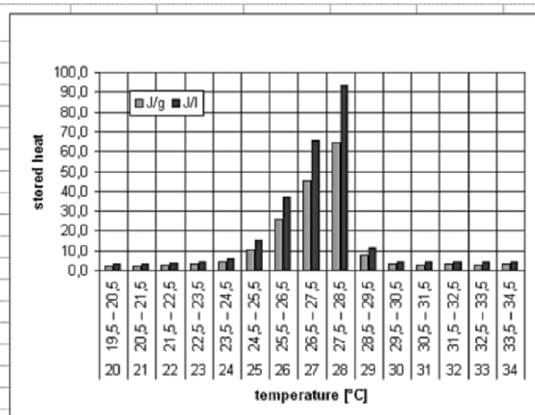

Figure 155. Stored heat of a PCM in fixed temperature intervals

When measuring the stored heat in a PCM using any kind of calorimeter some important and general things have to be considered:

- We need a representative sample size.
- Sample preparation is very important and should be done correctly and repetitively.
- There should be a correct measurement of the temperature by the sensor.
- The heat flow should be correctly measurement.
- Thermodynamic equilibrium in the sample has to be ensured. That requires that the sample is isothermal, otherwise the heat flux cannot be attributed to the single temperature indicated at the sensor. Furtheron, the sample has to be in reaction equilibrium, so there should be no subcooling of the sample.

To perform such a measurement, four methods are available: adiabatic calorimetry (or adiabatic scanning calorimetry), differential scanning calorimetry (DSC), the T-history method, and in-situ measurements. These methods are described here.

18.4.3. ADIABATIC SCANNING CALORIMETRY

Adiabatic scanning calorimetry is a standard method used in some other applications. In this method, a known electric heating power is fed into the sample. Heat losses into the surroundings are minimized by matching the environmental temperature. The sample is therefore in an adiabatic environment. The sample's heat capacity may be obtained from the heating power and the heating rate. Heats of transition are determined from the time integral of the heating power. During a first-order transition, the calorimeter's behavior is quasi-isothermal.

The main advantages of this method when used to analyze PCM are that the samples used are bigger than with DSC (here, the sample size is about 10 ml), and its high precision. Its disadvantages are the high cost of the equipment, its operation is complicated and interpretation of the results is not straight forward, and most important, cooling is not possible with this system.

18.4.4. DIFFERENTIAL SCANNING CALORIMETRY (DSC)

Differential scanning calorimetry (DSC) can be performed in heat compensating calorimeters (as the adiabatic calorimetry), and heat-exchanging calorimeters (Hemminger, 1989; Speyer, 1994; Brown, 1998).

The *power compensating DSC* (Figure 156) uses a sample and a reference. During a controlled temperature program the sample and the reference are held at equal temperatures by adjusting the heat supplied to them by separate

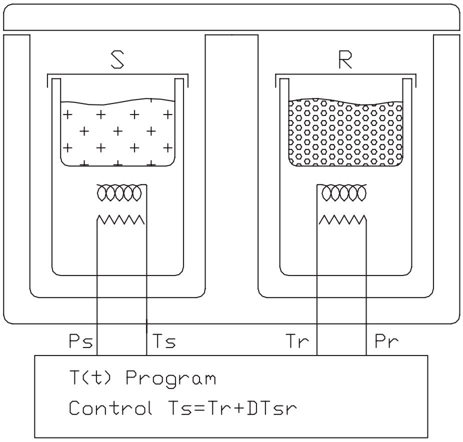

Figure 156. Power compensating DSC (picture: Universitat de Lleida)

electrical heaters. The difference in heat supplied is equal to the heat supplied to or by the sample material.

Heat-exchanging calorimeters are also called *heat flux DSC*. The principle of measurement is that the temperature of the ambient is changed and the heat flux to the sample measured. Now two containers/supports with sample and reference sample are provided with temperature sensors which measure the temperature difference between the specimens. This temperature difference is proportional to the difference in the heat flow rates flowing from the surroundings (furnace) to both specimens. Due to the twin construction, a direct measurement of the temperature difference between sample and surroundings is not necessary; the main heat flow must not be registered, only the differential one. Heat flux DSC can be constructed as disk-type (Figure 157) where the heat flux is realized and measured through the disc shaped support or as cylinder-type (Figure 158), where this is done on the surface of the cylindrical sample and reference.

When measuring PCMs with any DSC, the following aspects should be considered:

- Heating and cooling ramps should be very small to ensure thermal equilibrium within the sample; a maximum of 0.5 °C/min is recommended.
- All characteristics of the test and the analysis should be given with the results.

18.4.5. *T*-HISTORY METHOD

The *T*-history method, proposed by Yinping et al. (1998), is a simple and economic way for the determination of the main thermophysical properties of

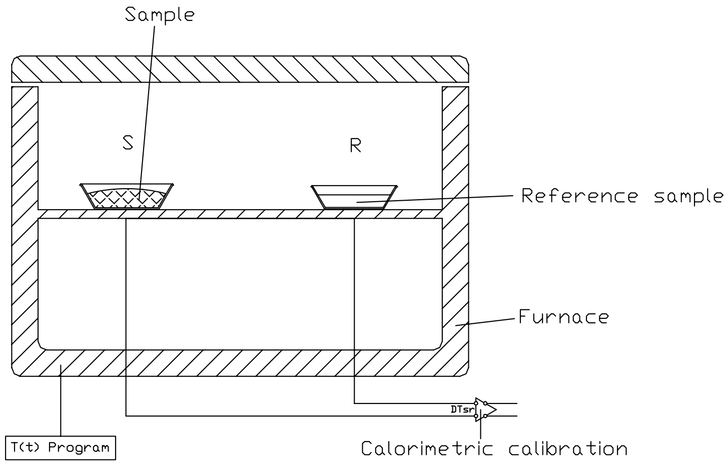

Figure 157. Disk-type heat flux DSC (picture: Universitat de Lleida)

materials used in thermal energy storage based on solid–liquid phase change. It is based on comparing the temperature history of a PCM sample and a sample of a well known material upon cooling down (Figure 159).

The PCM sample and a sample with known thermal properties are subject to ambient air. Their temperature history upon cooling down from the same initial temperature to room temperature is recorded (Figure 160). A comparison

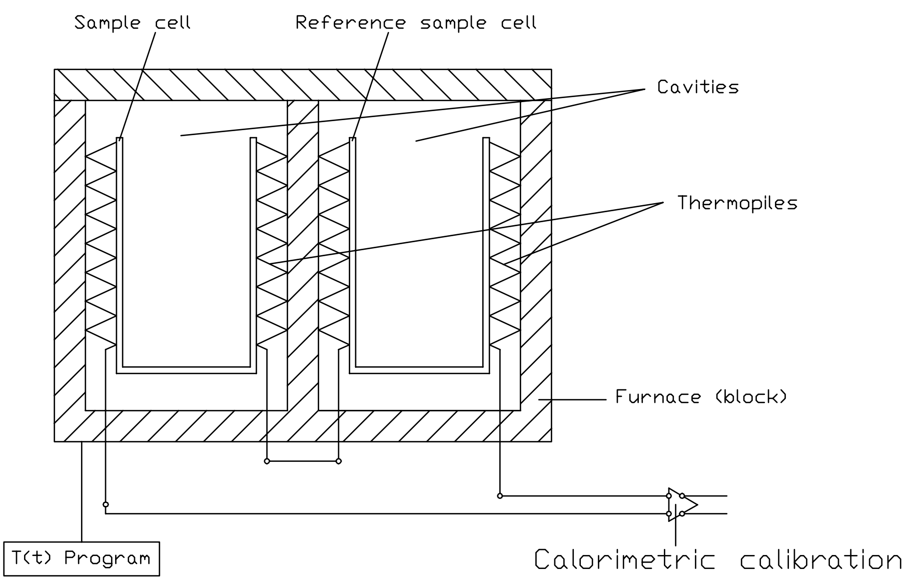

Figure 158. Cylinder-type heat flux DSC (picture: Universitat de Lleida)

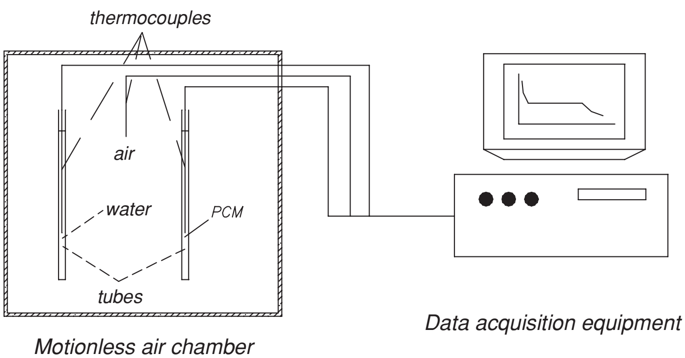

Figure 159. Schematic representation of an experimental T-history set-up (picture: Universidad de Zaragoza)

of both curves, using a mathematical description of the heat transfer, allows the determination of the heat capacity, c_p, and the enthalpy, h, of the PCM from know c_p of the reference material.

The geometry of the tubes allows the heat transfer being considered one dimensional, and each tube to be a lumped system in front of the ambient air. This two conditions are fulfilled when $Bi < 0.1$ (Biot number; $Bi = \alpha R/(2\lambda)$, where R is the radius of the sample, λ its thermal conductivity and α the heat transfer coefficient between the tube and the environment). Once the temperature–time curves of the PCM and the reference substance are obtained (Figure 160), the data can be used to determine the thermophysical properties of the PCM.

The mathematical treatment of the method (Marin et al., 2003) starts with the energy balance of the PCM from the beginning of the experiment to the

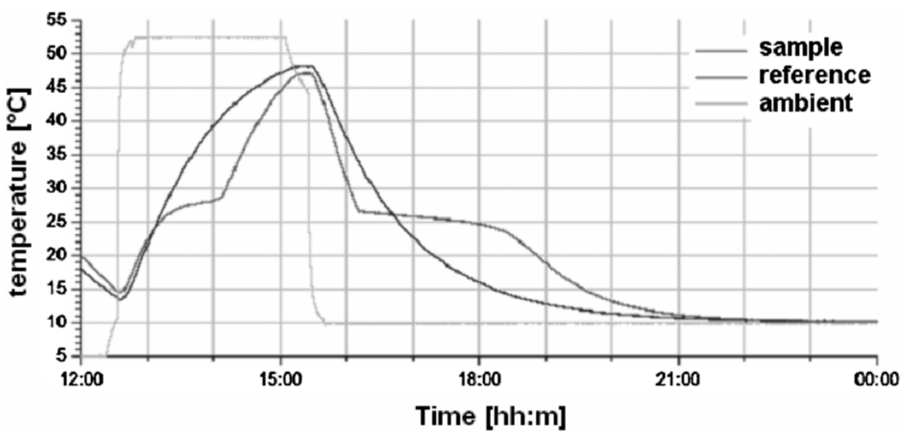

Figure 160. Temperature–time curves obtained in a T-history experiment

moment when the phase change starts ($t1$, T_S), is

$$(m_t \cdot c_{p_t} + m_p \cdot c_{p_l}) \cdot (T_0 - T_s) = \alpha \cdot A_t \cdot \int_0^{t1} (T - T_\infty)\mathrm{d}t = \alpha \cdot A_t \cdot I_1. \tag{43}$$

Hereby I_1 is the integral of the temperature difference (PCM temperature, T, minus ambient temperature, T_∞) vs. time. It considers constant specific heat for the substance and tube, and constant external convection coefficient.

The energy balance of the PCM from the phase change beginning to its end can be expressed as

$$m_p \cdot h_{sl} = \alpha \cdot A_t \cdot \int_{t1}^{t2} (T - T_\infty)\mathrm{d}t = \alpha \cdot A_t \cdot I_2 \tag{44}$$

where I_2 is the integral of the temperature difference vs. time between $t1$ and $t2$.

Finally, the energy balance of the PCM from the end of the phase change to the finish of the experiment would be

$$(m_t \cdot c_{\mathrm{pt}} + m_P \cdot c_{\mathrm{ps}}) \cdot (T_s - T_r) = \alpha \cdot A_t \cdot \int_{t2}^{t3} (T - T_\infty)\mathrm{d}t = \alpha \cdot A_t \cdot I_3. \tag{45}$$

If the same procedure is applied to a tube only containing distilled water, the energy balance is shown in the next two equations:

$$(m_t \cdot c_{\mathrm{pt}} + m_w \cdot c_{\mathrm{pw}}) \cdot (T_0 - T_s) = \alpha \cdot A_t \cdot \int_0^{t1'} (T - T_\infty)\mathrm{d}t = \alpha \cdot A_t \cdot I_1' \tag{46}$$

$$(m_t \cdot c_{\mathrm{pt}} + m_w \cdot c_{\mathrm{pw}}) \cdot (T_s - T_r) = \alpha \cdot A_t \cdot \int_{t1'}^{t2'} (T - T_\infty)\mathrm{d}t = \alpha \cdot A_t \cdot I_2'. \tag{47}$$

This set of equations allows the calculation of the thermophysical properties of the studied substance, the PCM, from the known properties of the reference substance, distilled water:

$$c_{p\mathrm{s}} = \frac{m_w \cdot c_{pw} + m_t \cdot c_{p\mathrm{t}}}{m_p} \cdot \frac{I_3}{I_2'} - \frac{m_t}{m_p} \cdot c_{p\mathrm{t}} \tag{48}$$

$$c_{p\mathrm{l}} = \frac{m_w \cdot c_{p\mathrm{w}} + m_t \cdot c_{p\mathrm{t}}}{m_p} \cdot \frac{I_1}{I_1'} - \frac{m_t}{m_p} \cdot c_{p\mathrm{t}} \tag{49}$$

$$h_{ls} = \frac{m_w \cdot c_{pw} + m_t \cdot c_{p\mathrm{t}}}{m_p} \cdot \frac{I_2}{I_1'} \cdot (T_0 - T_s). \tag{50}$$

For non-pure materials, when the phase change temperature is not constant, the phase change enthalpy can be calculated using the following expression, which includes the temperature change (between T_{m1} and T_{m2}) during phase change:

$$h_{ls} = \frac{m_w \cdot c_{p\mathrm{w}} + m_t \cdot c_{p\mathrm{t}}}{m_p} \cdot \frac{I_2}{I_1'} \cdot (T_0 - T_s) - \frac{m_t \cdot c_{pt}(T_{m1} - T_{m2})}{m_p}. \qquad (51)$$

Another very interesting methodology for storage heat analysis of PCMs is in-situ measurement. In this method, a close loop air is used connected to a small energy storage continent where the samples are located. The air can be heated and cooled, and temperatures and flow are monitored. The data treatment is the same as in the T-history method.

When PCM-objects (building components, PCM composites, PCM in bags, in balls, etc) have to be tested, the measurements can be done with some changes in the T-history set-up and with the in-situ measurement technology.

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19. TEMPERATURE CONTROL WITH PHASE CHANGE MATERIALS

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Abstract. Temperature control is a very suitable application because there you can take advantage of the high capacity of PCMs in a small temperature range. In the case of transport boxes, PCM modules to keep the internal temperature constant within a few degrees for a long time have already penetrated the market. Further applications that are currently under development or in a first stage of market introduction are textiles and clothes, electronic equipment, medical applications, cooling of newborns, catering and even a laptop that includes PCM!

Keywords: PCM, temperature control, applications

19.1. Introduction

Phase Change Materials offer the possibility of thermal protection due to their high thermal inertia. This protection could be used against heat and against cold, during transport or during storage. Protection for food, beverages, pharmaceutical products, blood derivatives, electronic circuits, cooked food, biomedical products, and many others, is possible. Here some of these applications are presented, especially looking for the ones that are already in the market and that emphasize the potential of Phase Change Materials.

An application already in the market is the temperature control utilizing PCM for transportation of pharmaceutical goods or other temperature sensitive goods. Also in the market is the utilization of PCM for cooling or heating of the human body, since it has been demonstrated both for personal comfort and for medical therapy. Passive cooling of buildings and of telecom cabinets are examples of widespread applications.

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 315–321.

19.2. Strategy

When talking about temperature control, what do we mean? An example from everyday life is the storage of frozen food like pizza, ice cream, vegetables or others. In many cases, the package indicates the maximum storage time depending on the temperature of storage. If the food is not kept below a certain temperature it often has to be eaten at the day of purchase.

Let us think that a temperature sensitive good is inside a well insulated box. To stabilize the temperature at the desired value, the thermal mass of the interior is strongly increased by adding PCM with suitable melting point and mass. Everybody has done this before when adding ice packs to a picnic basket.

19.3. Containers for Any Kind of Temperature Sensitive Goods

The best known application of PCMs for transport or conservation of materials at a constant temperature are containers with removal parts containing a PCM (usually water, nowadays many other products) that must be kept in the refrigerator before use, and that keep a low temperature in the container for a period of time (Figure 161).

Some companies only commercialize the PCM pads (Figure 162). Such pads can be used to keep products warm (the pad must be conditioned in an oven and/or microwave oven) or cold (conditioning is done in the refrigerator) during shipment.

Figure 161. Rigid encapsulator (picture from Va-Q-tec)

Figure 162. Soft encapsulator (picture from Climator)

PCM transport boxes have also been adopted in vacuum boxes to improve their performance (Figure 163).

19.4. Containers for Beverages

One application that has been commercialized is the so-called “isothermal water bottle” especially developed for cycling, but that could be used by any other sporting person. It is a double wall bottle, with ALCAL® being the PCM. The bottle has capacity for about 0.5 l, and it has to be held in the refrigerator for the PCM to solidify and then the bottle will keep the beverage cold.

This concept could be used in many other products, such as isothermal maintenance of fresh drinks (isothermal container for champagne, cava, wine, etc) and warm drinks (soups, tea, coffee, etc).

Figure 163. Transport box (picture from Va-Q-tec)

19.5. Containers for Food

In many catering applications, cooked meals are produced in one point and have to be transported to another place where they are eaten. Some examples are transport of cheese, salads, frozen deserts, confectionery, or fish. PCMs containers could also be used to avoid breaking the cold chain during transportation of precooked meals, foie-gras, smoked salmon, milk derivates, ice-cream, and many others.

One example of such an application that has already been comerzialized are pizza-heaters. The use of a container with PCM with the right melting temperature allows multiplying by three the length of time the food is kept above 65 °C.

The same concept could be used for food distribution in hospitals, schools, etc, or in devices to heat up feeding bottles, and other food containers.

Other developments are a container for hot food transportation at controlled temperature between 70 °C and 85 °C, and a container for ice-creams to be held below −8 °C.

19.6. Medical Applications

The transportation of blood and its by-products is very critical with respect to temperature. Some products need to be transported between 20 and 24 °C, others between 2 and 6 °C, and others between –30 and –26 °C. A container was designed and produced to allow the transport of such blood products between the hospital and the transportation vehicle (which is already conditioned at the right temperature), and between the vehicle and the final destination.

Another medical application are hot or cold pads to treat local pain in the body (Figures 164 and 165).

One more application for medical purposes is a mattress for operating tables, which can be use to avoid the decrease of the body temperature during

Figure 164. Hot cushion for medical purposes, (picture from Rubitherm)

Figure 165. Rubitherm cold product for cooling therapy

long operations, or during operations of burned people. The mattress would be heated up electrically before its use (the PCM would be with a melting temperature of about 37 °C), and it would release the heat during the operation. This application is being tested as a prototype.

A new product being developed currently is a mattress to treat hypoxia (lack of oxygen) in newborn babies. It is known that reducing the body temperature of the new born has a positive effect; therefore they are treated with cold towels, medicine, chilled rooms to keep the temperature. This treatment could be substituted by the treatment with a PCM mattress, thereby optimizing and prolonging the temperature regulating effect.

Heat can be a problem for professional athletes both in training and at competition. Heat can also be a problem for other people like elderly people, people with some diseases and children. If the body-temperature can be reduced most people would feel more comfortable, perform better, be able to keep concentrate better and for a longer time and the risks of dehydration and fatigue is reduced. New products to address this problem were developed, such as wrist-cooler, a neck-cooler, a cap and a vest (Figure 166). All these

Figure 166. Wrist-cooler, neck-cooler, cap and vest with PCM for thermal protection (pictures from Climator)

Figure 167. Left, PVC packet with TEAP PCM; center, top view of Battery Jacket; right, side view of Battery Jacket

products are being used by more and more athletes and professional workers in extreme conditions.

19.7. Electronic Devices

This is one of the most developed and commercialized applications of PCMs. For example, a protection for electronic devices from heating up by solar isolation when installed outdoors has been studied. A PCM developed to change phase at 35 °C would absorb solar radiation during day, and thereby prevent the electronic device to reach certain security temperature. The heat would be released during the night, allowing the cycling performance of the PCM. Nowadays, France Telecom has a prototype which is being tested.

Many telecommunication equipments must be located outdoors and powered by batteries. The company TEAP, together with Power Conversion Products and MJM-Engineering, has developed a battery jacket that minimizes the effects of peak heat loads in the day (Figure 167). The use of TEAP TH29 allows the heat loads to be absorbed in the daytime and released during night.

A passive cooling system for communications equipment has been developed by Climator AB, starting with a collaboration with Ericsson Telecom. This work lead to the production of ClimSel Cooling Systems that nowadays have been in operation for several years, with 100% function and no

Figure 168. PCM pad for laptops (picture from Climator)

maintenance needs. This system has been installed in other sites and applications, with similar good results.

Another problem studied and solved with PCM products is the overheating of laptop computers. One of the possible alternative methods to the used the noisy traditional fans is to add a PCM pad under the computer chassis (Figure 168). When the temperature of the computer chassis rises to the melting point of the PCM, the temperature of the computer will be kept close to this temperature.

Finally, we can report that a shelter for telecommunications with over 300 kg of PCM for thermal protection is commercialized in India.

20. APPLICATION OF PCM FOR HEATING AND COOLING IN BUILDINGS

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Abstract. Research for the application of PCM in buildings has been focused in recent years on three fields. The first one is the reduction of temperature swings of lightweight buildings by increasing their thermal mass. This is done by incorporation of PCM into building materials. The second one is the cooling of buildings through intermediate storage of cold from the night or other cheap cold sources. If the cold is for free, as with cold from night air, this is also called free cooling and very promising with respect to energy saving. A third field of application is for heat storage in space heating systems. Here the main advantage is a reduction of storage volume by a factor of three or more. This section describes the basic strategies for different systems for heating and cooling of buildings as well as the state of the art in R&D.

Keywords: PCM; phase change; latent heat; melting; heat storage; cold storage; comfort requirements; space heating; space cooling, free cooling

20.1. Human Comfort Requirements

The energy demand to ensure a comfortable environment for humans in buildings is increasing worldwide. In recent years, especially the energy use for cooling and air conditioning is rising fast and causing increasing problems. Heating and cooling in buildings is strongly connected to human comfort requirements. The criteria considering the human comfort with respect to indoor climate all relate to the heat exchange between the human body and the environment via thermal conduction, radiation, evaporation (sweating) and convection caused by temperature differences. The main criteria and recommendations for human comfort are:

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 323–348.

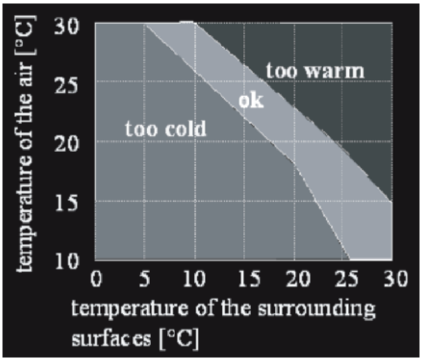


Figure 169. Connection between the influence of the temperature of the air and room surfaces on the human comfort

Temperature

- temperature of the air: 22–26 °C are recommended in moderate climates.
- temperature of the room surfaces which exchange heat with persons by radiation: changes <5 °C vertically, <10 °C horizontally.
- temperature of the air and room surfaces in combination: Figure 169 shows the influence of these temperatures in combination, that is with rising air temperature the temperature of the surrounding surfaces has to be reduced to maintain human comfort and vice versa. The absolute values of the temperatures and the location of the boundaries between acceptable and unacceptable conditions is however subject to intense discussion.

Relative humidity

- a relative humidity of 30% to 70% is recommended; about 50% is considered to be best.

Air velocity

- an air velocity <9 m/min in winter and <15 m/min in summer is recommended.
- A minimum air movement is necessary for moisture removal, especially in summer and in humid climates.

PCM offer unique possibilities to improve energy systems in the field of heating and cooling in buildings. The potential for applying PCM comes from the small temperature range that is regarded to be comfortable for humans. If a temperature change of ±5 °C is taken as an acceptable upper boundary, water as storage medium can store $2 \times 5\,°C \times 4\,J/g\,°C = 40\,J/g$. An average PCM has already a melting enthalpy of 200 J/g, which is a storage density five times that of water. For a temperature change of ±2 °C, as is recommended above to maintain human comfort, the ratio is more than 12. That means PCM can store about 12 times as much heat per mass as water within the comfortable temperature range. In contrast to ordinary building materials that store heat or cold sensibly, PCM can also stabilize the room temperature as they release heat automatically when the ambient temperature is below the melting temperature and store heat when it is rising above the melting temperature.

20.2. General Comments

20.2.1. SPACE HEATING AND COOLING

Before trying to solve a heating or cooling problem in any application using PCM, it is advisable to restrict the heat or cold flux that should be buffered beforehand. For the case of a building, this means that the construction should be done in a way that heat input in summer and loss in winter are already minimized. This can be done preferring a building shape with small surface to volume ratio or multiple unit buildings.

The orientation of the building should be with small surface to the dominating wind direction to minimize convection heat loss, large area to the south to maximize solar gain in winter, and with sleeping rooms in the northern part of the building as lower temperatures are desired for sleeping. The outer walls of the building should be white to reduce solar heat input and the roof construction should be as shown in Figure 170.

20.2.2. DOMESTIC HOT WATER

Domestic hot water is a field of application for PCM that poses two severe problems. The first problem is that drinking water regulations usually strongly

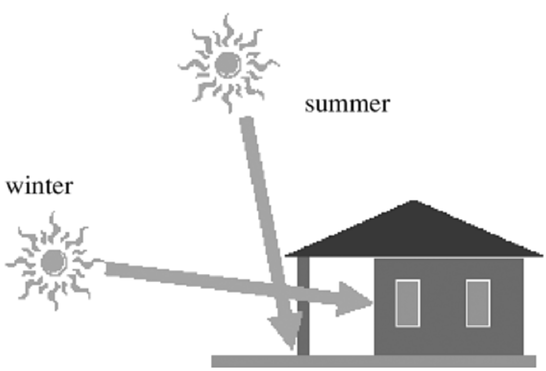


Figure 170. Intelligent roof construction to reduce solar heat input on the south facing wall in summer and maximize it winter

restrict materials that can be used in contact or with possible contact to drinking water. This might be avoided if an additional heat exchanger is used to separate the heat transfer fluid in the storage from the drinking water. The second problem is the high power requirement for domestic hot water. A water flow of 1 l/min ($\cong$1 kg/min) heated up from 20 °C to 40 °C requires 20 °C $\times$ 4 kJ/kg °C $\times$ 1 kg/min = 80 kJ/kg $\times$ 1 kg/60 s $\cong$ 1.25 kW of heating power! The competition to hot water storage with direct discharge is therefore very strong and usually PCM is not used to supply domestic hot water.

20.3. Applications for Space Heating

20.3.1. GENERAL CONSIDERATIONS FOR SYSTEMS WITH HEATING FROM A WARM SURFACE

For a heating system that heats from a warm surface, the heating power $q = A \cdot \alpha \cdot \Delta T$, where A is the surface area, α the heat transfer coefficient and ΔT the temperature difference between the surface and the ambient air. Typical values are:

- Range of ΔT: ΔT can be increased by raising the surface temperature, but at the expense of reduced efficiency where the heat is generated. $\Delta T =$ 10–20 °C for low temperature heating, usually using large surface areas A. $\Delta T =$ 20–40 °C for regular heating systems with separate heating units.
- Heat transfer coefficient α (for free convection): for heating from a wall $\alpha =$ 3–6 W/m^2 K (surface vertical), for floor heating $\alpha =$ 6–10 W/m^2 K (surface horizontal) and no heating from ceilings, as the temperature gradient would suppress convection!

This also includes heating with hot water, as the hot water usually is used to raise the temperature on a heating surface.

20.3.2. GENERAL CONSIDERATIONS FOR SYSTEMS WITH HEATING BY SUPPLYING HOT AIR

For a heating system that heats by supplying hot air into the room, the heating power is $q = T \cdot \Delta V/\Delta t$, where $\rho \cdot c_p$ is the volumetric heat capacity of the air, $\Delta V/\Delta t$ the volume flow and ΔT the temperature difference between room air and supply air temperature. Hera

- ΔT can be raised, but at the expense of reduced efficiency.
- Heat capacity of air $\rho \times c_p \cong$ 1 J/LK (water has approx. 4 $\times 10^3 \cdot$J/LK).
- Large volume flows $\Delta V/\Delta t$ create noise and can lead to feeling of uncomfort if the air moves too fast.

A big advantage of such systems is however that heating and ventilation can be done at the same time.

20.3.3. EXAMPLES FOR SPACE HEATING WALL ELEMENTS FOR EXTERIOR WALLS

Wall elements for exterior walls to heat a building have been investigated for several decades. The basic idea is explained in Figure 171.

On an ordinary wall (Figure 171 left), the temperature gradient within the wall is dominated by heat loss from the heated interior to the cold outside. Solar heat gains on the outer surface are lost through free convection or forced convection in case of wind. To reduce the loss of solar input "transparent insulations" were developed. A transparent insulation consists of a material (called transparent insulation material = TIM) or a layered structure that lets the solar radiation through to be absorbed on the surface of the regular wall, but reduces the transfer of the heat by convection to the outside. The heat flow through the wall to the outside is thereby minimized or even reversed. In that case the wall element acts as a heater (Figure 171 center).

Transparent insulations are nowadays state of the art and available as commercial products. One of the main disadvantages of the combination of ordinary walls with transparent insulation is, that to ensure enough storage capacity the wall needs to have a sufficient thickness. PCM offer here a very promising advantage.

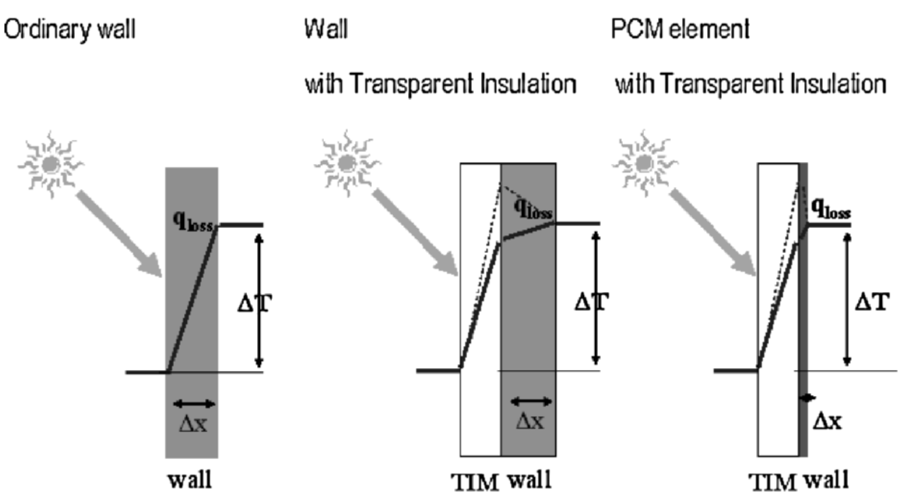
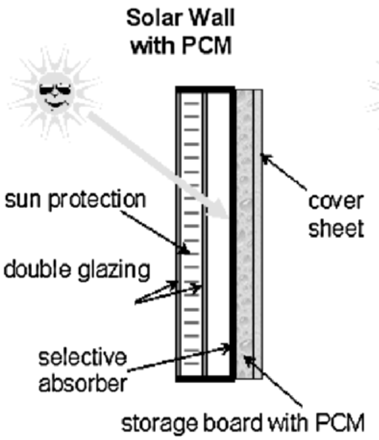


Figure 171. Different wall constructions to use solar radiation for space heating



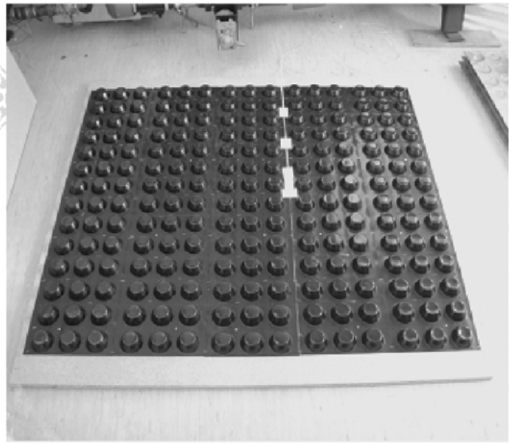


Figure 172. Sketch of the solar wall (left) and PCM-wall using TEAP PCM (right)

20.3.3.1. *Solar Wall with PCM*
Within the R&D project "Innovative PCM technology" (funded by the German Ministry of Economics and Labor, BMWA; Nr. 0327303) the Glaswerke Arnold and the ZAE Bayern investigated the replacement of the ordinary wall by a PCM-wall (Figure 171 right). The system design and one of the investigated concepts for the PCM-wall are shown in Figure 172.

A double glazing was used as transparent insulation. Figure 173 shows some experimental results from an installation in a test building.

The PCM starts to absorb heat and melt as soon as the absorber is heated above about 25 °C. The melting is completed before the absorber temperature

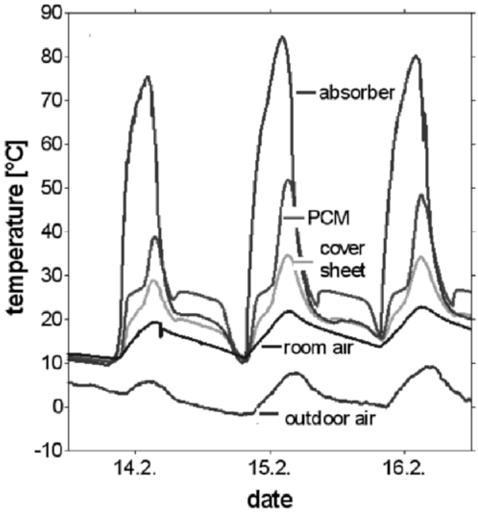


Figure 173. Experimental results of a solar wall with PCM from an installation in a test building

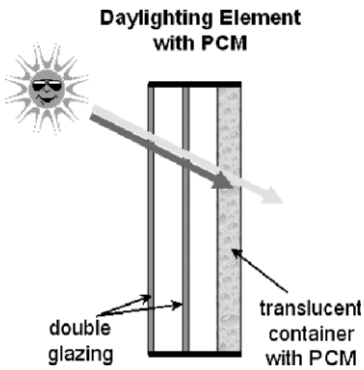


Figure 174. Sketch of the daylighting element

falls below the melting point again. After a short time, where the PCM is subcooled, it releases the heat slowly throughout the night. The cover sheet on the inner side of the wall is thereby kept close to 20 °C at almost all times. The temperature of the cover sheet is always above the room air temperature and the room is thus heated by the wall.

20.3.3.2. *Daylighting Element*

The idea of the solar wall with PCM can be developed further. If the PCM is at least partly transparent, the whole system becomes transparent and can in addition to the heating of the building also be used for illumination. This idea has also been investigated by the Glaswerke Arnold and the ZAE Bayern within the R&D project "Innovative PCM technology". Figure 174 shows a sketch of the investigated system, again with a double glazing as transparent insulation.

Figure 175 shows two concepts for the transparent PCM wall. Both show a diffuse illumination and a transparency which changes to some degree between the solid and liquid state of the PCM.

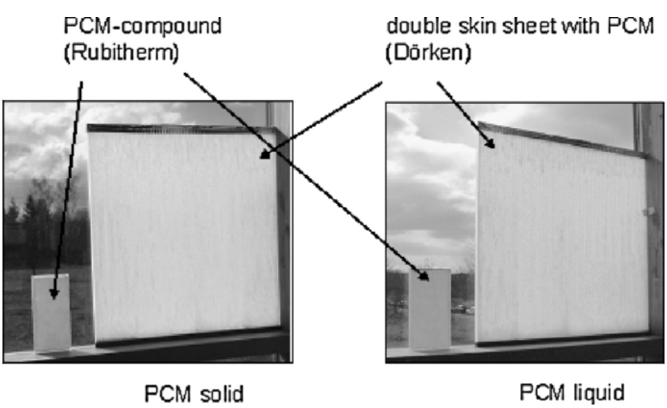


Figure 175. Concepts for a transparent PCM wall for the integration in a daylighting element

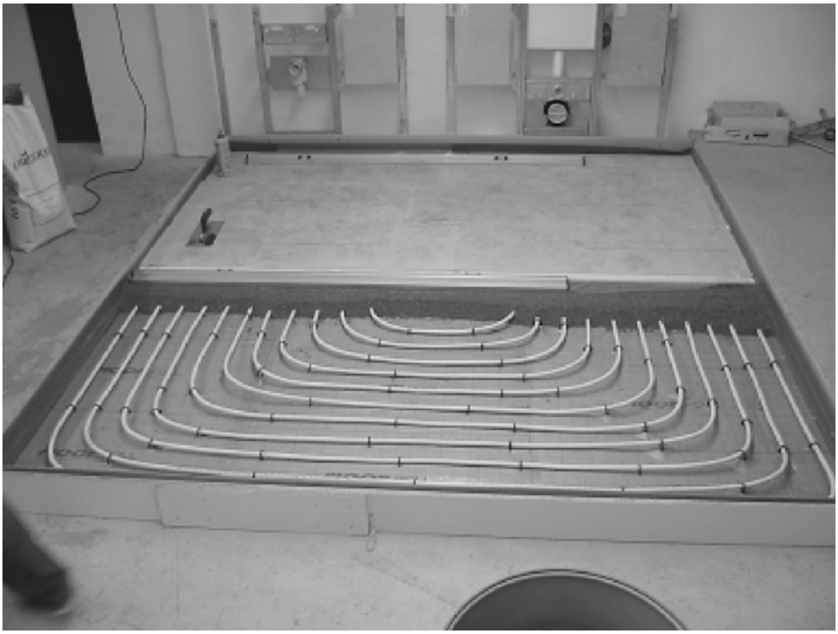


Figure 176. Floor heating system with integrated PCM storage (Rubitherm PCM granulate) using hot water as heat supply (picture: Rubitherm/Germany)

A similar system has also been investigated by the company Inglas. More information can be found at http://www.inglas.de/glas/heiz/pcm.html.

20.3.3.3. *Floor Heating Systems*

Floor heating systems have already been used by the Romans. Nowadays, in addition to warm air from a fire as heat supply, they can also use solar hot water or electricity.

20.3.3.4. *Floor Heating System with Hot Water*

Figure 176 shows a floor heating system developed by Rubitherm GmbH (http://www.rubitherm.de).

The storage material, a granulate infiltrated with paraffin, is heated by hot water pipes. Even though the floor thickness is reduced, 0.5 kWh/m^2 of heat can be stored in the floor. An additional advantage of the concept is that no drying of any component after installation is necessary, thereby reducing installation time and cost compared to common systems.

20.3.3.5. *Floor Heating System with Electrical Heating*

A floor heating system with electrical heating has been developed by Sumika Plastech (Japan, http://www.sumikapla.co.jp/e/main_pd05.html). The

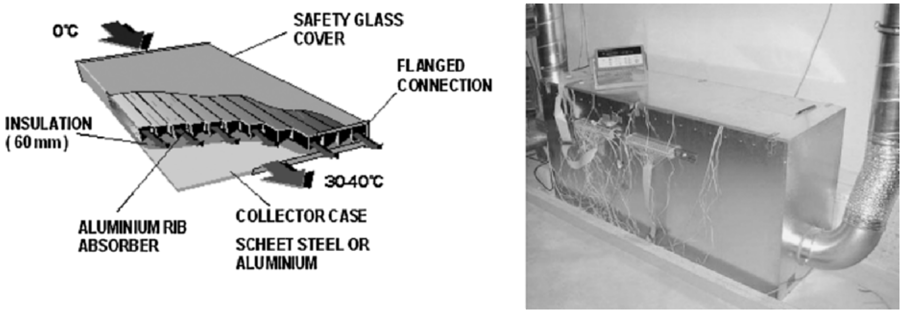


Figure 177. Sketch of a solar air collector (left) and storage block prototype (right)

working principle is similar to the example above, however here the electrical heating wires are in contact with PCM-modules instead of PCM-granulate.

20.3.3.6. *Solar Air Systems*

Solar-air systems are well suited for heating fresh air circulated into buildings. However, due to the low heat capacity of the heat transfer medium air, the temporal difference between heat demand and solar heat availability is a problem. The use of PCM seems to be very promising, as the storage density is high and PCM will also have a regulating effect on the supply air temperature. Within the R&D project "Innovative PCM technology" the company Grammer and the ZAE Bayern investigated the application of PCM is such solar air systems. PCM was tested in four different solar-air system components:

- collector absorber.
- ventilation pipes (double pipe with PCM in the partition).
- hypocaust (hot-air floor heating with a PCM layer as heat storage).
- storage block with PCM to heat air via a heat exchanger connected to the collector.

It was concluded that the storage block is the most promising application. A prototype with approx. 4.1 kWh latent heat storage capacity was then built and is currently operated at Grammar. The storage (Figure 177) is charged via a 20 m^2 collector with an air flow of 180–300 m^3/h.

20.3.3.7. *Storage for Heating with Hot Water*

Storages for space heating using water as heat transfer medium have been investigated frequently during the past decades. For regular heating systems with heating units they work in the temperature range of 40–60 °C. Due to the good heat transfer coefficient to water of several 100 W/m^2 K and the high heat capacity of water, the bottle neck for heat transfer is usually within the PCM. Therefore, different ways to improve heat transfer in the

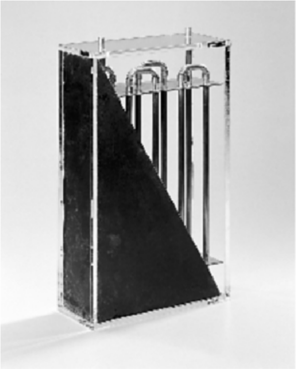


Figure 178. Model of the developed heat storage for space heating (picture: SGL Technologies)

PCM have been investigated in the past. Within the R&D project "Innovative PCM technology" Bosch, SGL Technologies, Behr, Merck, and the ZAE Bayern developed a new heat storage concept. The storage is based on a PCM-graphite-compound as heat storage medium to ensure a high storage density and good heat transfer. The storage contains a pipe heat exchanger and is filled with the PCM-graphite-compound (Figure 178). Currently more than 1,500 cycles have been tested on the storage and no performance loss was observed.

20.4. Applications for Space Cooling

20.4.1. GENERAL CONSIDERATIONS FOR SYSTEMS FOR SPACE COOLING

The demand for space cooling has been rising strongly during the past decade and is expected to continue this way. Reasons for the rising demand are an increasing number of heat sources in buildings (TV, computer, light, . . .), improved insulation of buildings to reduce energy demand for heating, a trend in architecture to use glass facades that lead to large solar gains and to use light weight construction (Figure 179). This can save cost and increase the flexibility of construction, however in contrast to massive construction with stone walls there is little thermal mass to buffer temperature fluctuations. These fluctuations increase the demand and the maximum necessary cooling power.

Strategies to reduce energy demand for cooling can therefore be

- reduction of temperature fluctuations by increasing the thermal mass of a building.
- reduction of energy consumption in cold production by increasing the efficiency or by using natural cold sources.

Figure 179. Typical light weight construction with wooden frame and glass windows

The selection of the PCM is a very critical point for space cooling. On the demand side, 26 °C that are considered the maximum comfortable temperature (Figure 180). If a temperature difference of 3 °C is used to transfer heat from the PCM to air, the maximum melting temperature of the PCM is 23 °C. The lower the melting temperature of the PCM in the cold storage, the higher the cooling power on the demand side. On the other hand, the lower the melting point, the more difficulties arise in the search for a suitable cold source on the supply side.

Cold sources are selected with respect to

- cold source temperature.
- availability and reliability.
- efficiency (running cost) and investment cost (initial cost).

Some cold sources are “natural” sources (Figure 181):

- cold night air; in Germany usually below 22 °C, but this is not reliable!
- evaporative cooling; depends on humidity, maybe down to 12 °C.

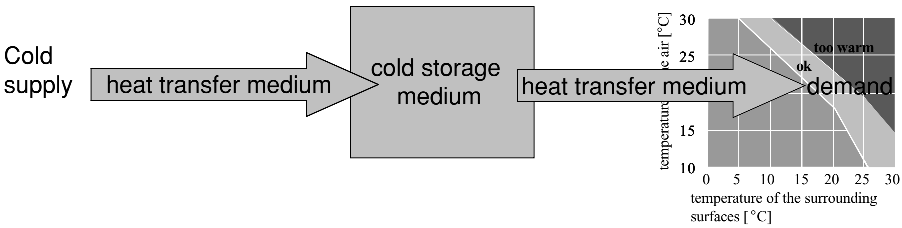


Figure 180. Connection between cold supply, storage and demand for space cooling

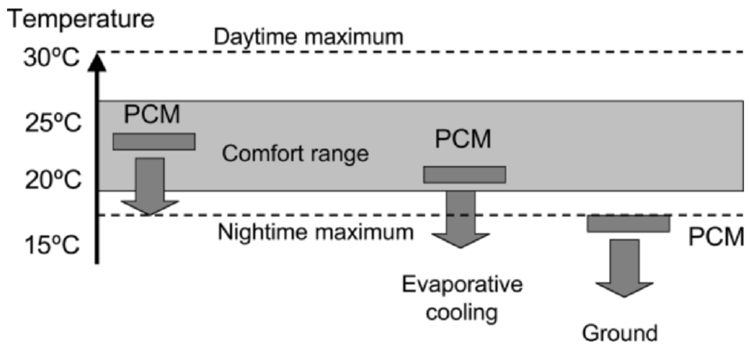


Figure 181. Natural cold sources as heat sinks to reject heat stored in the PCM

- ground water; maybe down to 10 °C.
- natural ice; if available in sufficient amounts and with storage space. That might not be reliable every year.

Natural cold sources show the highest energy efficiency as cold is for free or with little energy consumption. However, they might have restrictions with respect to reliability (warm summer nights) and cooling power (small temperature differences).

Some cold sources are "artificial" sources (Figure 182):

- chillers can go down to below 0 °C. They allow cold storage in artificial ice and therefore high storage density and cooling power and good reliability.

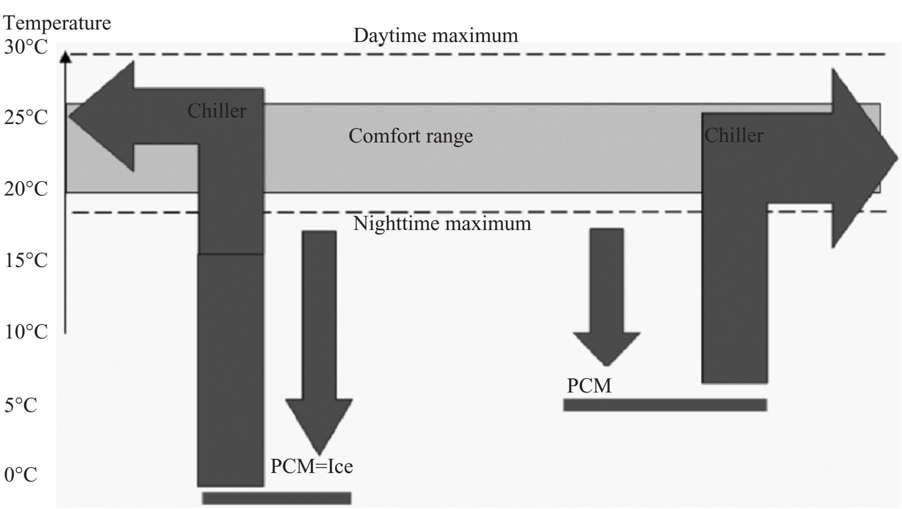


Figure 182. Chillers producing cold to reject heat stored in the PCM

- with other PCM having higher melting temperatures than ice, the chiller efficiency can be increased. The disadvantage is the higher cost for the storage material compared to ice which is for free.

Artificial cold sources show a lower efficiency than natural cold sources, as low temperature cold has to be produced. On the other hand, they are reliable as the chiller works independent of other climatic conditions.

Applications of thermal energy storage (TES) in cooling systems would be one of the most popular fields in warm countries such as Japan. In Japan, they had started in 1960s in order to reduce size of refrigeration machines that had high proportion of HVAC installation cost at that time. In late 1970s, the peak demand of electricity began to increase as comfort cooling was widely used in many buildings. Consequently, electric utility companies offered nighttime discount to shave the peak. TES system could take advantage of discount electric tariff as well as reduction of refrigeration machines.

Table 23 shows the electric tariff of Tokyo Power Electric Company in Japan. They offer 70% of discount in nighttime for TES contract. This policy gives users additional economical merits

Therefore peak shaving has following advantages.

- Lower machine capacity.
- Machines can be operated in high efficiency.
- Reduce demand cost for facilities.
- Leveling of power generation throughout a day or seasons.

TES applications in cooling systems could be achieved by sensible and latent storage. The sensible heat storage usually uses water as storage medium. Water is safe and inexpensive, and has large specific heat. As it also has large heat of fusion, it is used for latent heat storage, or ice storage.

For the sensible heat storage using water, water should not be mixed during operations, because temperature difference must be maintained for sensible storage. To keep water temperature separate, multi-connected tanks and a stratified water tank are used.

TABLE 23. Electric tariff in Japan

	Daytime	Night time	Discount rate
Summer	11.08 JPY (0.15 EUR)	3.24 JPY (0.044 EUR)	0.707
Other seasons	10.07 JPY (0.14 EUR)	3.24 JPY (0.044 EUR)	0.678

Night time is from 22:00 to 8:00. 100 JPY = 0.73 EUR (2004 November).

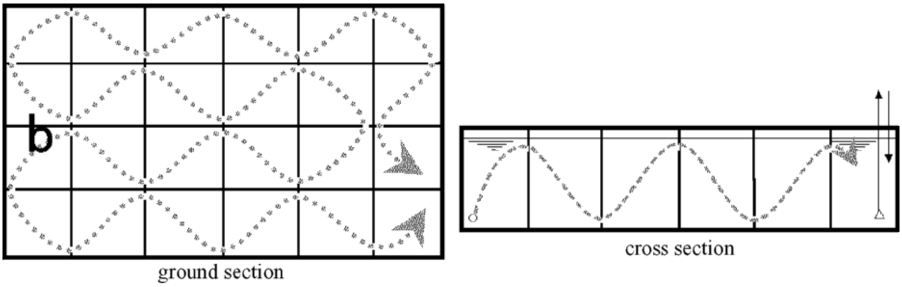


Figure 183. Diagram of multi connected water tanks

There is a structure in basement, which should be built against earthquake in a Japanese building. The space among this structure can easily be used as a water tank if it is made waterproof and insulated. The storage tank consists of several small spaces shown in Figure 183. Water flows through small tanks changing its inlet and outlet to avoid making short circuit. It is important not to mix temperature among tanks so that the storage efficiency increases. The flow pattern of the whole tank becomes similar to piston flow if a lot of tanks are connected.

The temperature stratification is used to avoid mixing in a water storage tank. These tanks are not only deep ones but also shallow ones as shown in Figure 184. The degree of stratification depends on momentum of water and density or temperature difference at inlet. The appropriate condition at inlet can stratify water in a tank whatever the depth of a tank is. As the deep tank has difficulties of construction, the shallow tanks in basement are often applied.

Figure 185 shows an office building equipped with stratified water tank. The deep water tank is installed in unused space of the building. The volume of tank is 375 m^3 that can store 3,500 kWh of cold heat with a 90 kW heat pump chiller. Inside of the water tank, an apparatus, which decreases inlet velocity of water is installed.

20.4.2. ICE STORAGE

20.4.2.1. *Natural Ice*

In some climate, if outdoor air is cold enough, it can be used for making ice. Snowfall is another source of cold storage. The snow/ice can be stored to the summer and used for cooling applications. Snow and ice storage for cooling application is an old technique. In some countries, the fallen snow and the ice in lake were gathered in a storage house or cave and used as cold source in the summer before the utilization of refrigeration machines.

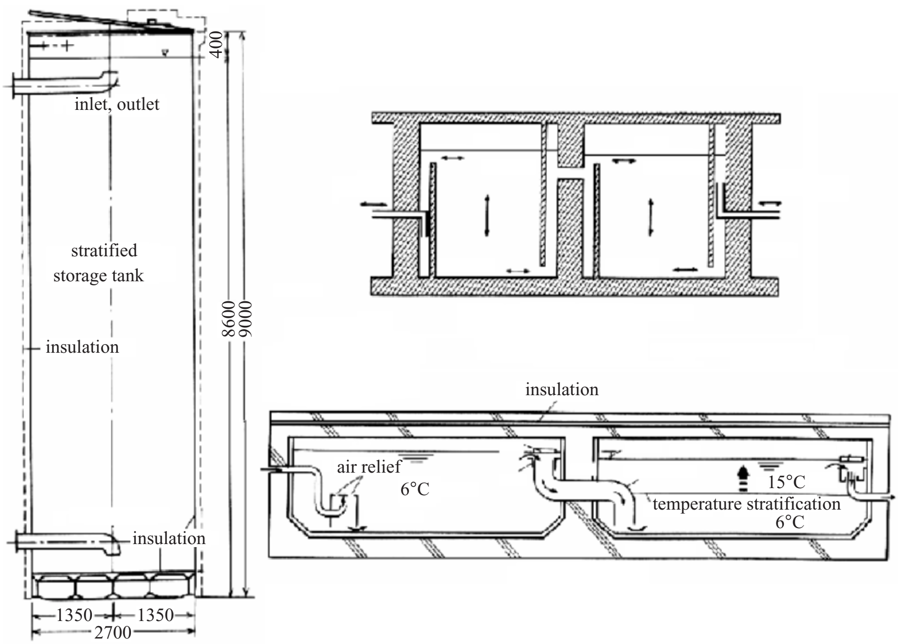


Figure 184. Stratified water tanks

To have snow/ice in buildings, there are two ways of making. One is a natural convection type. In this type, there is no fan and snow is stored in the storage room. The other one is forced convection type. This system has fans to circulate air between storage room and snow room. Many small holes are made in the stored snow perpendicularly and air come from storage room is cooled during pass through these holes from the upper part to the bottom. Recent large systems adopted this forced convection type.

There are two types of snow cooling system. One is direct heat exchanging system and the other is indirect heat exchanging system. Direct heat exchanging system is the same as the forced convection type. Room air is circulated between a room and a stored snow room by fans and cooled directly. The advantages of this system are

1. Simple.
2. Purification of indoor air (dust and water-soluble gases such as CO_2 and ammonia).

However, serious problems are considered due to use of air and common ducts.

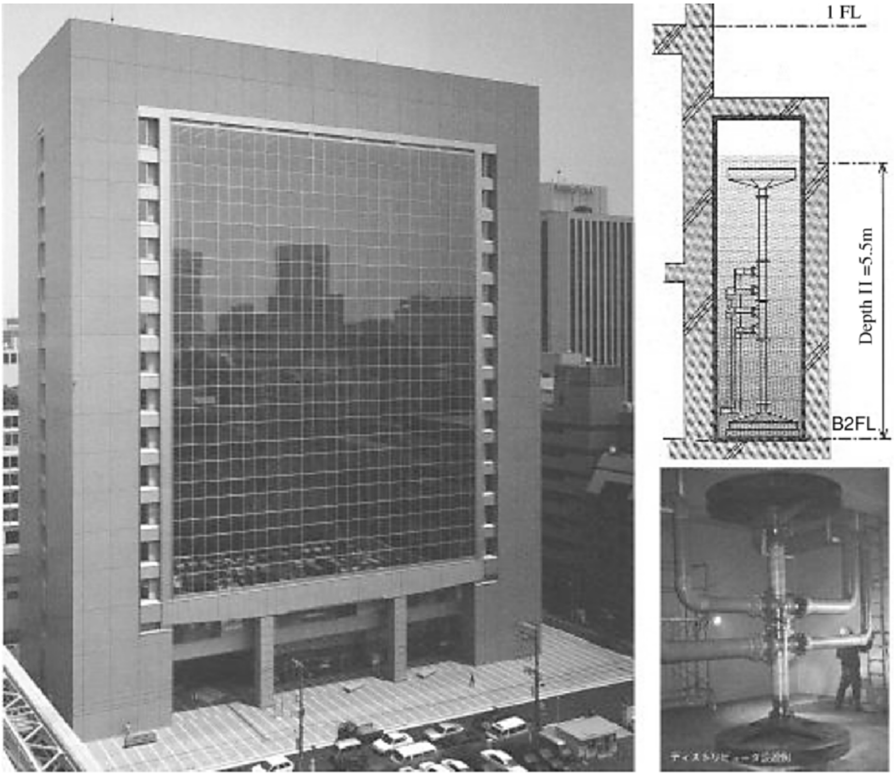


Figure 185. An office building with stratified water tank

1. Smell of snow.
2. Noise.
3. Power consumption of air circulating fans.

From these reasons, almost large snow cooling systems adopted indirect heat exchanging system. In indirect system, a water–water-type heat exchanger is used between primary and secondary system. Figure 186 shows an application of direct heat exchange system.

20.4.2.2. *Artificial Ice*

Ice storage systems become very popular recently. The volume of the tanks is smaller due to large latent heat of ice so that they can be applied to the small buildings that do not have enough space for water storage. Although ice making is established technology in food industry, adopting it to buildings several types are developed as shown in Figure 187.

Figure 188 shows how much the volume of tank would be reduced by latent heat of water. Although the reduction ratio depends on temperature

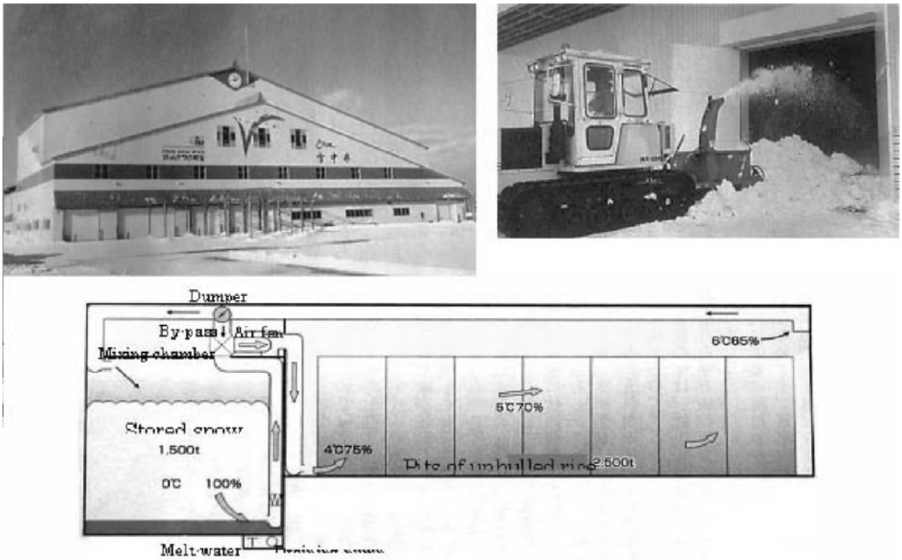


Figure 186. An application for snow storage to rice storage

difference for secondary system, size of the tank is reduced drastically for small ice packing factor, which is ratio of ice volume to tank volume. Since PCM has large heat of fusion, size of storage tank could be reduced as ice storage tanks.

20.4.2.3. *Ice On Coil*

A set of curved tube or coil in which refrigerant or glycol solution run is installed in a tank and cooled inside of the coil. Ice is formed on the surface of the coil and melted from boundary between ice and water. Stored heat is usually recovered by supplying water directly to secondary system. Some systems use heat exchanger to reduce water head loss. This type of ice storage

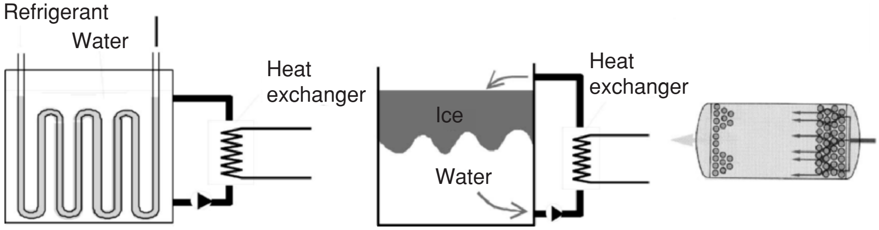


Figure 187. Types of ice storage tanks, Left: Ice on coil type, Center: Slurry type, Right: Encapsulated type

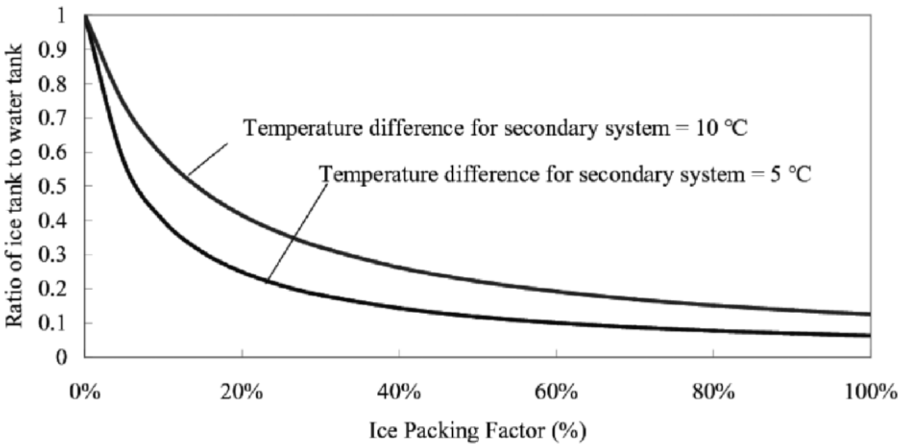


Figure 188. Volumetric reduction of storage tank by ice

is also called static type, since ice in the tank is statically fixed. In freezing process, as thickness of the ice increases, ice layer will be thermal resistance so that performance of the system decreases.

In Japan, a system in which storage tank and compressor are packed is very popular as shown in Figure 189. Several manufactures produce such package system in various sizes.

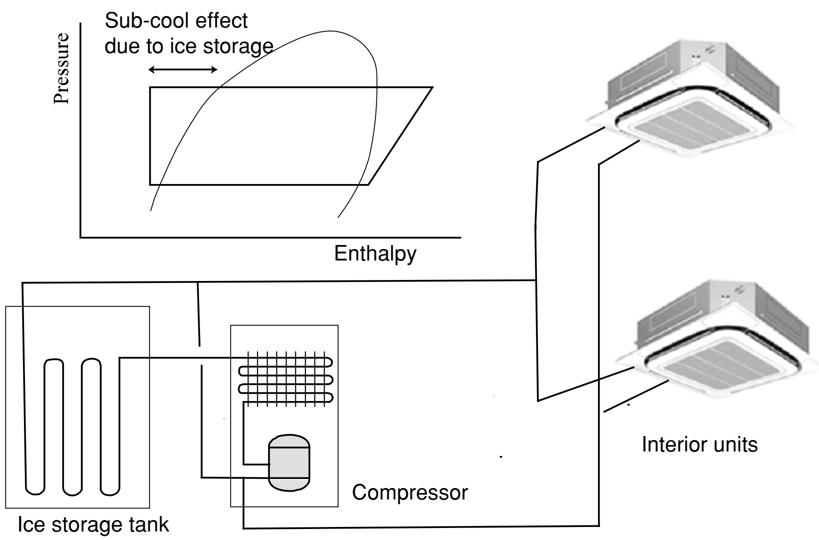


Figure 189. A schematic diagram of package ice storage system

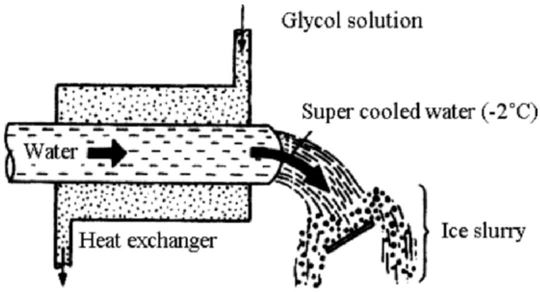


Figure 190. Slurry ice system using super cooled water

20.4.2.4. *Slurry Type*

Fresh water or thin glycol solution is cooled by heat exchanger. Fresh water becomes super cooled water, which is sub zero temperature, then a certain mechanism such as collision or vibration releases super cooled state so that some part of the water becomes ice. Figure 190 shows outline of making super cooled water.

In the case of thin glycol solution, solution becomes ice and thicker solution by cooling. Figure 191 shows one example of such ice-making system. These ice-making method don't have problem of thermal resistance of growing ice as ice on coil type.

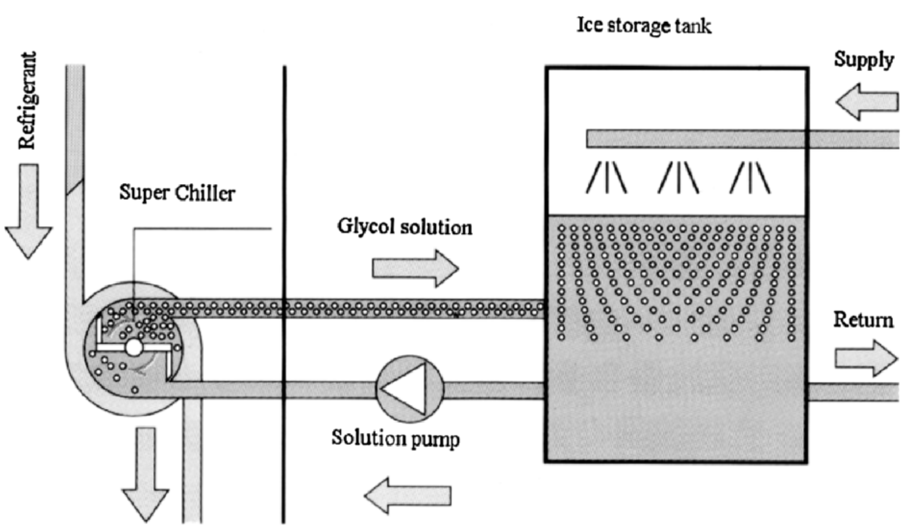


Figure 191. Slurry type using thin glycol solution

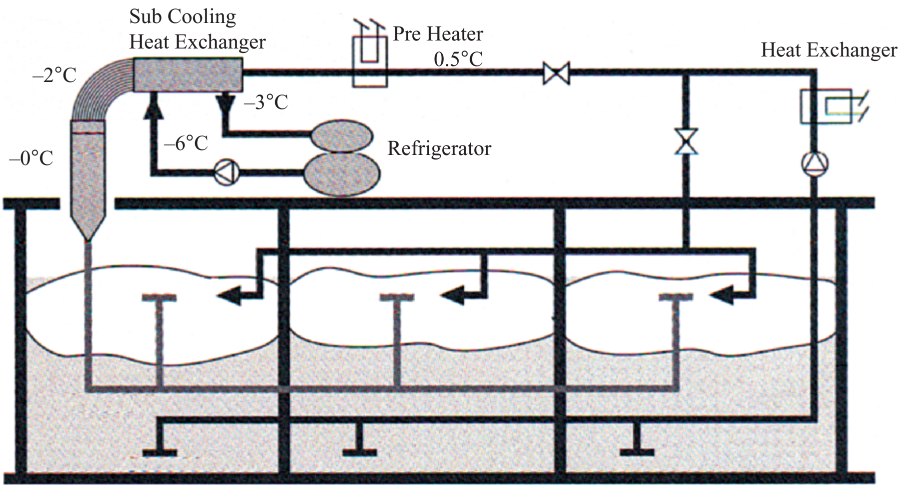


Figure 192. Schematic diagram of storage system

The large scale of slurry ice storage system is installed in Kyoto station building. The building of 237,689 m^2 floor area is air conditioned with 4 absorption chillers of 5,830 kW and slurry ice tank of 1,700 m^3. Sub cooled water is projected into tube shape inlet where ice is made by release from sub cooling state. Slurry ice is conveyed to separate tanks through a pipe (Figure 192).

20.4.2.5. *Encapsulated Type*

Small balls which contain storage medium, water is most popular, are packed in a vessel and are cooled and heated by operating liquid. According to the melting temperature of medium inside the balls, operating temperature of system will be identified.

Figure 193 shows Ibaraki Prefectural Office which is located the north of Tokyo. The facilities for renewable energy that are solar cells, grand water heat pump and natural ventilation are equipped. Energy supply for HVAC is done by both electricity (50%) and gas (50%). Half of cold heat generated by the electricity is stored in water storage tank (2,820 m^3) and rest is stored in ice storage tank. Ice storage is sometimes used together with water storage. COP of ice-making chiller is lower than water chiller because of low evaporation temperature. Although water storage is preferable, space for storage tank might not be prepared for some buildings. As internal heat gain, such as lighting and computers, is getting larger in recent years, the cooling load exceed the heating load in large extent. Ice storage is suitable to compensate the difference.

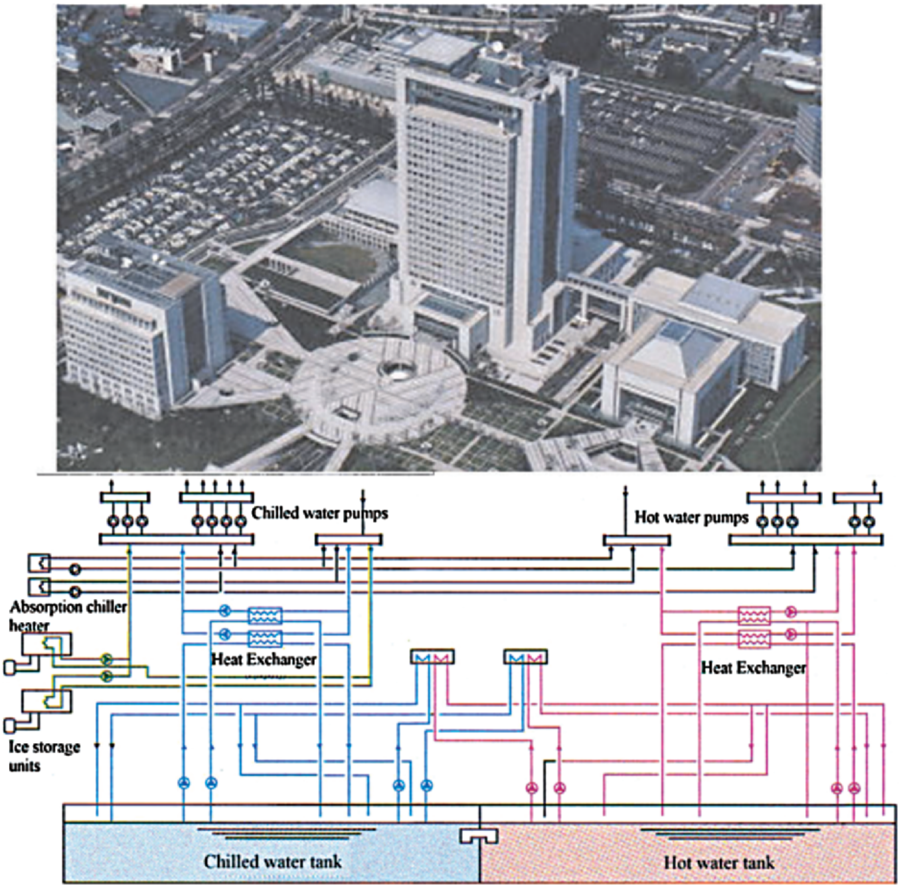


Figure 193. Ibaraki Prefecture Office

20.4.3. PCM OTHER THAN ICE

The applications of PCM (excluding natural or artificial ice) are usually using natural cold sources. The concepts can be divided into:

1. Passive systems to decrease temperature fluctuations. The PCM is integrated into building materials or building components and increases the thermal mass of the building.

   PCM in building materials
   Since the last oil crisis many attempts have been made to incorporate PCM into building materials like plaster, wood or fiber boards. At the beginning this was done by impregnation, recently micro encapsulation has been preferred as it reliably prevents leakage. The potential of PCM to reduce

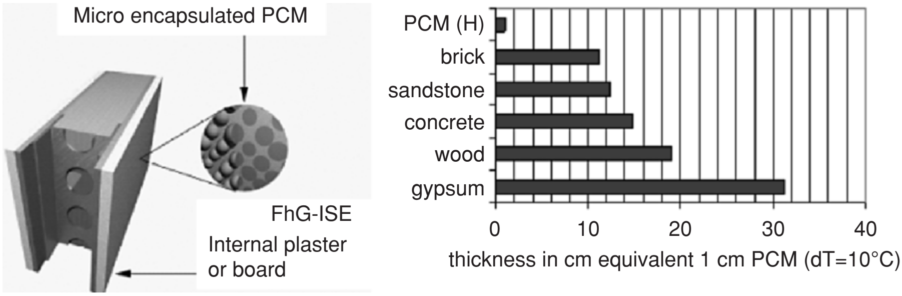


Figure 194. Integration of PCM into building materials by micro encapsulation (left) and comparison of different wall thicknesses to store a given amount of heat (right)

temperature fluctuations, especially in light weight buildings, becomes clear from Figure 194. Here different wall thicknesses are shown, which are necessary to store the same amount of heat based on a temperature difference of 10 °C.

It is shown that with PCM the wall thickness can be reduced to a fraction of the thickness (and weight) of ordinary walls. Figure 195 shows the implications of this fact on the interior temperature of a building. With PCM, a light weight building can behave similar to a massive building. The strong temperature variations, which are typical for lightweight buildings, can efficiently be suppressed.

Gypsum plaster boards with micro-encapsulated paraffin
An interesting combination of PCM with building materials is the example of gypsum plaster boards. They gypsum plaster boards are used on a large

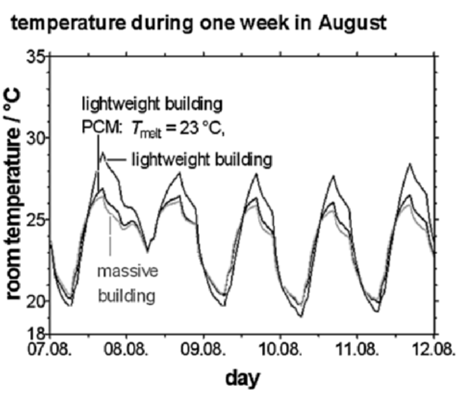


Figure 195. Simulation of temperature variation in a massive building, lightweight building, and lightweight building with PCM

scale in construction of lightweight buildings. The incorporation of micro-encapsulated paraffin into the plaster board during a conventional production process has been successively investigated by Knauf (Germany). Gypsum plaster boards with micro-encapsulated paraffin are commercially available from BTC Speciality Chemical Distribution GmbH. 1.5 cm of their Micronal® PCM SmartBoard stores as much heat as a 9 cm thick concrete wall or 12 cm of brick (http://www.micronal.de/).

Plaster with micro-encapsulated paraffin
The integration of microencapsulated PCM into plaster follows the same idea as for the gypsum board. Maxit, FhG-ISE have been developing a product based on BASF Micronal®. The plaster has been applied and tested in several buildings. It is now commercially available from maxit Deutschland GmbH as "maxit clima" (http://www.maxit.de/index1.php3?id=2&schalt=4). 3 cm of maxit clima store as much heat as 8 cm of concrete.

Shading-PCM compound system
External blinds are susceptible to strong winds and therefore have to be constructed with sufficient mechanical stability. This however raises the cost of the blind. Internal blinds do not have this problem. However internal blinds absorb some of the incoming sun light. This leads to a large temperature increase at the blind and the absorbed heat is released into the room (Figure 196). Warema and the ZAE Bayern have investigated the idea of reducing reduce and delaying the temperature rise of the blind by integrating PCM. Figure 197 shows the prototype of an internal blind with integrated PCM. The measurements in a test room under realistic conditions have shown that the temperature rise of the blind can be decrease by ≈10–15 °C.

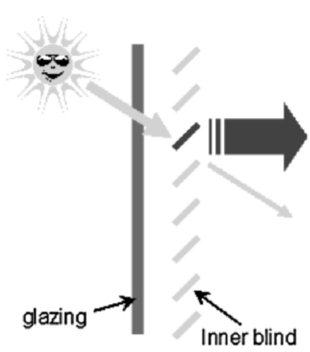


Figure 196. Ordinary inner blinds absorb solar radiation and release the heat into the room

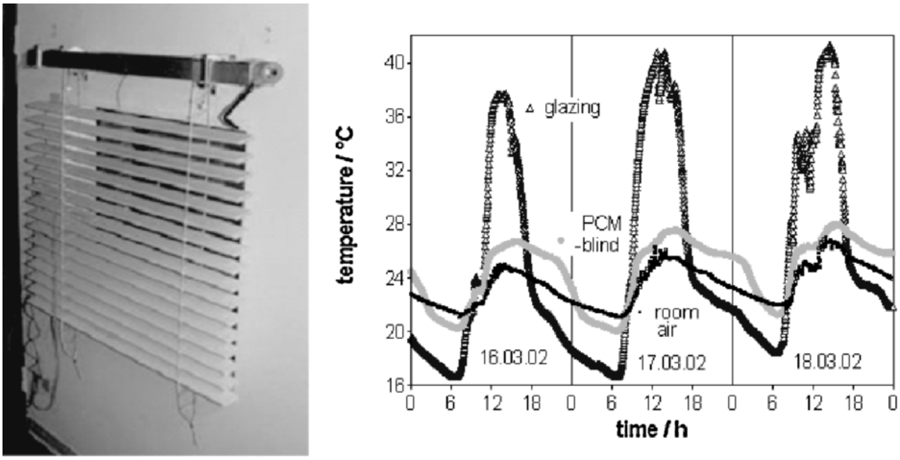


Figure 197. Prototype of a horizontal blind with PCM and experimental results

Further investigations by numerical simulation are very promising. They show a significant decrease of the operative temperature of the room by ≈3 °C and a time shift of the heat gains from noon to evening. The thermal comfort during working hours is therefore significantly improved.

2. Active systems as intermediate storage for cold with an actively moving heat transfer medium. Active systems use at least some energy usually

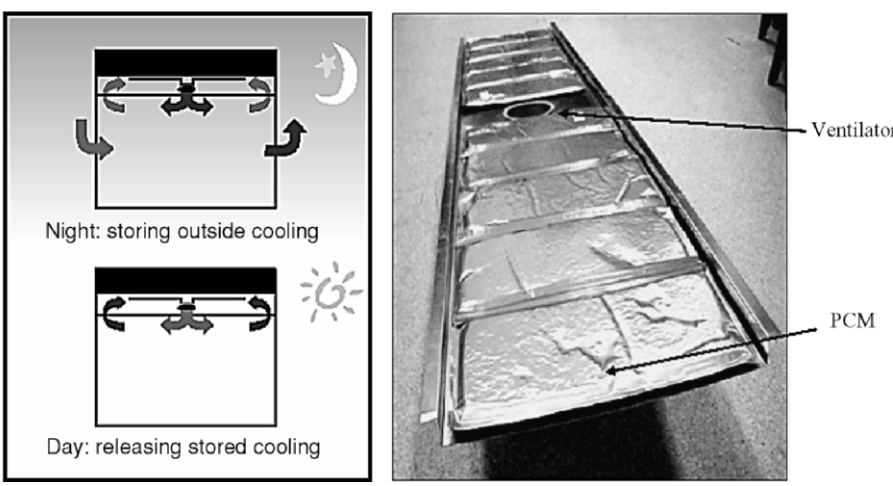


Figure 198. Free cooling by storing coolness from the night and use it for space cooling in daytime (left) and design of Climators CoolDeck (right)

for a fan or pump to move the heat transfer fluid. This can have several advantages:

- the system can be turned on and turned off. The stored cold is then available when it is really needed.
- heat transfer to the source and demand is better as the heat transfer medium is transported with a larger volume and mass flow rate.
- heat transfer coefficients also get higher.

Free cooling systems

Active systems that use the cold of the night air are usually called free cooling systems.

As the left side of Figure 198 shows, at night cold air from the outside is moved by a fan into the building. In the building, the cold night air cools down the PCM and the PCM is solidified. During the day, warm air from inside the building is blown by the fan across the PCM. The heat of the air is thereby melting the PCM and cooled down. This way, the cooling of the building is done without any cold production; the only energy consumption comes from the fan. The ratio of supplied cold to fan energy is in the order of 10–20. These systems are therefore very promising for saving energy. The right of Figure 198 shows a possible design of a free cooling system, the CoolDeck developed by Climator/Sweden. It has already been installed in several buildings and observations show a reduction of the

Figure 199. Installation of 1.100 m^2 plaster on capillary sheets in Berlin, Gotzkowskystraße (picture: BASF/Germany)

room temperature by 3–4 °C with very little energy consumption. Similar systems have also been developed by Rubitherm and Dörken.

PCM-plaster with capillary sheets connected to ground water
Free cooling systems as described above have two disadvantages. They rely on the availability of cold air at night and they have only a few °C temperature gradient, which limits the cooling power. As Figure 181 shows, the availability of "more reliable cold sources" like groundwater can remove these restrictions. An example of such a system has been installed by BASF in Berlin (Figure 199).

The storage consists of plaster with microencapsulated PCM (Figure 194). To release the heat, capillary sheets are integrated into the plaster and connected to ground water.

21. THE SUNDSVALL SNOW STORAGE—SIX YEARS OF OPERATION

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Abstract. Ice storage for cooling is an ancient technology which was common until the beginning of the 20th century, when chillers were introduced. During the past few decades new techniques using both snow and ice for comfort cooling and food storage have been developed. Cold is extracted from snow or ice by re-circulation of water or air between the cooling load and the snow/ice. The snow cooling plant in Sundsvall, Sweden, is used for cooling of the regional hospital. The stored natural and artificial snow is used for comfort cooling from May to August. It was taken into operation in June 2000 and is the first of its kind. Here the plant is described and the experience of its first six years of operation is presented.

Keywords: snow, cooling, energy storage, renewable energy, pilot plant, operation

21.1. Seasonal Snow and Ice Storages

In seasonal snow/ice storages frozen water is stored from winter to summer, when the cold is utilized. The snow/ice can be stored indoor, on ground, in open ponds/pits and under ground, Figure 200.

If snow/ice is stored indoor it is done in a more or less insulated building. In a cavern no insulation except the ground is needed. When the snow/ice is stored on ground or in ponds it is necessary with thermal insulation, henceforth denoted insulation. Both natural and artificial snow and ice may be used and there is no size limitation for snow cooling systems. This snow cooling plant in Sundsvall is an open pond with larger pieces of wood chips as thermal insulation.

The basic idea of such systems is that snow/ice is stored in a more or less water tight pond where a cold carrier is cooled by the snow, to utilize the large latent heat of fusion. For comfort cooling about 90% of the extracted energy is in the phase change, i.e., the melting. The cold carrier is either circulated between the load and the snow or rejected after it has been used for cooling.

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 349–366.

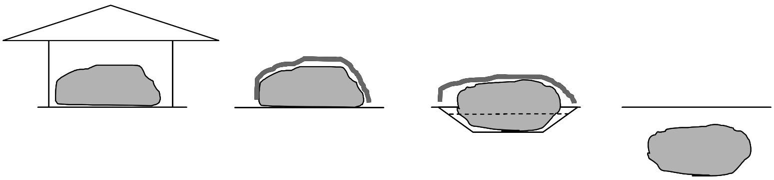

Figure 200. Outline of different snow storage methods; from left to right; indoors, on ground, in open ponds/pits, and under ground

In all implemented systems the cold carrier is in direct contact with the snow, but it is also possible to have closed systems where the cold carrier flows in pipes. Air, melt water, sea water, ground water or some other fluid might be used as cold carrier.

Snow is normally defined as precipitation formed of ice crystals and ice as solid water with hexagonal structure and density about 920 kg m^{-3}. In snow storage the main issue is to have enough amounts of frozen water at low cost why the only relevant distinction is the density. If natural snow or ice is too expensive or not available in enough quantity, it is possible to produce frozen water. Artificial snow and ice made with different types of water sprayers, including snow blowers (snow guns). The production rate depends on equipment, relative air humidity, and temperatures of the air and water.

Thermal insulation on snow can be loose fill, sheets and roof like structures. Loose fill insulation is different types of wood chips, rice shell, debris (mineral particles), etc. Sheets can be both plastic and filled tarpaulins, e.g. with straw. Superstructures are more or less rigid constructions that are partly or totally removed/opened during the winter.

There is always some heat leakage when storing snow in a surrounding above 0 °C. In pond storage the resulting natural melting is divided in ground melt, rain melt and surface melt. The surface melt occurs by heat transfer from radiation and ambient air. Ground melt is caused by heat transfer through the sides and bottom of the storage, Figure 201. The remaining snow is the volume that is used for cooling.

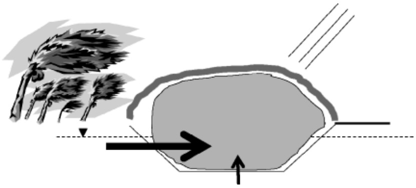

Figure 201. Natural snow melt in open pond snow storage

21.1.1. TECHNIQUES AND EXAMPLES

There are a number of suggested and implemented techniques of snow and ice storage for cooling applications. In Japan about 100 projects have been realized during the last 30 years, and in China there is about 50–100 snow and ice storage systems (Kobiyama, 2000). Also Canada, USA and Sweden have made efforts in the field. Below both realised and suggested techniques are presented.

21.1.1.1. *Himuros and Yukimoros*

A Himuro is a house, or a room, where vegetables are stored together with ice to prevent decay of the food. In a Yukimuro snow is used instead of ice. The house/room can be situated in or on the ground. This is the traditional way to use snow/ice for cold storage. The food is stored in shelves and the snow/ice is stored in carriages or in one section of the room. The food/snow ratio varies from 1/1 to 4/1 depending on climate and thermal insulation of the building. Cold distribution occurs by natural convection and can be roughly controlled by shutters, curtains, etc (Okajima et al., 1997). The temperature in a Himuro/Yukimuro is a few degrees above 0 °C and the relative humidity is about 90–95%. One important drawback is that humidity and temperature cannot be controlled with any precision (Kobiyama, 1997).

21.1.1.2. *Indoor Snow Storages*

In the Japanese All-Air-Systems air is used as the cold carrier. Warm air is blown through water made holes in the snow. The snow gets covered with a thin water layer where particles and gases are absorbed, i.e., the air is cleaned during the cooling. Hole spacing is about 1 m and the snow lies on a steel grid (Iijima et al., 1999). There are All-Air-Systems where both temperature and humidity is controlled (Kobiyama et al., 1997).

21.1.1.3. *Ice Box/Fabrikaglace*

The Icebox, or Fabrikaglace, is a Canadian invention. It consists of an uninsulated box inside an insulated shelter. Thin sprayed water layers, a few mm, are frozen in the inner box by cold air blown over the water. It is possible to form ice blocks more than 20 m thick during one winter and there is no size constraint. To extract cold the melt water is pumped to a heat exchanger and then re-circulated over the ice. Thermal expansion and ice creeping was handled by flexible walls. The tested Ice boxes varied from a few up to 250 MWh with cooling powers 8–1,600 kW but the high construction costs required large plants or high cooling powers (Buies, 1985; Morofsky, 1984). The coefficient of performance (COP) was about 90–100 (Abdelnour et al., 1994; Morofsky, 1982).

21.1.1.4. *Underground Storages*

Snow storage in underground caverns has so far not been implemented. Johansson (1999) investigated the prerequisites of snow cooling during the summer, with focus on under-ground storage. Storages of 25,000–150,000 m^3 were studied, assuming snow density 650 kg m^{-3}. Simulations gave that natural melting was about 3–6% the first years and 1–3% the tenth year. In many of the studied applications the pay-back time was less than one year.

In 1995 potatoes and rice were stored in an abandoned road tunnel in Japan. At the end of the winter the tunnel was filled with snow that was covered with aluminum coated tarpaulins. The products were stored both beside the snow piles and under the tarpaulins. Rice from the same harvest was stored in a grain magazine and at a research centre for comparison. The rice in the tunnel had best quality after two month. The potatoes under the tarpaulins kept its quality longest and it was found that the storage method was feasible over the whole year (Suzuki et al., 1997).

21.1.1.5. *Open Storages*

A number of different open pond snow and ice storage techniques have been suggested. In Ottawa a storage for 90,000 m^3 of snow in an abandoned rock quarry ($120 \times 80 \times 9.5$ m^3, $L \times W \times H$), was studied. The mean cooling load was 7,000 kW. A light colored PE plastic tarpaulin was suggested as insulation, with melt water re-circulation for cold extraction. The estimated payback time was 10 years (Morofsky, 1981).

Näslund (2000) investigated a district cooling system in Sundsvall, mid Sweden, with sea water and stored snow. The cooling load was 7,900 kW and 7,450–8,560 MWh. Natural snow from streets and squares were complemented with artificial snow made by snow guns or water spraying. The estimated snow proportion was 43.6–66.8% and 122,500 m^3 of snow was needed. Two layers of 0.01 m plastic sheets with thermal conductivity 0.04 W m^{-1} K^{-1} was recommended as insulation. The plant has not yet been realized (2005).

Näslund (2001) investigated a combined system with river water as base load and snow cooling as back up, for an industrial application. The cooling need was 1,500 kW at 5 °C and 1,500 kW at 15 °C, continuously. The needed snow amount was 78,800 ton, stored in a $120 \times 100 \times 3\,m^3 (L \times W \times H)$ pond with water tight asphalt bottom. The estimated investment cost was about 950,000 Ä.

André et al. (2001) investigated a snow cooling plant in northern Sweden for operation all year round with winter base load 2 MW and summer peak load 6 MW. The idea was to use the local snow deposit and combine with artificial snow. The system consisted of two main ponds and one smaller pond.

A Japanese study investigated the possibility to use snow for atomic power plant cooling, to increase electricity production. With 38,400,000 tons of snow the sea water temperature should be lowered 9 °C but only 1,800 MWh of extra electricity would be produced every year, because of few operation hours per day. It was suggested that a layer of 0.15–0.3 m rice husk should be used as insulation (Kamimura and Toita, 2004).

In a similar Swedish project with snow cooling of a waste gas power plant 2,080,000 ton of snow would increase the annual electricity production 30.7 GWh. The snow should be stored in a $400 \times 400 \times 20\,\mathrm{m}^3 (L \times W \times H)$ pond with water tight sides but open bottom. Here 0.2 m of wood chips was suggested as insulation (Falk et al., 2001).

21.1.1.6. *Water Purification*

Another application is to combine ice production and water purification since impurities are pressed out as water freezes. It is possible to freeze sea water or polluted ground water and use the cleaned melt water for both cooling and drinking. In Greenport, New York, sea water was cleaned from 3.0% to 0.00005% by freezing (Taylor, 1985).

21.2. The Sundsvall Hospital Snow Cooling Plant

The snow cooling plant at the Sundsvall Regional Hospital, mid Sweden, is an open pond solution designed for 60,000 m^3 of snow. The plant is owned and operated by The County Council of Västernorrland (CCV). The cooling operation started in June 2000. This is the first open storage with melt water re-circulation and a wood chip layer as thermal insulation. Monthly mean air temperatures during the cooling season (May to August) is 8–15 °C and daytime temperature often reaches above 25 °C. The hospital floor area is 190,000 m^2 and the plant is managed by the owner of the hospital.

Pre-studies and a small field experiment were made in 1998. In the theoretical study a cooling need of 1,000 MWh was assumed, and natural melting of a 15,000 and a 30,000 m^3 snow pile without cover and with 0.1 and 0.2 m of saw dust as insulation was simulated (Nordell and Sundin, 1998).

The simulations showed that 15,000 m^3 of snow was not enough in any case. An un-insulated 30,000 m^3 snow pile would be gone by mid-June and consequently the storage had to be thermally insulated. With 0.1 and 0.2 m of saw dust as thermal insulation the remaining snow volumes were 12,169 and 19,040 m^3, Figure 202.

21.2.1. CONSTRUCTION

The snow storage was built in 1998/1999. It is a shallow watertight pond (130·64 m) with slightly sloping (about 1%) asphalt surface, Figure 203. The

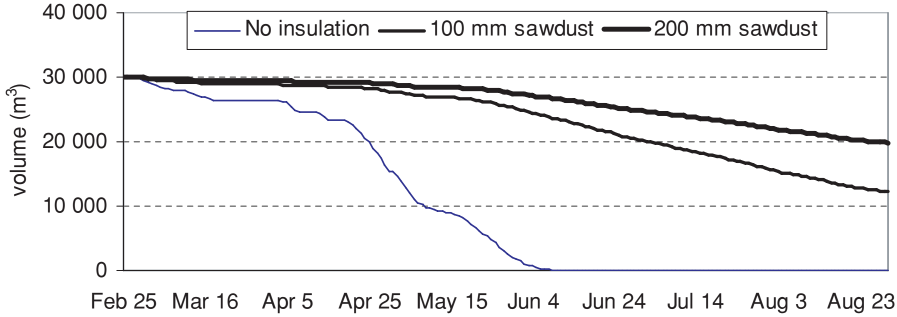

Figure 202. Natural melting of 30,000 m^3 of snow with 0.1 and 0.2 m of sawdust and without thermal insulation (from Nordell and Sundin, 1998)

0.1 m asphalt layer is overlaying 0.5 m gravel, 0.1 m insulation, and 0.8 m of sand.

Cold is extracted by pumping melt water through heat exchangers connected to the cooling system inside the hospital. The heated melt water is then re-circulated to the snow where it is cooled and new snow melts. The original re-circulation inlets are 36 valves located at the sides of the storage. The outlet is two openings in the pump house that is located in the lower corner of the pond. The water is cleaned by different filters, and pumped by two pumps (0.035 and 0.050 m^3 s^{-1}) to the heat exchangers (1,000 + 2,000 kW), Figure 204. The comfort cooling system of the hospital also includes one 800 kW chiller. The system primarily runs on snow cooling and the chiller supports when necessary.

21.2.2. FIVE YEARS OF OPERATION

During five years of operation at a full scale demonstration plant many experiences are made. Here the main results and findings are presented.

Figure 203. Right: Part of the snow pond with pump house and half of the hospital. Left: Snow covered with wood chips in beginning of the cooling season. Picture taken from the pump house

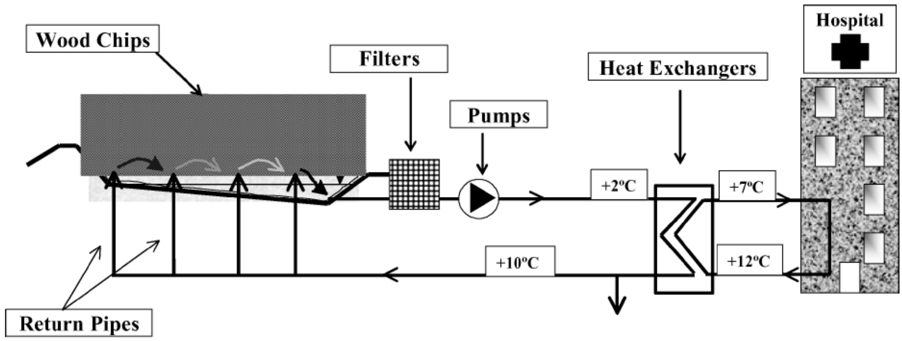

Figure 204. Outline of the Sundsvall snow cooling plant

1. *Snow*: Both natural snow from streets and squares and artificial snow, made with snow guns, was used. The artificial proportion was 38–59%. The stored snow volumes, measured with a geodetic total station, were 18,800–40,700 m^3. The snow density varied from 578–735 kg m^{-3}, with mean density about 650 kg m^{-3}. The first year snow and wood chips were spread with a tractor and an excavator. The other years a snow groomer was used which made spreading much more efficient.

   Different snow and ice making systems have been tested. The first years both fan type snow guns and one LowEnergyTower were used. The fan-type machines required a lot of maintenance and surveillance the first years and a lot of the snow from the LowEnergyTower landed outside the storage. The last year's two fan type snow guns have been used, mounted on 5 m high towers and now only operated in the wind direction. This worked well.
2. *Cooling*: The cooling needed varied largely. More than 75% of the total cooling load was delivered by the snow system and unnoticed 2000, when snow cooling started late, the snow cooling portion have increased. The maximum cooling load increased because the cooling system inside the hospital was increased, as planned, Table 1. It will continue to expand to about 3,000 kW and 3,000 MWh in about 2010 (Larsson, 2005).

   The snow cooling system mostly met the total cooling demand until the end of June/beginning of July when also the chiller run, because the re-circulation system was too small. Snow cooling stopped between middle of August and beginning of September all years. With increased operation experiences the snow cooling has met the total cooling load during a larger part of the cooling season, Figures 205 and 206.

   The total Coefficient of Performance (COP_{total}) was the ratio between the delivered cooling energy and total energy needed to produce the cold. This included total operation energy and yearly material depreciation, defined as the total energy required manufacturing and constructing the system

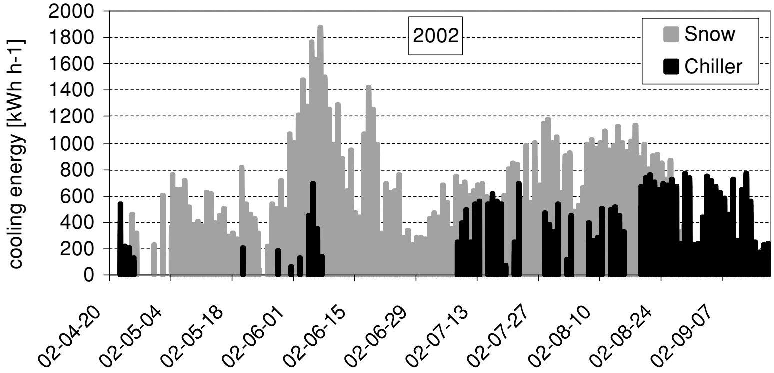

Figure 205. Hourly cooling amounts from snow cooling system and chiller in 2002

divided by estimated technical lifetime (Hagerman, 2000; Wichmann, 2003). The COP_{total} was based on the assumption of the same total cooling amount, from extrapolation.

The amount of artificial snow was calculated as the ratio of water for snow making over estimated snow amount, i.e., snow making loss was not included.

3. *Thermal insulation*: The snow is thermally insulated with a 0.2 m layer of larger pieces of wood chips, 20–150 mm. The wood chips decayed fast and it was necessary to add new material every year. After three years all wood chips were rejected, and new wood chips were bought. It might be necessary to replace all insulation also coming years because the insulation qualities deteriorates when wood chips mix with sand and gravel from the snow.

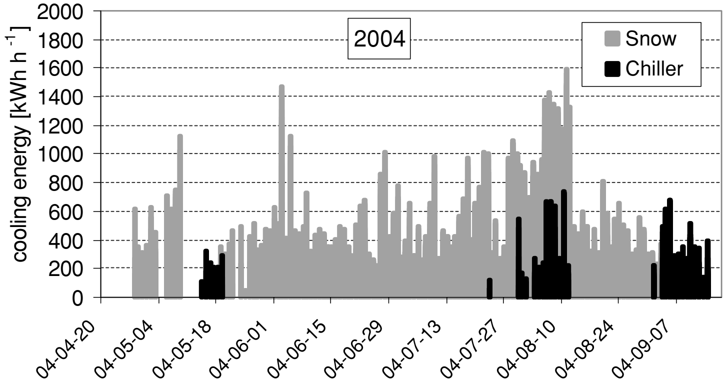

Figure 206. Hourly cooling amounts from snow cooling system and chiller in 2004

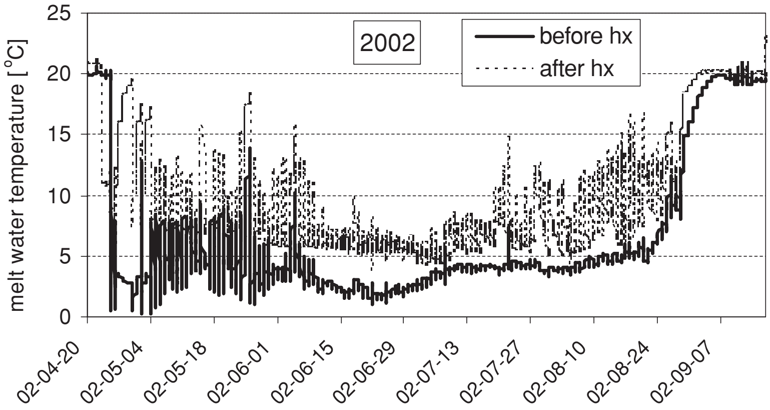

Figure 207. Melt water temperatures before and after the heat exchangers (hx), in 2002

In 2003 CCV made a minor experiment with 2·5 m insulation sheets, U-value 13.6 W m^{-2} K^{-1}, covering an area of about 10·10 m. In the beginning the insulation functioned well but as the snow started to melt uneven the sheets slid apart. The staff had to adjust the sheets frequently and it was concluded that this solution was less good than wood chips (Larsson, 2005).

4. *Melt water re-circulation*: The melt water re-circulation system is important since it delivers the cold to the hospital, but the system in Sundsvall however caused problems. The melt water temperature before the heat exchangers was mostly 2–5 °C until the end of July/middle of August. Then temperatures started to increase because of snow decrease and shortages in the re-circulation system. Around the inlet valves snow melted quickly why wood chips fell off, resulting in both increased natural melting and that re-circulated water found short cuts along pond sides. To decrease side-melting new re-circulation hoses were installed. The first year the hoses were placed on top of the snow, resulting in holes and exposed snow. In 2001 the hoses were placed at the far side, and in 2002–2004 the hoses were placed under the snow. In 2002 the hoses were damaged by the snow groomer but in 2003 and 2004 the outlet water temperature decreased, Figures 207 and 208.

   The average cooling load, melt water flow, melt water temperature before the heat exchangers and melt water temperature increase in June + July 2001–2004 are seen in Table 24. The melt water flows during these periods corresponded to a detain time of 1.5–3 days, assuming that 2/3 of the volume beneath the water surface was occupied by snow.

   The temperature increase in the heat exchangers was usually 1–8°C. The decreased temperature difference in 2002–2004 was due to algae growth

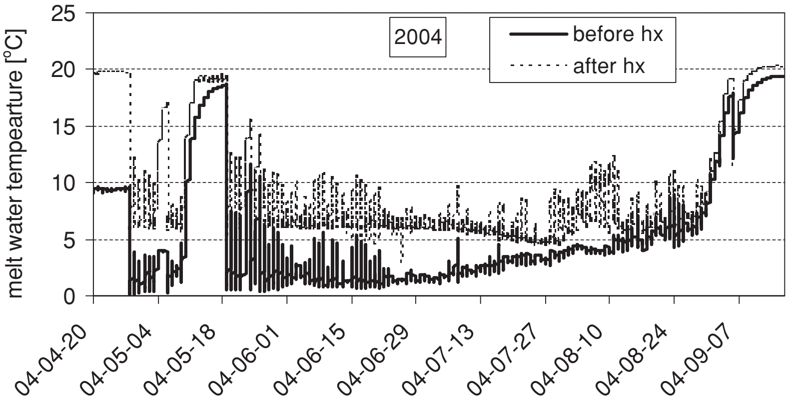

Figure 208. Melt water temperatures before and after the heat exchangers (hx), in 2004

in heat exchangers and decreased temperatures on the secondary side of the heat exchangers, from 13.0 °C in 2001 to 8.1 °C in 2004, before heat exchangers. The algae growth decreased the heat transfer and maximum flow rate. The heat exchangers were cleaned with lye one to three times per summer, and then the temperature difference increased again. CCV believed that increased decay rate of the old wood chips and high temperatures in heat exchangers caused the growth. Algae growth decreased radically during 2003 when new wood chips were used.

It is possible to avoid short cuts and decrease outlet melt water temperature by spraying re-circulation melt water on top of the snow instead, but so far CCV has avoided this because they do not want pollutions carried in the air to the hospital. For 2006 the melt water circulation will be extended with sprayed ground water and water outlets at the pond bottom centre, to decrease the water temperature further.

5. *Weather:* Air temperature, relative humidity, precipitation, solar radiation, air velocity and wind direction were measured by a weather station at the snow storage pump house. The summers of 2000–2004 were warmer and rainier than mean summer of 1961–1990. In 2002 the summer mean temperature was 4.1°C warmer than the average year. In 2001 and 2002

TABLE 24. Melt water outlet data for 2001–2004

June+July	2001	2002	2003	2004
Average snow cooling load (kW)	457.5	453.4	476.6	276.9
Average melt water flow (m^3d^{-1})	1,842.6	2,567.7	2,272.7	1,595.0
Average melt water outlet temperature (°C)	4.1	3.3	3.0	2.2
Average melt water temperature increase (°C)	4.9	3.6	3.8	3.7

TABLE 25. Monthly mean temperatures and total monthly precipitation during the cooling seasons of 2000–2004 at the Sundsvall Regional Hospital Snow Storage. Mean values between 1961–1990 are from the Sundsvall airport 30 km away

	Mean temperature (°C)				Precipitation (mm)				
	May	June	July	August	May	June	July	August	Total
2000	9.5	12.6	16.5	15.0	45.4	72.3	146.4	37.3	301.4
2001	9.3	15.1	17.8	15.4	33.0	15.4	68.8	221.8	339.0
2002	11.4	17.6	18.4	19.4	84.0	95.0	62.0	21.2	262.2
2003	8.7	14.1	18.9	15.6	41.0	79.6	13.0	134.2	267.8
2004	9.3	14.0	16.5	16.7	26.8	52.8	97.0	88.2	264.8
1961–1990	7.8	13.4	15.3	14.0	35	41	58	64	198

there was 52.2% and 71.2% more precipitation than the average year, Table 25.

Mean values of 1961–1990 are from the Sundsvall Airport, located about 30 km from the snow cooling plant. There was good correlation between measured air temperature and precipitation and values from the airport climate station, compared during 60 days the summer of 2000. The airport climate station is run by the Swedish Meteorological and Hydrological Institute.

The ground temperature under the storage pond was measured both above and below the ground insulation at 0.6–0.7 m depth. Above the insulation the ground temperature followed the air temperature with a slight trailing when no water or snow covered the storage bottom, otherwise it was 1–5 °C above zero. The ground temperature below the insulation was about 4–5°C from spring to end of snow cooling season, and then increased a few degrees.

6. *Environment:* A number of melt water quality analyses of heavy metals, hydrocarbons, oxygen demands, and nutrients were made. From September 2001 to September 2002 measurements were made in the snow storage, in the stream where melt water is discharged and in the recipient, totally at seven locations. Reference measurements were made at a nearby location not affected by the outlet water. The results were compared with Swedish environmental quality criteria (SEPA, 1990; SEPA, 1999).

The concentration of non-biodegradable compounds (measured as COD-Mn) was rather high, especially in the second half of the cooling season. These were supposed to origin from the wood chips (Ericsson, 2003). The content of phosphorus and some heavy metals in the snow were high, but in the outlet water most substances were considerably reduced. This was related to particle adsorption, as particles settled in the snow basin, which agrees with performed studies (Viklander and Malmqvist, 1993). The effect on the recipient was limited or non detectable for all parameters, relative the reference location (Ericsson, 2003). The phosphorus content in the snow was probably caused by droppings from dogs and birds (Malmqvist, 1983).

An unknown amount of the substances described above were lost in the oil and gravel separator and by flushing of the fine filters. An attempt to make a full mass balance failed.

21.2.3. ECONOMY

The total investment cost of the Sundsvall snow cooling plant was about 14.5 MSEK, or 1.59 M€ . Since this full scale operation plant was made also for experiments, research and demonstration it was difficult to distribute the cost in an appropriate way, e.g. the cost of reconstructions. An estimated cost split up is shown in Table 26 (Larsson, 2005).

The cooling operation costs during 2002/2003 and 2003/2004 are seen in Table 27. The cost of the first three years was not available, but CCV estimates that the total operation cost has decreased each year. The “Snow handling incl. production” was expensive in 2002/2003, when a contractor was responsible. The depositing of urban snow became an income from the winter 2002/2003, since CCV received a fee of 150 SEK per lorry-load of snow. “Cold production” means operation cost during the summer, excluding electricity and water. “Post season work” includes removing and storing wood chips, removing sediments and cleaning the pond.

“Electricity” depends largely on the amount of artificial snow. The cost of water has decreased due to their private well being used since 2002, on the other hand the electricity usage increased. The municipal water cost was 5 SEK m^{-3}.

TABLE 26. Investment costs for Sundsvall Regional Hospital snow cooling plant

	(kSEK)	(k€)
Pond construction	4,800	527.5
Ground thermal insulation	1,000	109.9
Fence + vehicle approach	700	76.9
Pump house	1,000	109.9
Pumps, pipes, etc	4,000	439.6
Electrical installations	1,000	109.9
Control system	600	65.9
Planning	1,400	153.8
Total	14,500	1,593.4

TABLE 27. Operation cooling cost from the Sundsvall Regional Hospital snow cooling plant (Larsson, 2005)

	2002/2003	2003/2004
Artificial snow share	37%	52%
Artificial snow amount (m^3)	13,984	18,408
Snow handling incl. production (SEK)	1,170,030	733,624
Natural snow (SEK)	−141,000	−98,100
Cold production (SEK)	199,781	167,932
Post season work (SEK)	147,652	134,291
Electricity (SEK)	35,551	45,893
Water (SEK)	34,587	33,485
Operation cooling cost (SEK kWh^{-1})	1.62	1.27

Larsson (2005) predicts that the operation cooling cost in 2004/2005 will be 0.85 SEK kWh^{-1}, and about 0.50 SEK kWh^{-1} in 2010. Thanks to the commitment of CCV, occurring problems have been solved, necessary reconstructions have been made and the plant has worked satisfactorily.

21.3. Boundary Conditions for Snow Cooling

The boundary conditions for snow storages on ground are social, political, economical, environmental, natural and system related.

The usage of snow cooling systems presupposes that the technique is known. However, the awareness of modern snow and ice cooling systems is limited to a fairly small number of countries. Also in countries that have snow cooling systems there is generally a low knowledge about the technique, both among engineers and authorities at all levels. There is also a lack of education in system construction and design.

The *political* boundary conditions are connected with financial support, taxes and regulations and laws. For instance the Kyoto Protocol might be an incentive to an increased number of snow cooling plants, since they to a large extent uses renewable energy (Paksoy, 2003).

The *economical* boundary condition varies largely over the world and generally depends on cost of land, water, electricity, fuel and labour. Natural snow is a cost if the snow is collected expressly for cooling but might be an income if snow depositors pay for getting rid of the snow. Besides this snow cooling can both be an alternative to other technologies but also enable alternatives not relevant without natural snow and ice, e.g. in developing countries. For Swedish conditions the estimated economical conditions of a new snow cooling plant was good (Skogsberg, 2005).

The *environmental* boundary conditions both concerns chillers and snow cooling. Reduction of green house gases, elimination of ozone depleting gases, noise reduction, aesthetic concerns and peak shaving (i.e., reducing electricity demand during maximum usage periods) are connected to chillers. Pollutions in snow, noise from snow making, transports and land/ground usage are related to snow cooling (Paksoy, 2003:2). Snow pollutions might be both a draw back and a benefit for the technique. The negative aspect is that pollutions are concentrated at one location and the positive is that pollutions can be measured and controlled, if the storage is water tight.

The *natural* boundary conditions are air temperature, air humidity, precipitation, solar radiation, water availability, ground water flow, ground water level, soil properties and ground water usage. In a snow cooling plant it is primarily necessary with enough amounts of snow, either natural and/or artificial. If the amounts of natural snow vary too much it is necessary with a cold period long enough to produce snow/ice. This production is benefited by low temperatures and low air humidity. In general it is possible to produce enough amounts of snow/ice if it is possible at all, since short and/or warm winters can be compensated by more snow guns.

During the cooling season the climate affects both natural melting and cooling demand why the difference in needed snow amounts between a normal year and a warm year is double influenced. Natural variations in the climate must be carefully considered during dimensioning. Furthermore seasonal snow storage is a long-term investment; estimated life time is 40 years, why also the global warming influences predictions.

The water availability is important if artificial snow is used. It is however possible to save water in the pond during a period and use it for snow making when the temperature is low enough. It is also be possible to reuse water if pollution problems can be handled.

The ground water flow and level influences both natural melting and suitable storage constructions. Large ground water flows increases the ground melt of both water tight and permeable ponds, if cold pond water flows into the ground. Soil properties also influence the pond construction. In water tight ponds snow pollutions are concentrated and treatable. If the pond on the other hand is permeable it is necessary to study how pollutions migrate. This is more important if nearby ground water is used for drinking or irrigation.

The *system* conditions concerns load characteristics and needed energy, temperature and humidity. This affects the cold distribution system and how suitable snow cooling is for a certain project. Melt water from snow is about 0 °C at normal conditions why snow cooling for freezing applications are not directly applicable. It is however possible to run freezers towards melt water instead of warm air, which is beneficial both since the temperature difference between the cold and the warm side decreases and because heat transfer to a liquid is better than to a gas. It is however necessary with further studies of chillers to enable condenser temperatures of about 5 °C (Hill, 2004). The needed cooling energy and maximum cooling load influences the plant size and design, but in general a snow cooling system has no power limit since the cold carrier is circulated through the snow at the same rate as it is pumped to the object that is being cooled. With open water circulation systems there is however a delay between water inlet and outlet why a melt water buffer in the pond is necessary to meet load variations. There are snow cooling systems where humidity can be controlled, if air is used as cold carrier.

21.4. Conclusions

In seasonal snow/ice storages frozen water, snow or ice, is stored from winter to summer, when the cold is extracted. The basic idea is that a cold carrier (water or air) is cooled by the snow, to utilize the large latent heat of fusion, and then delivers the cold. The cold carrier is either circulated between the load and the snow or rejected after it has been used for cooling.

There are a number of suggested and implemented ways to store the snow/ice. If the snow is stored underground it is not necessary with thermal insulation. Otherwise a more or less insulated building or insulation directly on the snow is needed. There are different types of insulations; loose fill, sheets and superstructures, with different advantages and disadvantages.

Both natural and artificial snow and ice may be used and there is no size or power limitation for snow cooling systems. Here the main issue is to have enough amounts of frozen water at low cost why the only relevant snow/ice distinction is the density. If natural snow or ice is too expensive or not available in enough quantity, it is possible to produce frozen water. Artificial snow/ice is made with different types of water sprayers, including snow blowers (snow guns). The production rate depends on equipment, relative air humidity, and temperatures of the air and water.

At present (2005) there is one modern Swedish snow cooling plant at the regional hospital in Sundsvall. It has been in operation for six years, and it has mostly worked well. The plant is an open shallow pond with water tight asphalt bottom and larger pieces of wood chips as thermal insulation. The outlet water is cleaned in some steps, and after cooling the heated melt water is re-circulated back to and through the snow, where it is cooled again. To keep a constant water level in the pond some heated water is rejected gradually. Based upon the experiences from Sundsvall it is estimated that a new large-scale plant could deliver cold at competitive cost, with a considerably higher COP.

There are a number of boundary conditions for snow storages; social, political, economical, environmental, natural and system related. Except that the usage of snow cooling systems presupposes that the technique is known the prerequisite as financial support, taxes, regulations and laws are important. The relevant economical boundary conditions are cost of land, frozen water, electricity, fuel and labor. The environmental boundary conditions concerns reduction of green house gases, elimination of ozone depleting gases, noise reduction, aesthetic concerns, peak shaving, snow pollutions, noise from snow making, transports and land/ground usage. Two other important conditions concern the climate, both for receiving snow/ice and cooling needs, and ground conditions. The water availability is important if artificial snow is used. There are however possibilities to save and reuse water. The system related boundary conditions concerns load characteristics and needed energy, temperature and humidity.

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22. DEVELOPMENT OF THE PCM FLOOR SUPPLY AIR-CONDITIONING SYSTEM

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Abstract. Floor supply air-conditioning system using PCM can enhance the building mass storage. In this study, diurnal cooling load is aimed to be covered by stored cold energy in PCM and building during night. Results from actual scale experiments and performance predictions are given.

Keywords: Floor supply air conditioning, granular PCM, diurnal, building mass

22.1. System Concept

A new floor supply air-conditioning system was proposed using phase change material to augment building mass thermal storage [1, 2]. Figure 209 shows conceptual diagrams of the system. In this system, latent heat is stored in PCM that is embedded just under OA floor boards in the form of granules with several millimeters in diameter. The feature of the system is that heat exchange occurs through direct contact between the packed bed of the granular PCM and air flowing as the heat medium. This allows outstanding heat exchange efficiency and little space needed for storing PCM then increase of the TES capacity in the entire system [3]. The whole diurnal cooling load aimed to be covered by stored cold energy in an embedded packed bed of the granular PCM and the building frame during night. In addition, the use of the granular PCM can lead to improvement of the indoor thermal environment in comparison with that in conventional systems due to thermal radiation from the floor surface area, which can be maintained around the phase change temperature.

22.2. Used PCM and Numerical Prediction of System Performances [4]

Figure 210 shows a flowchart of our study and development of actual system. As a result of some trials, PCM granules, which consist of microcapsules with a diameter of a several micrometers containing paraffin wax PCM, named

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 367–373.

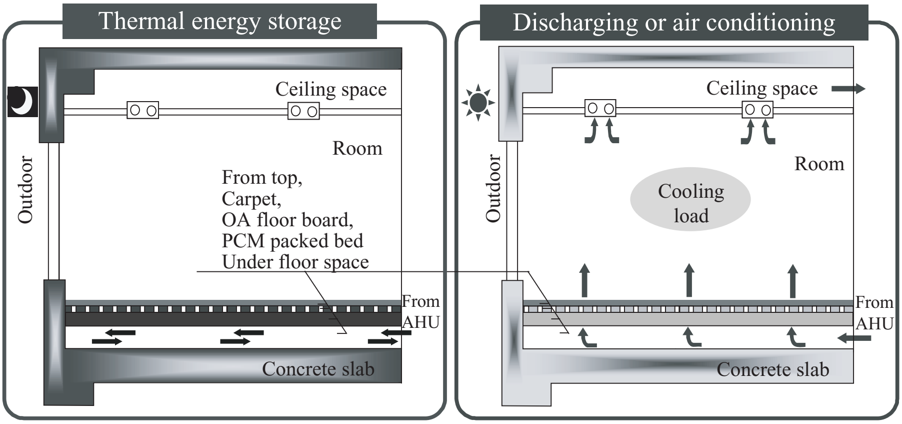


Figure 209. Conceptual diagrams of the developed system

FMC-PCM (Flocculated Microcapsules PCM) as in Figure 211 have been applied in this system. Figure 212 indicates a cross section of the FMC-PCM taken by a scanning electron microscope. Operating conditions were discussed by a developed computer simulation program, which includes not only heat balances for each component under the floor board and for the room space according to the air movement but also analyses of radiative thermal environment in the target room using the Monte-Carlo method. The values of heat transfer coefficient of each part were determined as compared with measurements. Calculation predicted that the use of a FMC-PCM, which shows phase change temperature between 20.0 °C and 22.9 °C and latent heat

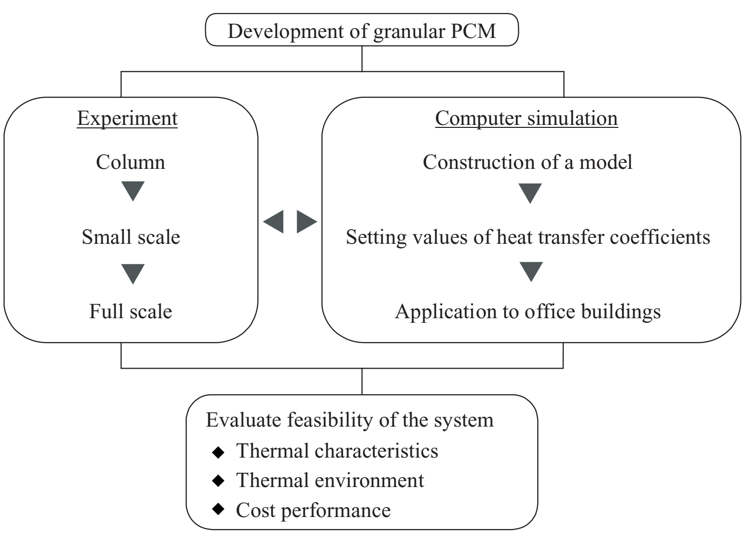


Figure 210. A flowchart of the development

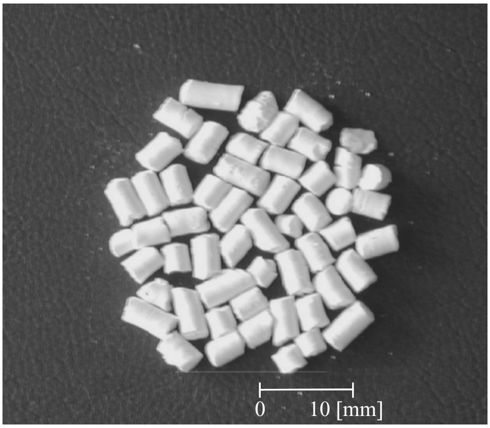


Figure 211. Appearance of FMC-PCM

of 130 kJ/kg, led to a load shifting ratio η_s of 100% in the case of the packed bed with a thickness of 25 mm and a weight of 12.5 kg/m^2. Additionally heat radiation from the floor face resulted in comfortable thermal environment even at a room temperature of 28 °C in office hours.

22.3. Results of Actual Scale Experiment and Discussions

On the basis of calculation results, full scale experiments were conducted in a test room with a floor area of 9.2 m^2 shown in Figure 213. In this experiment, packed beds of FMC-PCM, which shows phase change temperature between

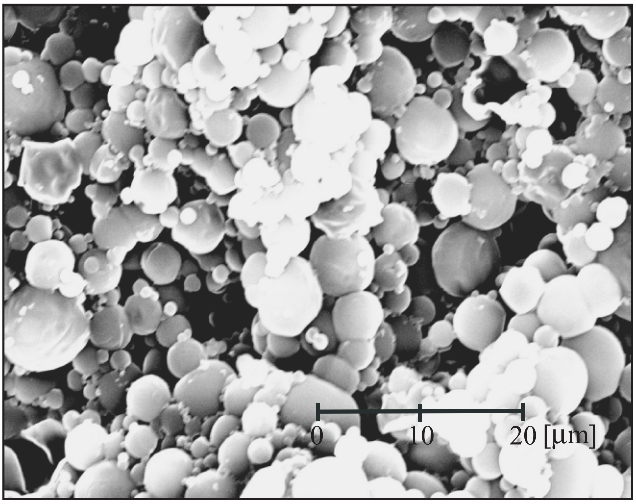
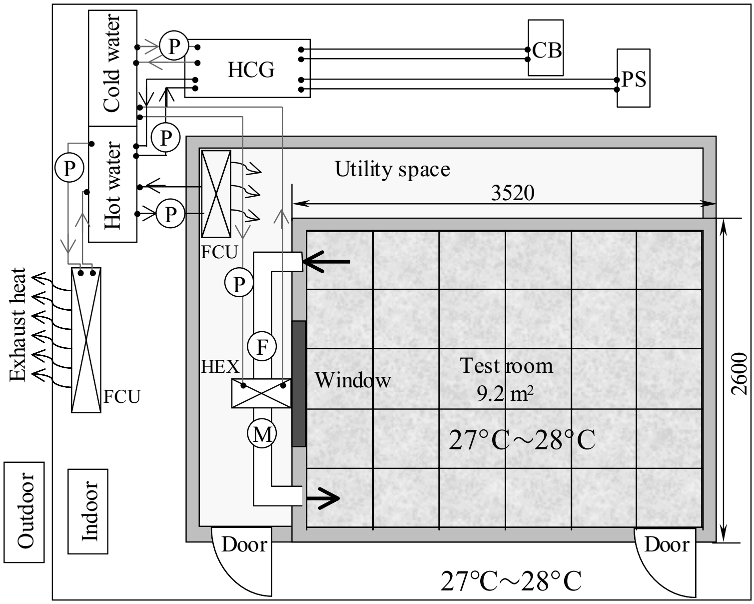


Figure 212. Cross section of the FMC-PCM



CB: Control board, F: Fan, FCU: fan coil unit, HCG: Hot and cold water generator,
HEX: Heat exchanger, M: Flow rate meter, P: Pump, PS: Power supply

Figure 213. Plane of the full scale experimental room and air-conditioning units

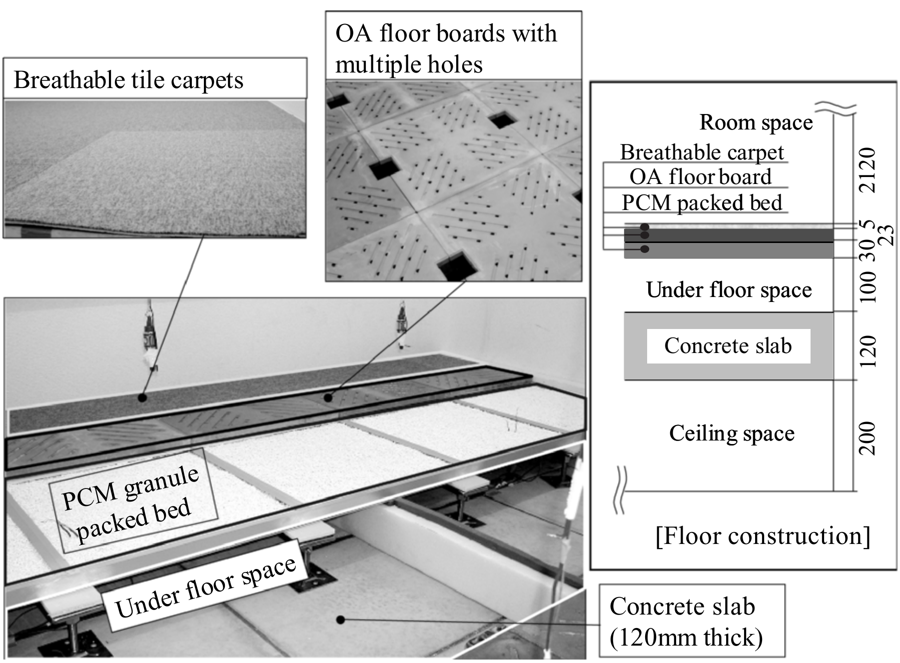


Figure 214. Detail construction of under floor ventilation system

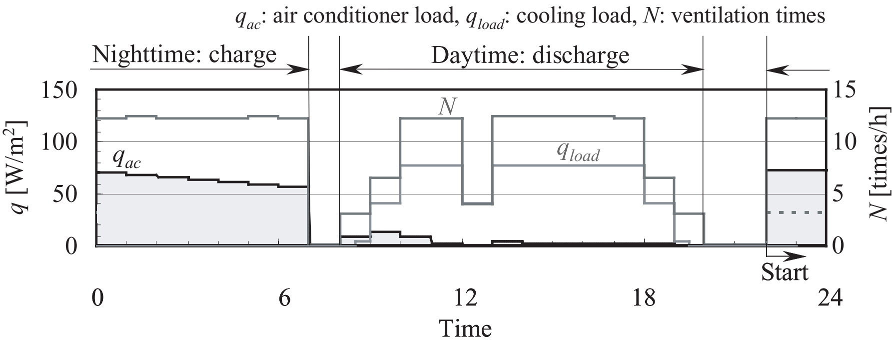


Figure 215. Variation of air conditioner load indicates that almost all diurnal cooling load can be covered by stored cold energy in the nighttime

18.2 °C and 21.4 °C and latent heat of 136 kJ/kg, were installed under the OA floor board with a thickness of 25 mm in 12.5 kg/m^2 (Figure 214). The authors made some experiments under different conditions and explain one successful example.

Cool air at 12 °C was flowed into the under floor space between OA floor boards and concrete slab in order to store thermal energy during night. On the other hand, room air circulated through the PCM packed bed at the air change rate of 12 times per hour during office hours. The air change rate was decreased before 10:00 in the morning, during the lunch time and after 18:00 according to the cooling loads. An air conditioner was operated when the room air temperature exceeds 27.5 °C. Stored cold energy was estimated 2.1 MJ/m^2 during night and the air conditioner supplied only 0.2 MJ/m^2 during daytime. Consequently, this system could achieve a load shifting ratio η_s of 92% as shown in Figure 215, whereas another condition with conventional thermal mass storage, that is, without PCM, shows the η_s of 50%.

22.4. Indoor Thermal Environment

Previous under floor ventilation systems have indicated a disadvantage of low room temperature at the beginning of office hours, in which only sensible thermal energy storage is applied by using building frame such as concrete slab. The use of the FMC-PCM is expected to improve this drawback. Figure 216 illustrates variations of room air temperature T_a and mean radiative temperature T_r. Both T_a and T_r were kept around 24 °C even at 9:00. In addition though T_a reached 28 °C at 6 PM, the maximum T_r was 27 °C due to radiative influence from the lower floor surface temperature. PMVs, that is one of the representative indexes for thermal sensation, were −0.4 at 8:00 and

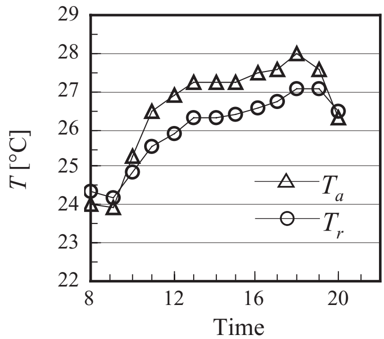


Figure 216. Hourly variations of room air temperature T_a and mean radiative temperature T_r

0.4 at 13:00 lying within the neutral zone. The thermal sensation tests using some young subjects resulted in the answers of "Comfortable" throughout the daytime. These results indicate that this system can provide the comfortable thermal environment as well as the high rate of the load shifting.

22.5. Cost Analysis and Further Challenges

At this moment, the price of the used granular PCM exceeds 10 EURO/kg due to the test production stage. Our calculation showed the cost payback time can be less than 10 years when the PCM cost would be reduced to 4 EURO/kg under the electrical utility rate condition in Japan. Further cost reduction will be needed to promote the actual system. In addition we have to care of flammability of paraffin wax as PCM in use of inside the buildings. Fortunately, PCM is placed under the OA floor board made of fireproof cement-mortal and above the concrete slab in this system. It may not have any problems under the Fire Defense Law in Japan. However, further development of noninflammable PCM granules, for example micro encapsulation of mixture of inorganic and organic PCM, is required.

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PART VI. THERMOCHEMICAL ENERGY STORAGE

23. CHEMICAL ENERGY CONVERSION TECHNOLOGIES FOR EFFICIENT ENERGY USE

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Abstract. Energy conversion technologies using chemical reaction are introduced. Thermal energy conversion by chemical heat pumps and a hydrogen production system is shown mainly as efficient energy utilization technology utilizing chemical reaction ability. Chemical reaction would be useful for thermal energy management, because heat density of chemical changes is relatively higher than one of physical changes, which are used in conventional conversion system. Then, reversible chemical reaction is expected to have potential for thermal energy conversion, storage and utilization process in the next generation. The know-how of development of energy conversion system utilizing reversible chemical reactions is explained using chemical reaction equilibrium analysis. Chemical heat pump for thermal energy storage and conversion, and hydrogen production utilizing separation process are reviewed as practical example. Possibility of chemical energy conversion methodology would be understood from this section.

Keywords: chemical energy conversion; energy storage; chemical heat pump; separation; hydrogen production; reaction equilibrium

23.1. Introduction

Energy conversion is important technology for modern life. Mechanical energy conversion technologies using internal and external heat engines offered great contribution on 20th century industry and people's life. We in 21st century are required to reduce the emission and also efficient energy use from the standpoint of reduction of carbon dioxide emission. Paradigm shift is required in energy conversion technology field to realize the emission reduction. Chemical reaction is attractive process for the technology. Energy conversion utilizing chemical reaction has possibility to realize higher efficient compared with usual energy conversion system. Chemical energy conversion has already been popular in battery, fuel cell and so fourth. On the other hand,

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 377–391.

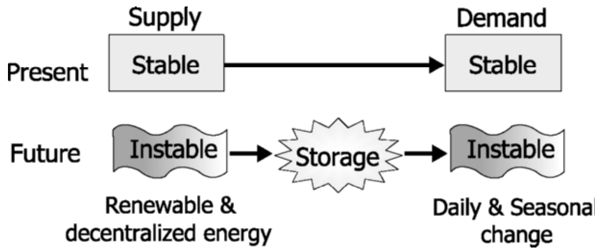


Figure 217. Trend shift of energy chain between energy supply and demand sides

application of the energy conversion on thermal energy utilization field has still frontier. Energy storage is also important key technology to realize efficient energy usage especially for thermal energy use, because it is hard to be stored efficiently. Figure 217 shows trend shift of energy supply chain from present to future. In the present or previous energy system, energy demand side is relatively stable, and then supply side just delivers produced energy to demand side directly on time. However, instability of daily and seasonal change of energy demand becomes more visibly recently. On the other hand, energy production from renewable energy and decentralization of power plant induce instability of energy demand side. Some energy storage process will be required for the connection between instable supply and demand, because generated energy in the supply side cannot deliver to demand side directly. Future energy system needs not only high efficient energy produce and usage system, but also efficient energy storage system.

Chemical reaction has potential for both efficient energy conversion and storage. This part shows possibility of efficient thermal energy storage and thermal energy conversion methodologies by chemical reaction.

23.2. History of Thermochemical Energy Storage

The world oil shock of 1973 triggered the development of heat pump technologies for enhancement of economical and efficient energy usage. Thermochemical heat energy storage using chemical heat pump was started to be developed as one of candidates at the moment. Variety of chemical heat pump systems had been discussed in many countries which have independent environmental and energy situations each others. Inorganic/ammonia systems of NH_4Cl, LiCl, $MgCl_2$, $CaCl_2$, $MnCl_2$ and $FeCl_2$/ammonia were well discussed in Sweden [1] and Swiss [2] in 1970s. In the United States, the Brookhaven National Laboratory conducted program for chemical heat pump and chemical energy storage [3] from 1975 under contract with the Department of Energy. Five subjects of (1) Calcium chloride/methanol, (2) magnesium-chloride/water, (3) sulfur oxide/water, (4) Ammoniate/ammonia, (5) metal hydride were studied in the program.

Magnesium oxide/water system (1977) [4] and Calcium oxide/water (1981) [5] system were discussed firstly as candidates in inorganic reaction system group for high-temperature thermal energy storage for solar and industrial waste heats. Organic reaction system has advantage that it can be operated generally in continuous flow cycle compared with that gas/solid inorganic system is operated generally batch-wise. Acetone/2-propanol/hydrogen system [6, 7] and benzene/cyclohexane/hydrogen system were well discussed [8] in 1980s. In Germany, EVA-ADAM long distance thermal energy transportation [9], which was proposed in 1970s and demonstrated in 1980s, was also one of practical example of thermochemical heat storage.

In Japan, the Agency of Industrial Science & Technology conducted the project of "Super heat pump and energy integrated system" [10] during 1984–1992. Ammoniate/ammonia of $FeCl_2$, $NiCl_2$, NaI and $NaScN/NH_3$, and halogenated inorganic reactant/water of $MgCl_2$, $CaCl_2$, $CaBr_2$ and $MnCl_2/H_2O$ were discussed as candidates for chemical heat pump in this project. The project catalyzed heat pump research in Japan. After principal proposal of chemical heat pump, practical discussion of them was examined in 1990s. CaO/H_2O chemical heat pump was studied thermal energy transfer enhancement [11]. Development of practical material which had high-durability to cyclic repetitive operation was studied in MgO/H_2O [12]. A twin chemical reactions type heat pump using CaO/CO_2 and PbO/CO_2 was proposed [13]. Thermal conductivity enhancement in solid reactor bed becomes one of important subjects. Enhancement of reactor bed thermal transfer using mixture of reactants and heat transfer enhancer were discussed in France [14], UK [15] and Japan [16]. Tert-butanol/iso-buntene/H_2O was proposed for lower quality waste thermal energy recovery [17] at less than 60 °C as an organic reaction type. This history says that development of new reaction and material technologies creates new possibility of thermochemical energy technologies.

Open and closed sorption systems were investigated concerning their application as thermal energy storage devices. The storage capacity of these systems is usually related to the volume or the mass of the dry sorbent. Selvidge and Miaoulis [18] a range of different sorption materials was tested concerning their possible storage capacity. The result was, that Zeolites as a class of solid adsorbents are able to store an almost two time higher amount of thermal energy than Silicagels.

Alefeld [19] worked with closed Zeolite/water systems. They reached storage capacities of about 250 kWh/*t* related to the dry mass of Zeolites in their experiments. The change in the water uptake was about 25% of the Zeolites dry mass. The load levelling within a district heating net was identified as a possible application for such systems. This can be realized at the power plant (centralized) or at the consumer side (decentralized). An application as a seasonal storage system was excluded for economical reasons.

Thermal energy storage in open sorption systems was investigated by Close and Pryor [20], Clos and Dunkle [21] and Pryor and Close [22]. They put emphasis on the high storage capacities of these systems compared to sensible heat storages and the possibility to support drying processes by the application of this technology.

Gopa et al. [23] reported about open Zeolite/water systems for the storage of solar heat. In their work they were presenting stability tests, methods for the definition of the adsorption enthalpy in dependence on the adsorbed amount of water and the possible heating power during the discharging process.

23.3. Why Chemical Reaction?

Although there are a lot of ways for thermal energy conversion and storage, thermal performance of an energy system depends on thermodynamic properties of the used energy media. Higher-energy storage density and reversibility are required on the materials for thermal energy conversion and storage. Figure 218 shows relative relationship of energy densities of physical and chemical changes [24]. Energy density of chemical changes is relatively higher than one of physical changes. Sensible heat and phase changes in physical changes are popular system for conventional energy conversion and storage technologies, such as steam engines, because of well reversibility of their changes. On the other hand, chemical changes such as oxidation are irreversible and hard to apply for repetitive heat storage operation. Then, reversible chemical reaction is expected to have potential for energy conversion

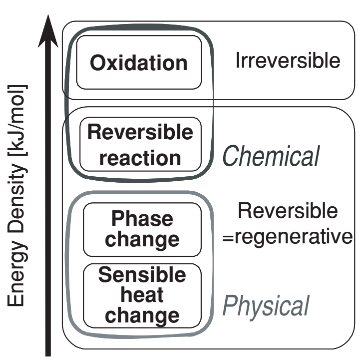


Figure 218. Energy density of physical and chemical changes

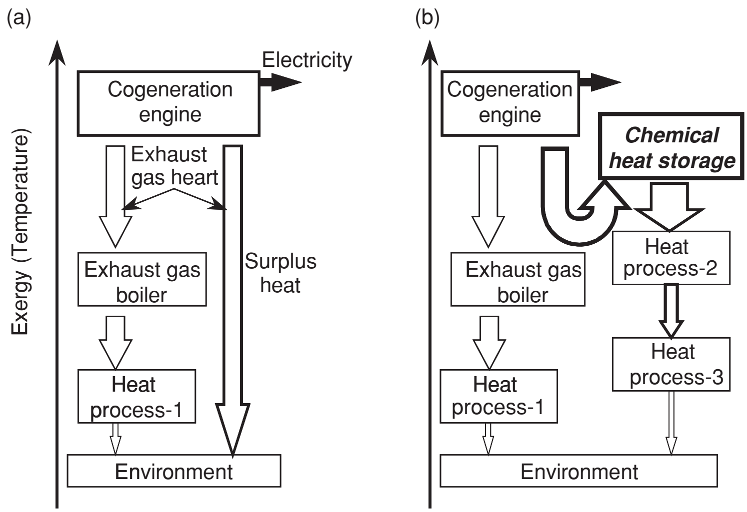


Figure 219. Contribution of chemical heat storage process on cogeneration system; (a) conventional cogeneration system, (b) combined system with chemical heat storage process

and storage process in the next generation because of its relative higher-energy density and reversibility.

A merit of chemical energy conversion is the possession of efficient energy storage performance. Especially the performance is advantageous for thermal energy storage. Physical thermal storage gradually loses thermal energy by heat conduction and radiation, however, chemical storage can store energy as reactants with small loss.

Figure 219 shows a contribution of chemical heat storage process on cogeneration system [25]. A conventional cogeneration system shown in Figure 219(a) uses a shaft work and exhaust heat of a gas, a diesel engine or micro gas turbine for electrical and heat output, respectively. The high temperature exhaust gas of the engine is generally used to generate steam at an exhaust gas boiler. However, since the demand for the electrical output is generally inconsistent with that for the heat output, a large amount of surplus heat output is occasionally discharged into the atmosphere as shown. Chemical heat storage has a possibility to enhance the energy use efficiency of a cogeneration. The proposed system is shown in Figure 219(b). The system consists of a cogeneration engine and a chemical heat storage system.

The heat storage system is operated in batch mode between a heat storage mode and a heat output mode. In the heat storage mode, an endothermic reaction process of the chemical heat storage system proceeds by consuming surplus waste heat generated from the engine. In the heat output mode, an exothermic reaction process of the storage system proceeds generating a reaction heat output. Because the heat storage system could store heat for longer period as chemical reactants, and the heat output temperature could be variable by choosing reaction conditions, then the heat output would be supplied at various temperatures, when it is required. In consequence, the surplus heat is utilized more efficiently than a conventional heat storage system.

23.4. Chemical Heat Pump

Chemical heat pump is a representative of chemical thermal energy conversion and storage systems. This section shows fundamental of chemical heat pump. The knowledge would be applicable for other chemical energy conversion system.

23.4.1. HOW TO FIND AND USE REACTION—CHEMICAL EQUILIBRIUM

When subjected heat source for chemical energy conversion has been targeted, an appropriate reaction system has to be searched for the conversion. The temperature range of the reaction should fit the temperature of the heat source. The turning temperature of reaction, T_{turn} (K), would be a measure to find the reaction system. Gibbs's free energy change of reaction, ΔG (kJ mol^{-1}), is obtained from reaction enthalpy change, ΔH (kJ mol^{-1}), and enthalpy change, ΔS (kJ mol^{-1} K^{-1}).

$$\Delta G = \Delta H - T\Delta S \tag{1}$$

ΔG has relationship with the reaction equilibrium constant, K.

$$\ln K = -\frac{\Delta G}{RT} = -\frac{\Delta H}{R} \cdot \frac{1}{T} + \frac{\Delta S}{R}. \tag{2}$$

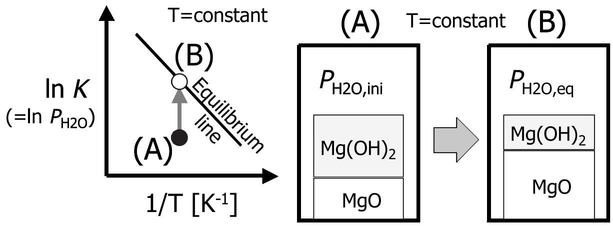
Reversible reaction condition is established around unity of K, that is, $\Delta G = 0$. Then, T_{turn} is reduced by

$$T_{turn} = \frac{\Delta H}{\Delta S}. \tag{3}$$

When a reversible chemical reaction has a temperature of T_{turn}, the reaction can be applicable for thermal energy utilization at around the temperature.

Now, the following gas – solid reaction of magnesium oxide/water (MgO/H_2O) is used as an example reaction,

$$Mg(OH)_2(s) \Leftrightarrow MgO(s) + H_2O(g). \tag{4}$$



(a) (b)

Figure 220. Reaction direction from a state (A) to an equilibrium state (B) in a reaction of MgO/H_2O

The reaction equilibrium of it is defined as

$$K = \frac{a_{\mathrm{MgO}} b_{\mathrm{H_2O}}}{a_{\mathrm{Mg(OH)_2}}} \tag{5}$$

a_i is an activity of a reactant. Activity of solid and gas material regards as unity and partial pressure, respectively. Then, Equation (7) is reduced to

$$K = P_{\mathrm{H_2O}} \tag{6}$$

$$\ln P_{\mathrm{H_2O}} = -\frac{\Delta H}{R} \cdot \frac{1}{T} + \frac{\Delta S}{R}. \tag{7}$$

Equation (7) shows the reaction temperature corresponds to the reaction pressure in Equation (4) type gas–solid reaction system. Relationship between $1/T$ and $\ln K (= P_{\mathrm{H_2O}})$ is linear at a range in which changes of ΔH and ΔS are negligible, and is called as a reaction equilibrium line.

Figure 220(a) shows the line and the state change of the reaction state. At a constant temperature, an initial state at (A) moves to an equilibrium state (B) on the line finally. Figure 220(b) shows a schematic of state change in a reactor from the state (A) to the equilibrium state (B). The reaction proceeds to attain the state in the equilibrium condition, and reaction pressure changes with the reaction progress, when the pressure attains an equilibrium state, the reaction is terminated.

23.4.2. CHEMICAL HEAT PUMP OPERATION

Chemical heat pump uses chemical reaction for thermal energy storage and conversion. The heat pump operation is based on reaction equilibrium relationship, and has two operation modes. Figure 221 shows equilibrium relationship of chemical heat pump cycle for MgO/H_2O system at (a) heat amplification and cooling mode and (b) heat transformation mode [26]. Figure 222 shows

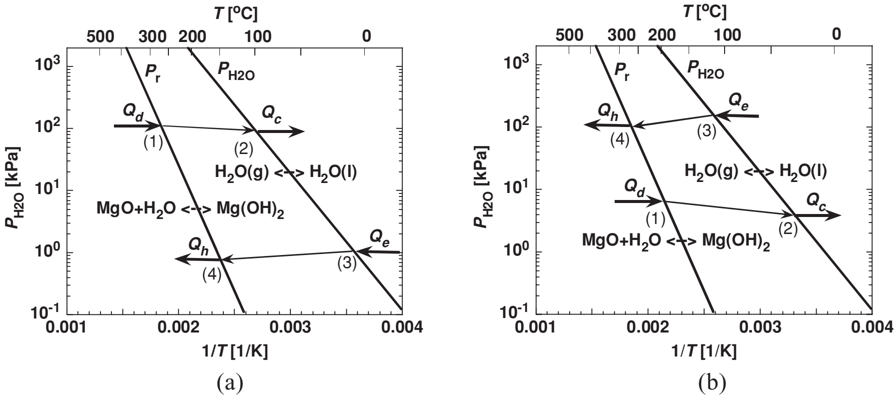


Figure 221. Equilibrium relationship of chemical heat pump cycle for MgO/H_2O system; (a) heat amplification and cooling mode, (b) heat transformation mode

heat pump structure and operation at heat amplification and cooling mode. Numbers at the equilibrium line in Figure 221(a) are corresponding to the state of the reactor at the same number in Figure 222. Generally, a chemical heat pump needs two reaction equilibriums and two reactors for realizing both reaction equilibriums. At heat amplification and cooling mode (Figures 221(a) and 222), high-temperature heat source (1) is stored in the system by endothermic dehydration, and condensation heat of vapor is used at heat demand side at (2) in the heat storage mode. At the heat output mode, hydration of magnesium oxide (4) proceeds consuming the vapor, then evaporation of water (3) is enhanced, and cooling output is generated at (3). Because produced heat output ($Q_h + Q_c$) is larger than heat source amount (Q_d), then this mode is called as heat amplification mode. The heat transformation mode consumes heat sources at middle-temperatures (1, 3) to generate

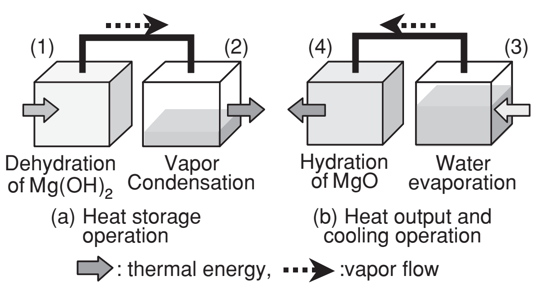


Figure 222. MgO/H_2O chemical heat pump operation in heat amplification mode with cooling operation

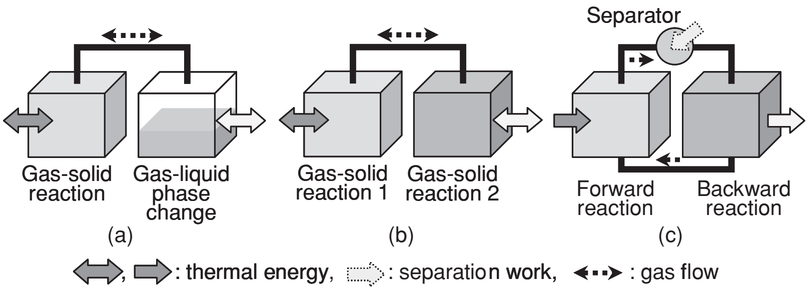


Figure 223. Types of chemical heat pump; (a) reaction & phase change batch type, (b) reaction & reaction batch type, (c) reaction & separation work continuous type

heat at higher temperature (4). Operation cycle resembles to one of adsorption heat pump. The advantages of chemical heat pump compared with other heat pumps are higher heat storage density, the wider operation temperature around near ambience temperature to over 1000 °C, controllable operation temperature by choice of reaction condition and thermal drivability with high efficiency.

23.4.3. VARIETY OF CHEMICAL HEAT PUMP

Chemical reaction having reversibility can be used for chemical heat pump. Figure 223 shows types of chemical heat pump [24]. The structure of a chemical heat pump depends on reaction used in the pump. Generally, the heat pump consists of two reactors. Gas–solid reaction is applicable for batch operation. Driving force for heat pump operation comes from pressure difference between two reactors. Gas–liquid phase change or secondary gas–solid reaction [27] are used for the driving force (Figures 223(a) and (b)). Table 28 shows a classification of the heat pump. Some driving force is required to operate a heat pump. Continuous operation Figure 223(c) is realized by gas–liquid reaction systems [28]. Forward reaction and reverse reaction are proceeded in different catalytic reactors. Driving force of the operation is concentration difference arisen by separation process using distillation, pressure difference generated by compression process, or combination of both the differences. Reaction systems in Table 28 are discussed practically. Reaction for a chemical heat pump is needed reaction reversibility, reactivity, and durability to the repetitive reaction firstly, and safe for material and operation, low-cost, compactness and light weight secondary.

A plenty of rooms to find new reaction systems would still remain because there are thousands reversible reaction and chemical material performance is developing every time.

TABLE 28. Classification of chemical heat pump

Operation	batch		Continuous	
Reaction phase	Gas-solid		Gas-liquid	
Driving operation	Gas-liquid phase change	Chemical reaction	distillation, membrane separation	compressor
Driving force	Reaction pressure	Reaction pressure	concentration	Reaction pressure
Reactor system	(a)	(b)	(c)	
Reaction example	$CaCl_2/NH_3$, CaO/H_2O, MgO/H_2O, Adsorption heat pump	$CaO/PbO/CO_2$, Metal hydrate/H_2	Acetone/H_2, Absorption heat pump	Isobutene/H_2O, benzene/H_2

23.5. Energy Media Transformation—Hydrogen Production

23.5.1. SEPARATION PROCESS FOR PRODUCTION ENHANCEMENT

Energy media transformation is one of performance of chemical energy conversion. Hydrogen production is a practical candidate. Its requirement is growing rapidly as fuel cell technology advance. Hydrogen production using chemical energy conversion technology is introduced as energy media transformation in this section. Separation process is key technique for efficient energy media transformation. Figure 224 shows concept of separation processes to enhance production yield using hydrogen production from water as an example. The yield of H_2 is restricted by equilibrium in state (A) in

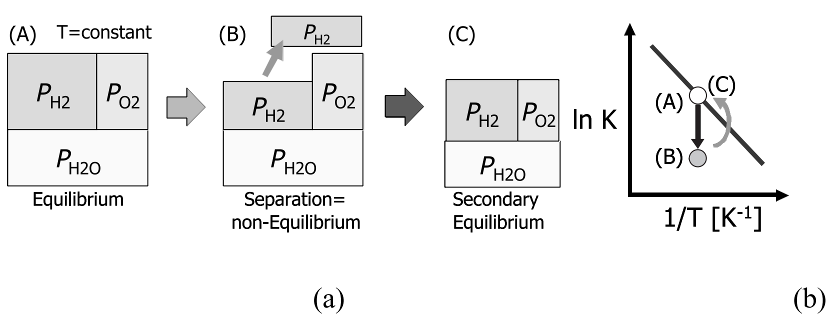


Figure 224. Separation process for enhancement of energy media transformation, (a) Schematic of the process, (A) an original equilibrium, (B) separation of hydrogen, (C) secondary equilibrium, (b) Relationship between separation process and reaction equilibrium line

Figures 8(a) and (b). Removing H_2 from the system induces the system into non-equilibrium state (B), and hydrogen production is going on to establish the next equilibrium state (C) by Le Chatelier's principle. The yield of H_2 will be enhanced in the result.

23.5.2. FUEL REFORMING FOR HYDROGEN PRODUCTION SEPARATION PROCESS FOR FUEL REFORMING

Fuel reforming is popular way for hydrogen production for fuel cell use. Hydrocarbons are used for the fuel resource. Methane (CH_4) steam reforming process consists of the following two gas phase reactions with various catalysts.

Methane steam reforming:

$$CH_4(g) + H_2O(g) \leftrightarrow 3H_2(g) + CO(g), \Delta H^{\circ}_{\text{reform}} = +205.6\,\text{kJ/mol} \tag{8}$$

Carbon monoxide (CO) shift reaction:

$$CO(g) + H_2O(g) \leftrightarrow H_2(g) + CO_2(g), \Delta H^{\circ}_{\text{shift}} = -41.1\,\text{kJ/mol} \tag{9}$$

Separation process is key technique for high-efficient fuel reforming. Figure 225 shows concept of the separation processes to enhance hydrogen yield. In conventional steam reforming in Figure 225(a), hydrocarbons attains to equilibrium and produces H_2, carbon dioxide (CO_2) and carbon monoxide (CO). The yield of H_2 is restricted by equilibrium. Removing H_2 from reforming reaction system in Figure 225(b) induces the system into non-equilibrium state, and hydrogen production is going on to establish the next equilibrium state. The yield of H_2 will be enhanced in the result. Removing

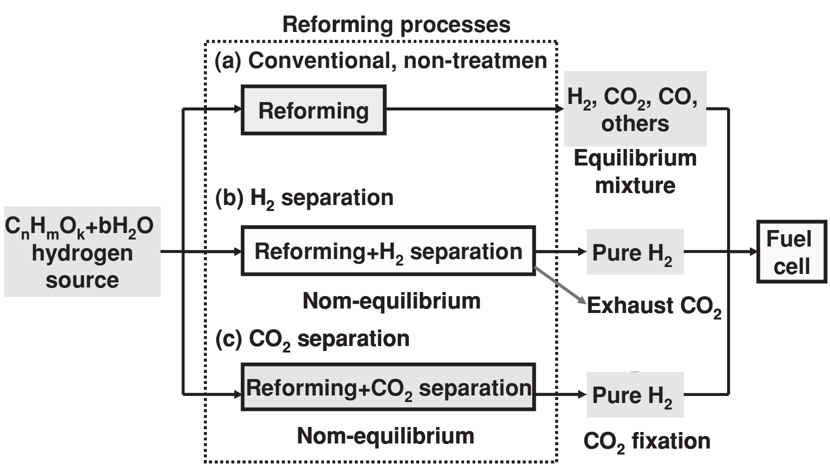


Figure 225. Separation processes for enhancement of H_2 yield in fuel reforming

CO_2 in Figure 225(c) induces also non-equilibrium state and enhances CO_2 production, then H_2 productivity and purity are also enhanced. These separation processes would realize not only high-yield of H_2, but also decrease of temperature of the endothermic reforming. It means that the separation process is important methodology for energy media transformation and chemical energy conversion.

23.5.2.1. *Carbon Dioxide Separation for Fuel Reforming*

Carbon dioxide separation reforming in the above mentioned is one of useful methodologies for efficient hydrogen production [29]. Calcium oxide (CaO) carbonation can absorb CO_2 from the reformed gas and fix it.

Carbonation of calcium oxide:

$$\mathrm{CaO(s)} + \mathrm{CO_2(g)} \leftrightarrow \mathrm{CaCO_3(s)}, \Delta H^\circ_{\mathrm{CaO}} = -178.3\,\mathrm{kJ/mol} \tag{10}$$

Equations (8)–(10) reactions are able to cause in the same reactor at once. These reactions, taken as a whole, are defined as regenerative reforming, because the reforming is regenerative thermally [30].

Regenerative reforming:

$$\mathrm{CaO(s)} + \mathrm{CH_4(g)} + 2\mathrm{H_2O(g)} \leftrightarrow 4\mathrm{H_2(g)} + \mathrm{CaCO_3(s)}, \quad \Delta H^\circ_{\mathrm{regen}} = -13.3\,\mathrm{kJ/mol} \tag{11}$$

Conventional steam reforming is depicted in Figure 226(a). CH_4 and H_2O react by Equation (8) in a catalytic reformer, and the generated CO is shifted by Equation (9) into CO_2 and H_2 in a catalytic converter. The endothermic reforming process needs a heat supply of $\Delta H^\circ_{\mathrm{reform}}$. A proposed reforming with CO_2 separation process is shown in Figure 226(b). This process consists of a reforming process (Figure 226(b-1)) while the vehicle is driving and a regenerating process (Figure 226(b-2)) for CaO regeneration and CO_2 recovery while the vehicle is turned off. CaO and a reforming catalyst mixture are packed in a regenerative reformer. Reactants are reformed by Equation (8), and generated CO_2 is removed from the gas phase by the CaO carbonation of Equation (10). The CO shift reaction of Equation (9) is enhanced under the non-equilibrium condition realized by the CO_2 removal. Purified H_2 is generated from the reactor finally. The whole reaction of Equation (11) is exothermic, hence the reaction needs no heat supply and can proceed spontaneously. A zero CO_2 emission drive is possible due to CO_2 fixation resulting from the carbonation. In the regenerating process, $CaCO_3$ is decomposed endothermically into CaO in the reactor using high-temperature heat, which is assumed to be supplied as heat from high temperature process, or as joule heat generated from off-peak electric output generated from nuclear power plants and also renewable energy system. The reformer is regenerated, and

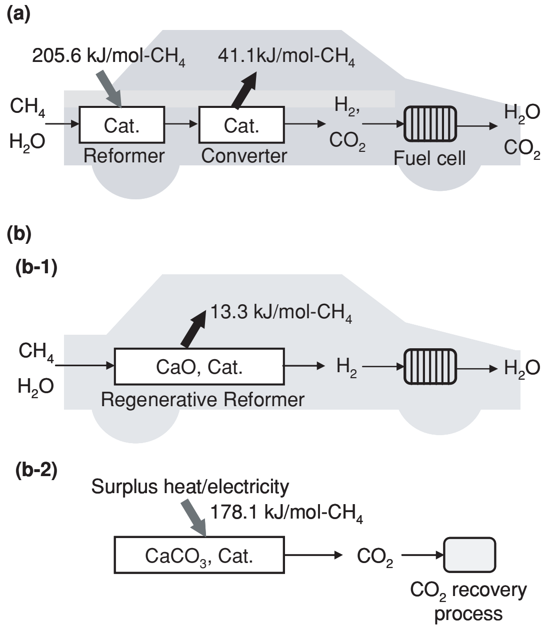


Figure 226. Concept of a zero CO_2 emission vehicle using a thermally regenerative reformer; (a) conventional reforming, (b) proposed thermally regenerative reforming, (b-1) reforming mode, (b-2) regenerating and CO_2 recovering mode [6]

used again for the reforming. The proposed regenerative reformer is intended to be contained in a removable package for use in a fuel cell vehicle. The package is loaded into and recovered from a vehicle at a regeneration station that supplies new packages and regenerates used ones. Regenerated CO_2 is managed according to a CO_2 recovery process. When the recovered CO_2 is regenerated in fuel hydrocarbons by hydrogenation process using H_2, which is generated from water electrolysis consuming the power plant output, and the regenerated hydrocarbons are reused cyclically in the vehicles, then, a comprehensive zero CO_2 emission system is established. It is one of examples of possibility of chemical energy conversion.

23.6. Summary

Chemical energy conversion has potential to realize efficient energy usage. It is important to find appropriate reversible chemical reaction for the temperature range of subjected energy source. Every reaction would have possibility to develop every original energy conversion process. Not only to find new reaction system, but also to improve proposed reaction is key point for the heat pump development, because chemical material performance is developing continuously every time. Proposing new chemical energy conversion system would contribute on efficient energy utilization and sustainable our life on the globe.

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24. SORPTION THEORY FOR THERMAL ENERGY STORAGE

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Abstract. The theory of sorption processes and its relevance for thermal energy storage (TES) concepts shall be introduced. Starting from the thermodynamics of TES systems a motivation for sorption storage systems will be developed. The adsorption theory is based on the adsorption equilibrium. The equilibrium can be described by isotherms (curves of equal temperature), isobars (curves of equal water vapour pressure) and isosteres (curves of equal water concentration within the adsorbent). From the adsorption equilibrium the adsorption enthalpy or heat of adsorption can be calculated. The heat of adsorption describes the amount of energy involved in the process. The ratio of discharged to charged thermal energy and the possible storage capacity of different applications derived from the adsorption equilibrium and the heat of adsorption will be defined. A similar method—from the equilibrium to the storage capacity—will be shown for liquid sorbents.

Keywords: Adsorption, absorption, thermal energy storage, adsorption equilibrium, heat of adsorption

24.1. Introduction

Thermal Energy Storage can be realized by utilizing reversible chemical reactions. The number of possible reactions for this application from first principle is huge, however only very few are suitable concerning a usable reaction temperature. The process of adsorption on solid materials or absorption on liquids is the most investigated one. Figure 227 shows the process schematically.

Adsorption means the binding of a gaseous or liquid phase of a component on the inner surface of a porous material. During the desorption step—the energetic charging step—heat is put into the sample. The adsorbed component, in this example water molecules, are removed from the inner surface. As soon as the reverse reaction—the adsorption—is started by adding water molecules to the sample, the molecules will be adsorbed and the heat, brought into the system during desorption will be released. The adsorption step represents the discharging process.

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 393–408.

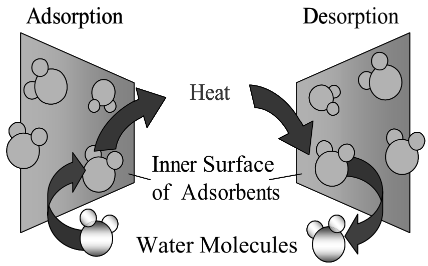

Figure 227. Adsorption process of water vapor on solids

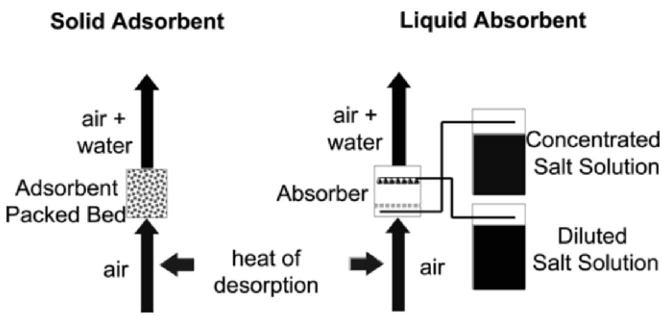

Figure 228. Examples of open sorption storage systems during desorption/charging

Figure 228 shows the examples of liquid and solid open sorption storage systems. In both cases the Desorption is activated by an hot air stream carrying the heat of desorption. For the solid a packed bed of adsorbent pellets and for the liquid solution a reactor are blown through, leaving the packed bed dry and the solution concentrated.

TES can be achieved by separating the desorption step (charging mode) from the adsorption step (discharging mode). After desorption the adsorbent and the absorbent can theoretically remain in the charged state without any thermal losses due to the storage period until the adsorption process is activated.

Figure 229 shows schematically the discharging of open sorption storages. Humid air blown through the storage becomes dry and can be used for

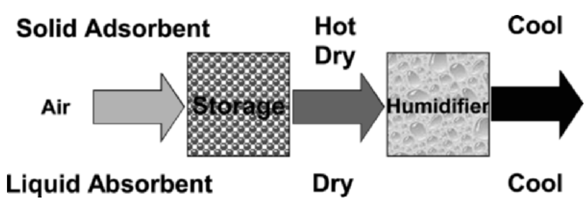

Figure 229. Examples of open sorption storage systems during adsorption/discharging

dehumidification or, by adding an humidification step, for cooling (desiccant cooling systems). If solid adsorbents are used the air might be very hot after the adsorption. This heat can be used for heating purposes.

24.2. Thermodynamics

In this chapter the question "why are sorption systems interesting for thermal energy storage?" should be answered. The reason for ongoing research is the possibility of high storage capacity, or energy density of the storage medium by the utilization of chemical reactions.

It is useful to distinguish between *direct* and *indirect* thermal energy storage (TES) based on the thermodynamics of the process [1].

A direct TES is charged by a heat flux from a heat source to the storage (see Figure 230). In general a heat flux Q is coupled to a flux of entropy S. Both are in related by the equation $Q = TS$, where T is the temperature within the storage. Thus heat at high temperature is coupled to a smaller amount of entropy as heat at flow temperature. As a consequence it can be said, that the capacity of a thermal energy storage is at a given temperature of the heat source depending on the capacity of entropy uptake.

One known disadvantage of direct TES is the fact, that they have to be at a higher (or lower) temperature as the ambience. Due to this temperature difference (their exergy content) the are able to operate as heat (or cold) storage. A thermal insulation is necessary to avoid losses over the storage period.

The limitation of the storage capacity is, as mentioned before, caused by the limitation of entropy change ΔS within the storage (see Figure 4). For sensible and latent heat storage (so-called "direct" thermal energy storage) this is defined by the specific heat

$$c_P = T\left(\frac{\partial S}{\partial T}\right). \tag{1}$$

Within a temperature interval $\Delta T = T_2 - T_1$ the stored heat is

$$Q_{sens} = \int_{T_1}^{T_2} c_P \cdot \mathrm{d}T = \int_{T_1}^{T_2} T\left(\frac{\partial S}{\partial T}\right) \cdot \mathrm{d}T = \bar{T} \cdot \Delta S_{21} \tag{2}$$

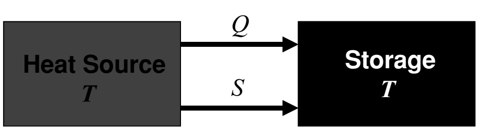

Figure 230. Direct thermal energy storage (Q thermal energy, S entropy, T temperature)

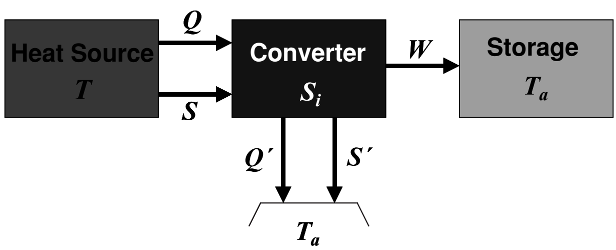

Figure 231. Indirect thermal energy storage by the conversion of thermal energy into work (S_i entropy production due to internal irreversibilities, Q' and S' waste heat and entropy of the converter, T_a ambient temperature, W work)

where Q_{sens} is the sensible heat,

$$Q_{lat} = \int_{T_1}^{T_2} c_P \cdot \mathrm{d}T + \Delta H_{ls} = \bar{T} \cdot \Delta S_{21} + T_{ls} \cdot \Delta S_{ls} \tag{3}$$

where Q_{lat} is the sensible and latent heat stored and ΔH_{ls} is the heat of fusion at the phase change temperature T_{ls}.

The reachable temperature difference ΔT is determined by the charging temperature T_C, which is given by the heat source.

One possibility to avoid this limitation is the conversion of heat into another kind of energy like mechanical or electrical energy. In this case (see Figure 231) the converter is producing entropy free work, which can be stored without theoretical limitations. Examples are pump storages, where water is pumped to a higher level, or flywheels, where kinetic energy can be stored.

In these systems the converter is producing waste heat, which has to be released to the ambient connected to an entropy flow caused by the irreversibilities within the converter. The discharging process will be a heat pump process, where the entropy has to be taken from the ambient. Therefore it is obvious that these systems have to be coupled to the ambient conditions. Such a storage is not self-sufficient. These systems are called "indirect" thermal energy storages.

Another possibility to reach high storage capacities is the utilization of reversible chemical reactions. An ideal reaction scheme is a reversible dissociation of a solid or liquid compound AB to a solid or liquid component A and a gaseous component B.

$$\mathrm{AB} \leftrightarrow \mathrm{A} + \mathrm{B_g}$$

The component B should preferably be gaseous for various reasons:

- It is from the process engineering point of view much easier to separate a gaseous component from a condensed. This is necessary to prevent the reverse reaction and to provide a thermal energy storage without degradation.
- The evaporation of B is causing a significant increase in the reaction entropy. As we stated before is this necessary to reach a high energy density within the storage.
- A high reaction entropy increase influences the reaction temperature of the thermochemical dissociation equilibrium. Assuming that the reaction enthalpy and the reaction entropy have no significant temperature dependence, this simplified equation can be derived.

$$T_R = \frac{\Delta H}{\Delta S} \tag{4}$$

where T_R is the reaction temperature, ΔH the reaction enthalpy and ΔS the reaction entropy. That means, a high entropy change is keeping the reaction temperature low.

The energy density of the storage is defined as $E_V = (V_A + V_B)$. If $V_A \ll V_B$ (because B is a gas) it can be simplified to $E_V = \Delta H/V_B$. The reaction enthalpy ΔH cannot be influenced, but the volume of B can be reduced by different processes:

- The gaseous component B can be condensed. Such a system is shown in Figure 232.
- The component B can be stored by a chemical condensation at a lower temperature. A storage medium like MgH_2 (at 400–500 °C is an example of such a system.
- The gas B is part of the atmosphere, like water vapor. In this case it can be stored in the ambience and its volume is not taken into account concerning the energy density. Such systems are called "open" systems (see Figure 233).

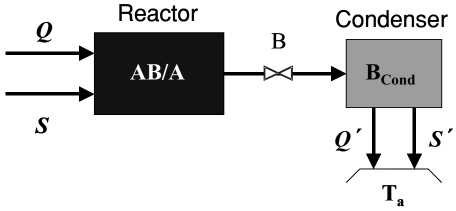

Figure 232. Closed system with condensation of the gaseous component B

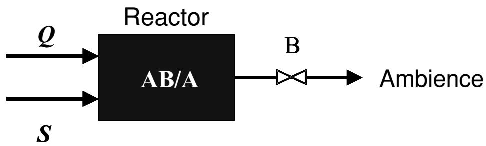

Figure 233. Open system releasing the gaseous component B into the ambience

A sorption process on the surface of a porous material, like Zeolite and other solid adsorbents, or within a concentrated salt solution, like LiCl and others, are examples for such chemical reactions for thermal energy storage.

24.3. Sorption Storage Systems for TES

24.3.1. CLOSED SORPTION STORAGE SYSTEMS

A closed sorption system is shown in Figure 8. It is based on the same physical effect as the open storage. However the engineering is quiet different from open sorption systems. Closed system could be more precisely described as evacuated or air-free systems. The operation pressure of the fluid to be sorbed can be adjusted in theses systems. In closed systems components, which are not existing in the atmosphere, can be used, because there is no connection to the ambience.

Figure 234 is showing a closed sorption system using water vapor as adsorptive. The heat has to be transferred to and from the adsorbent by an heat exchanger. This holds also for the condenser/evaporator. Heat has to be transported to the adsorber and at the same time the heat of condensation has to be distracted from the condenser in order to keep up the water vapor flow

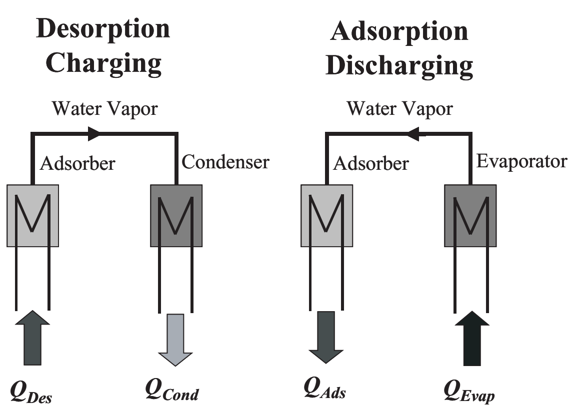

Figure 234. Closed sorption system

from the adsorber to the condenser during the desorption. During adsorption the heat of adsorption has to be taken from the adsorber and the heat of evaporation has to be delivered to the evaporator. Is this not possible, the sorption process will reach thermodynamic equilibrium and the flow of water vapor comes to a stop.

The main problem in the system design is the heat and vapor transport in and out of the adsorbent. Advanced heat exchanger technologies have to be implemented in order to keep up the high energy density in the storage, which would be reduced by the amount of "inactive" heat exchanger material.

Thermal energy storage can be realized by closing the valve between adsorber and condenser/evaporator after the desorption. The energy density expected is reduced compared to open sorption storages due to the fact, that the adsorptive (water vapor in this case) is part of the storage system and has to be stored as well. In the case of Zeolite or Silicagel as adsorbent this is about 30% to 40% of the weight of the storage material.

Closed systems are able to reach higher output temperatures for heating applications compared to open systems. Furthermore they can supply lower temperatures for cooling, e.g. it is possible to produce ice in the evaporator.

24.3.2. OPEN SORPTION STORAGE SYSTEMS

In an open sorption storage system air is transporting water vapor and heat in and out of the packed bed of solid adsorbents (see Figure 235) or a reactor where the air is in contact with a liquid desiccant. In desorption mode a hot air stream enters the packed bed or the reactor, desorbs the water from the adsorbent or the salt solution and exits the bed cooler and saturated. In adsorption mode the previously humidified, cool air enters the desorbed packed bed or the

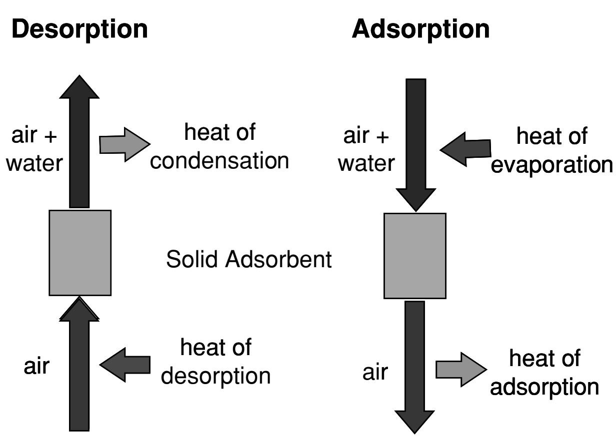

Figure 235. Open sorption storage system using a packed bed of solid adsorbent

concentrated solution. The adsorbent or the solution (or absorbent) adsorbs (or absorbs) the water vapor and releases the heat of sorption. The air exits warm and dry. In case of a solid adsorption it can be very hot. In case of a liquid absorption the dehumidification of the air is the main purpose.

The desorption energy Q_{Des} is the energy input to the thermochemical storage system, whereas the heat of adsorption energy Q_{Ads} can be used for heating. The heat of condensation Q_{Cond} can be additionally used, if it is available on a usable temperature level, which is depending on the inlet air conditions. The energy for evaporation Q_{Evap} has to be available at a low temperature level, which cannot be used otherwise (right scheme of Figure 235). The desiccant cooling process is based on the dehumidification of the air during the adsorption mode only.

Thermal energy storage is achieved by separating the desorption step (charging mode) from the adsorption step (discharging mode). After desorption the adsorbent can theoretically stay in this desorbed state, being referred to as charged in the following, without any thermal losses until the adsorption or absorptions process is activated.

24.4. Theory of Adsorption for TES

The theory of sorption storage systems will be explained for solid adsorbents. The basic results can be transferred to liquid absorbents.

To run adsorption storage systems efficiently the appropriate adsorbent has to be used. The right choice is possible on the basis of the measured adsorption equilibrium. The adsorption equilibrium of water vapor and different adsorbents (zeolites and silica gels) was experimentally found [3,4]. The differential heat of adsorption (ΔH_d) was calculated from the equilibrium data.

For the characterization of solid sorbents in thermal applications like heating, cooling and thermal energy storage the following criteria are defined: The possible temperature lift (and drop in humidity ratio), the breakthrough curves (responsible for the dynamics of the process), the thermal coefficient of performance and, for the thermal energy storage application, the energy density referring to the volume of the adsorbent. All these criteria can be calculated from the adsorption equilibrium as properties of the adsorbent.

24.4.1. ADSORBENTS

The most common classes of solid adsorbents are Zeolites and Silicagels. The main difference between the two is the way they are built. Zeolites have a crystalline structure and therefore a certain pore size. Silicagel have a pore

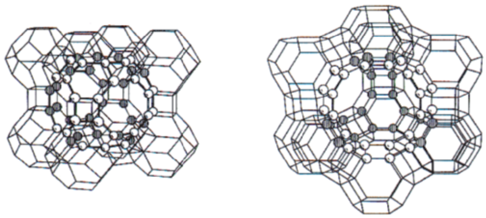

Figure 236. Crystalline structure of different Zeolite types

size distribution. Figure 236 shows two Zeolite types A and X and there crystalline structure. The pore size diameter of the A type is about 4 Å, while the diameter of the X type is about 10 Å. The chemical composition of the Zeolite types is given in Table 29. Some data describing Zeolites of type A and X are given in Table 30.

Figure 237 shows the pore size distribution of narrow pore (A) and wide pore Silicagel (B).

Silicagel is built of 99% SiO_2, while the rest are OH groups together with changing amounts of integrated water. The properties of Silicagel are shown in Table 31.

Concerning the application of these adsorbents as thermal energy storages the amount of water, which can be adsorbed is the most important property. Figure 238 shows the maximum water uptake of some commercially available adsorbents. Zeolite A can reach 25% and Zeolite 13X up to 32% of its dry weight. Narrow pore Silicagel can adsorb 38% water. Two special adsorbents Sizeo, which is a mixture of Zeolite and Silicagel, and SWS, which is a wide

TABLE 29. Chemical composition of Zeolite

Zeolite	Composition	Pore diameter	SiO_2/Al_2O_3
Type A	$Na_{12}[(AlO_2)_{12}(SiO2)_{12}] \cdot 27H_2O$	4.1Å	2.0–2.5
Type X	$Na_{86}[(AlO_2)_{86}(SiO_2)_{106}] \cdot 264H_2O$	7.4Å	2.0–3.0

TABLE 30. Properties of zeolites

Property	Type A	Type X
Inner surface (m^2/g)	800–1,000	800–1,000
Specific heat (kJ/kg K)	0.8–0.9	0.8–0.9
Heat conductivity (W/m K)	0.58	0.58
Packed bed density (kg/m^3)	750	700

TABLE 31. Properties of Silicagel

Property	Wide	Narrow
Inner surface (m^2/g)	300–500	600–800
Pore diameter (Å)	25–50	10–15
Specific heat (kJ/kg K)	0.92–1.0	0.92–1.0
Heat conductivity (W/m K)	0.14–0.2	0.14–0.2
Packed bed density (kg/m^3)	450	700

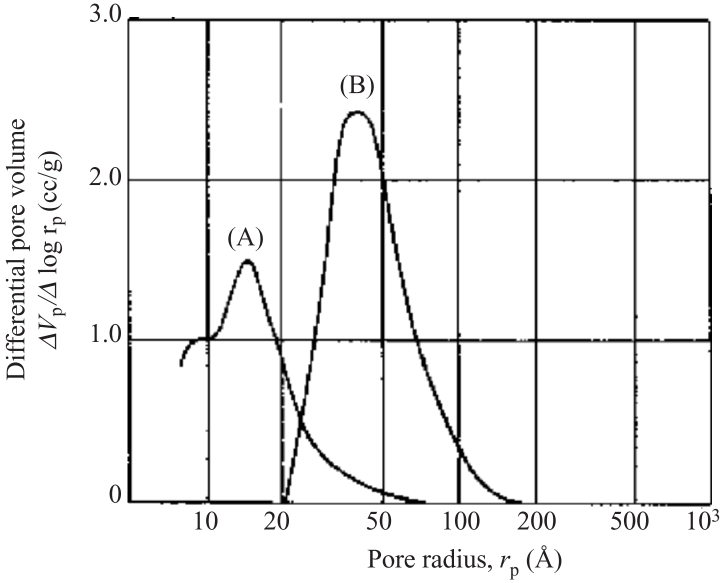

Figure 237. Pore size distribution of Silicagels

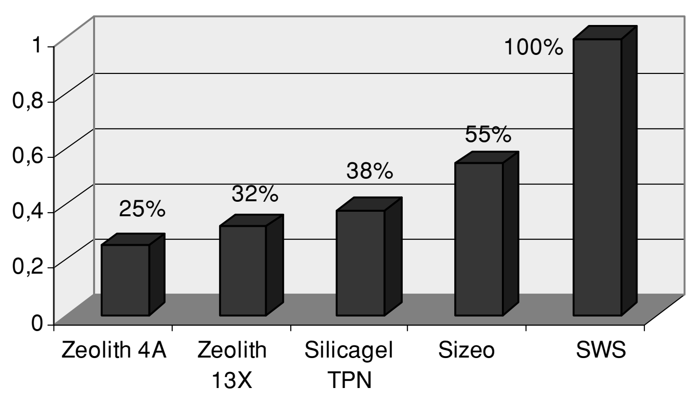

Figure 238. Water uptake of different adsorbents

pore Silicagel filled with the liquid absorbent Lithium Chloride, can reach much higher values.

24.4.2. TEMPERATURE LIFT

The temperature lift is defined as $\Delta T = T_{air\ out} - T_{air\ in}$. The possible ΔT is crucial for the design of sorption systems for heating applications. The temperature lift of each adsorbent can be very different under the same adsorption conditions. The temperature lift can be calculated as

$$\Delta T = \Delta x \cdot \frac{\Delta H_{ads}}{c_{p\,\mathrm{air}} - \frac{\Delta x}{\Delta C} \cdot c_{sorb\,eff}} \tag{5}$$

where $\Delta x = x_{in}\ {}^{-}x_{out}$ is the humidity ratio difference, ΔH_{ads} is the integrated differential heat of adsorption ΔH_d between C_{ads} and C_{des}, $c_{p\ air}$ is the heat capacity of the air, $\Delta C = C_{ads} - C_{des}$ is the difference in water concentration of the adsorbent and $c_{sorb\ eff} = c_{sorb} + (C_{des} \cdot C_{\mathrm{H_2O}})$ is the effective heat capacity of the adsorbent.

Experiments showed an almost linear relation between T_{out} and x_{out} for the variation of desorption conditions at fixed adsorption conditions. Zeolite 13X and silica gel were used. Figures 239 and 240 show the experimental data. In the background the isosteres of each adsorbent are drawn. Starting from high x_{out} values (and low T_{out} values) going to high T_{out} values, each point represents a desorption with a higher temperature and a lower equilibrium water concentration of the adsorbent. The upper test sequence was found with adsorption using 20 °C saturated inlet air. The lower points were measured at

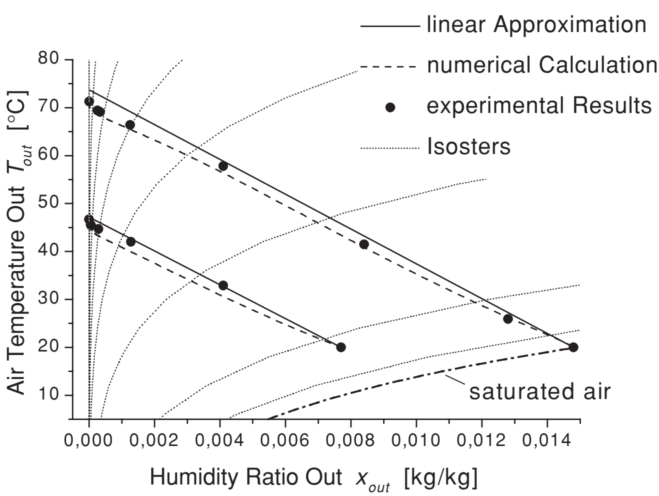

Figure 239. T_{out}/x_{out}-Diagram for Zeolite 13X

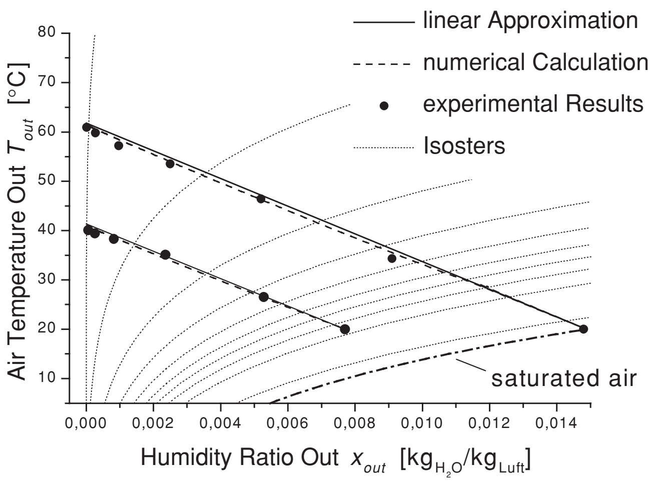

Figure 240. T_{out}/x_{out}-Diagram for Silicagel

a adsorption with $T_{in} = 20$ °C and $x_{in} = 7.72$ g/kg. Both curves are parallel and almost linear for each adsorbent

The linear approximation in Figures 239 and 240 was found by assuming, that no change in the incoming air conditions T_{in} and x_{in} will occur, if the adsorbent has reached its equilibrium at T_{in} and x_{in}. This case is represented by the dot at $T_{in} = 20$ °C and $x_{in} = 14.88$ g/kg, $x_{in} = 7.72$ g/kg respectively, in Figures 228 and 229.

For the second point of the linear approximation the maximum ΔT has to be found. Assuming that the highest temperature $T_{out\ \max}$ can be reached after a complete desorption and that all of the water vapor within the air stream will be adsorbed ($\Delta x = x_{in}$), $T_{\max}$ can be written as

$$\Delta T_{\max} = x_{in} \frac{\Delta H_{ads}(C_{\max})}{c_{p\,air} - \frac{x_{in}}{C_{\max}} \cdot c_{\text{Adsorbent}}}. \tag{6}$$

This point can be easily calculated from the adsorption equilibrium of each adsorbent. As shown in Figures 2 and 3 the linear approximation gives a sufficient accurate estimates (±5% of the experimental values) within a range of realistic conditions to predict T_{out} and x_{out} under given de- and adsorption conditions for each adsorbent [4].

24.4.3. BREAKTHROUGH CURVE

The time dependent changes in the properties of the outlet air of an adsorber is called the breakthrough curve. In most applications, like gas drying, it is referring only to the changes in the water content. For thermal applications also the temperature change is important.

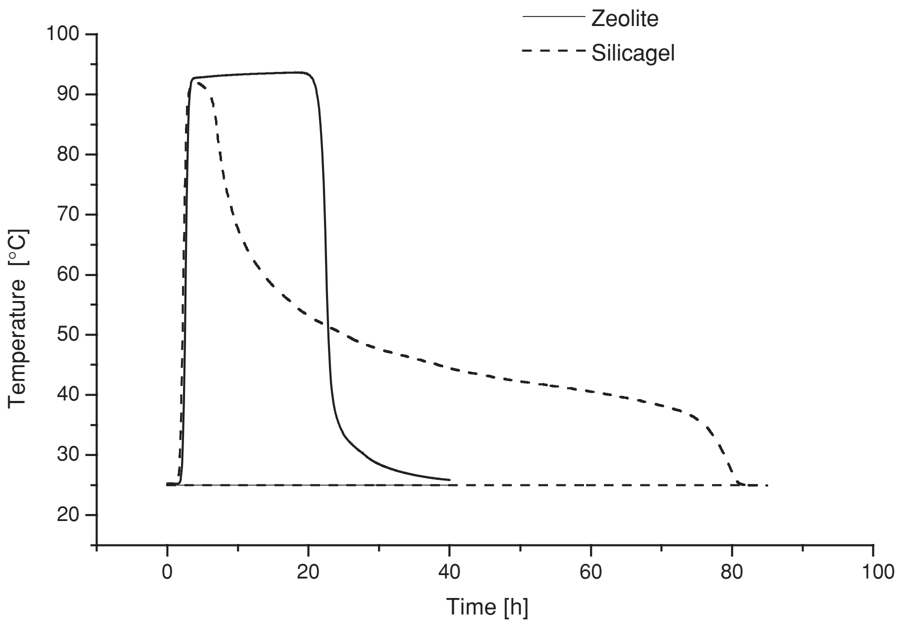

Figure 241. Thermal breakthrough curves (adsorption) for Zeolite and Silicagel

Figure 241 shows the shape of two thermal breakthrough curves for Zeolite and Silicagel in adsorption. The adsorption is following a desorption using 130 °C and the inlet air is saturated with water vapor at 25 °C.

Zeolite is reaching its maximum outlet temperature and is keeping that almost constant until the adsorption is over. Silicagel has a falling outlet temperature just after reaching the maximum. The adsorption using zeolite is about half long as the silicagel adsorption.

The shape of the breakthrough curve is depending on the behavior of the so called mass transfer zone (MTZ). Figure 242 shows schematically the MTZ within a packed bed of adsorbent. Within the MTZ the properties of the incoming air are changed to the outlet air properties.

The dimension of the MTZ within the packed bed can be constant, expanding or shrinking. The Zeolite curve is caused by a constant or slightly shrinking MTZ, whereas the Silicagel curve is caused by e expanding MTZ. With the expanding MTZ cooler and more humid air is reaching the end of the bed. This leads to the falling outlet temperature and a rising water content, which can be observed (Silicagel curve in Figure 242).

24.4.4. THERMAL COEFFICIENT OF PERFORMANCE AND ENERGY DENSITY

The thermal COP can be defined according to Figure 235 (neglecting Q_{evap}) as $\mathrm{COP}_{th} = (Q_{cond} + Q_{ads})/Q_{des}$. The energies are defined per mass of ad-

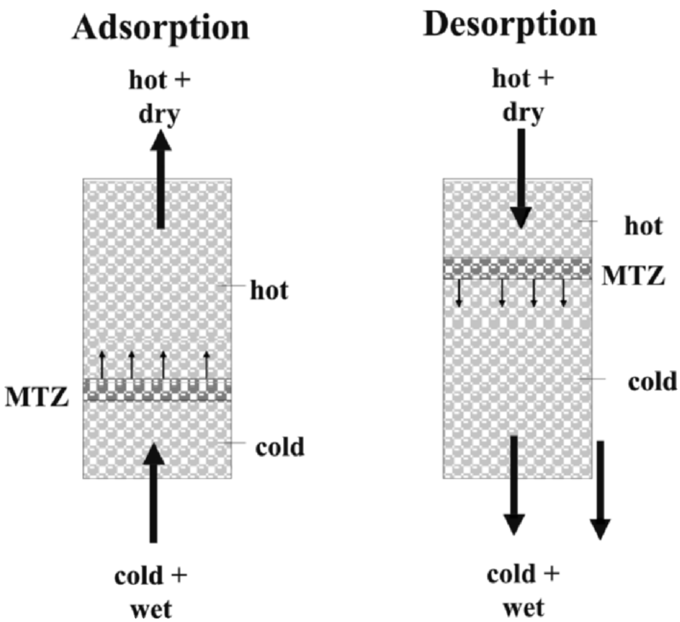

Figure 242. Mass transfer zone for ad- and desorption

sorbent. They can be calculated from the adsorption equilibrium in order to find the theoretical COP$_{th}$ of the system. The energy for desorption Q_{des} can be divided into three different parts:

$$Q_{des} = Q_{cond} + Q_{bind} + Q_{sens} \tag{7}$$

The sensible heat Q_{sens} has to be brought into the system to heat up the packed bed of adsorbent pellets to T_{in}. Q_{sens} is defined as $Q_{sens} = \Delta T_{sorb}\, c_{sorb\ \ eff}$. The condensation energy Q_{cond} and the binding energy (caused by the adsorption forces) Q_{bind} is defined according to Figure 243. Q_{cond} and Q_{bind} only depend on the differential heat of adsorption and the water concentration C of the adsorbent at the end of the de- and adsorption process:

$$\begin{aligned} Q_{cond} &= (C_{ads} - C_{des}) \cdot L(T) \\ Q_{bind} &= \int_{C_{des}}^{C_{ads}} (\Delta F + T\Delta S) \cdot dC \end{aligned} \tag{8}$$

where $L(T)$ is the heat of evaporation for water vapor, $(\Delta F + T\Delta S)$ is the heat of binding taken from Dubinins theory of volume filling for vapor adsorption [5], which can be determined from the adsorption equilibrium.

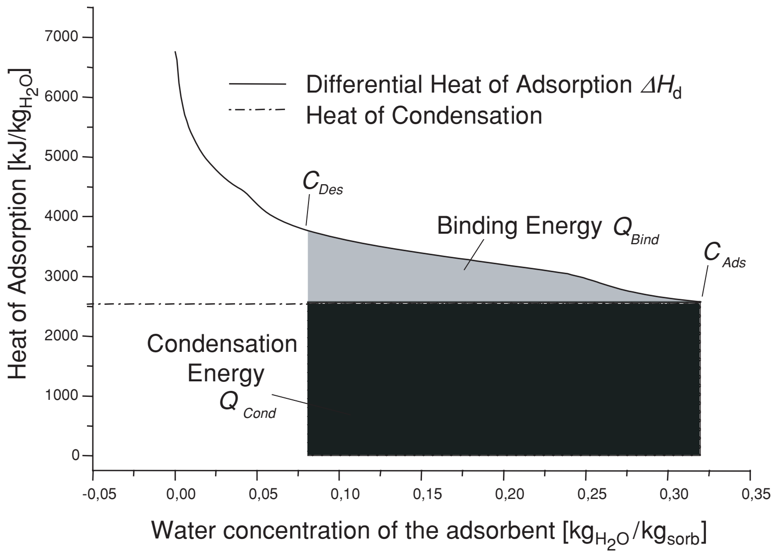

Figure 243. Definition of Q_{cond} and Q_{bind} for Zeolite as an example

Q_{ads} depends on the actual application. For the heat pump is $Q_{ads} = Q_{des}$. For long-term TES, Q_{sens} cannot be used due to thermal losses. For a desiccant cooling system only Q_{cond} can be used during adsorption.

The thermal COP$_{th}$ can be calculated directly from the adsorbents equilibrium data, the differential heat of adsorption respectively. Losses in the usable energies Q_{cond} (during desorption) and Q_{ads} (during adsorption, see Figure 235) due to the dynamic profile of the outlet temperature $T_{out}(t)$ have to be taken into account for a final characterization of the investigated adsorbents. Only energy above (or below) a temperature limit, given by the heating (or cooling) system of the building, can be utilized. This fact can lead to drastic reductions in COP$_{th}$ especially for adsorbents with low energy of binding Q_{bind} like silicagel [4].

The energy density ρ_Q is defined as

$$\rho_Q = \frac{(Q_{cond} + Q_{bind}) \cdot m_{sorb}}{V_{sorb}} = (Q_{cond} + Q_{bind}) \cdot \rho_{sorb} \tag{9}$$

where m_{sorb} is the mass, V_{sorb} is the volume and ρ_{sorb} is the density of the adsorbent. ρ_Q can be determined according to the COP$_{th}$ for an experimentally found density of the adsorbent ρ_{sorb}.

24.4.5. CONCLUSIONS

Using the methods presented in this chapter, adsorbents for energy applications can be evaluated on the basis of the adsorption equilibrium. The possible temperature lift and the dehumidification of the air, the dynamic behavior of the air properties, the thermal COP and the energy density of a TES can be calculated in advance. Boundary conditions of the actual application, e.g. temperature of the heat source or usable temperature level of the buildings heating system can be directly included in the calculation and influence the choice of the appropriate adsorbent [4].

The methods described in this chapter can be transferred to liquid sorption processes with slight modifications. The example of concentrated salt solutions and water absorption is described by Kessling [6].

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25. ADSORPTION SYSTEMS FOR TES—DESIGN AND DEMONSTRATION PROJECTS

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Abstract. Adsorption systems for thermal energy storage can be designed as closed or open systems. The two possibilities are described in chapter V.2. In this chapter some examples of complete systems will be given. There will be two examples for closed systems. One is a commercially available self cooling beer keg (ZeoTech Zeolite Technology, http://www.zeo-tech.de) and the other one is a seasonal storage for solar heat. For open systems one adsorption storage is described, which is installed in the district heating net and is able to provide heat for heating purposes in winter and air conditioning in summer.

Keywords: Adsorption, thermal energy storage, Silicagel, Zeolite

25.1. Closed Adsorption Storage Systems

25.1.1. SELF-COOLING BEER KEG

The self cooling beer keg is based on the principle of closed sorption systems shown in chapter V.2 Figure 8. Figure 244 shows the beer keg from the outside and the inside.

During the adsorption—the discharging—cold is produced in the evaporator. In the beer keg the evaporator is located in the lower part of the keg and is in contact with the beer. As soon as the valve is opened the water in the evaporator will evaporate and starts getting cold. The heat of evaporation will be extracted from the beer and by this process the beer will be cooled. Figure 2 shows the falling temperature of the beer down to 8 °C. The process can be stopped and restarted by closing and opening the valve.

At the same time the adsorber will become hot by the released heat of adsorption. The adsorber is located at the outside of the keg. The heat of adsorption will be released through the outer surface of the keg. Figure 245 shows the temperatures in the adsorber. They can reach more than 80 °C.

Looking at the beer keg sorption system as an indirect heat storage, the system would look like Figure 246. The first part of Figure 3 shows the charging

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 409–427.

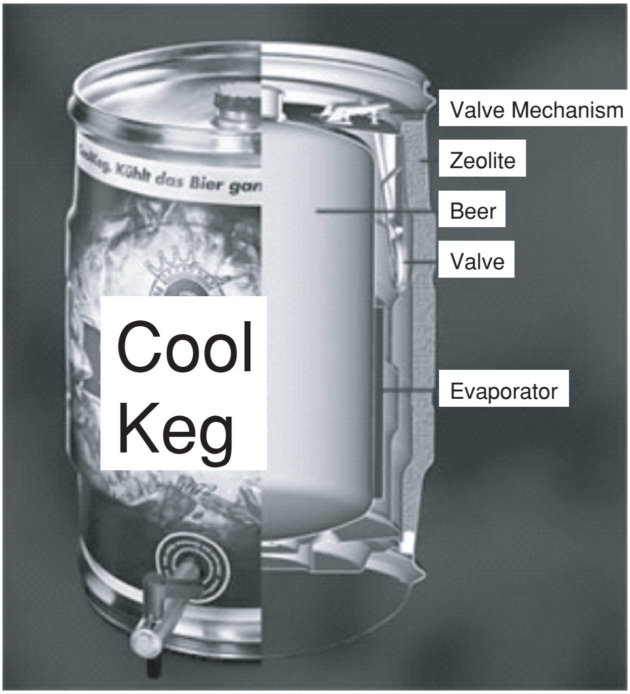

Figure 244. Outside and inside the beer keg

process. For this purpose a big oven was built at the brewery. Discharged kegs refilled with beer are slowly moved through an oven with a temperature of around 500 °C. The Zeolite on the outside adsorber will be desorbed and the evaporated water vapour will be condensed at the condenser/evaporator inside the keg. The beer, in good contact with the condenser/evaporator, is providing the cold side, which is important for an efficient charging process. The heat of condensation will be distracted by the beer until the beer reaches a temperature of about 40 °C. At that point the Zeolite is almost completely desorbed, the water vapour is condensed in the condenser/evaporator and the charging process is over.

The lower part of Figure 246 shows the discharging process, where the heat for evaporation is taken from the beer and the heat of adsorption is released through the surface of the keg.

25.1.2. SEASONAL STORAGE OF SOLAR ENERGY

Seasonal storage with sorption storage systems is strongly influenced by the changes in ambient temperature between winter and summer. A reduction of the thermal COP_{th} (beside the reduction due to the irreversibilities of the converter at charging and discharging) will be demonstrated in the following

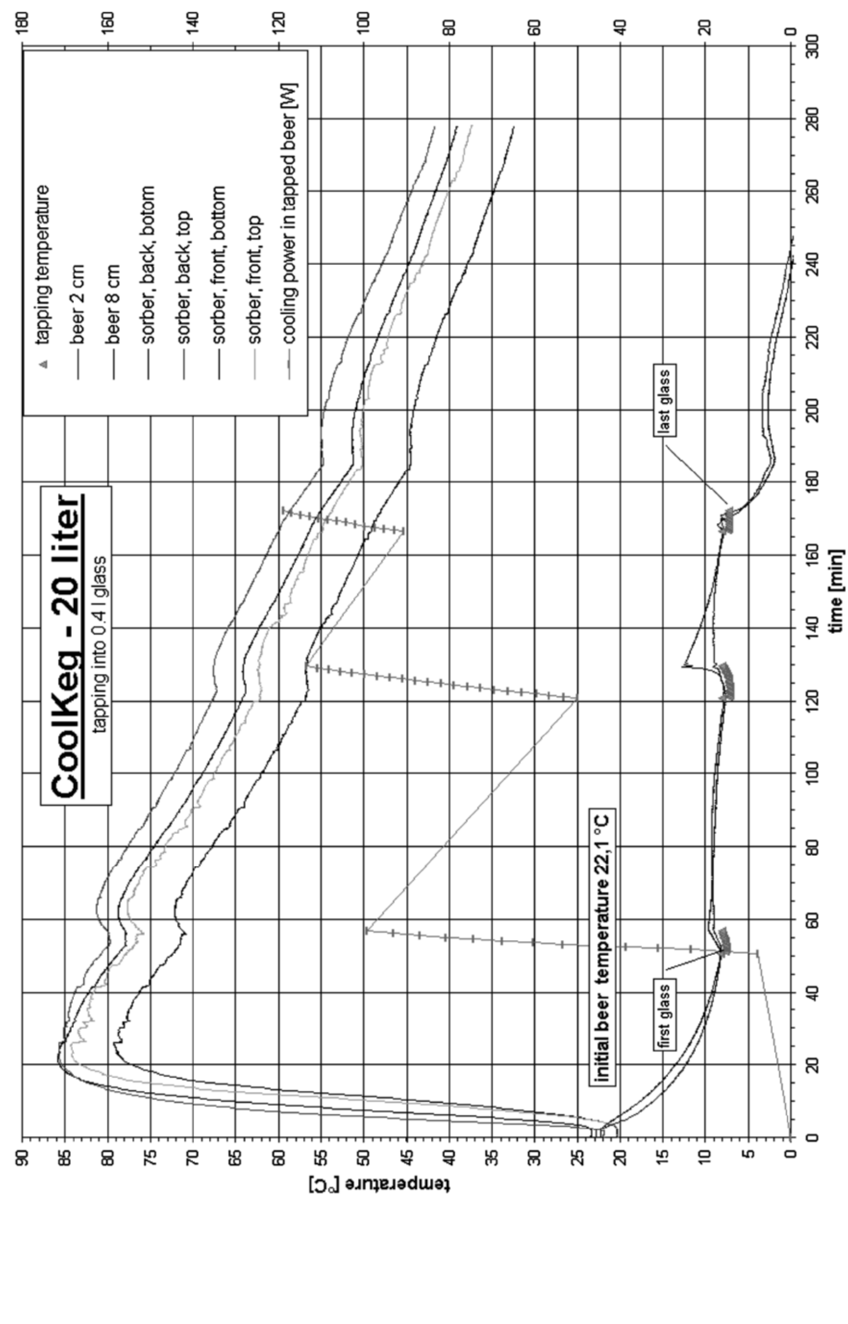

Figure 245. Temperature during the adsorption/discharging of the self cooling beer keg

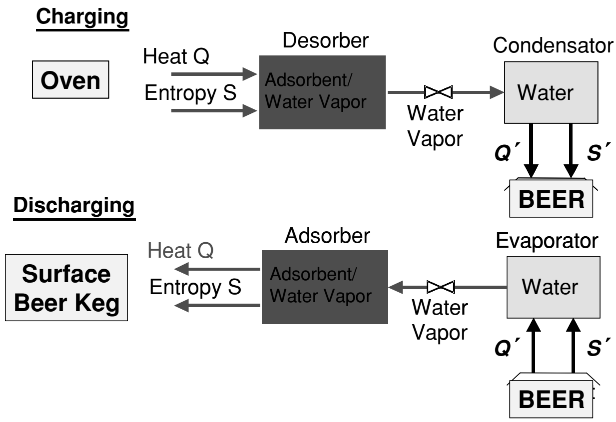

Figure 246. Self-cooling beer keg as an indirect heat storage

example: The charging process takes place in the summer time at ambient temperature $T_{AC} = 30\ ^\circ\mathrm{C}$, while the discharging is happening in winter at $T_{DC} = -20\ ^\circ\mathrm{C}$. This leads to a reduction of the ideal ratio between the discharged amount of heat Q_D and the heat charged to the storage Q_C:

$$\frac{Q_D}{Q_C} = \frac{\left(1 - \dfrac{T_{AC}}{T_C}\right)}{\left(1 - \dfrac{T_{AD}}{T_D}\right)} = 0.6. \tag{1}$$

In this example [2] the charging T_C and the discharging temperature T_D are both chosen to be 100 °C. In most applications $T_C > T_D$, which leads to a further decrease in the COP_{th}. The system is schematically shown in Figure 247.

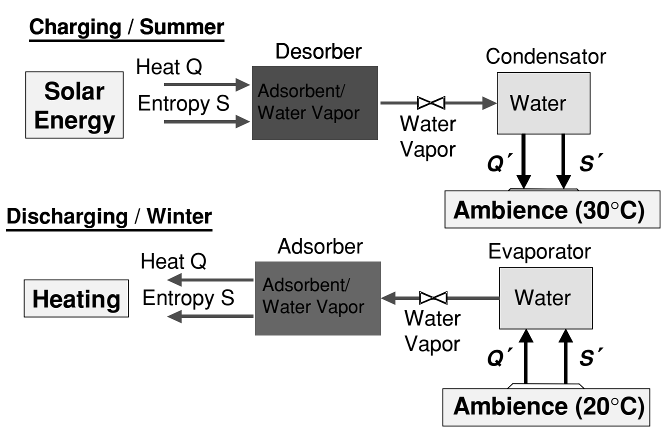

Figure 247. Seasonal storage of solar energy as an indirect heat storage

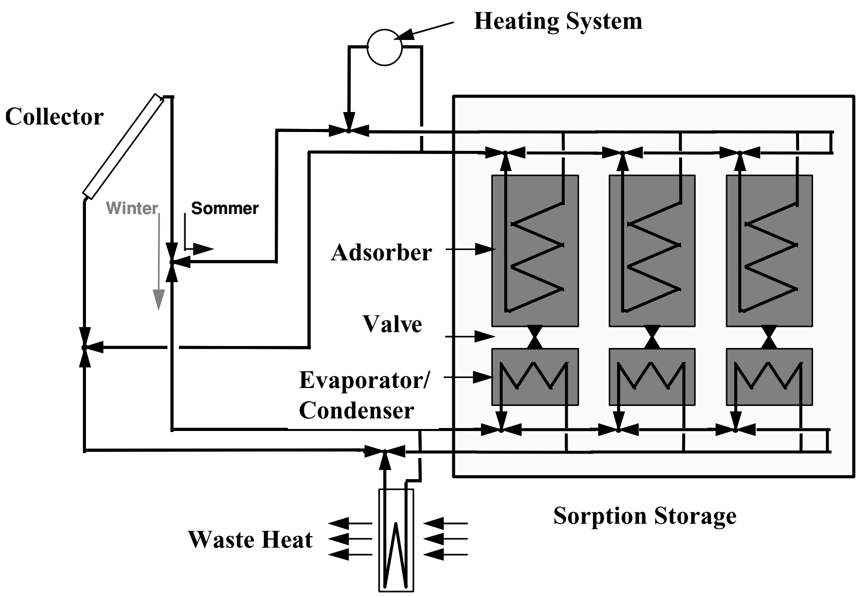

Figure 248. Closed sorption storage system for seasonal storage

Nevertheless long-term heat storage is one of the main challenges for an effective year round use of thermal solar energy. Therefore, high energy density heat stores are the focus of an increasing amount of research efforts.

In the period from 1998 to 2001 the European Union funded the Project "*High Energy Density Sorption Heat Storage for Solar Space Heating*" (HYDES). The major objectives of the project HYDES were the development of a high energy density heat storage system based on closed cycle adsorption processes suitable for the long-term storage of low-temperature heat and the testing of this system in the application of seasonal storage of solar thermal energy for space heating purposes under different climatic and system conditions [3].

The complete system is shown in Figure 248. In summer, during the charging of the storage, heat from the solar collectors is delivered to the three adsorbers. The heat input has to be realized by heat exchangers, which are located within a packed bed of Silicagel. The desorbed water vapour is condensed at the evaporators/condensers and the heat of condensation is transferred as waste heat to a cooling fan. During the discharging in Winter low temperature heat from the solar collectors is used for the evaporation of the water in the evaporators/condensers. The heat of adsorption is collected by the inner heat exchangers within the adsorbers and is delivered to the heating system of the building. In this constellation the heat needed for evaporation during the

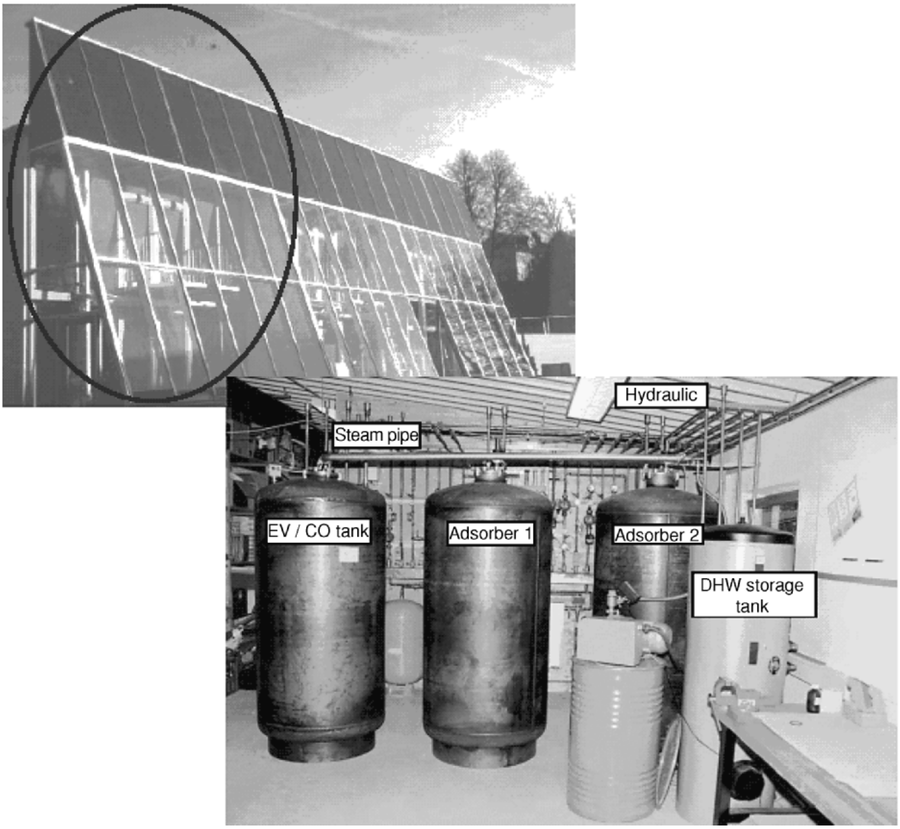

Figure 249. Pilot plant in Gleisdrof/Austria

discharging is not coming from the ambience (see Figure 247), because the temperature level would be too low. It is provided by the solar collectors at a higher temperature.

In order to study the performance of the sorption heat storage in different climatic conditions two prototype systems were planed and constructed. One test plant was installed in Austria (Figure 249). This system is operated by solar collectors and provides heating and domestic hot water production for a low energy house next to the office of the Austrian partner AEE. Due to the expected load profile of this long-term heat storage application a prototype system consisting of two adsorbers and one evaporator/condenser was build. A thermal solar plant with an aperture area of 20.4 m^2 is available for the test plant as the primary source of energy. The solar plant faces to the South with a 10° deviation towards the West. The collectors have an inclination of 72° and are optimized for their use in the autumn and winter.

For desorption the vapor desorbed from the silica gel has to be condensed. For this reason a low temperature heat sink is required. The hydraulics of the plant provides two heat sinks: a 10 m^3 rain water cistern and the thermal solar plant. With these heat sinks two desorption modes could be carried out: a desorption with simultaneous condensation of the vapor an a second mode in which the desorption and condensation were not done simultaneously. If desorption and condensation of the vapour occur at the same time, then the condensation heat is rejected via the rain water cistern. The condensation heat can also be removed by the solar plant when the desorption and condensation operation are discontinuous. In this case the solar plant heats up the adsorber during daytime but no condensation is done. The condensation take place through the solar system during the night given correspondingly low outside temperatures.

To achieve a high rate of utilization when operating the sorption storage tank, a heating system which operates with low temperatures is essential. This can be a wall or floor heating system or a combination of both. The floor and wall heating in the test apartment is a low temperature heat delivery system, with design temperatures of 40/30 °C at –12 °C ambient temperature and a maximum heating power of 2.2 kW. In addition the heating system of the laboratory room is connected to the hydraulic network of the test plant so that the radiator heating, designed for 45 °C/35 °C, can be used as alternative heat sink for the energy from the adsorber tanks.

Due to delays in manufacturing the adsorber heat exchanger containers the test period had to be shortened. The installation of the plant and start of the measurement was in May 2001, therefore a coupling to the system with room heating was not possible. In the first test runs the heat exchanger characteristics and the operational capability of the plant hydraulics were examined and initial adsorption and desorption tests were carried out. It could be shown that the test plant in general and the adsorption heat storage system in particular is a technically feasible solution for long-term storage. Due to the short duration of the tests, the adsorbers were not charged in the best possible way. The experimental results up to now show that the achieved energy density is about 20% below the value expected from simulation (150 kWh per m^3 of silica gel). The planned monitoring over the heating period could not be carried out within the time schedule of HYDES.

Seasonal heat storage leads in general to severe problems concerning the economics of storage system. If it is not a very inexpensive technology, like the storage of sensible heat in a large water tank, one of the most important factors is the number of charging/discharging cycles. A large number will reduce the actual cost for the stored thermal energy.

Recently a follow-up project entitled “Modular High Energy Density Sorption Heat Storage” (MODESTORE) was approved by the European

Commission. The work in this project will start in April 2003 and will continue for three years. One of the objectives is the development of a gas fired adsorption heat pump coupled to an underground storage system. That means the size of the sorption storage system is reduced significantly, a large number of cycles is possible and the actual seasonal storage is now realized by the UTES system. The economics of this concept are expected to be much better.

25.2. Open Adsorption Storage Systems

25.2.1. TES FOR HEATING AND COOLING IN A DISTRICT HEATING SYSTEM

25.2.1.1. *Introduction*

An open adsorption storage was installed in Munich/Germany connected to the local district heating net [4]. To use this thermochemical storage system for heating in winter and air conditioning in summer leads to an increase in operation time. This can provide substantial economic advantages. A thermochemical storage using Zeolite as adsorbent has been installed in order to heat a school building in winter and to cool a jazz club in summer time. The school building and the jazz club are connected to the district heating system of Munich.

About one-third of the total energy consumption of Germany is used for space heating. The use of district heating systems with cogeneration is a possibility to save primary energy and reduce pollution. Local district heating systems are frequently operated at the upper limit of their capacity. Therefore new technologies are required to connect additional consumers without expanding the system. Decentralized thermal storage devices installed close to the consumer can shift the on-peak demand into off-peak periods. The power demand of the district heating system can be averaged and the number of consumers may be increased. The Zeolite system is designed to shift the peak demand for one day of the school building from day to night.

For the heating of buildings the heat of adsorption can be used in the adsorption mode. Depending on the used adsorbent and the desorption and adsorption conditions temperatures up to 160 °C can be reached. Under certain desorption conditions thermal energy can be delivered to the buildings heating system in the charging mode as well.

Looking at open sorption system as an indirect heat storage, the system would look like Figure 250. The charging process is driven by the district heat input. While the district heat return flow, the condensate, is providing the low temperature heat input to the humidifier for the evaporation.

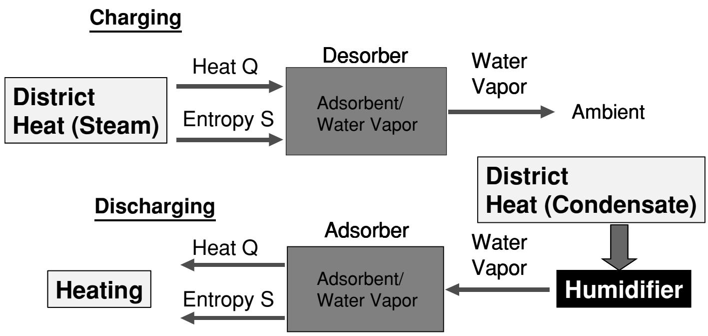

Figure 250. Open sorption storage for heating connected to a district heating system

The need for air conditioning is increasing remarkably over the last years. Reasons for that are architectural motives (like the use of large glass elements, a rising internal load caused by technical equipment and last but not least higher comfort standards. Cooling load consists of two part: The latent and the sensible load. The latent cooling load can be matched by dehumidification of the buildings supply air. Sensible cooling can be achieved by lowering the air temperature in the building. The cooling demand of the jazz club connected to the sorption storage system is caused by the large number of persons in the room. Therefore about ¾ of the cooling load is latent, which makes it an ideal application for an open adsorption system.

The application of open sorption systems can provide dehumidification by the adsorption of water vapor and sensible cooling by adiabatic humidification (after a cold recovery for the dried air) at temperatures between 16 °C and 18 °C. Conventional systems have to reach temperature as low as 6 °C or lower in order to start dehumidification by condensation. For comfort reasons this cold air has to be heated up to about 18 °C before released into the building. This shows that open sorption systems can provide in general an energetically preferable solution.

25.2.1.2. *Adsorption Process*

1. *Heating:* In order to reach a coefficient of performance COP_{heat} of about 1.0 or more it is important to utilize the heat of condensation Q_{cond} during the charging process.

   The thermochemical storage system can be charged off-peak. At on-peak times it uses only low temperature heat extracted from the return flow of the district heating system. By lowering the temperature of this return flow, the power transported is increased and heat losses of the net are reduced. In addition to that, thermochemical storage systems offer high energy storage densities without degradation due to heat losses in long-term storage.

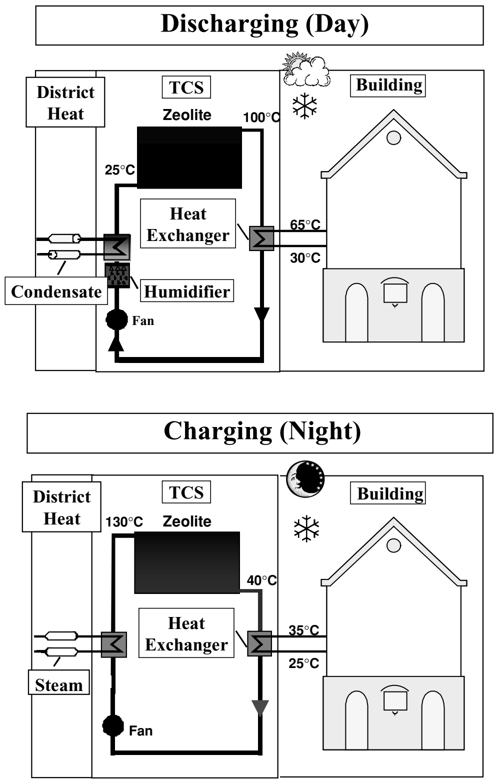

Figure 251. Charging and discharging mode for heating

Figure 251 shows heat fluxes during charging and discharging mode. At night Zeolite is charged by air, heated up to about 130–180 °C using the steam line of the district heating system (heat of desorption Q_{Des}). Under these conditions the final water content of the Zeolite reaches $0.09\ kg_{Water}/kg_{Zeolite} - 0.05\ kg_{Water}/kg_{Zeolite}$.

The waste heat of the charging process (temperature level 35–40 °C and above) and the heat of condensation Q_{Cond} is supplied to the heating system of the school.

The storage system is discharged in times of peak power demand. At first the air is heated up to 25–30 °C and saturated with water vapor by a humidifier. The energy for this process is provided by the low temperature return flow of the district heating system (heat of evaporation Q_{Evap},). The steam line of the district heating system is not used. The humidified air is blown into the tank containing desorbed Zeolite. The air temperature is raised to 100 °C (heat of adsorption Q_{Ads}). The thermal energy is transferred

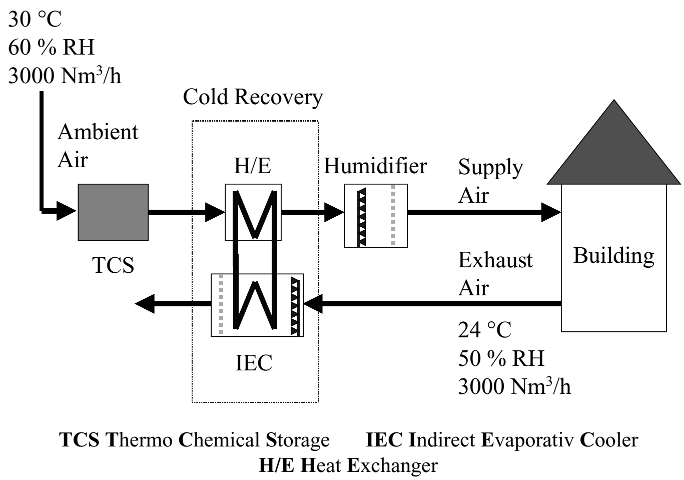

Figure 252. Additional components for air conditioning

to the heating system of the school (flow temperature 65 °C) by heat exchanger.

2. *Cooling:* In order to use the Zeolite storage system as a desiccant cooling system two additional components have to be integrated. Figure 252 shows the complete desiccant cooling system including the cold recovery device and the supply air humidifier. As shown in Figure 252, the air stream exiting the Zeolite bed has to be cooled down (cold recovery device) before entering the supply air humidifier.

   The cold recovery device consists of an exhaust air humidifier with an integrated heat exchanger and the supply air heat exchanger, which are connected by a fluid circuit. The first can be described as an indirect evaporative cooler. The cold recovery device is able to transport 83% of the maximum possible enthalpy difference from the exhaust air to the supply air.

25.2.1.3. *Adsorption System*

1. *Thermochemical storage:* The storage system is designed to cover a heat load of 95 kW over a period of 14 h (from 7 a.m. to 9 p.m.) each day. This was achieved by using 7,000 kg of Zeolite 13X for the storage tank. The tank consists of three connected cylinders arranged in a horizontal line (see background of Figure 253). The storage system is connected to a combined air/radiator/floor heating system. The result is a flow temperature of 65 °C and a return temperature not higher than 35 °C. The maximum thermal power is 130 kW, the storage capacity is 1,300–1,400 kWh at

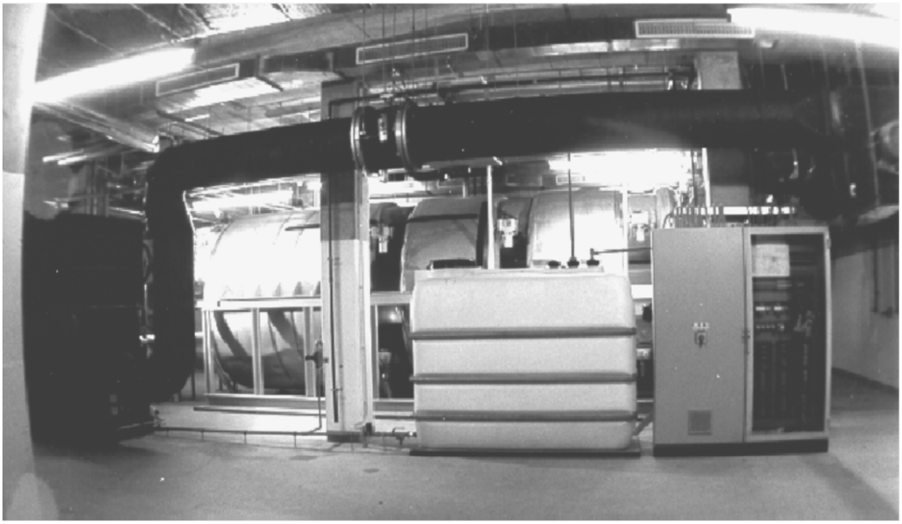

Figure 253. Thermochemical storage system (Humidifier, water tank and control unit in the front from left to right, three Zeolite modules in the back)

130 °C charging temperature.[4] Table 32 shows the relevant data of the thermochemical storage.

The additional components for the air-conditioning application are shown in Figure 254.

2. *School building:* The thermal energy storage was installed in a school building in Munich, Germany, by 1996. The school building is a former brewery in Munich, converted by the *Münchner Gesellschaft für Stadterneuerung (MGS)* and is connected to the district heating system (guaranteed steam temperature of 130–140 °C). The thermochemical energy storage system is used as a buffer between the district heating system and space heating system of the school. Table 33 shows the relevant data of the school building.
3. *Jazz club:* A jazz club with a reasonable cooling load is located close to the storage system. The club has a floor area of 160 m^2 and a volume of 800 m^3.

TABLE 32. Data of the thermochemical storage

Mass of zeolite	7,000 kg
Max. air flow	6,000 m^3/h
Max. heating power	130 kW
Max. cooling power	50 kW
Energy density (heating)	Up to 200 kWh/m^3
Energy density (cooling)	Up to 100 kWh/m^3

Figure 254. Heat exchanger and humidifier of the supply air and the indirect evaporative cooler of the exhaust air

The maximum capacity is about 200 persons. The room temperature should not exceed 26 °C at a relative humidity of 50%. The maximum cooling load was calculated assuming 4.5 kW for lighting and 1 kW for other technical equipment. The result is a maximum latent cooling load of 22 kW (73%) and sensible of 8 kW (27%).

25.2.2. RESULTS OF THE DEMONSTRATION PLANT HEATING

In the beginning of 1997 the automatic operation of the storage system started in the heating application. Since then the required room temperatures and heating power were covered by the storage system. Problems detected in the first operation period were removed. The control strategy during discharging mode was constantly improved.

The upper part of Figure 255 shows on a desorption /charging process of two storage modules (indicated above the diagram). During this process the heating system of the building is supplied at a temperature level of about 40 °C (see flow line temperature). As soon as this temperature starts rising the next module is charged. The thermal energy supplied to the heating system during desorption keeps the building from cooling down at night time. Therefore a heating demand peak in the morning can be avoided.

The discharging process, at the lower part of Figure 255, shows inlet and outlet air temperatures of the storage. Saturated air from the humidifier enters

TABLE 33. Data of the school building

Heated floor space	1625 m^2
Max. heat load (at −16 °C)	95 kW
Specific heat demand	65 kWh/m^2 a
Heating system	radiator/floor /air heating
Fresh air flow rate	30 m^3/h person

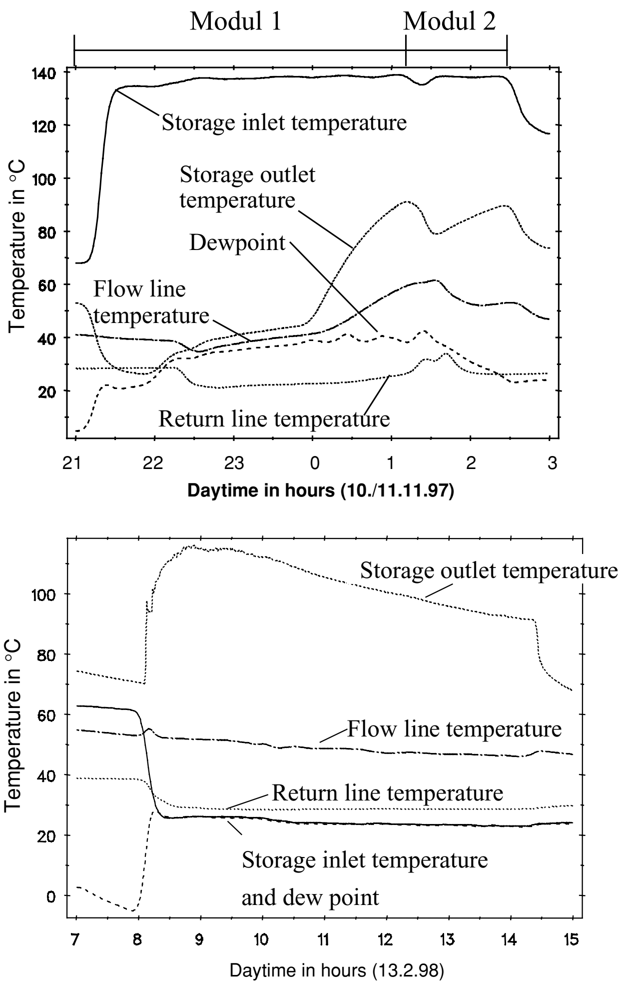

Figure 255. Experimental data of a desorption and an adsorption (air temperatures and dew points, flow and return line temperatures of the heating system)

the storage at 25 °C and exits at about 100 °C or more. The hot air transfers heating power to the heating system at about 50 °C. The discharging of the first module lasts for 6.5 h.

The thermal coefficient of performance COP_{heat} is defined as the ratio of thermal energy supplied to the building (heat of condensation Q_{cond} and adsorption energy Q_{ads}) and thermal energy input for charging the storage (desorption energy Q_{des}). The utilized thermal energy in the experiments is the actually transferred heat to the heating system including all losses. The experimentally achieved thermal energy density is the actually transferred

TABLE 34. Comparison of experimental and theoretical results of the thermochemical storage system for heating

Thermal coefficient of performance COP_{heat}	
Theoretical COP_{heat}	1.07
Experimental COP_{heat}	0.92
Energy density ρ_Q	
Theoretical ρ_Q	552 MJ/m^3 (153 kWh/m^3)
Experimental ρ_Q	446 MJ/m^3 (124 kWh/m^3)

thermal energy divided by the volume of the Zeolite. Table 34 compares the experimental results to the theoretically calculated values[4] for the given conditions.

Table 34 shows that 86% of the theoretical maximum COP_{heat} and 81% of the maximum thermal energy density were reached in the demonstration plant.

25.2.2.1. *Cooling*

Simulations of the storage have shown that desorption temperatures below the possible 130 °C from the district heat can lead to higher COP_{cool}. Experiments with 130 °C, 100 °C and 80 °C were carried out. Figure 256 shows the 100 °C desorption as an example.

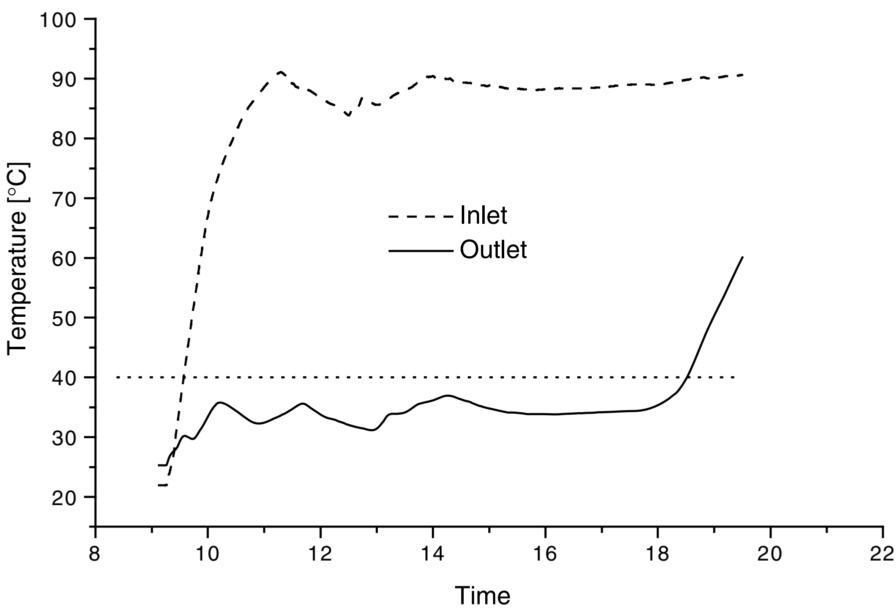

Figure 256. Desorption at 100°C for air conditioning

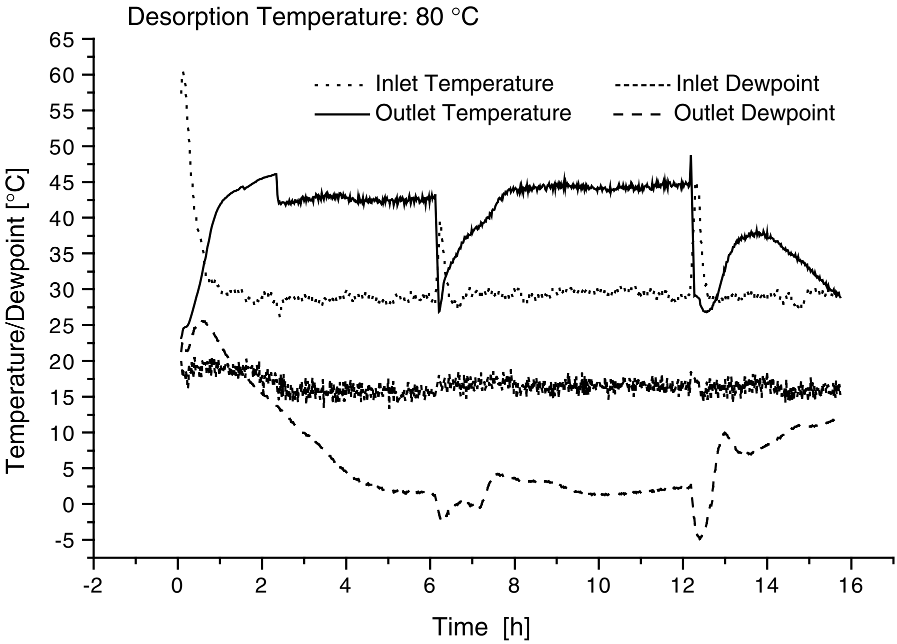

Figure 257. Adsorption following an 80 °C desorption

The inlet air temperature shown is measured right on top of the Zeolite bed. It has already cooled down to about 90 °C between the district heat exchanger and the Zeolite. The outlet air is almost saturated at about 35 °C for about 9.5 h. The desorption was stopped after the outlet air temperature has reached 60 °C. At the same time the humidity ratio of the outlet air has dropped and the equilibrium is almost reached. A continuation of the desorption is not efficient.

Figure 257 shows an adsorption following an 80 °C desorption. The adsorption was stopped after 6 h. This is a typical period for cooling demand in an office building in Germany. The adsorption was continued the next day. Figure 14 shows two and a half of these air-conditioning periods (in the picture directly succeeding one another). The starting phase of each period (of about 45 min) was included in the calculation of the thermal COP.

The inlet air had a temperature of 29 °C and a dew point of 16 °C (a relative humidity of about 47%). The outlet air temperature was rising to about 45 °C, while the outlet dew point went down to less than 5 °C.

Two different coefficients of performance for dehumidification COP_{dehum} and for cooling COP_{cool} were defined:

$$\mathrm{COP}_{dehum} = \frac{Q_{dehum}}{Q_{Des}} \quad \text{and} \quad \mathrm{COP}_{cool} = \frac{Q_{cool}}{Q_{Des}}. \tag{2}$$

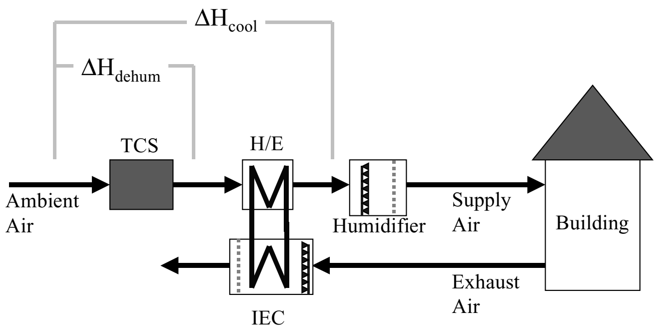

Figure 258. Definition of ΔH_{dehum} and ΔH_{cool}

where Q_{dehum} and Q_{cool} are:

$$Q_{dehum} = \int_{t_i}^{t_f} \Delta H_{dehum} \cdot \dot{m}_{air} \cdot dt$$

$$Q_{cool} = \int_{t_i}^{t_f} \Delta H_{cool} \cdot \dot{m}_{air} \cdot dt$$

Figure 258 shows this definition schematically. ΔH_{dehum} is the enthalpy difference of the air stream caused by the adsorption within the zeolite tank only. ΔH_{cool} includes the cooling effect caused by the cold recovery.

The energy density ρ_{cool} is defined as

$$\rho_{cool} = \frac{Q_{cool}}{V_{Zeo}} \tag{3}$$

where V_{Zeo} is the volume of the packed bed of Zeolite.

The results for COP$_{dehum}$, COP$_{cool}$ and ρ_{cool} for the different desorption temperatures were shown in Table 35.

The best performance was measured using a desorption temperature of 80 °C. Below that temperature almost no dehumidification can be observed.

TABLE 35. Experimental results of the thermochemical storage for cooling

	Desorption temperature		
	130 °C	100 °C	80 °C
COP$_{dehum}$	0.45	0.48	0.50
COP$_{cool}$	0.67	0.80	0.87
ρ_{cool}	168 kWh/m^3	105 kWh/m^3	100 kWh/m^3

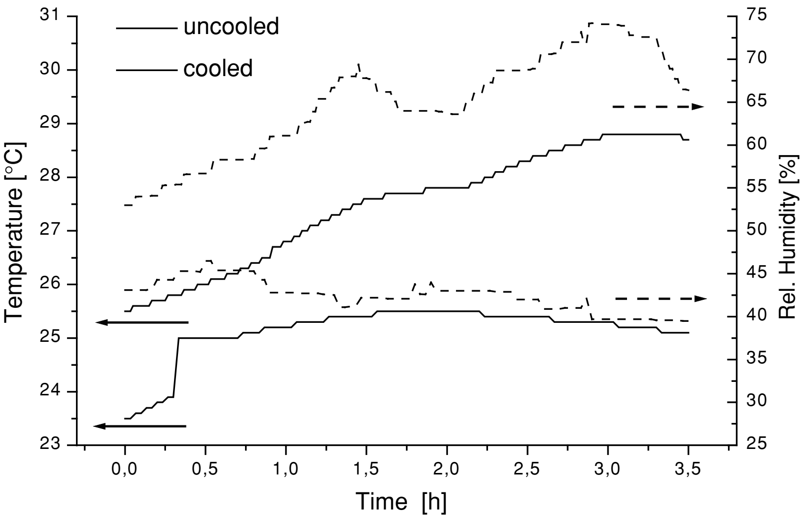

Figure 259. Temperature and relative humidity in the jazz club during a concert

There is a remarkable increase from COP_{dehum} to COP_{cool}, which is caused by the heat recovery. This can be explained by the high outlet air temperatures during adsorption, which are supporting the cooling by the indirect evaporative cooler very effectively. At a desorption temperature of 80 °C a value of 0.87 could be reached. That means 87% of the district heat were converted into cooling energy and were delivered to the jazz club.

Figure 259 shows temperature and humidity inside the jazz club over a sold out evening concert without any air conditioning (ventilation only) and with desiccant cooling by the thermochemical storage. In the first case temperature and humidity are rising until the end of the concert (there is a break at about 22:45 when humidity is decreasing for half an hour), which indicates that sufficient air conditioning cannot be provided by the ventilation system only. The maximum values of almost 29 °C temperature and 74% relative humidity are far outside the comfort zone (26 °C/50% R.H.). By using the Zeolite system temperature and humidity could be kept at comfortable values over the complete concert.

25.2.3. CONCLUSIONS

The demonstration project on thermochemical energy storage showed results in the heating and air-conditioning application with good correspondence with the theoretical calculated values. The operation control strategies have to be improved and simplified in the future.

A rough economic evaluation showed that the pay back time depends strongly on the price reduction for the off peak thermal energy, the investment costs and the number of storage cycles. Under the assumption of an 40% reduction in winter (which is already negotiated) and a 60% reduction in summer for the district heat energy price, investment costs of 60.000 Euro (for the described system) and 150 heating and 100 cooling cycles per year, a pay back time of 7–8 years was found. This shows that thermochemical energy storage systems can be competitive in the near future due to their large development potential.

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26. OPEN ABSORPTION SYSTEMS FOR AIR CONDITIONING AND THERMAL ENERGY STORAGE

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Abstract. Open absorption systems for thermal energy storage have been investigated over the last years. Open sorption systems using liquid desiccants like Lithium chloride are able to dehumidify an air stream. By adiabatic humidification this dry air can be cooled down and used for air conditioning of buildings. These systems provide cool and dry air to the rooms. At the same time these systems are able to store thermal energy very efficiently. The thermal energy can be stored within the difference of salt concentration between the diluted solution (after absorption) and the concentrated solution (after regeneration).Examples of demonstration projects will be given. A solar application for the dehumidification of an office building and an application connected to a district heating net for the cooling of a jazz club will be presented in detail.

Keywords: Absorption, thermal energy storage, desiccant cooling, Lithium chloride

26.1. Motivation

External cooling loads of buildings can be reduced efficiently by shading, insulation and cold recovery. Internal heat gains have to be removed actively. So a significant incoherence of solar irradiation and cooling load can result, even if air conditioning of the building is necessary only at daytime. In this case, energy storage can increase the solar fraction of the energy supply significantly. Energy for dehumidification can be stored efficiently and none dissipatively in desiccants.

Figure 260 shows the primary energy demand depending on electricity net efficiency on the upper x-axis for conventional vapour compression chillers and on the solar fraction for solar thermal driven air-conditioning systems. Each curve is representing a vapour compression chiller or a single (or double) effect absorption chiller with different thermal COPs. It is interesting to see that single effect absorption chillers using solar energy have to reach high solar

H. Ö. Paksoy (ed.), Thermal Energy Storage for Sustainable Energy Consumption, 429–444.

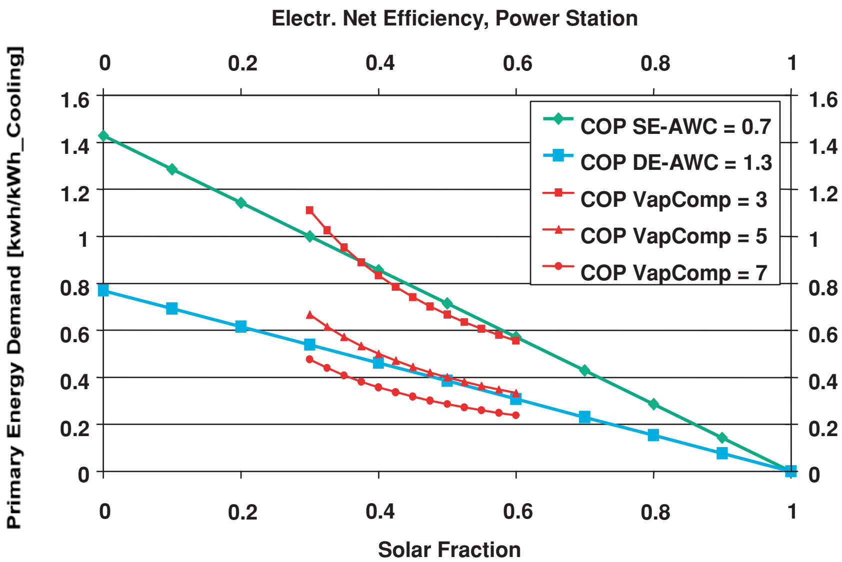

Figure 260. Primary energy demand for vapour compression and absorption chillers

fractions around 0.8 in order to compete with conventional systems. Solar fractions of 0.8 cannot be reached without efficient thermal energy storage.

The water removed from the air $\Delta Y = Y_{abs\ in} - Y_{abs\ out}$ is absorbed by the desiccant. The system of absorbed and regenerated desiccant has the potential to remove the latent enthalpy $\Delta H_{lat} = M_{air\ reg} \cdot \Delta Y \cdot h_{lat}(T)$ from the air flow. Neglecting the small amount of sensible heat of the absorbed vapour, the storage capacity SC for dehumidification enthalpy per volume desiccant V_{des} can be defined as

$$\mathrm{SC} = M_{air\ reg} \cdot \Delta Y \cdot h_{ev}\,(T = 0)/\,V_{des} \tag{1}$$

where in the case of a liquid desiccant the volume of the diluted solution is taken $V_{des} = V_{dil.\ sol.}$.

26.2. Absorption Process

Figure 261 shows the absorption and the regeneration process schematically. During Absorption the concentrated salt solution is distributed over an exchange surface, which is in contact with an air stream. The air will be dehumidified and the salt solution will be diluted by the absorbed water vapour. During regeneration the diluted solution becomes concentrated again by desorption from a hot air stream.

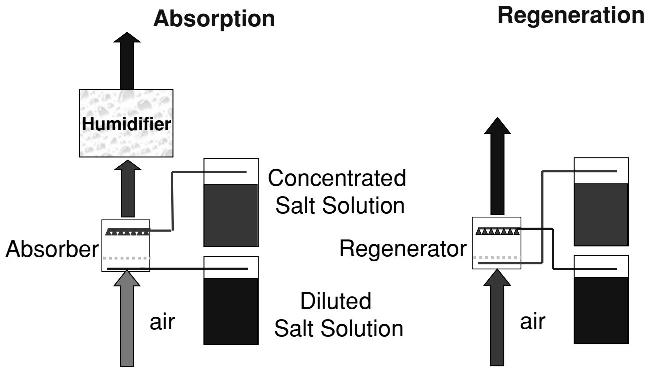

Figure 261. Absorption and regeneration of an open absorption storage

In open cycle desiccant cooling systems, DCS, air is dehumidified by solid or liquid desiccants and cooled subsequently by heat exchange and water evaporation. The desiccants absorb water when drying the air and have to be regenerated to continue the process. In open cycle systems the desiccants have to be regenerated against the ambient water vapour pressure, corresponding to the dew point of the ambient air [1, 2]. Closed cycle absorption systems have to regenerate against a temperature well above the ambient wet bulb temperature [3]. Consequently open cycle regeneration requires lower regeneration temperatures, which results in higher collector efficiency. A different method to achieve lower regeneration temperatures is cooling the absorption process [6, 7].

In Figure 262 plots for a cooled (P_1) and an adiabatic sorption process (P_2) are compared in a psychrometric chart, with parametric lines indicating different concentrations (C_1, C_2) of an arbitrary desiccant. Both processes dehumidify air from ambient humidity ratio of 14.5 g/kg to a humidity ratio of 8 g/kg (solid lines). The dotted lines continuing the process lines, point to the different concentrations, (C_1, C_2), required to drive the processes. If the absorption process is cooled, low humidity ratios can be achieved using a desiccant of concentration C_1. At a given ambient humidity ratio the desiccant can be regenerated with an ideal equilibrium regeneration temperature Treg1. In the adiabatic absorption process the temperature increases significantly and a higher concentration is required to achieve the same humidity ratio. Consequently T_{reg2} a significantly higher (equilibrium) temperature is required to regenerate the desiccant. This statement is almost independent of the nature of the desiccant. For example, C_1 corresponds to a 40% solution of LiCl–H_2O, as well as to Silicagel of a water content of 10.5% as well as to Zeolite M13X of a water content of 25.5%. C_2 corresponds to Silicagel of a water content of 3.1% or to Zeolite M13X of a water content of 20.3%. A LiCl–H_2O solution at a concentration corresponding to $C_2 = 65\%$ will crystallize.

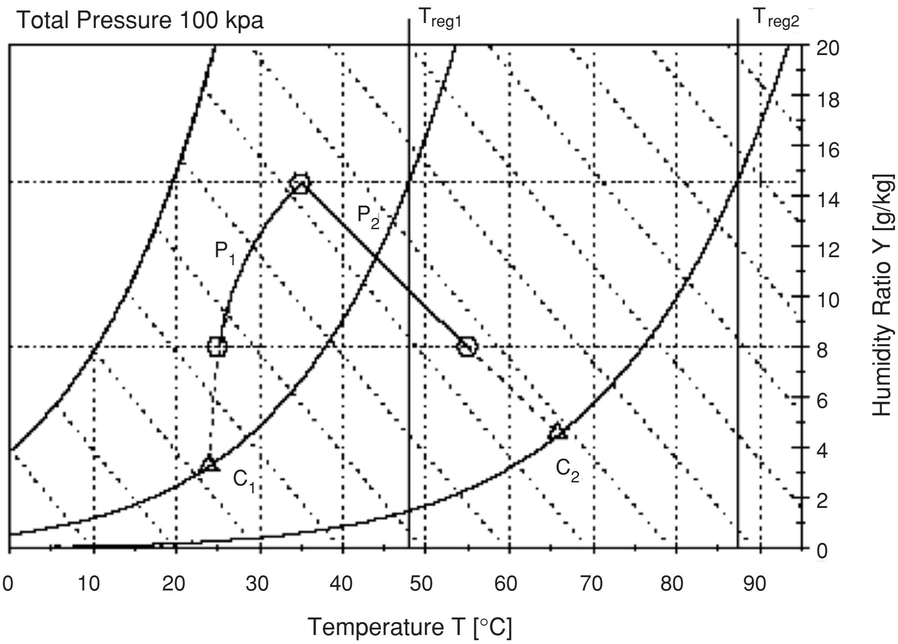

Figure 262. Comparison of cooled (P_1) and uncooled (P_2) absorption process. T_{reg} indicate the equilibrium regeneration temperatures at ambient humidity for different concentration C

Most of the commercial systems installed use rotary dehumidifier wheels with solid sorbents as e.g. Silicagel for air dehumidification. This is a design, which cannot be cooled easily. Absorption processes using liquid desiccants can easily be cooled, either by precooling the desiccant and supplying a sufficiently large desiccant flow, or by spraying the desiccants over a heat exchanger. These systems are also commercially available [4].

For ideal cooled absorption processes (cooling temperature 20 °C) a comparison of air dehumidification and energy storage capacity of different solid and liquid desiccants is given in Figure 262 [5]. In cooled absorption processes, the equilibrium humidity ratio is only a function of the temperature T_{reg} at which the desiccant has been regenerated (at given air humidity ratio). There is no difference in air dehumidification potential between different desiccants, either liquid or solid (left diagram of Figure 261). The energy storage capacity, however, achieved in liquid desiccants, is significantly greater compared to solid desiccants (right diagram of Figure 261) [6]. The highest applicable regeneration temperature to prevent from crystallization is indicated by symbols.

The absolute values of the storage capacities of liquid desiccants are high compared to other storages used in air-conditioning systems, in general 3–5

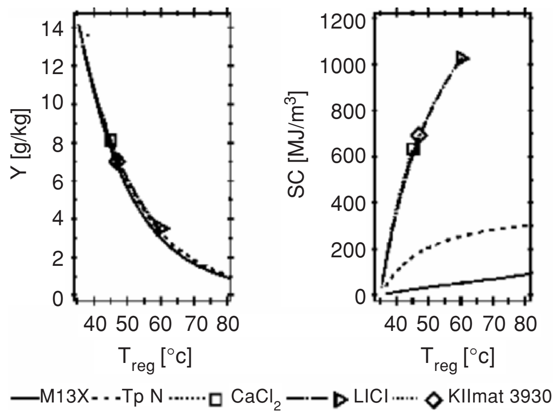

Figure 263. Dehumidification and storage capacity for different liquid and solid sorbents

times higher as the storage capacity of an ice storage. As Figure 263 shows the energy storage capacity in solid desiccants is not competitive.

In Figure 264 the dehumidification potential ΔY of a 40% LiCl–H_2O solution is plotted as a function of the air to solution mass flow ratio MR for certain operating conditions and ideal mass exchange, solid line (1). In addition the energy storage capacity SC is plotted, dotted line (2). Up to a

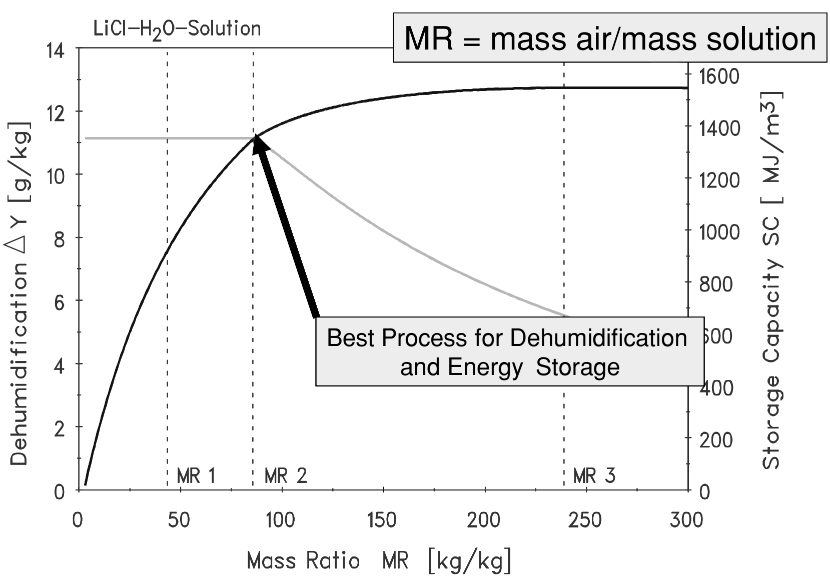

Figure 264. Air Dehumidification and energy storage capacity in an ideal absorption process as a function of the air to solution mass ratio (cooling temperature 24°C, inlet humidity ratio 14.5 g/kg, LICL–H_2O solution)

mass ratio MR 2 = 84 the theoretical dehumidification potential ($\Delta Y = 11.2$ g/kg) is not affected by the mass ratio, the solution flow is still sufficiently large to dehumidify the air. On the other hand the storage capacity increases with the mass ratio and reaches about 1,360 MJ/m^3 which is about 87% of its maximum value of 1,560 MJ/m^3 at MR 3. In ideal cooled absorption processes with 40% LiCl–H_2O solution, inlet air humidity of 12–20 g/kg and cooling temperatures of 20–35 °C energy storage capacities of about 800–14,00 MJ/m^3 are achievable. For efficient dehumidification the temperature of the absorption process and the concentration of the salt solution are of major importance, where as for high energy storage capacities a high air to solution mass ratio 50–100 is decisive [6]. High MR correspond to very small specific solution flows of about 0.2–0.5 l/(h m^2)! Therefore serious technical problems have to be solved, developing adequate exchanger surfaces which are reliably wetted by this very small solution flow in order to achieve high energy storage capacities [6, 7].

26.3. Absorption System

For solar applications, according to the reasons mentioned before: low regeneration temperature, high cooling system COP and high energy storage capacity, the ZAE Bayern suggests a liquid desiccant cooling system dehumidifying air by a small flow of a concentrated salt solution, Figure 265.

The air-conditioning system on the right hand side of Figure 6 (wet bulb temperature of dehumidified air: 15.6 °C) can be driven if concentrated e.g.

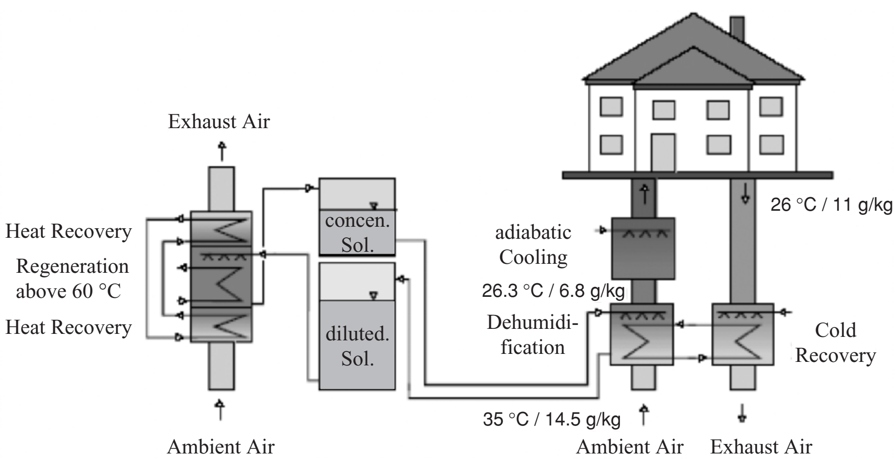

Figure 265. Example of a liquid desiccant cooling system L-DCS with energy storage in the desiccant solution

40% LiCl–H_2O solution is available. Diluting the solution to 26% an energy storage capacity of SC = 1,000 MJ/m^3 is achieved. If heat for regeneration (e.g. solar energy) is available the diluted solution can be concentrated (heat recovery 0.6). The $COP_{th}(M_{air\ cool} = M_{air\ hyg})$ is 1.1.

The absorption process is cooled by the exhaust air using an indirect evaporative cooler. A numerical example for achievable air states is given in Figure 6. Supply air and return air are coupled by a water circuit, dehumidifier and regenerator are coupled by a solution circuit. So the devices can be installed at separate locations within the building or outside. Regeneration is possible at temperatures above 60 °C under the specified ambient conditions (see Figure 265). Figure 7 shows the required specific regenerator surface as a function of the regeneration inlet temperature. Higher temperatures reduce the required surface. The optimal regeneration temperature can be found by a cost optimization analysis taking into account specific costs for collectors and exchanger surfaces.

Regeneration is possible at temperatures above 60 °C under the specified ambient conditions (see Figure 265). Figure 266 shows the required specific regenerator surface as a function of the regeneration inlet temperature. Higher temperatures reduce the required surface. The optimal regeneration temperature can be found by a cost optimization analysis taking into account specific costs for collectors and exchanger surfaces.

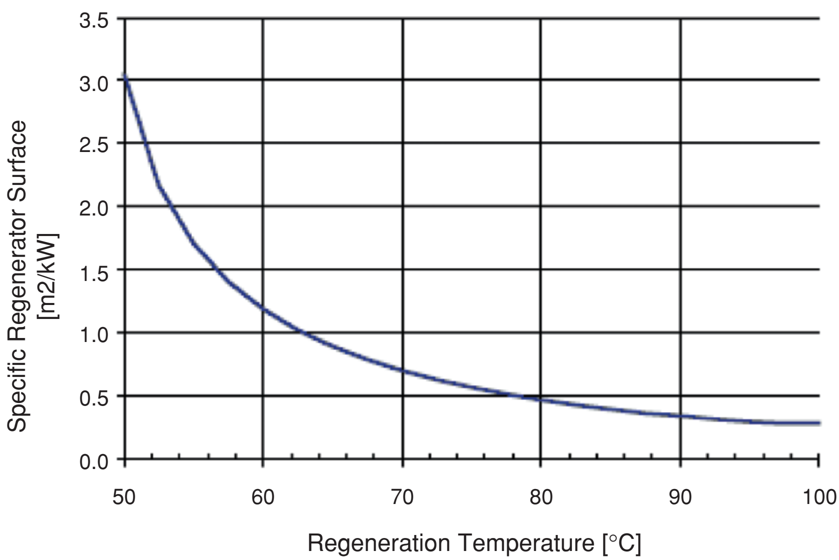

Figure 266. Specific regenerator surface as a function of the regeneration inlet temperature. According to the example of Figure 263, the diluted LiCL–H_2O solution of 26% is regenerate to 40%. The heating water is assumed to be isothermal

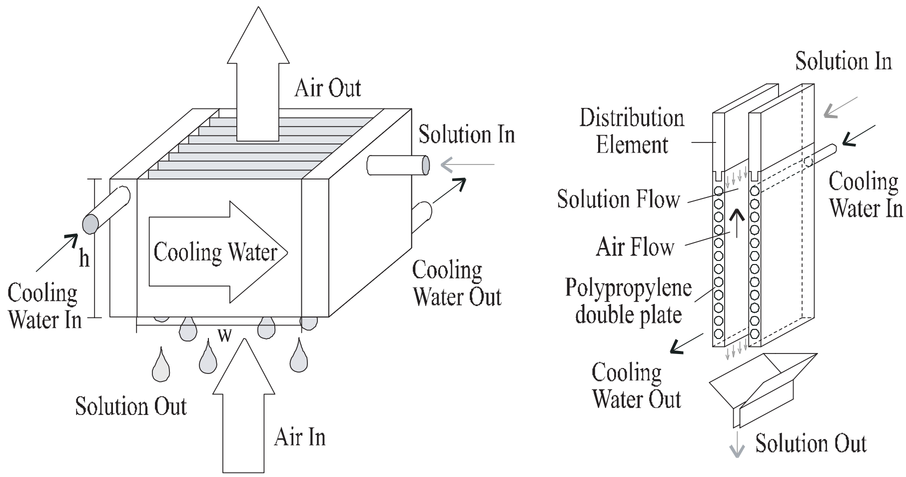

Figure 267. Ddehumidifier design and single channel exchanger design

Key component of the system is the dehumidifier shown in Figure 267. The absorptive dehumidifier has to cool the salt solution sufficiently to guarantee a low water vapour pressure. A small specific solution flow has to be distributed uniformly over the dehumidifier surfaces to achieve a high energy storage capacity. Furthermore the dehumidifier has to withstand the corrosive forces of the salt solution and has to be build of inexpensive materials which can easily be manufactured.

First prototypes have been built of separate water cooled exchanger plates, based on polypropylene double plates, which have been developed and tested within the last years. Since 1997 the tests has shown very satisfying results. A special coating of the plates and a special solution distribution element (no sprays are used) provide a uniform and almost complete coverage of the surface by the solution.

26.4. Example: Office Building Amberg/Germany

26.4.1. Abstract

An office building of 5,700 m^2 floors space has been built in Amberg, Germany, by architects Hart & Flierl for Prochek Immobilien GmbH. The innovative air-conditioning concept using solar energy has been worked out by M. Gammel engineering consultants. The comparatively low heating (35 $kWh/m^2/a$) and cooling demand (30 $kWh/m^2/a$) of the building is covered by thermally activated ceilings, assisted by appropriately conditioned ventilation air.

Well water of 12–14 °C with a cooling capacity of 250 kW is used for cooling the ceilings in summer. A solar driven liquid desiccant cooling system, developed by ZAE Bayern, dehumidifies outside air by a liquid desiccant, a concentrated salt solution, LiCl–H_2O, with a capacity of 70 kW and cools 30.000 m^3/h of supply air with a capacity of 80 kW by cold recovery from evaporatively cooled exhaust air. The liquid desiccant is regenerated by solar thermal energy from a 70 m^2 flat plate collector array at 70–80 °C with a maximum capacity of 40 kW. Solar energy for air conditioning is stored efficiently in about 10 m^3 of desiccant solution. Summer air conditioning uses only solar energy, except from electricity for pumps and fans.

26.4.2. INTRODUCTION

Prochek Immobilien GmbH in Amberg, Germany, has built an office building called the "email-fabrik", with a floorage of 5,700 m^2, see Figure 268. The building has been designed by architects Harth & Flierl in Amberg, Germany. The consulting engineers M. Gammel in Abensberg, Germany, prepared an energy saving heating and air-conditioning concept, using solar energy. Due to this concept the building should have an annual energy demand for heating as low as 35 kWh/m^2. The high comfort standard requested and the internal heat loads require air conditioning in summer. The predicted specific annual cooling and dehumidification load is about 30 kWh/m^2.

Thermally activated ceilings are used to heat and cool the building. In summer the ceilings are cooled by well water. Therefore humidity control is required to prevent humid air from condensating at the ceilings. The humidity is controlled by a solar driven liquid desiccant cooling and dehumidification

Figure 268. Office Building "email-fabrik" in Amberg, Germany

system. A special dehumidifier, developed by ZAE Bayern, using evaporative cooling from the exhaust air flow, provides high system efficiency and low desiccant regeneration temperatures, which can be efficiently delivered by economic flat plate collectors. Extremely low desiccant flow rates in the dehumidifier provide a high, non dissipative energy storage capacity in the desiccant. Thereby a high solar fraction and a maximum saving of fossil energy can be achieved. Both systems, the thermally activated ceilings and the desiccant cooling system, need only small amounts of electricity for pumps and controls. The Bavarian Ministry of Economics, Traffic and Technology has funded a demonstration project in which the liquid desiccant cooling system has been built as a prototype. The building and the active ceiling system is in use since June 2000. The solar desiccant cooling system has been installed and will start operation in spring 2003. Component tests and long-term system monitoring are planned and prepared. Details of the system concept and projected performance data are outlined in this article, the state of the project is reported.

26.4.3. THERMALLY ACTIVATED CEILINGS FOR HEATING AND COOLING

The structure of the building has been thermally activated by molding polypropylene pipes into the concrete ceilings, see Figure 269. The thermally active ceilings provide base load heating and cooling. Peak load is covered by separate air handling units for different bureau departments in the building. The hygienically necessary air flow is sufficient to deliver the additional

Figure 269. Plastic pipes are molded into the concrette ceilings for heating and cooling

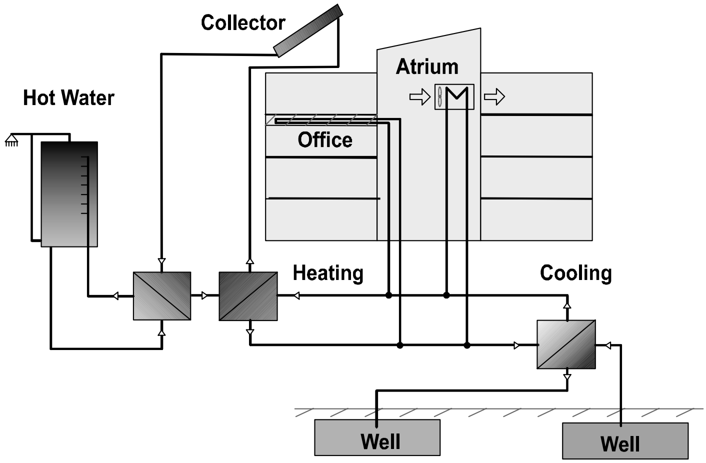

Figure 270. Sketch of the heating, cooling and hot water system, where ell water is used for cooling

heating or cooling. So, only small air channels are required. The heating is fueled by natural gas. Cooling is provided by well water.

A sketch of the system for heating, cooling and hot water production for a restaurant within the building is shown in Figure 270. The thermally activated ceilings are divided into 16 zones, the air handling system into 20 zones which allows separate cost calculation for different departments even if the division of the floor space should be changed in future. In summer the ceilings are cooled to about 18 °C. The maximum cooling capacity delivered by the well water is 250 kW.

26.4.3.1. *Solar Liquid Desiccant Cooling System*

In summer the ventilation air has to be dehumidified to keep the required comfort and to prevent from condensation at cold ceilings. The air dehumidification is done by a liquid desiccant dehumidification and cooling system, sketched in Figure 271. Warm and humid outside air is cooled and dried in a special dehumidifier by a concentrated Lithium Chloride salt solution ($LiCl–H_2O$) before it is blown into the atrium of the building. From there several air handling units draw the air into the offices and provide additional cooling on demand.

The exhaust air of the building is collected in three exhaust air handling units. Indirect evaporative coolers exploit the remaining cooling capacity of the exhaust air and cool the supply air in the dehumidifier via a water loop. This cold recovery makes the system more efficient. Depending on ambient

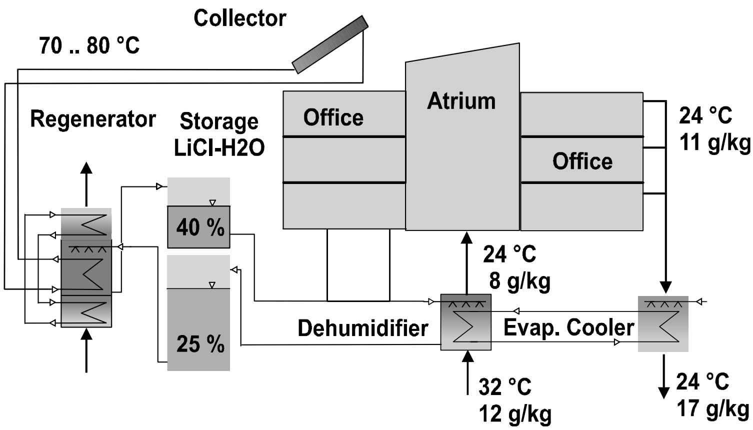

Figure 271. Air dehumidification by a liquid desiccant storage system

conditions the predicted thermal coefficient of performance of the system is 1.2–2[1]. The thermal coefficient of performance, COP, is defined as the enthalpy difference between outside and supply air related to the thermal energy used to drive the system.

A special low flow technique enables the dehumidifier to dilute the desiccant significantly when drying the air. The salt concentration changes from 40% to about 28% wt. Concentrated and diluted solution are stored separately. The dehumidification process can be operated as long as concentrated solution is available. The system of concentrated and diluted solution stores energy very efficiently. The energy storage density reaches up to about 300 kWh/m^3 related to the volume of the diluted solution. Since a chemical potential is stored, the storage is non degrading. No insulation of the storage tanks is required.

When solar energy is available the diluted solution is regenerated to its original concentration in a regenerator, at temperatures of 70–80 °C. At this temperature water evaporates from the desiccant solution and is taken to the ambient by an air flow through the regenerator. The Lithium Chloride does not evaporate. It remains in the solution and in the cycle. Heat recovery for the air flow is used to keep up the thermal coefficient of performance.

26.4.3.2. *Cooling Capacity, Solar Collector and Storage Size*

The desiccant cooling system is designed for a maximum air flow of 30.000 m^3/h. The design point for cooling is defined as 32 °C and 12 g/kg outside air state and 24.5 °C and 8.5 g/kg supply air state. Under this conditions the air cooling demand is about 80 kW the air dehumidification demand is 70 kW. A total air-conditioning capacity of 150 kW is required.

[1] In hot and humid climates the COP will be close to 1.

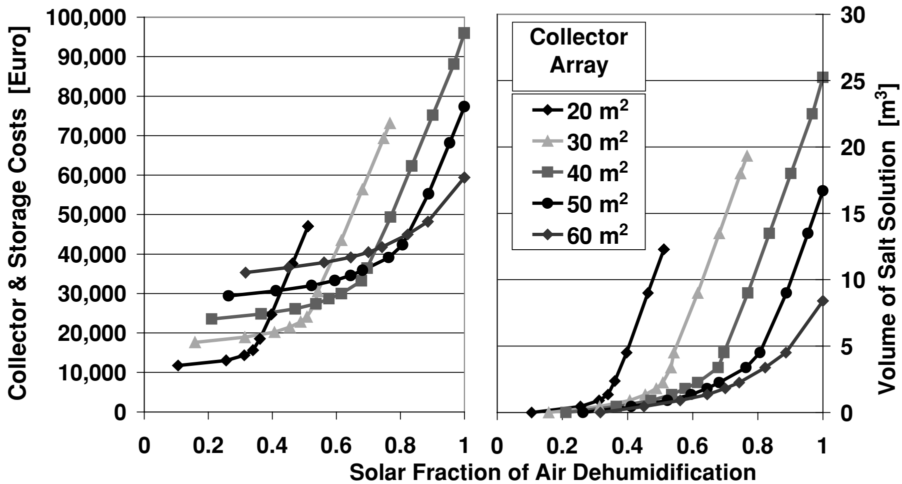

Figure 272. Investment costs of collector array and desiccant storage as a function of the solar fraction and the collector array size

The system concept demanded for a system driven solely by solar, no additional fossil fuel should be used. Therefore, the required storage volume and the investment costs for collector array and storage have been calculated as a function of collector array size and solar fraction. A computer simulation of the system has been made evaluating the seasonal performance of the system under the meteorological conditions of Amberg. Figure 272 shows the results.

On the right hand side of Figure 13 lines of constant collector array size indicate the storage volume needed to achieve a certain solar fraction. The larger the collector array size, the smaller is the required volume of the stored desiccant for a given solar fraction. The left hand side of Figure 13 shows the related investment costs. A collector array size of 60 m^2 and a storage volume of 8.5 m^3 turn out to be the most economic solution to achieve 100% solar operation.

A solar collector array of 70 m^2 of highly efficient flat plate collectors has been installed, providing a maximum thermal power of about 40 kW. Solar energy is collected during sunny periods in the early season and stored for several weeks until the energy is needed in short dehumidification periods in July and August. Separate tanks of 12 m^3 volume are used to store diluted and concentrated solution, containing 3,000 kg of Lithium Chloride salt and a varying amount of water.

26.4.3.3. *Predicted Energy Balance and Investment Costs of the Desiccant Cooling System*

The desiccant cooling system can provide up to 20 MWh per year of cooling and dehumidification energy. This includes the energy delivered by the cold recovery system. In addition to the regeneration of the desiccant solution the collector array has the potential to deliver about 11 MWh per year of hot water for the restaurant or the heating of the building. A connection to the heating system of the building, however, is not yet installed.

The necessary electrical energy for operating the desiccant system has been calculated to be about 1.5 MWh per year. Compared to a conventional system using vapor compression cooling and gas heating about 6 MWh of electrical energy and 11 MWh of thermal energy per year can be saved.

The well water cooling system provides a cooling energy of 150 MWh per year and needs about 10 MWh of electrical energy. A conventional vapor compression system would need about 50 MWh of electrical energy per year.

The total investment costs of the desiccant cooling system including collector array, cold recovery, storage, and controls have been planned to be about 300,000 Euro, this is 2,000 Euro per kW respectively 10 Euro per (m^3/h). The final costs have not yet been evaluated.

26.4.3.4. *State of the Demonstration Project*

The building is in operation since June 2000. The thermally activated ceilings and the air handling units are in operation since the beginning. With high outside air humidities in summer, however, the cooling has to be reduced to prevent from condensation at cold surfaces.

The collector array and the components of the desiccant cooling system, such as dehumidifier, regenerator, storage and desiccant handling system, have

Figure 273. 70 m^2 collector array, in the background: casing of dehumidifier and regenerator

Figure 274. Installation of the dehumidifier

been installed in 2001. The dehumidifier and the regenerator are prototypes. Figure 273 shows the collector array consisting of 70 m^2 highly efficient flat plate collectors (Wagner Eurosolar 20HT). Figure 274 shows the installation of the dehumidifier in October 2001. The dehumidifier is mounted into an air handling unit already installed on the roof of the building.

The desiccant solution has been mixed on-site from salt and water and filled into the tanks in November 2001. The heat and cold recovery system is in automatic operation since December 2001. The dehumidifier is part of this system and is used as heat exchanger in winter.

The measurement and control system of the desiccant system has been set into operation in February 2002. The collector system runs in automatic mode since March 2002 and produces hot water for the restaurant.

The liquid desiccant cooling system is designed, built and installed. A series of problems occurred and caused a significant time delay. Due to that time delay the demonstration project has to be stopped and no reliable data of the component test and the system operation could have been recorded.

26.4.4. OUTLOOK

The presented project was not able to deliver reliable data for the validation of the preliminary simulation results or the experimental results in the laboratory. In 2002 another research project has been started to investigate the cold storage for air conditioning by liquid desiccants. This project deals with cold water production for fan coil units and is integrated in the district heating net of Munich/Germany. The thermal energy from the district heat will be used for the regeneration, the charging of the TES respectively. Experimental results from that demonstration project can be expected end of 2005.

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