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Optimization of Cool Thermal Storage and Distribution

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Preface

Introduction

The International Energy Agency (IEA) was established in 1974 in order to strengthen the co-operation between member countries. As an element of the International Energy Programme, the participating countries undertake co-operative actions in energy research, development and demonstration.

District Heating offers excellent opportunities for achieving the twin goals of saving energy and reducing environmental pollution. Its is an extremely flexible technology which can make use of any fuel including the utilisation of waste energy, renewables and, most significantly, the application of combined heat and power (CHP). It is by means of these integrated solutions that very substantial progress towards environmental targets, such as those emerging from the Kyoto commitment, can be made.
For more information about this Implementing Agreement please check our Internet site www.iea-dhc.org/

Annex VI
In May 1999 Annex VI started.
The countries that participated were:
Canada, Denmark, Finland, Germany, Korea,The Netherlands, Norway, Sweden, United Kingdom, United States of America.

The following projects were carried out in Annex VI:

Title project	ISBN	Registration number
Simple Models for Operational Optimisation	90 5748 021 2	S1
Optimisation of a DH System by Maximising Building System Temperatures Differences	90 5748 022 0	S2
District Heating Network Operation	90 5748 023 9	S3
Pipe Laying in Combination with Horizontal Drilling Methods	90 5748 024 7	S4
Optimisation of Cool Thermal Storage and Distribution	90 5748 025 5	S5
District Heating and Cooling Building Handbook	90 5748 026 3	S6
Optimised District Heating Systems Using Remote Heat Meter Communication and Control	90 5748 027 1	S7
Absorption Refrigeration with Thermal (ice) Storage	90 5748 028 X	S8
Promotion and Recognition of DHC/CHP benefits in Greenhouse Gas Policy and Trading Programs	90-5748-029-8	S9

Preface

Benefits of membership

Membership of this implementing agreement fosters sharing of knowledge and current best practice from many countries including those where:

- DHC is already a mature industry
- DHC is well established but refurbishment is a key issue
- DHC is not well established.

Membership proves invaluable in enhancing the quality of support given under national programmes. The final materials from the research are tangible examples, but other benefits include the cross-fertilisation of ideas which has resulted not only in shared knowledge but also opportunities for further collaboration.

Participant countries benefit through the active participation in the programme of their own consultants and research organisations. Each of the projects is supported by a team of Experts, one from each participant country. The sharing of knowledge is a two-way process, and there are known examples of the expert him/herself learning about new techniques and applying them in their own organisation.

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Preface

This work was done under the IEA R&D Programme on District Heating and Cooling Research Topics for Annex VI by Electrowatt-Ekono Oy, Finland (chilled water and brine storage technology) and FVB District Energy, Sweden (ice storage technology).

An Experts Group provided valuable information and comments on drafts of this report.
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General

This report is not a complete guide to cool thermal storage or distribution, but it concentrates on suitable solutions for district cooling systems and specific technologies which have wide reference backgrounds and which were of special interest (for example: sodium nitrite/nitrate water solution and ice slurry).
The main emphasis in the report is put on presenting different cool storage technologies for daily use and on the selection of feasible and technically most interesting storage alternatives for cooling systems with district-cooling networks. The feasibility of selected storage alternatives was calculated and studied in cases having different loads and operating costs by using case district-cooling systems located in Stockholm, northern Europe and in Barcelona, southern Europe.

Key Contents

This work is focused on selecting the most feasible cool thermal storage, chiller and distribution system combinations for district-cooling (DC) systems. Information on the cool thermal storage, chiller and cooling distribution techniques on the market and under promising development was collected and analysed. The analysis concentrated on finding out the best commercial production/distribution/storage concepts in general and, especially, for the conditions of two case DC systems of 30MWcw in Scandinavian and Mediterranean climates.

The studies on optimising the size and operation of cool thermal storage concentrate on:

- Helping to understand the different economic variables of DC applications and their relations by using a range of operation (load) and economic (price of drive energy) conditions
- Providing examples of different cool thermal storage with chillers and distribution systems in accordance with the storage technology selected and load profiles of two case DC systems
- Providing case study examples with sensitivity analyses by using three peak/off-peak prices of electricity (flat- electricity tariff, 8cent/kWh/ 5cent/kWh and 11cent/kWh/ 5 cent/kWh)

Preface

Approach

The analysis of different storage technology on the market resulted in the selection of chilled water, sodium nitrite/nitrate-water solution, ice-on-coil (external melt), and ice slurry storage for the case study systems. Chilled water, sodium nitrite/nitrate water and ice-on-coil storage systems are all commercially available technologies and used in several locations. The ice slurry technology is still being developed but has promising advantages for reducing the size of the distribution piping. Centrifugal vapour compression chillers are chosen in all cases. However, in the case of chilled water storage LiBr absorption chillers are also feasible in this size of system when waste heat is available without excessive investments in a transmission system. In the cases studied the size of plant site did not limit the room for the chiller or storage installations and therefore, full advantage was taken of large chilled water tanks.
The results of the case studies with sensitivity analyses are presented as graphs, in which the total price of cooling ($/MWh) is shown as a function of storage output capacity (%/peak load demand).

Results

- The size and feasibility of cool thermal storage is most strongly affected by the customer demand profile and by the drive energy rate structure (on-peak/off-peak energy rates, demand ratchet, etc).
- With moderate cost of land, the investment cost of chilled water storage or sodium nitrate/nitrite solution storage is normally lower than the investment cost of chiller capacity. Regardless of the drive energy rate structure, this type of storages will thereby normally be economical as it reduces the daily peak load and consequently, the installed chiller capacity.
- Systems based on ice storage technologies tend to have a higher investment cost than chilled water storage systems unless the cost of land is high. The higher investment cost is partly due to a higher cost of the storage itself but also due to increased chiller cost because of the decreased chiller capacity at lower evaporator temperatures. More excess chiller capacity can also be needed in an ice storage system to cover unavailability for both ice-making chillers and water chillers.
- With increasing land cost, or a direct limitation of available land, the ice storage technologies can start to compete with plain chiller systems or chilled water storage systems. However, while the investment cost of ice storage systems normally is higher than that of plain chiller systems, the economic feasibility of an ice storage system is based on a variable drive energy rate structure.

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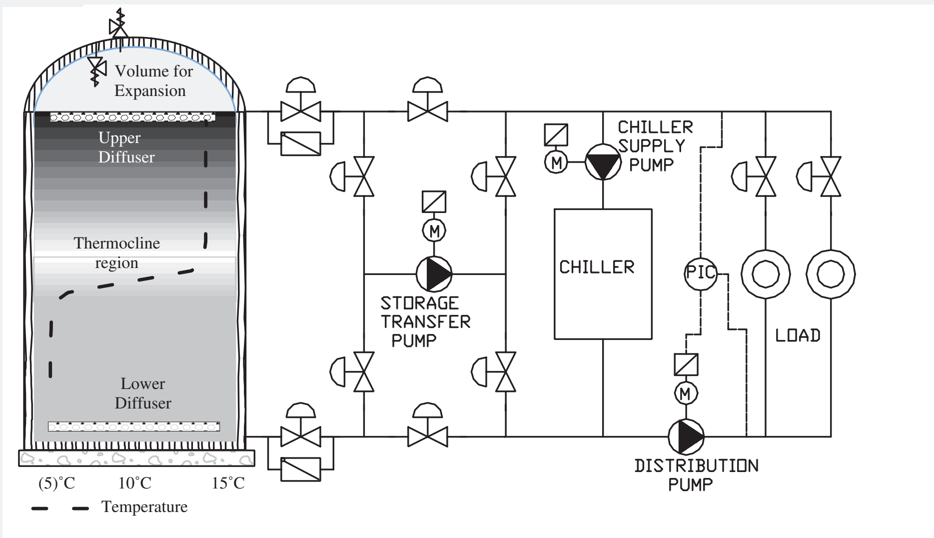
Introduction

Cool Thermal Storage in General

Thermal Energy Storage (TES) technology is a matured and accepted as a proven technology to improve energy efficiency by many in the Energy Management and Building Operations fields. Cool storage using water, ice or other phase-change materials as the storage media is most widely used (thousands of installations in operation) in the USA where the summertime cooling requirements are high. It is also used in Europe, where it is also connected to heat-recovery and hot water storage systems. Cool thermal storage is also used in Australia, Canada, South Korea, Japan, Taiwan, South Africa and several other countries.
Facilities that typically get the best benefit from Cool Thermal Energy Storage are those with low cooling load at night when off-peak electricity rates are in effect. Most commercial buildings fall into this category. TES systems are suitable for individual buildings from about 10000 m^2 as well as for district-cooling systems supplying entire downtown areas. They work best in facilities that are not fully operational at night, such as office buildings, schools, airport terminals, convention centres and sports complexes.

Figure 1
Principles of Cool Thermal Storage (chilled water)



Advantageous features a well-designed and correctly operated cool thermal storage are many:

- Installed chiller and cooling tower capacity can be reduced with a storage system.
- Cooling production can be "de-coupled" from the customer demand thereby making it possible to utilise lower off-peak drive energy rates, etc.
- A good storage system can provide better energy performance than a conventional instantaneous system because lower ambient temperatures are available to cooling towers at night when the storage is charged. Increased efficiency also results from all the machinery operating nearer the design points more of their operation time than in conventional systems.
- Storage designs make it easy to operate variable flow systems because the storage forms a buffer between the consumption and production and reduces the need for rapid changes of chiller operation.
- Storage systems provide higher reliability since they do not totally depend on instantaneous cooling production. In cases of short shutdown of a chiller it is possible to serve critical areas for a period of time from storage.

Additional information about cool storage is available from many sources. References and lists provided in this study give some guidelines for selecting appropriate cool storage alternatives. The following are useful sources of information:

- The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE)
- The Electric Power Research Institute (EPRI)
- The International Thermal Storage Advisory Council (ITSAC)
- The Thermal Storage Applications Research Center (TSARC)
- The International District Energy Association (IDEA)
- The Centre for the Analysis and Dissemination of Demonstrated Energy Technologies (CADDET).
- Vendors of thermal cool storages
- Magazines of heating, refrigerating and air-conditioning
- The Internet
- Utility companies

For many years the electric power industry used to support the installation of thermal storage systems to reduce on-peak loads of air-conditioning systems. This additional support from the utility side made it even more interesting to exploit the off-peak electricity rates by using cool storage. Hundreds of electricity utility programmes have been introduced worldwide to promote the use of storage technologies. Many of these are part of large Demand-Side Management (DSM) programmes. Such programmes can greatly influence the economic feasibility of installing thermal storage by offering financial rebates on equipment, information programs and special electricity rates for consumers. The deregulation of the energy market can be disadvantageous for the continuation of incentive tariff policy, because incentive programmes with special customer commitments are now widely prohibited by legislation. However, customers with thermal storage become more attractive to power generators anxious to effectively utilize their production capacity in competitive energy markets.

The retention of customer attraction is a primary aim of DSM programmes on the free energy market. The unique feature of thermal storage is its ability to offer customers cost savings without totally eliminating the energy sale to the utility. Thermal storage provides energy cost savings to the customer, energy efficiency and load management benefits to the power producer, and reduced pollution and fuel use to the environment. However, the payback period resulting from the analysis of a potential cool storage installation is often not attractive enough to the investor to give the project priority over other energy efficient technologies. All too often this is because full advantage was not taken of the many potential cool storage benefits or because thecool storage system sizing was not optimized. Under-sizing can result in poor levels of indoor comfort, while over-sizing results not only in higher than necessary initial costs but also in the potential wasting of electricity because of increased storage losses if more energy is stored than is required.

Introduction

The correct sizing of all equipment is the most important issue in optimising investment and operating costs. The load calculation must be accurate and indefinable safety margins should be avoided. The cooling load profile over periods from 24 hours up to 168 hours is as important as the peak hourly load for design of a cool storage installation. In a non-storage system, the capacity over 24 hours is 24 times the peak hourly load, which makes it possible to “top-up” the cooling demand after a of short period of overload. However, a cool storage system must be designed more carefully to be able to meet the extended loads over time as well as the peak hourly load. It is important to calculate the total consumption over the whole storage cycle, which is usually 24 hours but can be several days in systems having special load profiles.
In the following chapters there are examples of load calculation for cool storage systems. In chapter “Calculation and Simulation Software Tools” several software tools for precise calculation of building cooling load are listed. However, the daily load profile for a district cooling system is difficult to calculate precisely because of the numerous customers, each with their own load profile. Values based on experience from other existing systems should be considered unless the values measured from the particular system are available.

In centralized systems the diversification of customer behaviour helps to reduce the total installed chiller capacity. A normal reduction is 80...95% of the connected load. Due to the varying customer behaviour the demand profile of cooling is broader and larger base load units can be installed, which results in higher efficiency and better operation conditions (fewer operation periods under part-load conditions). The larger capacity of the base load units leads to lower cost per unit of capacity. The capital cost per unit of thermal storage also reduces as the stored capacity increases, this is applicable to centralized plants.

The centralization of cool production and district-cooling distribution are measures towards the well controlled, more cost-effective and environmentally sound cooling systems. The daily activation of a storage system can be automatic, although trained operating personnel are important in making time-of-day strategies successful.

Introduction

The supply guarantees of an individual customer can be realized in an easier way by district-cooling supply. A cost-effective backup and peak operation can be realized by installing an electric compression chiller and heat-driven refrigerating processes together. The electric compression chillers together with cool thermal storage can cover peak load demand. The heat-driven refrigerating processes can operate on base load and generate a large part of annual cooling demand. For a combined district-heating and district-cooling supply, the extension of CHP and a better use of waste heat, rejected by industrial processes, will be promoted, too. According to the aspects of environmental protection, this is an important contribution to the reduction of CO2 emissions and to pollution control.
The customer of centralized cooling facility need no longer take care of the maintenance of cold supply equipment, HCFCs or increasing space demand for a chiller and cooling tower while the old systems are replaced and enlarged.
In the inside urban area noise also represents a problem. It has to be reduced using expensive sound insulation. Low-noise cooling towers are expensive and their electricity consumption is also high. It is easier locate the centralized cooling plant where where the noise is not disturbing and the direction of sound can be better controlled.

As the distribution cost of cooling is easily as high as the production cost of cooling in many of existing centralised cooling systems, the transmission of chilled water or other cold media is a key factor in the economy of district cooling. The installation of large diameter chilled water pipes is a difficult and expensive task when building centralised cooling systems in densely populated urban areas. The full utilisation of high DT systems with variable flow and effective consumer equipment, and the optimisation of pipe insulation according to the distribution temperatures, result in the lowest life cost of the distribution system. A higher DT throughout the whole cool production, storage and distribution is one major goal in the development of cooling technology.

The following chapters introduce various technologies at the present level of commercially available systems.
A thermal storage system, properly selected, implemented and controlled, with high DT distribution creates a cooling utility which is competitive to respond to the market forces in an uncertain and unregulated power marketplace.
The feasibility of different thermal cool storage systems in combination with a right cool distribution network is demonstrated using case examples in the last chapters of this report.

Design of a Cooling Facility

The design and operating strategy for a cooling plant depends on a number of factors such as:

- Capital cost of chiller capacity and storage capacity
- Equipment efficiency
- Energy cost
- Demand cost
- O&M cost
- Space limitations
- Cost of acquiring land

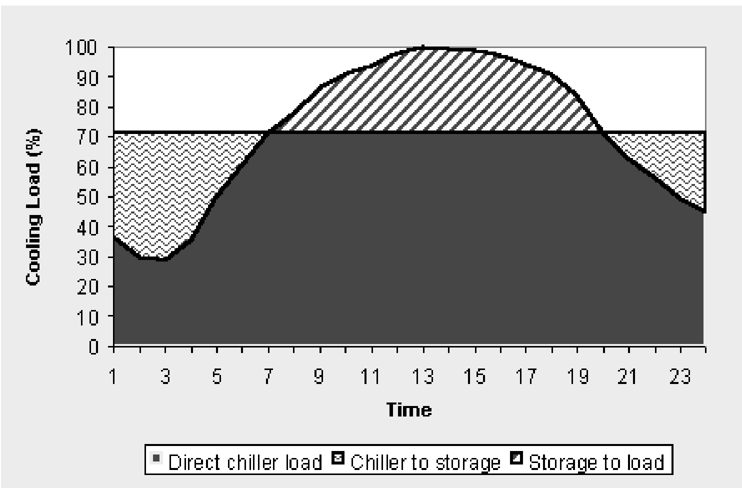
Thermal storage can provide the following advantages:

- Installed chiller capacity is reduced
- Power demand cost is reduced
- Chiller efficiency is increased due to less operation hours at partial loads and operation during the lower night temperatures
- Cooling is produced more cost-efficiently by charging the storage when electricity prices are low and discharging it when electricity prices are high.

Design and operating strategies will mainly vary depending on the energy demand cost and space limitations. Three main strategies are presented in the following:

With a relatively low demand cost or on-peak energy cost a “load-levelling” operation strategy may be preferable. With a load-levelling system the chillers operate at their maximum capacity continuously during the design day. When the customer load is less than the chiller output, the thermal storage is charged with the excess capacity (see Figure 2). When the customer load exceeds the chiller capacity, the necessary additional capacity is discharged from the thermal storage.
A load-levelling system minimises the required chiller capacity and reduces the on-peak cooling load by 20-40% depending on the design day load profile. With a load profile according to Figure 2, a 30% peak reduction can be achieved and the storage capacity is about 15% of the daily cooling load.

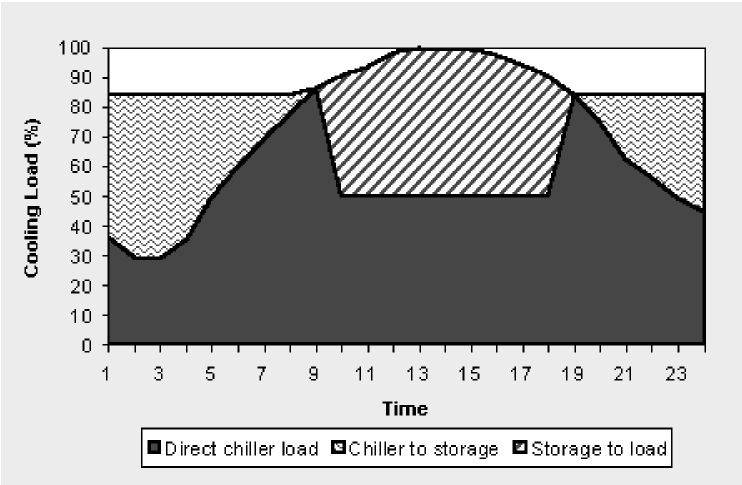
Figure 2
Load-levelling
thermal storage operation.



Introduction

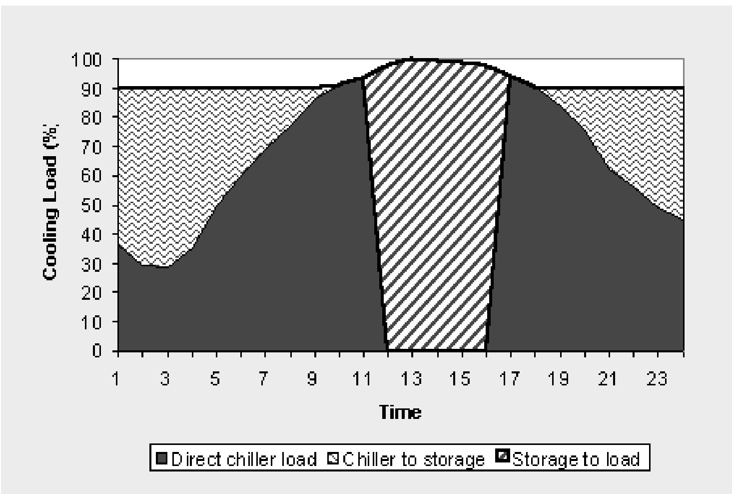
With increasing demand costs or on-peak energy costs it can be feasible to shift more capacity from on-peak hours to off-peak hours. With a "demand-limiting" operation the chiller capacity is limited specifically during on-peak hours (see Figure 3). Demand cost or on-peak energy cost can be reduced further compared to a load-leveling system but the required chiller capacity will increase. With a 50% reduction in the demand during nine on-peak hours, the necessary storage capacity is about 25% of the daily cooling load.

Figure 3
Demand-limiting thermal storage operation.



With even higher demand cost or on-peak energy cost, or when the on-peak period is short, a "full load-shifting" approach can be the most viable solution. With the full load-shifting system the entire cooling load is shifted from on-peak hours to off-peak hours. In cases of extremely high on-peak demand or energy costs a chiller capacity higher than the peak customer load can be optimal. With a 100% reduction in the demand during five on-peak hours, the necessary storage capacity is about 30% of the daily cooling load (see Figure 4).

Figure 4
Full load-shifting thermal storage operation.



The choice between full (full load shifting) or partial (load levelling or demand limiting) storage defines the size of storage to be needed and in some cases this selection is decisive or what storage technology is feasible for the system. In some cases many different storage technologies can be used, the selection of technology often depends on the size of storage.

Load Calculation

The design parameters of a chiller plant are determined by the demand profile (combined demand), available drive energy sources, economic factors and regulations of the authorities.

The cooling load of a building is determined by outside heat gain (solar radiation and heat conduction, when the ambient temperature exceeds that of the indoor temperature) and internal heat gain (people, equipment generated heat, technological processes, etc.). The cooling load of public buildings ranges from about 50 W/m² in northern Europe to 200 W/m² in the Middle East. The maximum cooling demands of buildings become lower because of the improvements in insulation, radiation selective windows and the use of more sophisticated architectural solutions. On the other hand, greater window surface and an increasing number of electrical equipment such as computers create more cooling load.

Most of loads are time-schedule-controlled, which causes a fast rise in the demand in the morning and rapid reduction at the office closing time. The schedules of occupancy, lighting and equipment use are important factors in the cooling load profile but the most significant factors are the pull-down loads acquired during the cooling shutdown periods and the external loads at prevailing weather condition. In buildings where the cooling is provided during the occupied hours only, accumulates the heat gains of external loads during the unoccupied hours are accumulated. This creates a significant pull-down load to be met during the first hour or two of the next cooling period. These pull-down loads and normal morning-time cooling loads are less than the peak-time load in the afternoon. Therefore, the pull-down loads normally have no effect on the sizing of non-storage systems, but in the sizing of storage systems these accumulated loads must be taken into account.

The construction material with high thermal capacity forms an in-built thermal storage for every room to be cooled. When designing an optimal cooling system this existing storage capacity must be taken into account in the dimensioning of building-based or centralised cool thermal storage. New precise computer simulation models of buildings help to dimension the air-conditioning and refrigeration systems to better meet actual cooling demand, which leads to lower peak time load as the thermal capacity of buildings is fully utilised to level off the cooling demand.
Less chiller capacity is usually required when thermal storage is installed because of fluctuations in the daytime cooling load. The cooling load is low when there is no sun or when the outside temperatures are lower. Because of changes in solar intensity, outside air temperature, and use of the buildings during the day, a typical chart of load versus time shows an increase in load during the day peaking in the mid-afternoon and returning to a low level overnight.

Introduction

When the installed peak chiller capacity is replaced with storage capacity it is crucial to find out long term load profile over periods from 24 hours up to 168 hours. The load profile over the whole storage cycle during the highest cooling period is as important as the peak hourly load for design of a cool storage installation. In a storage system there is less chiller capacity, which makes it possible to "top up" the cooling demand after a period of over load. Therefore, it is right to use conservative selection of design temperatures.
It is most advantageous for a district cooling system that there are different types of buildings connected to a central cooling plant. In different types of buildings the cooling loads do not peak at the same time. As a result, the production plant experiences a lower peak load than the total sum of connected building peak loads. This phenomenon is commonly called a diversification factor. The choice of diversification factor for any given district cooling area would depend on the diversity of cooling consumption patterns and demands. In general the diversification effects on district cooling tend to be smaller than those on district heating because the main load is from offices, stores and other public buildings with homogenous usage patterns. A typical diversification factor for district cooling systems is 0.9 - 0.95.

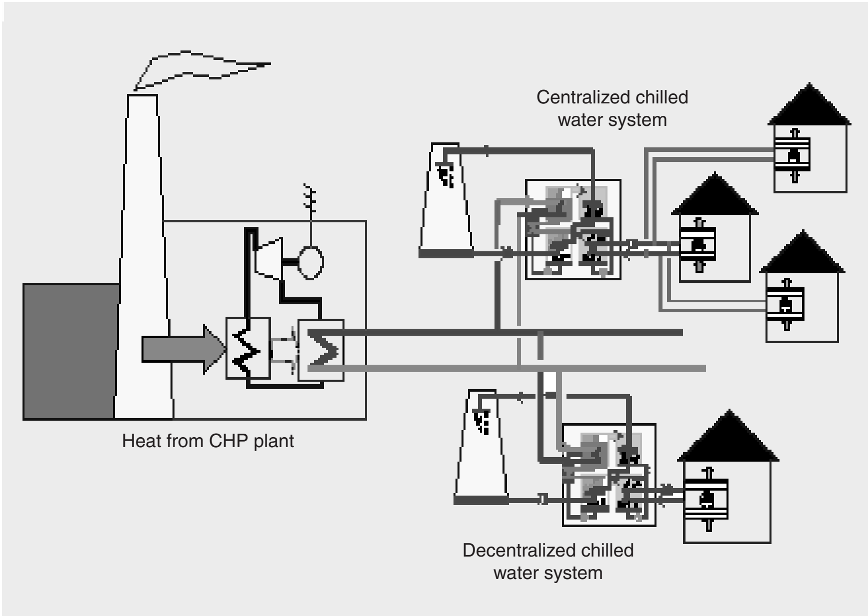
Operation Conditions and Practices

The type of air-conditioning system defines the supply temperature needed (+1…+16°C).
The range of supply temperatures is large and it has a major effect on overall design. In general, the higher supply temperature, the better for the economical construction and the operation of the chiller plant. Temperatures are fixed low in old air-conditioning installations, also air ducts are cheaper when low chilled water and air temperatures can be used.
The most common drive energy for chillers is electricity. With a coefficient of performance (COP) of about 5 for electric drive chillers and about 1 for heat-driven chillers the price of heat must be almost five times as low as the electric price to be competitive. When low-priced electricity is available, there is no chance for other sources of energy like hot water, steam or flue gas. These alternative energy must be wastes or by-products with a very low price. The prices of electricity and natural gas vary by season and by the time of day. Low-priced electricity is available during the night and low-priced gas can be obtained during the summertime. In the USA, due to the large air-conditioning load the tariff structure of electricity is generally divided into winter and summer charges.

The total economy of CHP and district heating favours the use of district heat in cooling - also, environmental regulations favour the use of alternative energy sources instead of peak electricity.

Introduction

Figure 5
The CHP integrated district cooling.



A district-heating network is used for distributing the CHP produced heat to the heating customers. The same network can be utilised to provide the primary energy to absorption chillers. In this case the absorption cooling units are located in individual buildings. Another way of providing the CHP produced cooling is to have centralized cooling units and a distribution network for the chilled water.

The noise from air-cooled cooling towers and compression chillers cannot be avoided altogether. Noise regulations and higher demands for comfort cooling guide designers towards the centralised district cooling systems where the noise is better controlled. The night-time storage charging noise caused by air-cooled cooling towers can be more disturbing. Special noise precautions must be taken even though the lower night-time ambient temperatures provide low condenser-water temperatures at lower air flows meaning lower fan power and noise.
In storage systems one should note that the size of cooling towers can be smaller and operating costs are also lower while they are operated at night when the outside air temperature is lower.

Cool Thermal Storage Technology

There are several technologies used in cool storage. Some of the methods are as old as the history of house building, however, new technologies are being developed all the time.
Thermal storage can be obtained in the temperature changes of almost any material. Concrete or other building materials are sometimes used to provide at least part of the building cooling during off-peak energy periods by sub-cooling a building during the night. Buildings are often sub-cooled by excessive air-conditioning during off-peak periods to reduce the cooling load during peak periods. These concepts affect the peak loads but should not be considered to be true thermal storage systems as they affect the room comfort. A true controlled thermal system does not overcool the building when cooling is not needed. Certain building element thermal storage systems have been designed to be controlled and reversible. When buildings are air-conditioned around the clock, the only variables are the gain factors. The mass of the buildings acts as a buffer against the changes in outside air temperature and solar gains. Many thermal storage systems have been built using gravel. The systems in some cases can be relatively inexpensive. The systems using gravel use cooler night air and/or evaporative cooling for production of cooling. Gravel has the advantage of being able to produce air temperatures very close to the storage temperature because of the large area of thermal contact, but air quality problems may exist, including damp and musty odours.

The storage of natural snow and ice was the first phase-change-based cool storage and this method can still be used in specific low demand conditions in places where seasonal temperature changes are large and the cooling demand season is short. Old mines and alike existing underground spaces can be used as long-term storage of snow and ice, which is cleaned off the streets. There are not often suitable storage rooms available in city areas, which reduces the potential.
Earth or rock strata have been used for thermal storage as themselves in some research applications. For example, a well and a recharge well combined with a cooling tower can be used to reduce the temperature of a confined underground gravel strata or contained aquifer during the winter period. During the summer, circulation of water through this cooled strata, might exchange energy to provide chilled water. Several projects using seasonal storage in aquifers have been carried out in the USA, Canada, Sweden, and China since the 1940s. There are many variables with this system and research is still needed to develop optimum applications. For daily storage, which is the subject of this review, the development of systems based on rock strata and aquifer may be of less interest. However, in Stockholm and Sweden, aquifers storage is used as daily storage for the downtown cooling system. The aquifer storage was chosen due to site constraints and the good conditions for such storage.

The phase-change materials include true phase-change materials such as organic waxes, salt hydrates and water/ice. Water/ice is the most common phase-change pair, even though the ice is expensive to produce because the low temperature needed causes low coefficient in refrigeration performance. Energy consumption per unit of cooling relative to requirements for conventional refrigeration cooling systems is up to 75% higher for ice storage systems due to storage losses and colder refrigerant temperatures for ice. The investment in a closed ice-making system is also high but the small space requirement makes it attractive in many urban areas where the price of building space is high.
The phase-change materials also include materials such as eutectic solutions, and materials that form high energy bonding structures (clathrates like R-22 and water). Phase change at a temperature point higher than that of water/ ice enabling higher refrigerant temperatures and saving energy in the charging mode. The investment in salt or organic material storage is higher than in ice storage and as the investment cost is the ruling factor in most cases ice storage is more competitive than other storage systems except chilled water.
The following chapters concentrate on ice, water and brine-based (solutions with water) technologies, all of which are commercial and also feasible alternatives for daily storage in large cooling systems. However, one must recognize that every potential thermal storage case is always different and that these differences affect the feasibility in any specific case. The only way to determine the best answer is to study the relevant alternatives.

Technology, Storage

Chilled Water Storage

Water is a common thermal storage material used today, because it is inexpensive, has high specific heat capacity and is easy to handle. Water is the most common cooling transfer medium regardless of the storage technology used. The main advantage is that a typical chilled water storage system uses the same pumps, chillers, and other components of the cooling system as a conventional cooling system. The cooling is transferred directly with the flow of stored water without heat transfer over any surfaces, which makes the output of chilled water storage extremely high compared with the heat exchanger or ice tank systems. A chilled water storage system is really just a variation of a decoupled chiller system, which separates the production and distribution of chilled water. The balance of flow between the constant volume production of chilled water and its variable volume distribution is handled with a bypass pipe commonly called a “decoupler.” The decoupler bypasses surplus chilled water when production exceeds distribution and borrows return water when distribution exceeds supply. In effect, the decoupler pipe itself can serve as a chilled water storage tank if its volume is large enough.

A chilled water storage system can provide chilled water at the same temperatures produced by the chillers, through the range of storage operation. This means that the chillers can be turned off for maintenance actions, or a system can be designed to operate the chillers only at night or during off-peak hours and provide the entire cooling capacity during on-peak demand periods. These systems also allow the chiller to operate at peak efficiency during the whole storage cycle. A chilled water thermal storage system is required to produce slightly colder chilled water than an instantaneous cooling system, which can be implemented just by setting the existing chiller controls at a lower setting.

In the cases of retrofit designs or expansion of existing systems, the ability to use the existing chiller and water systems can dramatically reduce the first costs of the storage system. Chilled water storage also provides low operating costs and long machinery life because of the simplest storage technology. Centrifugal water chillers and pumps, which are normally used, can last 20 to 30 years. Even though chillers may operate for more hours a day, that does not ordinarily reduce the equipment life since the machines experience fewer stops and starts. The water-filled thermal storage tanks normally require no routine service. Non-fuel operating and maintenance costs include costs caused by the treating and filtering of the system water, although the incremental cost maybe insignificant since the system water of the cooling system would have to be treated regardless of the presence of the storage system. The design lifetime of chilled water storage is at least 25 years. Other systems based on phase-change materials represent relatively new technology; so, there is more uncertainty about their expected lifetimes. These systems require maintenance and monitoring of operation conditions, but not additional personnel, as is the case in conventional district-cooling systems without chilled water cool storage. The direct energy consumption of the chilled water storage system is slightly more than what is required for conventional cooling systems mainly because of lower chiller temperature setting for storage charge and because of thermal and stratification losses of storage (<+10%).However, these losses can be partially recouperated by higher chiller efficiencies.

Most thermal storage systems ordinarily use non-pressurized tanks because of the high cost of large diameter pressure-rated tanks. Chilled water storage does not either benefit from an increased temperature span due to a pressurized tank, which is the case for hot water storage. A pressure control is necessary to separate the static head pressure of the directly connected chilled water piping systems of high buildings from the non-pressurized storage tanks. This is most commonly done with pressurizing pumps and pressure reducing valves at the storage tank. Back-pressure valves, back-pressure regenerators, gravity return systems, etc. in individual buildings could also be used to solve the pressurization problems. When water storage systems incorporate tanks with a water level within 5-10 metres from the highest point of the buildings, this usually eliminates the pressure problems. This will, however, make it possible to draw air into the system during pump stops. Indirect (via heat exchanger) building connection can also reduce the problem with system pressurization if the heat exchangers can be installed on the lower levels in the buildings.

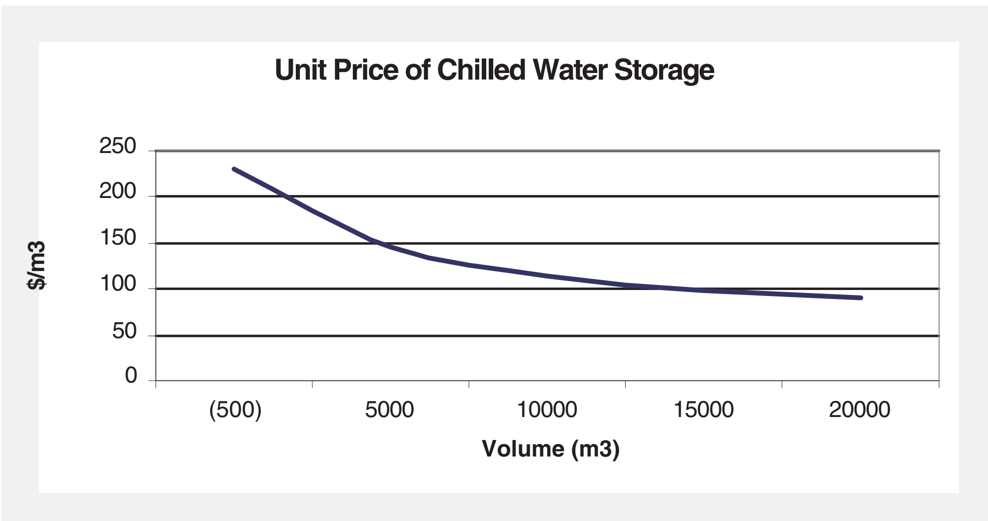
The greatest disadvantage of chilled water storage is the large storage volume required by water systems. The volume depends on the usable temperature difference (DT) of storage. To be effective, chilled water storage systems must raise the return water temperature to relatively high values. If the chilled water distribution system cannot achieve this, the capacity of the tank is severely impaired. Given its specific heat of 4.18 kJ/kg,K, about 0.86 m^3 of water are required to provide 10 kWh of cooling if the coil raises the water temperature by 10°C. The same amount of cooling can be provided with just 0.1 m^3 of ice, since each kg of ice absorbs 334 kJ as it melts. The actual size of ice storage is not small, because of the space needed for heat transfer piping and expension of ice (ice-storage constructions are described later), the volume of a chilled water storage system is 3 to 7 times that of ice storage systems.

Methods to obtain high temperature rise to the chilled water under various load conditions are most important when operating chilled water storage. Methods like variable chilled water flow, two-way control valves and chilled water return and coil valves that close when there is no air flow across the coils. Chilled water storage systems will not work well with systems with three-way control valves, which bypass unused low-temperature chilled water back to the tank. It also means that chilled water control valves must close when air-handling units are turned off. Failure of the valve to close causes the flow of unused chilled water from the supply to the return and dilution of the cooling potential of the remaining tank storage.

The sizable cost penalty imposed by the significantly larger storage tank volume required for chilled water is readily apparent in a cost-line comparison with ice storage. Keep in mind, however, that the cost of the water storage tank is a function of its surface area, while the capacity of the tank is a function of its volume. Therefore, when a larger chilled water storage tank is required, the MWhch cost of the tank decreases.

The installed cost curve of the chilled water thermal storage system in the following Figure 6 shows the significance of unit size to storage tank expense. The decreasing unit cost of chilled water storage systems can be very attractive to large central plants and industrial installations. For systems larger than 4-7 MWhch of storage, chilled water storage systems usually show the lowest overall ownership costs, when the system life, energy costs, first costs and maintenance costs are all taken into account. It should be noted when calculating the total investment cost that storing a large volume of water on site can be a valuable asset for fire/life safety systems. In fact, some system designs use sprinkler system water in their design.

Figure 6
The effect of increased volume on cost of chilled water storage in the range of 500 - 20000 m^3 by using above ground welded steel tank construction.



The following narrative addresses typical ways to provide temperature separation and some design considerations for chilled water storage.

Stratified Chilled Water Thermal Storage

Most chilled water storage systems installed today are based on designs that exploit the tendency of warm and cold water to stratify. It means that cold water can be added to or drawn from the bottom of the tank, while warm water is returned to or drawn from the top. A boundary layer or thermo-cline, 20 to 40 cm in height, is established between these zones. A large temperature difference in the chilled water system is needed to reduce the volume of storage and this difference between supply and return water temperature will represent a high density difference and thus increase the force separating the supply and return water. For this result, the return water temperature is more important than the supply temperature since the difference is not linear, but the difference increases more rapidly with increasingly warmer water.
Water should be introduced in a thin layer, uniformly throughout the tank. The velocity out of the diffusers should be very low to assure that the flow remains flat in the whole tank cross-section. There are several successful diffuser designs like octagonal pipe systems, radial disks and H-style pipe systems. The Reynolds number is the dimensionless ratio of inertial to viscous forces and it defines how turbulent a flow is at a certain environment. The limit value for a non-turbulent, a laminar flow is Re=2300. The inlet Reynolds number is calculated by:

$$Re_i = \frac{Q}{l \cdot v}$$

Q = flow per a diffuser (m^3/s)
l = the total effective length of a diffuser (m)
v = kinematic viscosity of inlet water (at 4°C v = 1.566 mm^2/s)

Mixing of chilled water storage is minimised by designing diffusers for low Reynolds numbers. The lowest Reynolds number is needed for short and small diameter tanks with sloping sidewalls, the lower limit of about 200 is recommended. For tanks deeper than 5 m the Reynolds numbers from 400 to 850 have been suggested. The Reynolds number can exceed 2,000 in tanks deeper than 12 m, but for design purposes a maximum of 2,000 for Reynolds number should be used. In the tests with many large tanks (Amy Musser; William P. Bahnfleth, ASHRAE (4294) 1999) it has been noted that higher Reynolds numbers (> 6,000) can be used. These larger Reynolds numbers enable low-cost diffuser constructions.

The Froude number is the dimensionless ratio of inertia force to the buoyancy force on a fluid. Small Froude number is important in the thermo-cline formation. Yoo et al (1986) verified that with a Froude number of 1 or less, the buoyancy force in the diffuser flow is more powerful than the inertial force, and gravity current is formed. With Froude numbers about 2 and above, the mixing becomes significant and the gravity current is destroyed.

The inlet Froude number is calculated as follows:

$$Fr_i = \frac{q}{\left[g \cdot h_i^3 \cdot (\rho_i - \rho_a) / \rho_a\right]^{1/2}}$$

where

q = volume flow rate per unit diffuser length (m^3/s, m)
g = acceleration of gravity (m/s^2)
hi = minimum inlet opening height (m)
ri = density of flow water (kg/m^3)
ra = density of ambient water (kg/ m^3)
q = Q/L (m^3/s, m)

where, Q = maximum flow rate of whole diffuser length (m^3/s)
L = effective diffuser length. The length of pipe that is perforated or slotted to act as inlet nozzle. For example, for a diffuser pipe having nozzles (holes) in two directions 180° apart the effective diffuser length is twice the pipe length.

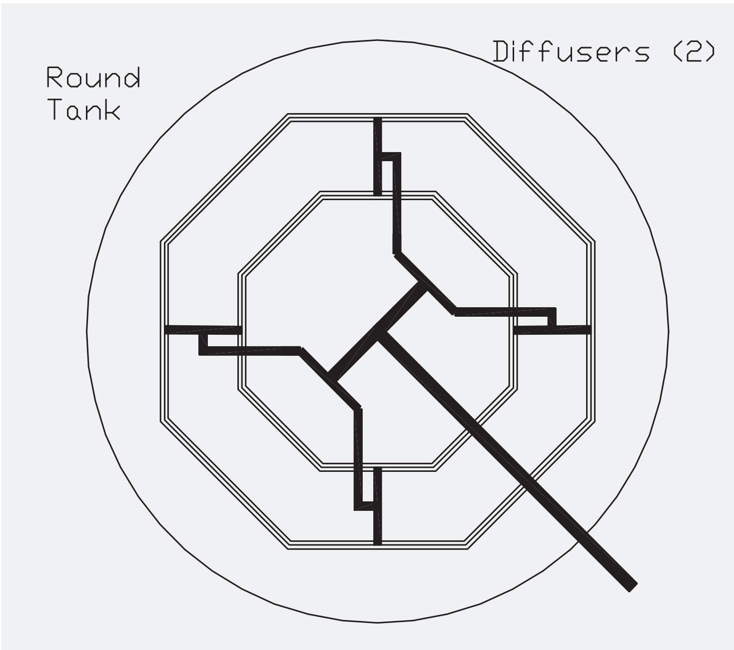
The Richardson number (Ri) is also used in evaluation of diffuser designs. It is the inverse square of the inlet Froude number and so, higher Richardson number than 1 is desired for a proper diffuser construction.

$$Ri = \frac{1}{Fr^2}$$

Diffuser Constructions

As mentioned earlier, several diffuser shapes are found to be good and provide favourable stratification performance. The performance varies in different shapes of storage tanks. Octagonal (or similar round-shape pipe system) and radial disk diffusers are the most suitable applications for round, cylindrical tanks, whereas the straight and rectangular shaped H and slot diffusers are best suited to square and rectangular tanks. Octagonal pipe diffusers, in single, double or multiple systems have been successful in controlled studies (Wildin and Truman 1989) and in many applications in large (~10,000 m^3) storage tanks. The configurations of single and double octagonal diffuser systems are illustrated in Figure 7 below.

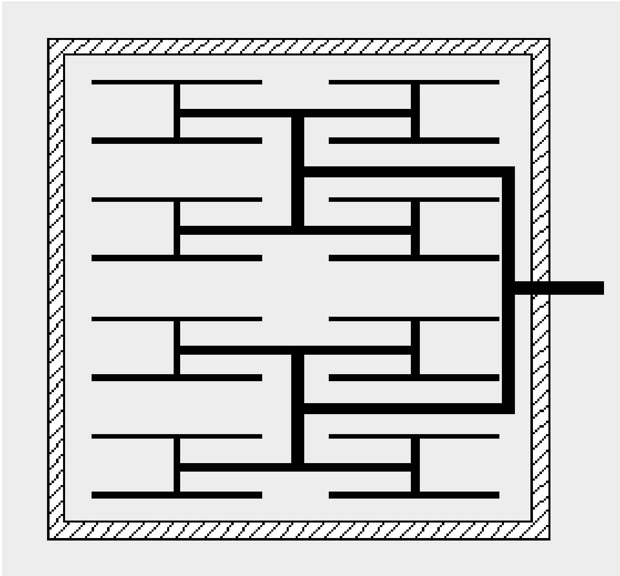
Figure 7
Octagonal diffuser system for a round-shaped tank



The pipe diffuser can be either single-pipe or dual-pipe design. Dual-pipe design has an inner carrier pipe and a slotted outer pipe. The dual-pipe approach enables independent control of flow and inlet fluid velocity along the whole diffuser length. However, single-pipe and radial disk diffusers designed to meet low Reynolds and Froude numbers can provide sufficient performance with much lower cost than more complex designs do.

Continuous horizontal slot diffusers with two discharge directions (180° apart) are often installed along the centreline of the tank. In large square or rectangular shape of tanks the total diffuser length is increased installing several diffuser bars side by side. This arrangement often forms H-shaped systems on square tank surfaces.

Figure 8
H-shaped diffuser system for a square storage tank

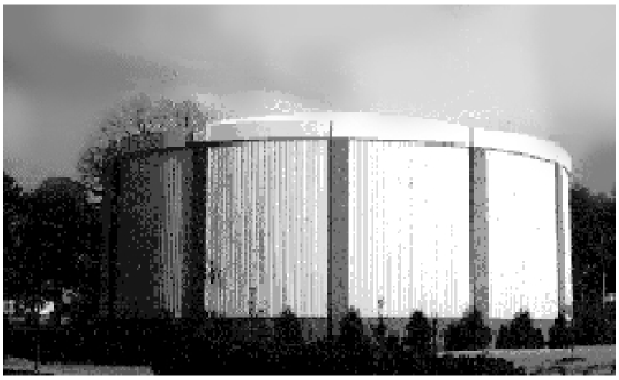


A radial disk diffuser is formed by two closely spaced disks, horizontally mounted. Radial diffusers have been used in many stratified water installations (chilled water and hot water). The radial speed out of a disk-shape diffuser is often higher than that of the octagonal pipe diffuser of the same circumference. The Reynolds number is reduced by increasing the diameter of disks, by increasing the inlet height of diffuser opening and by dividing the opening into several levels and/or sectors. Increasing opening height of the diffuser affects the effective capacity of the tank by reducing the dischargeable height.

Table 1
District-cooling systems with stratisfied chilled water storage

Name, Year of start-up	City, country capacity/m³	Water storage	Operating temperature
Bangi Itakan	Kuala Lumpur, Malaysia	15 000	
District Energy St Paul, 1994	St Paul, Minnesota, USA	9 500	5°C/ 10…°C
Climespace,	Paris City, France	13 000	5°C/ 10..15°C
Climadef, Suclim	Paris La Defence, France	3 800	4°C/ 14.5°C
Expo 98	Lisbon, Spain	15 000	4°C/ 12°C
TRIGEN, 1994	Chicago, USA	32 170	
TRIGEN, 1988	Trenton, USA	9 840	4.4°C/ 15.6°C
Edwards AFB, 1986	USA	16 000	
California State Univ., 1993	USA	10 000	
Västerås Energi och Vatten	Västerås, Sweden	4 000	
Vattenfall/Uppsala Energi	Uppsala, Sweden	4 000	

Figure 9
Chilled water storage of 2.25M-gal capacity chilled water storage (CWS) (Source: CERL)



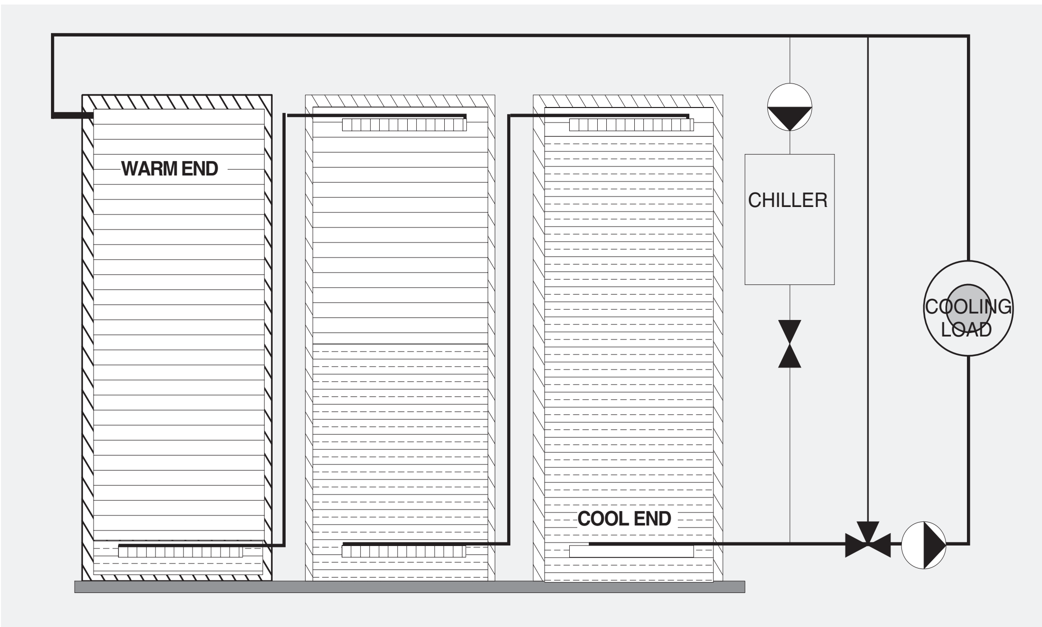
Brines for Low Temperature Stratification

Problems caused by relative high operating temperature, +4°C, of a stratified water tank can be solved by adding brines or other anti-freeze substances. The density of brine must grow as the temperature comes lower to make the stratification possible.
Calcium chloride, sodium chloride, glycol, sodium nitrite and/or sodium nitrate can be used to make a brine that stratifies right and has a freezing point below 0°C. Brines containing chlorides are corrosive and inhibitors are needed in most solutions.
Sodium nitrite and/or sodium nitrate brines are introduced by Trigen Energy Corp. (USA) to avoid the corrosion problems of chlorides and environmentally unacceptable inhibitors and glycols. With environmental concerns rising, the use of glycols or salts may become subject to stringent rules regarding chemical contamination of soils.

Multiple Tank Systems

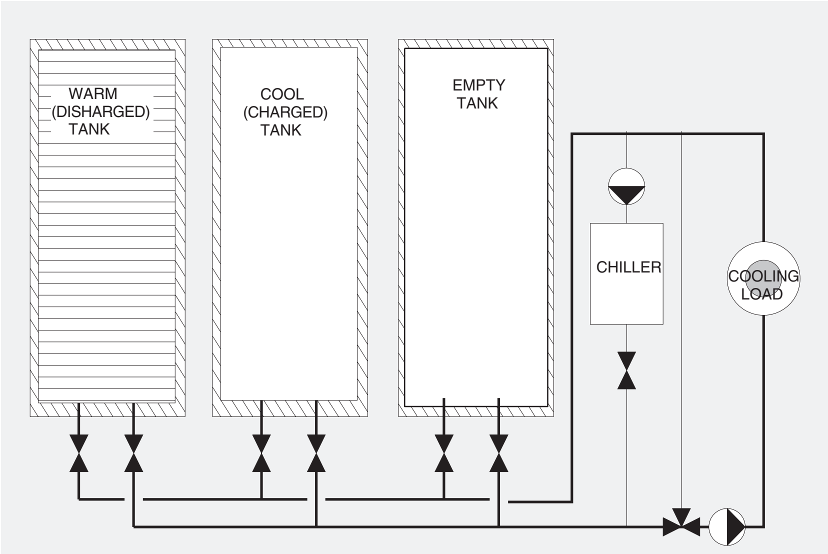
As the stratified tank takes high installation rooms and is very sensitive to charging/ discharging flows and temperatures the multiple-tank systems provide storage capacity which can be placed to low height rooms. The series tanks are typically connected with pipes from the bottom of a tank to the top of another tank. If water flows from the bottom of one tank to the top of the next the effect is similar to a very high tank. In multiple-tank systems the water charged goes first to the coldest chamber of the system and any possible mixing is limited within this one tank. Typical connections are simple baffles, which turn one large tank into several smaller tanks in series.

Figure 10
Multiple storage tank installation



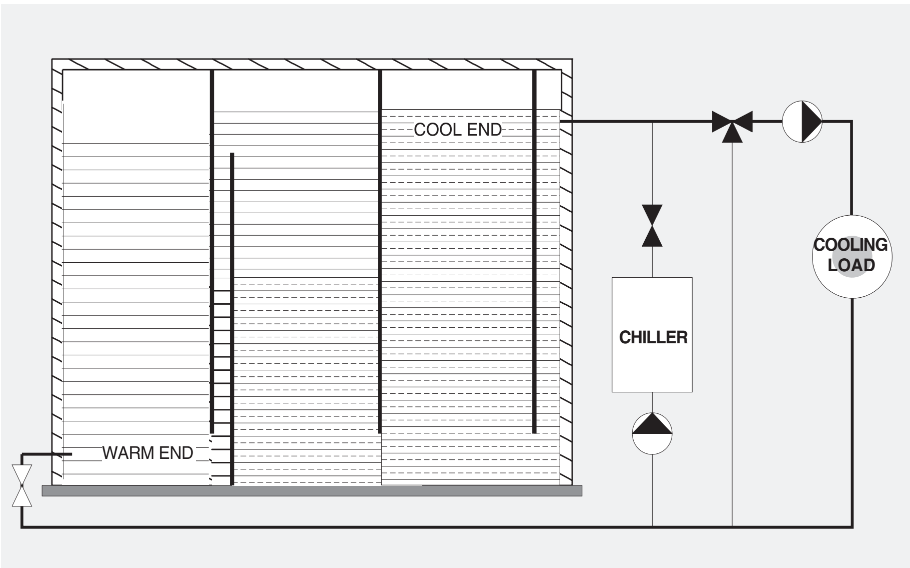
The mixing is almost totally avoided in empty tank systems. The empty tank system consists of two or more tanks, which are used sequentially, see Figure 11 below. One tank volume of the group of tanks is always kept empty for the return water. During the discharge of the storage the warm return water is pumped into an empty tank. By the time this first empty tank is filled the tank previously filled by chilled water is empty and ready to be filled by return water.

Figure 11
Empty tank installation



The cost of many tanks connected by piping increases the cost of storage volume but the savings in other structures building may compensate it (for example, when using underground concrete structures like a labyrinth as chilled and sprinkler water storages). A labyrinth solution is a design of many tanks arranged in a way that water from one tank flows to the next and forms a kind of series tanks described above. The two primary differences are the number of tanks and the locations of connections. Numerous labyrinth tanks are of type 'bottom to bottom' and 'top to top connected' Figure 12. Through the tanks the water flows from the bottom of the first to the bottom of the second and from the top of the second to the top of the third and so on through many tanks. This arrangement does not depend in any way on density difference between supply and return water, but rather depends on physical "plug-separation" of the chilled return, the mixed water interface, and the chilled water supply tank.

Figure 12
Baffle storage tank



In practice the labyrinth and many baffle designs suffer from mixing problems due to turbulence, high velocity and gravity currents and thermal conduction of numerous tank walls. It is often so that when the flow rate is low enough to avoid turbulence in a cell, large volumes of stagnant water are formed along the flow path, which reduces the effective storage capacity.

Membranes and Diaphragms in the Tanks

When flow control devices were not sophisticated enough to maintain the stratification undisturbed, rubber-like membranes and diaphragms were used in small size and low shaped single tank storage to separate the chilled water return and chilled water supply. The diaphragm moves up and down using the tank in a way that may be compared with the moving empty tank. This design concept is offered as a solution to the interface problem. However, it does not solve the problem because it does not eliminate thermal exchange between the hot and cold side, as the membranes are not insulated. Additionally, some space is needed to introduce and to withdraw the water, which initially is the interface zone and insertion of any membrane cannot eliminate that space.
A membrane mechanism is also subject to failure. In The University of New Mexico some flexible membrane systems have been tested and it was shown that the membranes do not substantially increase the performance over a good thermal stratification design without a membrane. Stratified and empty tank systems have beaten membranes and diaphragms in most cases. Especially, large dimension storage is suited to stratified tank design.

Brines for Low Temperature Storage and Distribution

Potassium formate (HCOOK) is highly water-soluble, making concentrated brines of high density and low viscosity. Formate solutions are an environmentally safe, high specific heat and low-cost alternative replacing glycol solutions. Corrosion inhibitors must be added to avoid corrosion by potassium formate. These potassium formate based refrigerants for indirect cooling systems are produced, for example, by Kemira Chemicals Oy, Finland under the trade name of Freezium TM and by Tyforop Chemie Gmbh, Germany under the name of Tyfoxit®.
Aspen Petroleum Ab (Sweden) presented brine-containing alkali salts of acetic acid and formic acid in 1999. This brine called Temper® has low viscosity and favourable corrosion properties.
Organic media like natural betaine (Thermera® by the Finnish Fortum Ltd.) enable an environmentally sound way of transmitting low-temperature cooling to the places where toxic anti-ice media cannot be used. The prices of alternative anti-ice media are in the beginning higher than the price of glycols because of the positive market situation of environmentally sound alternatives. The waste treatment cost of ethylene glycol helps the marketing of natural anti-freeze.
In any case, the cost of anti-freeze substances is significant in large volume district cooling systems, and the effects on heat transfer capacity and the operation of heat exchangers may reduce the economy of low-temperature liquid storage.
The reduction in the freezing point temperature of storage media must compensate the reduction in thermal capacity caused by anti-freeze media and the temperature difference over the heat exchanger needed to separate the storage liquid from the pure water distribution system. The use of brines makes it possible to use lower temperatures also in distribution, and higher DT reduces the flow rate needed. The viscosity of brines is often high, which reduces the turbulence and pressure loss in piping but unfortunately this also happens in heat exchangers at customer facilities. Large low-temperature distribution system systems take precautions on the customer side to avoid freezing at the secondary of the heat exchanger when secondary loop is a water circuit. A secured three-way control valve and a mixing pump on the primary side may be needed. The distribution and consumer systems are described in the following chapters.

Table 2
District-Cooling Systems with Brine storage

Name, Year of start-up	City, Country	Brine Storage Capacity / kWh (m^3)	Operating Temperature
McCormic Place,1994	Chicago, USA	430 000 kWh 32 000 m^3	-1°C/12°C
TRIGEN, 1998	Kansas City, USA	Low temp distribution network	0.5°C/12°C
Orlando Utilities, (under construction)	Orlando, USA (Florida)	880 000 kWh 64 000 m^3	1°C(4.5)/14°C

Summary of Chilled Water Storage Tank Design

Because of the decreasing unit cost of the tanks, chilled water storage can be economically attractive in larger systems. The disadvantages of chilled water storage are related to the large tank volume. The weight, location and space requirements of the storage tanks can pose some unusual problems in the design. The weight of a high water-filled tank can be 5,000 up to 30,000 kg/m^2, which requires solid foundations. By distributing the storage volume to more than one tank the problems of space and foundation can be helped. Storage tanks are normally installed at the ground level on an inflexible foundation secured against inclination and seismic activity when needed. The foundation of a non-pressurized tank is made of steel reinforced concrete cast. The base of a field erected welded steel tank is often secured by a layer of concrete but in certain cases just condensed and well drained gravel will do. On-site constructed water storage systems usually take at least three months to complete.

A welded steel tank is 100% waterproof. However, the condensation can make the outer surface wet if insulation is not made properly. Corrosion is exited by the wet surface and insulation like mineral wool can be soaked, which increases the thermal conductivity of insulation layer on extended areas. Plastics like cellular plastic and polyurethane are suitable insulation materials, because of low operation temperature of the storage tank and small insulation thickness needed. In case of fire the plastic insulation material is an increased risk.

Factory-assembled steel tanks are available in sizes under 1500 m^3. Larger above ground tanks are most economical to be made at site out of prefabricated (cut, bent and coated) plates and tank heads.

Tanks can also be made of concrete, either pre-cast, pre-stressed or cast-in-place concrete. The walls of concrete tanks have higher insulation properties than steel tanks and the structure is easy to locate underground - fully or partially depending on the place of installation. The investment cost of underground storage is often high, up to three times the cost of an above ground installation. The total cost of underground storage depends of course on the quality of ground and the further use of the top of the storage. When the cost of building ground is high, the price of saved building floor area can compensate the higher construction cost. An underground tank in well-drained soil reduces the external heat gain in warm atmosphere. The leakage of a concrete tank is small but it has effects on water treatment and insulation. The maximum allowed leakage in accordance with American Water Works Association (AWA) standard D110 is 0.1% of the tank volume per day for concrete tanks but for steel tanks no leakage is permitted. When the storage contains glycols or brines, the tank must be 100% liquid proof to avoid leakages to the ground and ground water. Plastic (polyethene, fibreglass) tanks provide structural and chemical strength and perfect tightness for use of chemical substances. For outdoor installation the material must be UV-stabilised plastic or the jacket of insulation should protect the tank against the sunlight.

When sizing the especially stratified storage systems one must note that heat is transferred across the interface between the warmer chilled water return and the colder chilled water supply, the amount of time the tank will sit idle should be held to a minimum. Water is a good insulator; nevertheless, it does conduct heat. Also, any heat gains to the storage tank can set up unwanted mixing currents within the tank when idle. The effective use of a smallest possible tank leads to the best storage efficiency.

Low bottom temperatures increase the volumetric capacity of chilled water storage but in when designing stratified water tank using pure water it must be noted that the density of water decreases as temperatures approach 4°C and that water starts getting lighter below 4°C. This does not mean that storage systems cannot be designed with lower than 4°C but it means that the return temperatures must be correspondingly higher to reach the density difference required. When temperatures are close, slightly warmer water in the interface may be denser than the colder supply water and will tend to sink down into the supply and cause some mixing until the maximum density point of 4°C is reached. In any case, the freezing of pure water is close when operating with low storage temperatures (+1°C). The freezing problem of water at +4°C can be overcome by using antifreeze substances, which also increases the density of solutions formed.

The biggest advantage of chilled water storage is the possibility to utilise (existing) standard water chillers operating at high COP. Low installed cost per MWh is advantageous with large storage capacity (> 5 MWh).

Table 3
Comparison of different types of water based storage

STORAGE TYPE	ADVANTAGES +	DISADVANTAGES -
Stratified tank (Chilled Water)	+ Low installation cost + Low operation cost + Easy to operate + Easy for retrofit installations	- Large space needed - High supply temperature (+4°C)
Stratified tank (Brines)	+ Low installation cost + Easy for retrofit installations + Relatively high capacity	- Space demand is high - Reduced heat transfer properties compared to chilled water
Multiple tank (Chilled Water)	+ Low operation cost + High charge/discharge potentia + Easy for retrofit installations	- Large space needed - Tank operations can be complicated - High supply temperature (+4°C) - High investment cost (depends on case)
Multiple tank (Brines)	+ High charge/discharge potential + Easy for retrofit installations	- Reduced heat transfer properties compared to chilled water - Tank operations can be complicated - High investment cost (depends on case)
Aquifer (Chilled Water)	+ Low installation cost, existing natural reservoirs are utilised (+Low operating cost) +Combination of heat sink and source (heat pump)	- Feasibility is much up to local ground conditions - High supply temperature (+4°C)

Ice Storage

Ice thermal storage is today a technical and commercial way to store cooling energy in the form of ice at its freezing point, 0°C. To produce this energy refrigeration equipment must operate at lower temperatures than usual to provide storage-charging fluid at -6 to -3°C, depending on the duration of charging cycle.

The working principle in the ice thermal storage is that the technology makes use of the latent heat of water (i.e. the heat required transferring water from a solid to a liquid state). Refrigeration equipment for ice storage technology must operate at temperatures well below the normal operating range for air conditioning applications. Depending on the ice storage technology, either special ice-making equipment or standard production chillers configured for low-temperature duty can be used.

The main advantages with thermal storage in district-cooling systems is the possible reduction of the peak electric demand and/or reduction of installed chiller capacity. This can be achieved with both chilled water storage as well as other storage such as ice storage. Normally the ice storage has a 15-20% higher initial cost than the chilled water storage. However, while the ice storage needs 4-6 times less volume than the chilled water storage, the ice storage still can be the optimal solution when building space is limited or expensive. The ice storage can also, as for chilled water storage, be advantageous when the difference between day and night electric rates are high and thereby providing an incentive to shift larger part of the cooling production from on-peak to off-peak periods. Due to the larger storage capacity per area unit compared with cool water storage, ice thermal storage can be preferred despite the higher investment.

There are a number of different technologies for ice storage. The main technologies, which will be described in this study, are:

- Ice-on-coil
- Sheet ice harvester
- Encapsulated ice
- Phase-change materials
- Ice slurry

While phase-change materials can be other than water and ice they are described under this section due to similar use of the latent heat in the material.

Ice on Coil

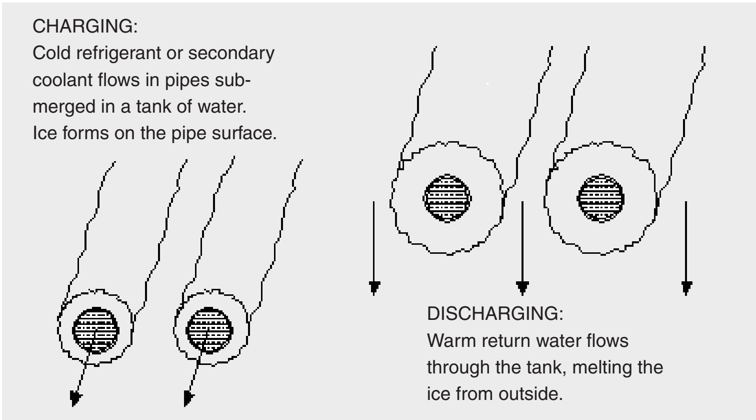
There are two applications of the ice-on-coil technology, external melt and internal melt. Both alternatives use the technique that ice is formed on the outer surface of pipes or coils, which are submerged in a water-filled tank. During the charging period a refrigerant or secondary coolant such as glycol solution (typically 25% propylene glycol and 75% water) or another freeze suppressing fluid is circulated through the coils of a temperature below 0°C and builds ice on the outside.

For the external-melt application the ice storage system requires a chiller capable of generating charging temperatures of -7 to -3°C for building 40 mm thick ice. For a system building ice to 65 mm thick, charging temperatures of -12 to -9°C are required. Internal melt systems use standard chillers selected to generate charging temperatures of -6 to -3°C.

External Melt

During discharge of the external-melt system warm return water generated by the heat load in buildings circulates through the ice tank melting the ice from the outside, whereby the water becomes chilled again. Air is bubbled through the tank at the beginning of the charging cycle and during discharging, to promote uniform ice building and melting and to equalise the water temperature.

Figure 13
Principle of operation of external melts



The ice is built on the coils to a thickness of up to 65 mm during the charging period. Control systems are used to limit ice thickness and also to prevent bridging of ice between individual coils. Bridging restricts free circulation of water and reduces the performance creating a higher leaving water temperature due to the reduced heat transfer surface.

Tank configurations

The storage tanks used for the external-melt system are open and non-pressurized, which requires some kind of static pressure control in the system, in order to maintain the desired static pressure in the district-cooling system and to prevent overflowing of the open tank. Control of pressure can be accomplished with heat exchangers or with pressure sustaining valves and pumps.

The required water treatment system for cool thermal storage is not fundamentally different from non-storage system. Special water treatment and corrosion protections may, however, be needed in external-melt configurations since the water in the tank is aerated. When using heat exchangers for pressure control the distribution system remains closed and special water treatment is not needed.

External-melt storage can also be arranged with multiple tanks. Multiple tanks are typically connected in parallel. A serial arrangement is sometimes used in systems requiring high discharge rates, where a longer residence time of water in the storage tank is desired. However, flow between tanks is driven only by difference in water levels. A connection with a large cross sectional area is necessary to minimize the pressure drop and the water level difference between the tanks. Series connections of more than two tanks are generally not recommended.

Operation strategies

External-melt ice storage technology is best suited to systems that require pure water without additives in the cooling loop or require supply water in the range of 1-2°C, and it is normally used in process-cooling and district-cooling applications.

Technology, Storage

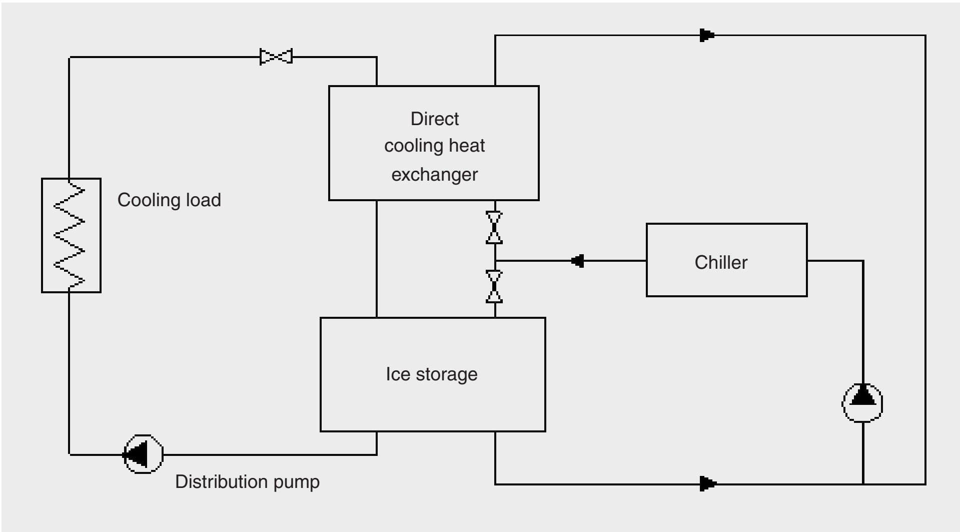
Two different configurations of external-melt systems are used:

- Systems that use refrigerant directly in the storage coils.
- Systems that build ice indirectly using a secondary coolant cooled by a chiller.

The direct refrigeration system builds ice with the storage pipes acting as an evaporator, meaning that the refrigerant flows through the storage. This is widely applied in industrial applications. The secondary coolant system is most common for HVAC applications, because the refrigeration plant is simpler to design, install and maintain, and because the volume of refrigerant is less. The direct refrigeration system is more efficient as there is one step less of heat transfer between the refrigerant and the ice-building surface.

Figure 14 illustrates one configuration of secondary coolant based external-melt ice storage. In this system configuration, the storage tank is directly connected to the distribution system. During discharge of the storage, a heat exchanger allows the chiller to provide direct cooling to support the cooling from the storage. A possible second chiller in the chilled water loop can also provide supplementary cooling.

Figure 14
Secondary coolant external-melt ice-on-coil configuration

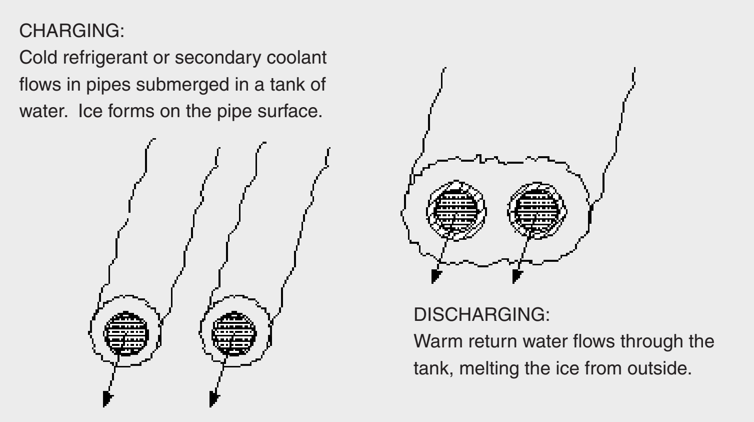


Minimizing bridging of ice between the coils in the tank and limiting the ice thickness during the storage-charging period are together two main concerns to the control of external-melt ice-on-coil technology. To maximise the efficiency and minimise the energy consumption and cost all ice should be melted at least once a week. Under partial storage load the charging cycle should be limited so that only the required amount of cooling energy is stored. Discharging all ice also prevents ice bridging. The ice thickness control system should have an ice thickness override control, which stops the compressor at the predetermined thickness. There are several methods of controlling the ice inventory, the most common is based on the fact that ice has a greater volume than water and a sensor that indicates that change in water level can be used to control the amount of ice stored. Another method consists of a fluid-filled probe positioned at a desired distance from the coil. As the ice builds, it encapsulates the probe, causing the fluid to freeze and apply a pressure that can be measured. At least one ice thickness device should be installed per ice bank.

Internal melt

The internal melt cool storage is discharged as the warm return water flow circulates through the coils, melting the ice from the inside. The ice in the boundary of the pipes is melted first and a ring of water is created between the pipe and the remaining ice. The discharge continues until the ice cracks and the water and ice mixes in the tank.

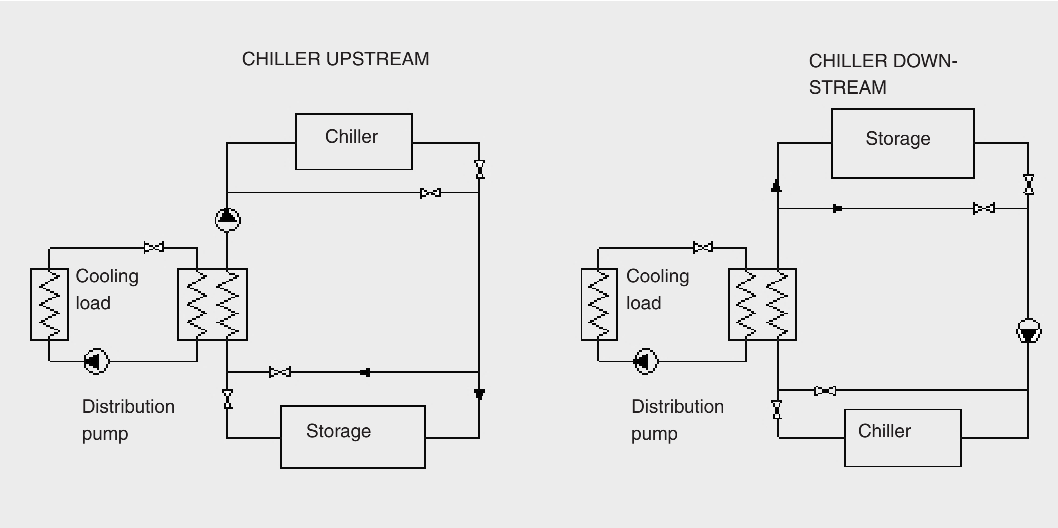
Figure 15
Principle of operation of internal melts



There are two configurations of an internal melt system, illustrated in Figure 16, where the storage is in series with the chiller, either in an upstream or downstream arrangement. For the upstream arrangement the chiller precools the warm return fluid from the building load before entering the storage tank. This results in a more efficient chiller operation due to higher operating temperature. However, the lower storage entering temperature reduces the storage capacity.

In the downstream configuration the return water enters the chiller after it has been chilled in the storage tank. This arrangement provides a higher usable storage capacity, as well as an assured constant supply temperature. The chiller efficiency is, however, lower and the chiller operates at a lower entering temperature.

Figure 16
Chiller upstream and downstream configurations



Materials in coil pipes

The most common material used for coils in commercially available internal-melt ice-on-coil storage tanks is polyethylene with poor thermal conductivity (0.31 W/m.K). Analysis of coil materials has been made to compare polyethylene with other materials with different thermal conductivities, materials with high thermal conductivity (372 W/m.K) as copper and with low thermal conductivity (1.73 W/m.K) as nickel, magnesium, or titanium alloy as well as different plastic materials. The result of the analysis is that coil materials made of metal alloys (e.g. nickel, magnesium, titanium) or formaldehyde plastics (e.g. urea, melamine, phenolic plastics, polyester) improve the heat transfer, decreasing the time to charge and discharge the tank. However, the cost of these materials is generally higher than polyethylene. Future developments may make metal alloys both inexpensive and even more flexible to form coils for use in internal-melt ice-on-coil storage technology.

Tank configurations

In thermal storage applications the cooling loop can be directly connected to the cooling loop of building or separated with a heat exchanger, as illustrated in Figure 16. In district-cooling applications separated systems are preferable. However, as internal melt uses a glycol solution in the coils it is well suited to the systems that utilise glycol in the building cooling loop and can therefore be more efficient than external melt.

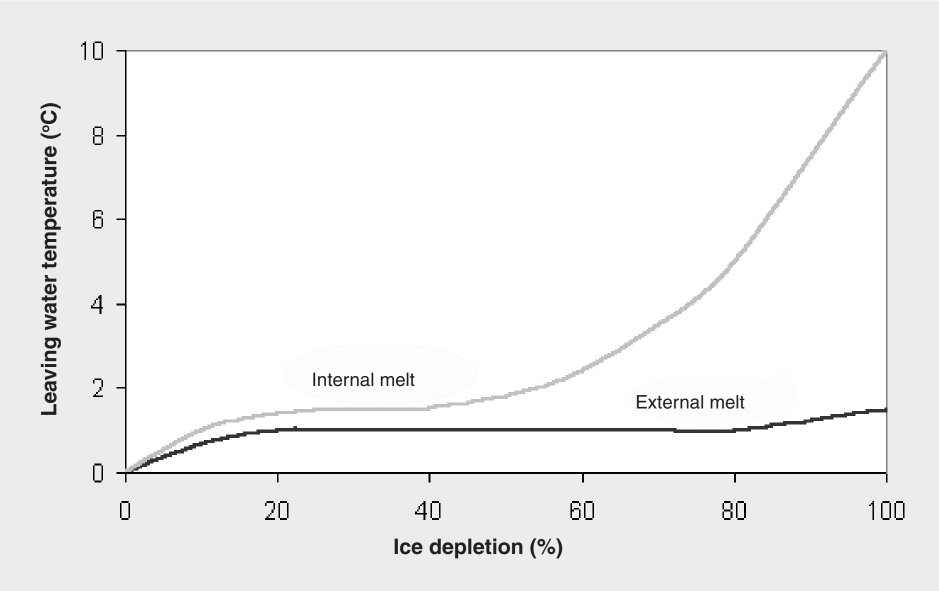
To have multiple small tanks connected together is more efficient than one single large tank due to the long coils and the large number of coils that comes with a large tank. The long coils have a detrimental effect on the thermal efficiency since the temperature gradient between the secondary fluid and the water/ice interface becomes small as the length of the coil increases. An increase in the number of coils causes a decrease in the flow rate per coil, resulting in a reduction in the connective heat transfer of the secondary fluid. With smaller tanks the time for charging and discharging will be reduced.

The internal-melt ice-on-coil storage tank also has low maintenance costs due to the lack of internal movable components. Additionally, the modularity of this system provides flexibility, addition of tanks in parallel is straightforward with low installation costs.

Charging and discharging: ice on coil

The difference between the two applications, external and internal melt, is the way to melt the ice. The difference can be illustrated in Figure 17 below, which shows the temperature change during the ice depletion mode. External melt provides a constant discharging temperature of about $+1^{o}C$, which is a result from air bubbles added in the tank to promote uniform ice building and melting that enhance the heat transfer. For the internal melt the ice near the coil melts and a ring of water is created between the coil pipes and the ice, which continuously grows as the ice melts. As the water isolates the pipes, which increases the resistance to heat transfer, the temperature of the glycol solution increases with time.

Figure 17
The leaving temperatures during the discharge period

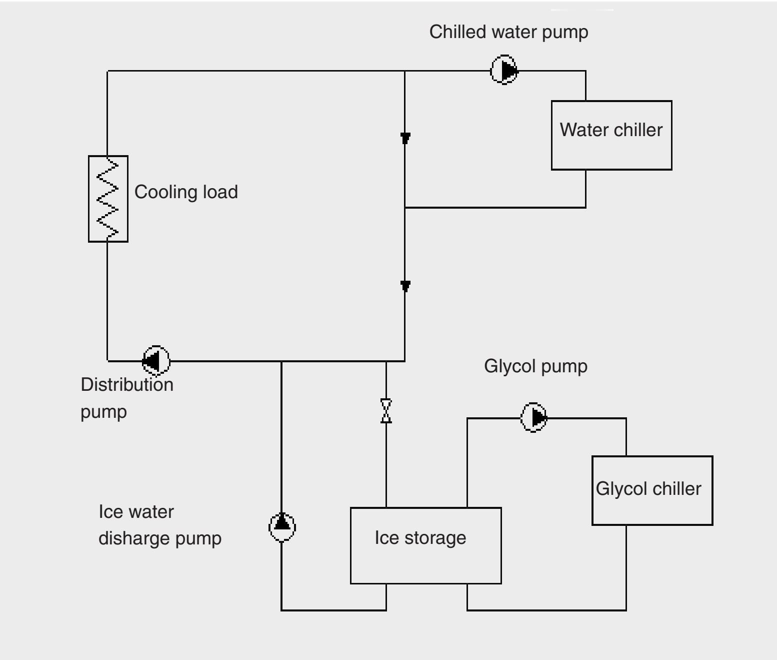


Existing system

In district-cooling applications the external-melt ice-on-coil storage is the technology most extensively used among all ice storage technologies. External melt is best suited to process-cooling applications and large district-cooling systems. Internal melt is more used for single HVAC applications such as hospitals, individual buildings and industries.

Figure 18 shows the district-cooling system in Windsor, Canada. The system has one glycol chiller that generates ice for the ice thermal storage. The external-melt ice-on-coil storage consists of two storage tanks with three respective four-coil bundles. There are also three water chillers that are directly connected to the district-cooling system. The storage is used to reduce the chiller operation during periods with high electric rates. At peak load conditions, when the capacity from the storage is less than the cooling load, the water chillers are also operated.

Figure 18
District-cooling system with ice thermal storage in Windsor, Canada.



The system used in Houston, USA (Figure 19) differs from the system in Windsor by having four screw chillers that both can operate as water chiller with a supply temperature of 4°C and as glycol chiller with a supply temperature at -4°C. The glycol chiller generates ice in an external-melt ice-on-coil storage. Further, there are four centrifugal water chillers that only produce chilled water at 4°C. The storage tank consists of one tank with five coil bundles and has a capacity of 264 MWh and a discharge temperature of 0°C. The plant peak capacity with chillers and storage is about 88 MW.

Figure 19
District-cooling system with ice thermal storage in Houston, USA.

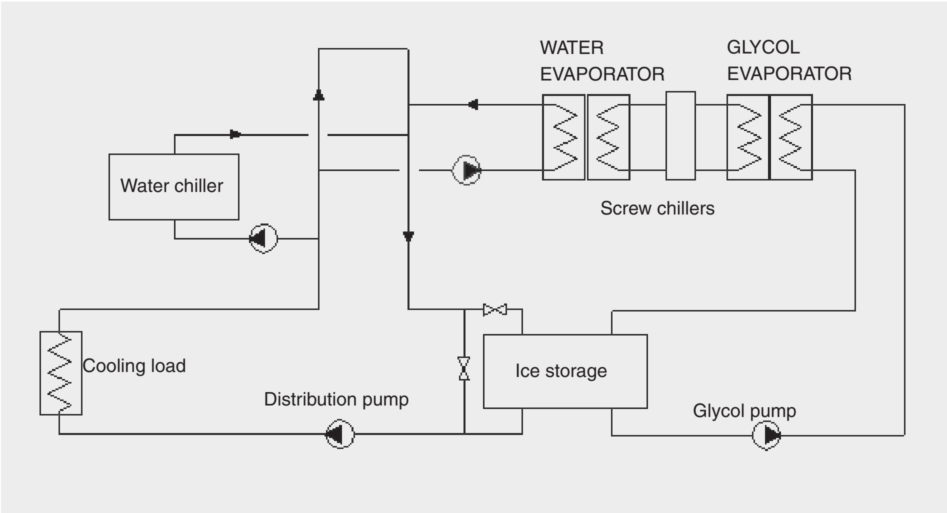


Table 4
Utility district-cooling systems with ice storage

Name	City, Country	Ice storage capacity/kWh
Franklin & Van Buren	Chicago, USA	440 000
Columbus & Randolf	Chicago, USA	341 440
State & Adams	Chicago, USA	232 300
Comfortlink	Baltimore, USA	165 960
Public Service of Colorado	Denver, USA	132 000
Northwinds Boston	Boston, USA	112 640
Cosmo Square	Osaka, Japan	103 136
Alamodome	San Antonio, USA	67 936
Ravens Football Stadium	Baltimore, USA	45 760
Chaffuage Urbain Prodith	Lyon, France	30 000
Windsor Ontario	Windsor, Canada	29 920

Sheet ice harvester

The sheet ice harvester builds ice on a flat plate evaporator or on the inside or outside of a cylindrical evaporator surface that is positioned above a water/ice storage tank. Ice is generated by circulating water at 0°C from the storage tank to the top of the evaporator plates where it flows freely down in a thin film outside the plates. The ice sheets are after released, often by circulating hot refrigerant gas in the evaporator, and falls down into the storage tank where it is mixed with the cool water. Other types of ice harvesters use mechanical means to separate the ice from the evaporator surface. The ice is released periodically when the thickness reaches 6-10 mm.

Figure 20
Ice harvester schematic

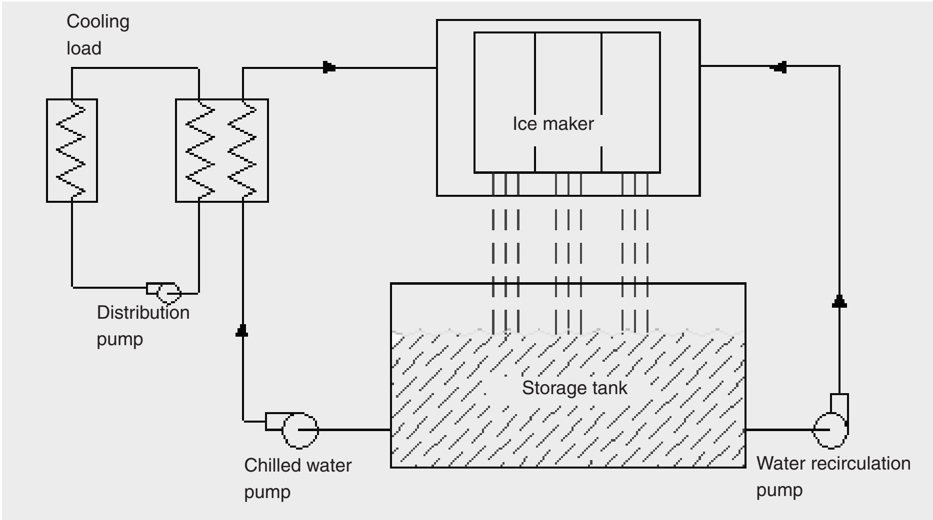


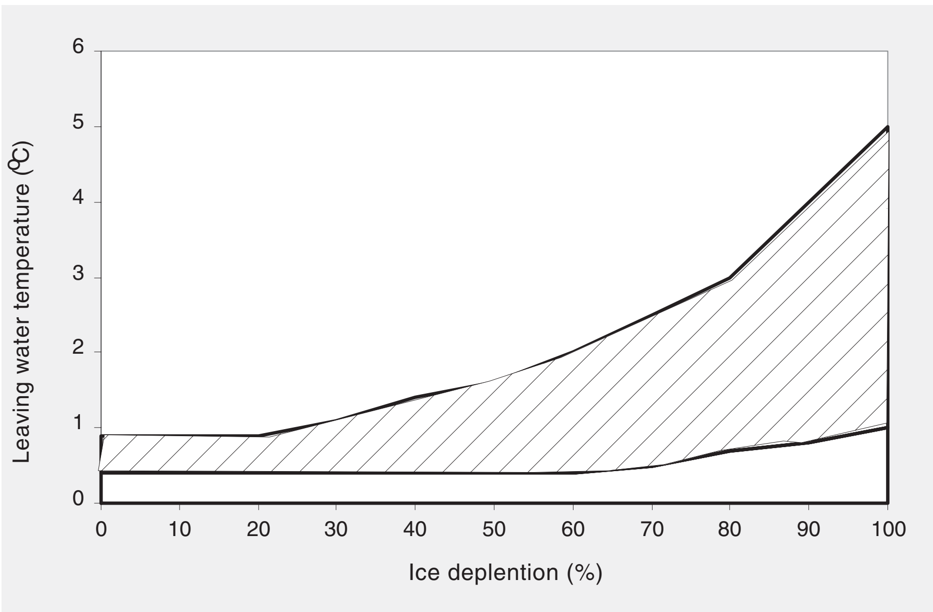
Figure 20 shows an ice harvester schematic. A recirculation pump is used to provide minimum flow for wetting the evaporator.

Charging and discharging

The charging performance of an ice harvester system remains the same throughout the charge cycle, independent of the amount of ice in the storage tank, which is unlike the other ice storage technologies. The ice charging cycle continues for 10 to 30 minutes. The defrost time depends on the control method, the evaporator configuration and the discharge conditions of the compressor, it is in the range of 20 to 90 seconds. The stored ice can then be melted very quickly, if it is properly wetted. A 24-hour charge of ice can be melted in less than 30 minutes for emergency cooling demand [5].

Discharge temperatures from a properly designed storage tank can remain about 1 to 2ºC until 80- 90% of the ice is depleted. At that point, the contact area between ice and water in the tank is reduced enough that the temperature rises. Figure 21 shows a typically range of discharge temperatures over the discharge cycle.

Figure 21
Ice harvesting discharging temperature range



The ice harvester can operate both as an icemaker and a water chiller. The selection of ice making or water chilling is made automatically, depending on the temperature of the water entering the evaporator. If the water is at or is near the freezing point, the ice-making mode is selected and the defrosting cycle is activated at intervals to release the ice. In the ice-making mode the ice harvester produces evaporating temperatures in the range of -9 to -3°C and in the chilling mode 1°C or higher.

The harvester separates ice generation from the storage, which makes it flexible to operate in off-peak periods. Since the ice is not stored at the ice-making surface it is not necessary to melt the complete ice inventory every day to maintain high operation efficiency.

The ice harvester and the storage tank are open to the atmosphere, which makes water treatment and corrosion protection necessary since the water in the tank is highly aerated due to repeated recirculation over the evaporator. The tank design is though simpler with no internal components compared to the ice-on-coil system. The tank geometry can, however, affect the storage capacity since the angle of repose of the ice when it fills the tank will leave some void space that does not contain any ice and thus consequently reducing the amount of ice that can be stored in the tank.

The tanks are often site-built of concrete in a rectangular shape. However, modular concrete, steel or fibreglass tanks can also be used. The evaporator plates are normally made of stainless steel.

Existing system

Table 5
Existing ice harvester system [8]

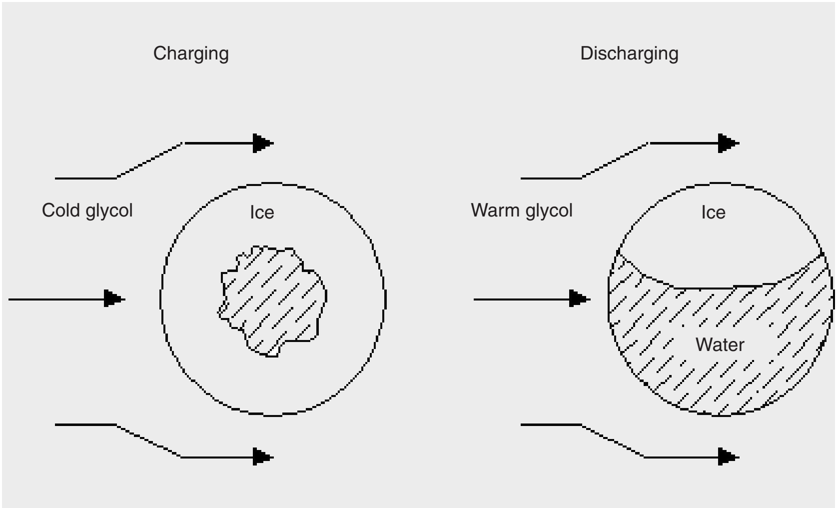
Name	City, country	Ice storage capacitykW	
General Mills Technical Center	Minnesota, USA	3 851	Office building
Rohm and Haas Research	Philadelphia, USA	4 502	Multiple building
Miller Electric-Arc Welding Equipment Manufacturing	Wisconsin, USA	2 286	Multiple building
National Science museum	Taiwan	1 407	Museum building
Texaco's Sargent Canyon	California, USA	1 477	Gas turbine inlet air cooling

Encapsulated Ice

This technique is based on small containers that are filled with water and that are submerged in a storage tank. The water inside the container freezes and the ice thaws as warm secondary coolant, such as glycol/water circulates through the tank. The charging temperature is between -6 and -3°C.

The plastic containers may be flat, rectangular, spherical or annular. In the US there are two designs available: rectangular containers (17 and 4.2 litre) and dimpled spheres about 100 mm in diameter. In Europe rigid spheres are also used with 95 mm and 75 mm in diameter. The number of containers required for an application depends on their individual storage capacity. For example, 3.5 kW storage can be provided with approximately 70 spheres with the diameter of 10 cm. Figure 20 shows a principle of operation of encapsulated ice.

Figure 22
Principle of operation of encapsulated ice



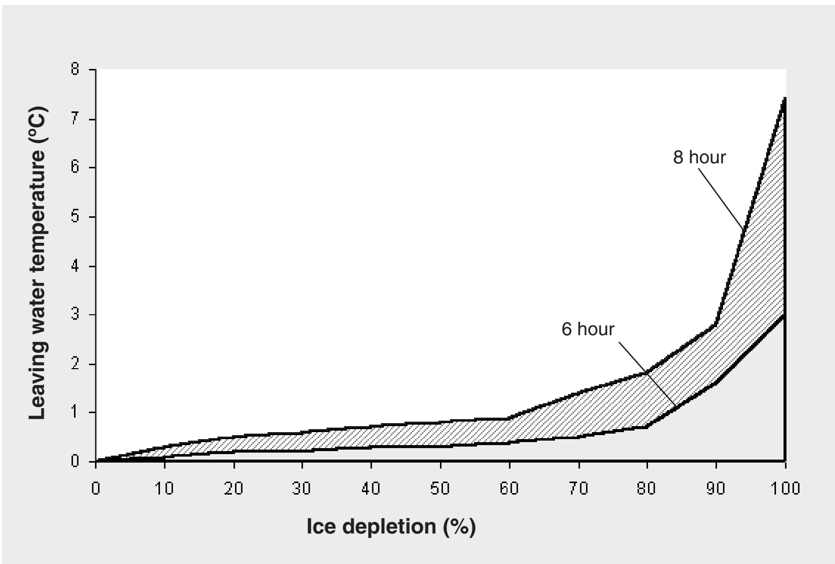
Encapsulated ice storage requires approximately 0.019 to 0.023 m^3/kWh tank volume of available stored capacity. The tank can be open and non-pressurized or pressurized made of steel, concrete, fiberglass, etc. The containers are made of high-density polyethylene and are designed to accommodate the expansion of the freezing ice. In tanks with spherical containers the secondary coolant flows vertical and with rectangular containers secondary coolant flows horizontal. The type, shape and size of the storage tank is limited only by its ability to achieve even flow of heat transfer fluid between the containers.

An encapsulated ice system can operate in either the chiller upstream or the chiller downstream configurations, both of which are equivalent to the internal-melt ice-on-coil technology, see Figure 16. Arrangements with the tanks in parallel with the chiller during discharge are also possible.

Charging and discharging

Encapsulated ice has a steadily falling discharge rate if a constant discharge temperature is maintained or a steadily rising temperature if a constant discharge rate is maintained. This characteristic results from the decreasing area of ice that is in contact with the container walls as the ice melts. Figure 23 shows a typical range of discharge temperatures for constant discharge rates over periods of 6 to 8 hours with an inlet temperature of 10ºC.

Figure 23
Encapsulated ice discharging temperature range



Encapsulated ice containers are subject to supercooling or cooling of the liquid water below its freezing point before the initiation of ice formation. Supercooling, which only occur in fully discharged containers where no residual ice remains results in decreased heat transfer rates at the beginning of charging. By adding nucleating agents the effect from supercooling can be decreased.

Existing system

Encapsulated ice storage technologies have not been commercialised on a large scale. There are some systems in the USA and China, see Table 6.

801 Tower in Los Angeles

Ice ball storage has been operating in a 24-storey office building (40 000 m^2) in Los Angeles, USA, since 1993. The storage capacity is 38 700 kWh cooling from a 681 m^3 storage tank located adjacent to the underground parking garage. The system includes a tank made of concrete with a polyethylene liner and approximately 750 000 spherical ice containers and two 2.5 MW three-stage centrifugal chillers.

Harold Washington Building

In Harold Washington Building in Chicago the existing water storage tanks were successfully converted to ice thermal storage. By altering four 31.8 m^3 tanks measuring 2 metres in diameter and 11 metres long the system has 6 900 kWh storage capacity compared to 1 400 kWh from chilled water storage.

Table 6
Existing system using encapsulated ice storage
(Source: Cryogel)

Name	City, country	Ice storage capacitykWh
801 Tower	Los Angeles, USA	38 700
Harold Washington Building	Chicago, USA	6 900
Beijing Daily	China	n.a
San Francisco Airport	San Francisco,USA	n.a.
Los Angeles Airport	Los Angeles, USA	n.a
Miami International Airport	Miami, USA	n.a

Figure 24 and Figure 25 show some storage tanks for encapsulated ice. Picture 24 shows an atmospheric storage tank by Cryogel. It measures 10 meter in diameter and is 18 metres tall. The storage capacity is 147 800 kWh. In Figure 23 there is a pressurized storage tank used for a hospital. The storage tank is 18 metres long and 3.5 metres in diameter and placed underground beneath a parking area.

Figure 24
Atmospheric tank
(Source: Cryogel)

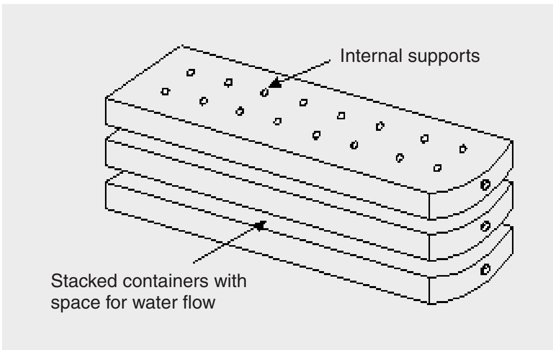
Figure 25
Pressurized storage tank for a hospital. (Source: Cryogel)

Phase-Change Materials

Eutectic salt is a chemical mixture, which changes phase from liquid to solid at a specific temperature. Eutectic salt phase-change materials have been used for various heat storage applications since the 1800s [3], but just recently they have been used as cool storage medium. The most common formulation for cool storage application is a mixture of inorganic salt, water and nucleating and stabilizing agents, which melts and freezes at 8.3°C. This formulation has a latent heat of fusion of 95.4 kJ/kg and a density of 1490 kg/m^3.

This phase-change material can be encapsulated in rectangular plastic containers as shown in Figure 26 or in spherical or annular containers. The storage tank can be open and non-pressurized or pressurized made of steel, concrete, fiberglass, etc.

Figure 26
Containers for phase-change material storage



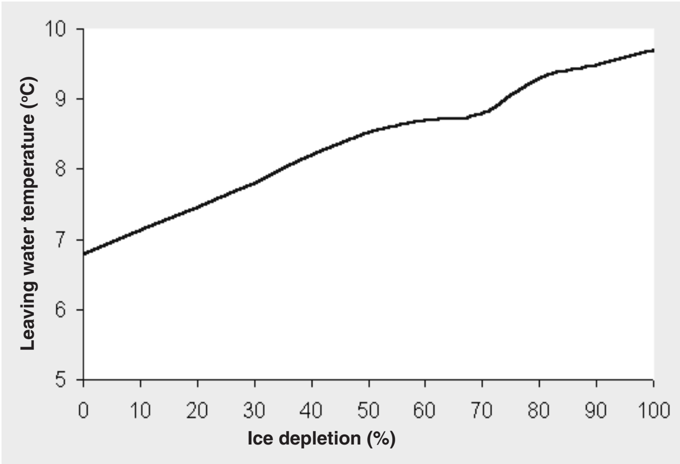
The eutectic salt storage can be charge with conventional chillers generating chilled water at 4 to 6°C. This allows the storage to be connected to an existing system with no modifications to existing chillers and few or no changes in the distribution system. Eutectic salt system can, as internal melt and encapsulated ice system, be arranged in chiller upstream or chiller downstream configurations.

A eutectic salt formulation that freezes and melts at 5°C is currently being developed. The discharge temperature of 6 to 7°C would be compatible with conventional distribution and air-handling systems. Eutectic salt mixtures for lowering the storage temperature are also available for ice storage systems. Internal melt ice on coil systems can utilize phase-change materials with additives where freezing temperatures of -2°C to-11°C can be generated.

The containers filled with phase-change materials do not float, as the density of eutectic salt is about 1.5 times the density of water. Neither does it expand on freezing so that there is no stress on the containers.

Figure 27 below shows the discharge temperatures typical of eutectic salt, the temperature rise steadily through the discharge period, from about 7ºC at the beginning of discharging to about 10ºC.

Figure 27
Eutectic salt discharging temperature range



Existing system

Shenzhen Electronic Science and Technology Building
In the early 1990s, China's first three thermal storage projects began operation, two ice-storage systems and one chilled-water storage system. The Shenzhen Electronic Science and Technology Building located in southern China was the first to install an ice storage air-conditioning system. Encapsulated ice, or ice balls, was used for a building floor area of 62 000 m2. The calculated cooling load was 11 250 kW and the ice storage capacity was 24 190 kWh.

Another example of encapsulated ice storage in China is the Bank of Communication in Hangzhou. The business building floor area is 19 913 m^2 with a cooling load of 1 919 kW. Two screw chillers and an ice storage tank of 138 m^2 are used to produce the ice.

Table 7
PCM storages in operation
(Source: Cryogel)

Name	City, Country	Ice storage Capacity kWh	
MM21 (DHC)	Japan	121 000	Distr. Cooling sys.
Bangsar Energy Plant	Kuala Lumpur, Malaysia	104 500	Distr. Cooling sys.
Aeroport Zaventern	Brussels, Belgium	33 000	
Aeroport Nice, zone 1 and 2	Nice, France	4 400	
Airport	Bangkok, Thailand	4 180	
Bank of Taiyo Kobe	Japan	11 000	
SBS Les Baurnettes	Lausanne, Switzerland	8 250	Bank
Shenzhen Electronic Science and Technology Building	China	24 190	
Hangzhou Bank	China	8 000	
Hangzhou Jinpeng building	China	16 700	
Nouveau Ministere Finances Sea Side Mornochi Cha Hospital	Paris, France Japan Seoul, Korea	24 640 30 525 22 715	Off. building Off. building, DHC

Ice slurry

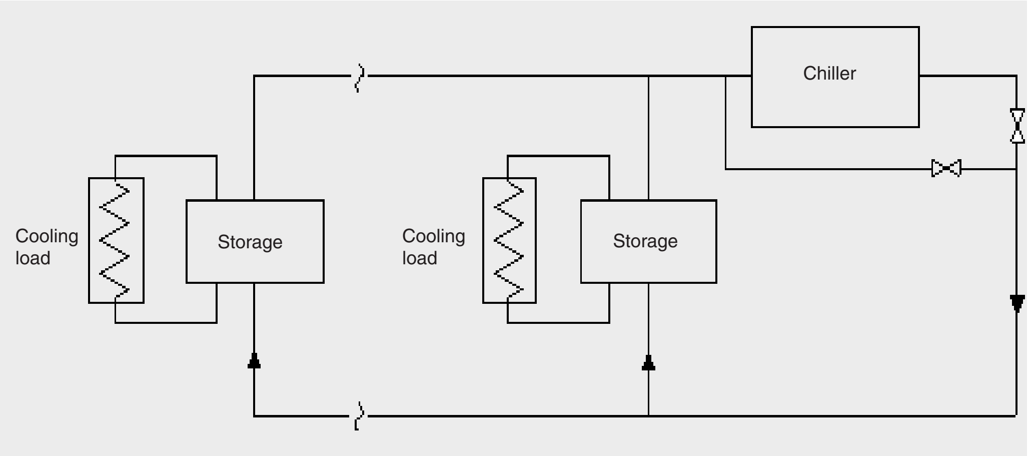
In any ice slurry thermal system, ice particles produced by a generator in a binary liquid (e.g.: water and glycol) are transferred to a storage tank. The particles are then stored with water in the tank. From the tank, this slurry is pumped to cooling loads where the ice fraction of the slurry is melted. The warm return water is then pumped back to the storage tank or ice generator to be re-cooled.

The storage system may have several configurations. Two common scenarios are distributed storage and central storage.

Distributed Storage Systems

In a distributed storage system, ice slurry is pumped to a number of tanks and stored. These tanks are located at each building or group of buildings throughout the district-cooling network. The ice slurry enters the tank where the ice particles are separated by gravity. The building cooling system can then use ice-free water from the storage tank to meet its cooling demand. This is demonstrated in Figure 28.

Figure 28
Distributed ice slurry storage system



The distributed storage tanks provide a buffer between the distribution system and the individual building-cooling load. This decoupling of the refrigeration system allows the distribution system to supply the average cooling load rather than the peak. If the average cooling load of a building is much lower than the peak, smaller-diameter distribution piping can be installed.

To meet the maximum design-cooling load, the refrigeration system must generate ice slurry continuously. This slurry is pumped to the distributed storage tanks. Depending on the time of day, if the building load is low, the ice slurry will accumulate in the storage tank. Later as the cooling load increases, the storage tank is simultaneously charged by the distribution system and discharged to meet the above average building-cooling load.

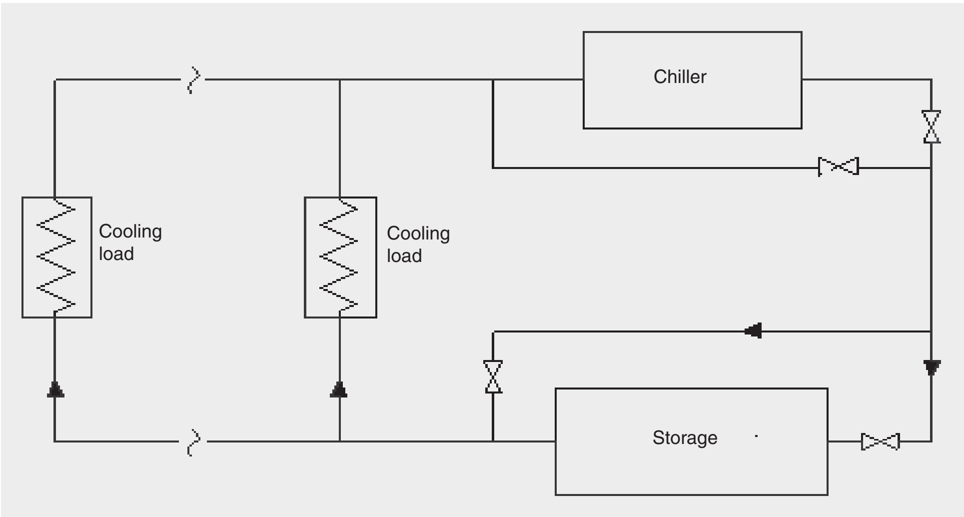
During periods when the building load is small, the return water may be at a lower than optimal temperature with a certain amount of cooling potential still unused. The loss of this sensible cooling during part of the load curve means that the maximum enthalpy difference cannot be continuously delivered to the storage facility. Therefore, the distribution pipes must be designed for a capacity somewhat greater than the average load.

Generally, distributed storage will be most economical if the peak-to-average load ratio is greater than 2 and the latent-to-sensible heat ratio of the ice slurry is greater than 1. Distributed storage also becomes more attractive if the base load is a substantial fraction of the average load i.e. when the peak load is of short duration.

Central Storage System

In the central storage approach, the district-cooling system is equipped with a central storage tank. This tank is located near the central refrigeration plant. The concept is shown in Figure 29. As with distributed storage, central storage retains the advantage of providing a buffer between the refrigeration plant and the actual cooling demand generated by the buildings on the district-cooling network. In a central storage system, the distribution system is not decoupled from the load and the distribution system must follow the actual cooling demand. Therefore, the diameters of the network piping will be larger than with a distributed storage system.

Figure 29
Central ice storage system



Central storage systems can be used with two operating strategies. Ice slurry can be stored but not circulated and used only to cool return water for re-distribution to the cooling load. With this strategy, near 0ºC water is circulated. Alternatively, ice slurry can be stored and fluidised during peak hours for distribution. With either strategy, the volume of storage is reduced compared with chilled water storage. Costs of storage tanks for ice slurry can be reduced by over 60% compared with tanks designed for conventional cooling systems. [13]

To meet the maximum design-cooling load, the refrigeration system must operate continuously. During periods of low cooling, the generated slurry is delivered to the storage tank from which slurry or cold water is removed to meet these low loads. Thus, at cooling loads below the average system load, the cooling capacity in the storage tank increases. As the cooling load reaches the ice slurry generator capacity, all the capacity supplied to the tank is removed at the same time and storage capacity remains constant. During periods when the cooling demand is above the average load, the capacity stored in the tank is removed to meet that load and the tank capacity decreases.

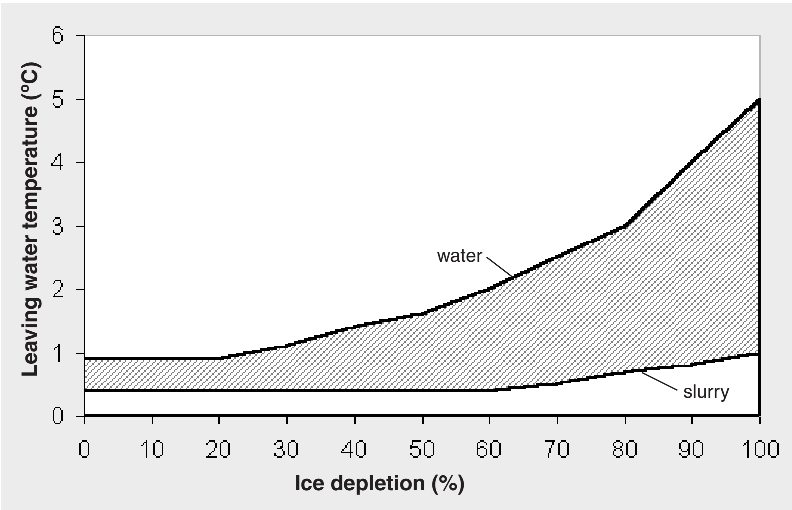
Ice Melting

In addition to the production of ice slurry, the capacity to melt ice requires special consideration. In certain cooling scenarios, water at temperatures just above the freezing point will be transported to the cooling loads rather than ice slurry. In such cases, the ice in the storage tanks must be melted with the aid of the returning liquid. Ice slurry has the tendency to agglomerate at the top of the storage tank, limiting the melting to the surface area closest to the entrance of the return water. Fluidising of the ice slurry through various methods is essential to generate a greater exposed surface area to facilitate melting. Mechanical methods such as stirring or an intake and pumping system, which keeps the ice slurry rotating in the tank, are possible mechanisms. This last system was successfully installed at the CANMET Energy Technology Center in Ottawa, Canada. Spray nozzles distributed the return water evenly to the top of the rotating ice mass, melting the ice evenly.

Charging and Discharging of Storage Tank

Ice particles in a storage tank tend to grow and agglomerate. There is a need for the ice particles to be properly fluidised before any effective ice melting or slurry transport can take place. The temperature of the water/slurry exiting the storage tanks depends on the amount of ice remaining in the tank. In design configurations that send ice slurry to meet cooling loads, temperatures of that medium are 0°C. Designs discharging only cold water have exit temperatures that are a function of the amount of ice remaining in the tank. These exit temperatures are similar, but somewhat lower, to systems using an ice harvester design. These exit temperatures are shown in Figure 30. All exiting temperatures will be lower for systems with freezing-point depressants in the cooling medium.

Figure 30
Discharging Temperatures from Storage Tank



Ice slurry is normally produced in a binary liquid of water and glycol. Other possible freezing-point depressants than glycol are salts. For example, seawater is currently used on board fishing vessels as a cooling medium. As well, a water solution consisting of a mixture of nitrite and nitrate has successfully been used in the CANMET ice slurry cooling system. The use of a combination of salts permits a higher total ionic concentration and hence a lower freezing point for the solution. A nitrite-nitrate mixture has the added advantage of having corrosion inhibiting qualities for some metals.

Operation strategies

Ice slurry generators now tend to be sized for modular systems. With this approach, building owners have more options with respect to the placement of smaller units. These ice slurry generators are constructed in increments of 17.6 kW.

The installed capacity at the Ritz Carleton Plaza in Osaka, Japan (one of the world's largest ice slurry systems) consists of 31 units of ice generators of 260 kW each and 16 sets of storage tanks. Capability to produce larger capacity ice slurry generators exists but ice slurry for district-cooling purposes is still under development.

Existing system

Sunwell Installation at Herbis Osaka
The Sunwell ice slurry based thermal storage system in the Ritz Carleton Plaza in Osaka, Japan was completed in February 1997. This building contains business offices, retail stores and the Ritz Carleton Hotel. The 136 823 m^2 building with its modern office automation equipment has a very high daytime cooling load, even in the winter months.

Based on the load profiles of the building complex, the installation included 31 heat-recovery ice generators of 260 kW each and 16 sets of ice storage tanks of either 140 m^3 or 70 m^3. The total thermal storage capacity of the system was 80 750 kWh. Environmentally safe R134a was used as a refrigerant. A binary solution of ethylene-glycol water was used as the ice-making medium of the system.

Table 8
Existing systems using ice slurry
**Ice-generating capacity*

Name	City, country	Ice storage capacity
Ritz Carleton Plaza	Osaka, Japan	80 750 kWh
Techno-Mart 21	Seoul, Korea	14 000 kW*
Commercial Center	Asia	1 760 kW
Middlesex University	London, England	370 kW
Virginia Commonwealth University	Virginia, USA	1 337 kW

Production plant

The fundamental idea for a district-cooling system is that the cooling is produced and distributed from a centralised production plant. The chilled water production in a central cooling plant is more efficient and cost saving than locally produced cooling. The operating and maintenance cost (O&M) per unit of energy will also be lower and furthermore, the central plant has the ability to use different energy sources for the cooling production such as hot water, steam and flue gas from combustion of, for example, natural gas or waste.

The key components in a centralized production plant for chilled water are:

- chiller,
- distribution pumps,
- process controls,
- cooling towers.

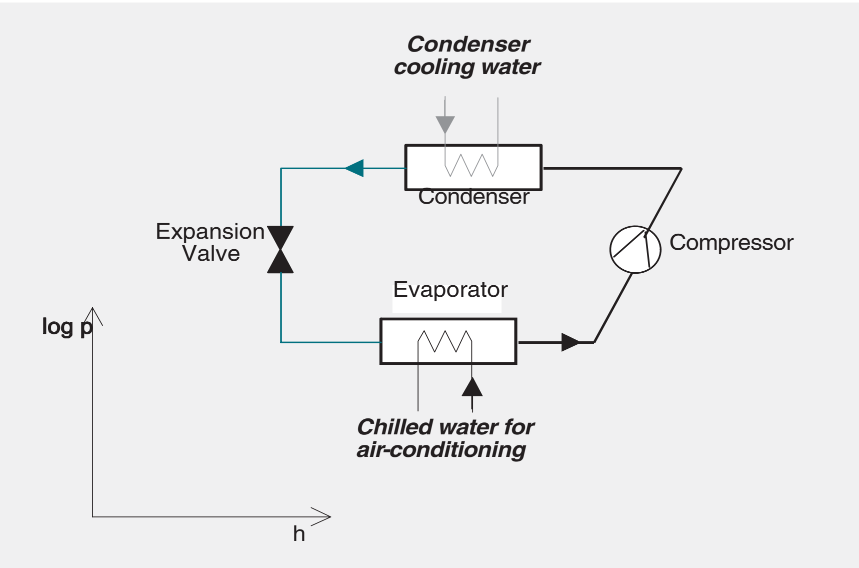
Chiller Technologies

This chapter describes the chiller technologies mostly used in air-conditioning and some potential alternative technologies.
The production of cold is mainly based on two principal techniques: absorption machines and vapour compression cycles. The key difference between vapour compression and absorption chillers is the type of energy normally used. Compression chillers typically rely on electrical energy (steam turbine, gas turbine or gas motor driven compression chillers are available but rarely used), while absorption units require a source of heat, hot water or steam or fuel, e.g. natural gas or light fuel oil. Absorption units are installed when the available heat source is more economical than electricity. They account for a very small portion of total installed capacity. Steam-jet compressors can be used in chilled water production when the available heat source is steam in usable pressure (300…1000 kPa gauge). Steam jets are rare in air-conditioning systems, they are mostly used for concentrating or drying food or chemicals.

Vapour Compression Chillers

Compression cycles form the basis of traditional cycles for cold production, used in particular for domestic refrigerators and air-conditioning covering the entire capacity range currently available on the market.

Figure 31
Flow schematic
of a Compression Chiller



Technology, Chillers

The types of compressors that are commonly used for large and medium-sized refrigeration plants:

1. Reciprocating
2. Screw
3. Scroll
4. Centrifugal

The most common type for large district-cooling scale comfort cooling is the centrifugal compressor. The proportion of screw compressors is however increasing. Reciprocating compressors are less occurring in the district-cooling sectors due to the limited capacity range. Screw chillers have a maintenance advantage over reciprocating chillers in the same size range, since these machines are more tolerant of liquid refrigerant in the suction line (ice storage applications with low evaporator temperatures).

The compressor employed in an application depends on the refrigerant and size and nature of the installation. Small units are based on reciprocating and screw compressors. Medium-sized compressors are screw or rotary coil type. Large units are based on centrifugal compressors. With the knowledge of the fact that chlorofluorocarbons (CFCs), refrigerants (R11, R12, R113, R114, and R115) contribute to ozone depletion (ODP) and the greenhouse effect, an international protocol to restrict utilization was signed in Montreal 1997. New environmentally safer refrigerants have been developed relatively fast with good results and the compressor chiller will still be the dominant technology for cooling. In district-cooling systems compressor chillers are the most common technology. Many countries have introduced laws that restrict the use of the detrimental refrigerants and all users have to change to more environmentally safer refrigerants.

Reciprocating chillers

Among the different chiller types the reciprocating chiller is the one used within the widest area of operation. The field of application stretches from refrigerators to smaller industrial plants with nominal capacities of 15-1500 kW. The COP is lower than that of other chiller types.

Screw chillers

Screw compressors generally use two intermeshed screws for compression. They are not as cost effective as either the small reciprocating units or the large centrifugal chillers. Screw compressors are becoming more common and are available in sizes up to 7 MWc, which is more suitable for district-cooling applications. Screw compressors have a high efficiency and are well suited to the high-pressure ratios of ammonia, although noise is a potential problem compared with centrifugal compressors. They are also very efficient compared with centrifugal chillers when using low temperature condenser water.

Scroll chillers

Scroll compressors are a relatively recent compressor development, which uses two spiral-shape scrolls for compression. One scroll is stationary and the other is moving radially (orbiting) by "crankshaft" drive. Scroll compressors are a new energy-efficient, low-noise and low-maintenance alternative in the 70 to 700 kWc range of cooling systems using R-134a, R-407C, R-410A refrigerants.

Centrifugal chillers

A centrifugal compressor rotates at high speed, compressing the refrigerant by centrifugal force. Because the design, they are well suited to large cooling loads, and they comprise over 95% of chillers with cooling capacities of 0.7- to 35 MWc. Centrifugal compressors are the most

commonly used compressors for district cooling as they are available in a wide range of sizes. The chillers are available in sizes ranging from less than 30 kWc up to 35 MWc. Centrifugal chillers can be used for large capacity requirements as they can handle large volume flows, 2 000 m^3/h or more.
Centrifugal units are most energy-efficient chillers, with COP in the range of 5-7 (at about 6°C evaporator temperature and 30°C condenser temperature).

One limitation is that they do not manage large pressure increases in one step.

The advantages of centrifugal chillers are mainly the following:
Centrifugal chillers are:

- Reliable and have low operating costs per MWh;
- Small and have low weight in comparison to capacity;
- Flexible and can handle oil-free compression with good efficiency;
- Relative vibration-free and have small demands on the fundament;
- Maintenance friendly and have long lifetime and few movable parts.

The disadvantages are:
Centrifugal chillers

- Cannot handle small volume flows; and
- Demand heavy refrigerants with low volumetric cold production.
- Cannot take advantage of very low condenser temperatures to improve efficiency.

General Features of the Vapour Compression Cycles

The compressor requires a drive, which provides the mechanical energy to the compressor shaft. These drives can be electrical motors, steam or gas turbines or reciprocating engines. There are basically two ways of connecting the drive to the compressor: hermetic and open (external drive). In a hermetic unit the compressor and the electric motor are enclosed in the refrigerant environment, so that the motor is cooled by the refrigerant. Only electric motors can be used in hermetic drives. Hermetic compressors are less expensive and they run more silently than external drive compressors. Maintenance of a hermetic welded compressor unit is, on the other hand, complex and expensive. Also semi-hermetic compressors are available. These are accessible for service because of their bolted construction.

The operating temperature of a compression cooling cycle can be adapted to different conditions by changing the pressure of evaporation and by using different working media. As low temperature (below 0°C) is easy to reach by compression cycles, ice thermal storage is possible and also the most attractive system in small-scale applications.
The purpose of all types of chillers is to provide chilled water at the rate and temperature required by the air-conditioning and thermal storage systems. There are numerous control strategies and devices to make the chiller adjust to the demand and to operate with as high efficiency as possible. The effects of varying the evaporator and condenser temperatures on the cooling output are shown in Figure 32 and Figure 33 below. In both figures the compressor speed is constant. The graphs show how determining the external temperatures are for the chiller operation. The effect of evaporator temperature is much stronger than that of the condenser: a 6°C fall in the evaporator temperature causes the capacity to drop by roughly 500 kWch as a 6°C rise in the condenser temperature reduces the capacity by about 300 kWch. The change of evaporator and condenser temperatures is an important issue when dealing with cool storage. Low-temperature storage means lower capacity and efficiency for the chiller plant but the night charging of storage can compensate for at least part of the losses.

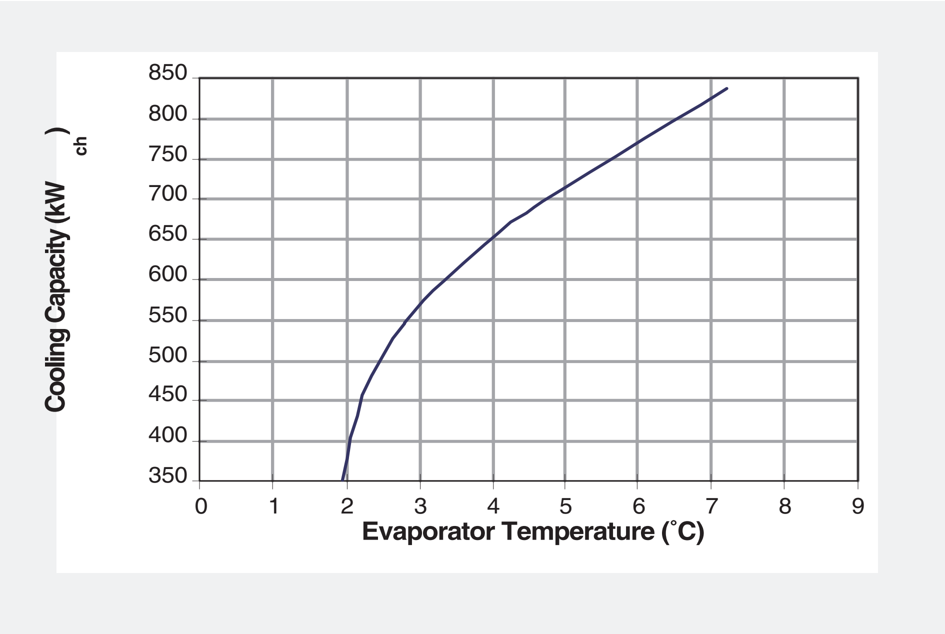


Figure 32
Effect of evaporator temperature on cooling capacity with constant condenser temperature (37.8°C) and constant compressor speed.

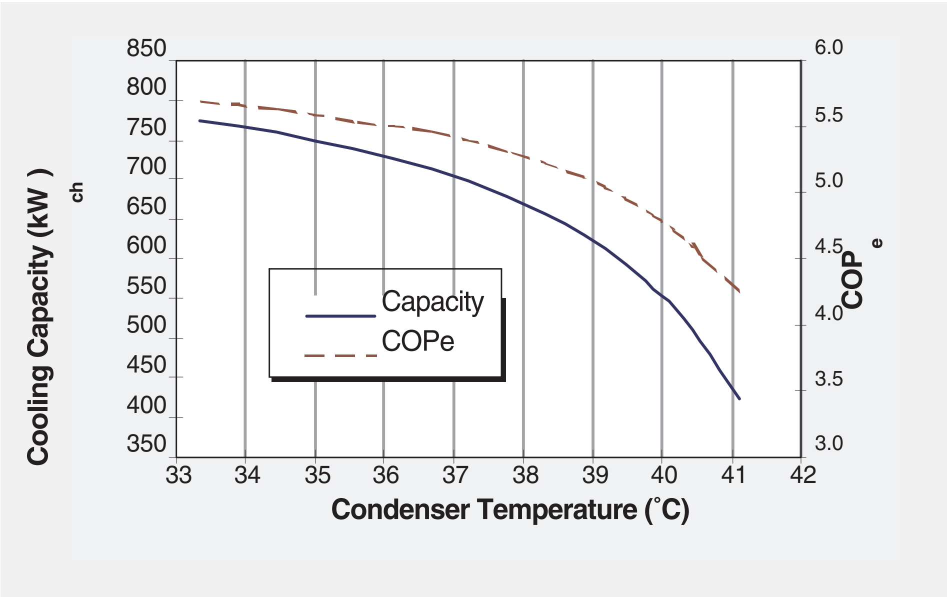
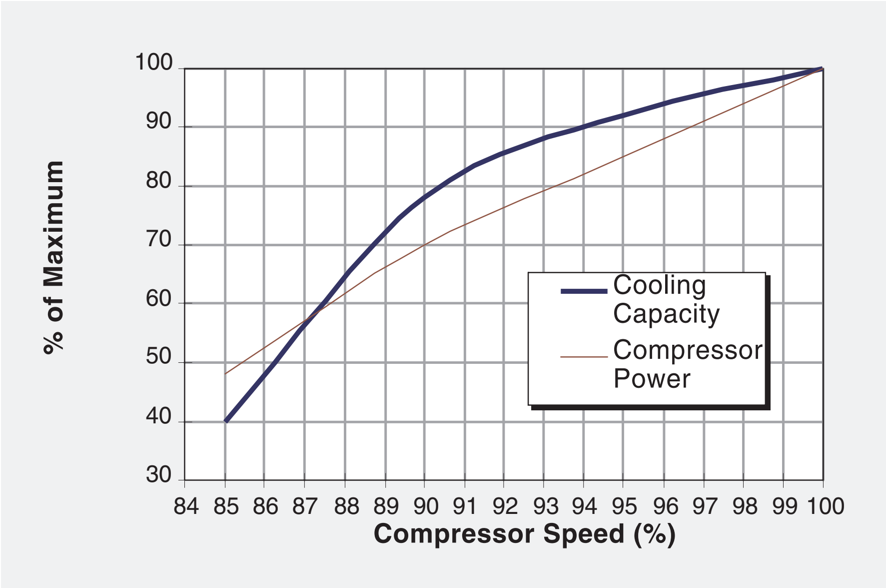


Figure 33
The effect of condenser temperature on cooling capacity and efficiency, COPe with constant evaporator temperature (4.4°C) and constant compressor speed. The COPe is based on brake power, not the total electricity consumption of the chiller.

The cooling capacity of a centrifugal chiller can be controlled by varying compressor speed as the evaporator and condenser temperatures remain constant. The compressor power requirement is decreasing in many cases faster than the cooling capacity up to a capacity level of 70…60% as seen in the next figure.

Figure 34
Capacity control by varying compressor speed with constant evaporating and condenser temperatures.



The types of compressors suitable for water chilling are collected in Table 9. The COP presented in Table 9 are in accordance with full-load ARI standard rating conditions. The COP can be significantly lower or higher at part-load conditions depending on the chiller design and conditions.

Table 9
Vapour compression refrigeration units for large-scale water chilling

Compressor	Cooling Capacity (MWch)	COPe (Wch/We)	Common Refrigerants
Reciprocating	up to 1.5	3.8…4.6	R22, R407c, R134a, R717
Screw	0.1…7	4.1…5.6	R22, R407c, R134a, R717
Scroll	0.1…1	4.6…7	R-134a, R-407C, R-410A
Centrifugal	0.3…35	5.0…6.7	R22, R134a, R123, R717

Centrifugal compressors dominate the market for 700 kWch and larger chillers. They come mostly with a semi-hermetic drive or an open drive with a shaft seal.

The price of the “bare” chiller consisting of a compressor, an electric motor compressor drive, an evaporator and a condenser is about USD 50/kWch for chiller sizes of 1000 kW and larger.

Absorption Cycles

Absorption machines differ from the mechanical compression cycles in the way they carry out the compression. The example of the water / ammonia cycle (NH3), illustrated in Figure 35, shows the operation principles of an absorption machine. In this type of cycle, the water plays the role of the absorbent and the ammonia is the refrigerant.

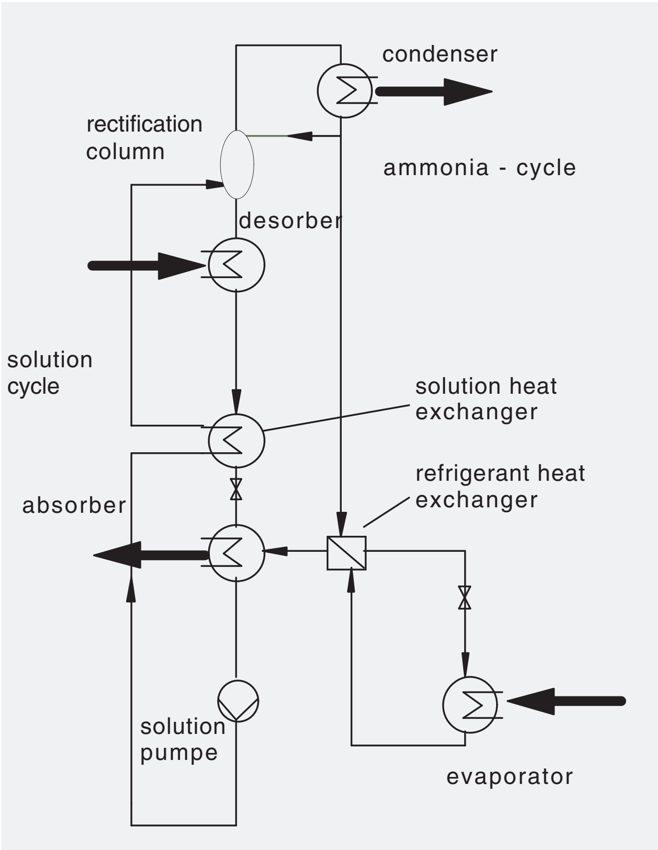


Figure 35
Flow diagram of an ammonia/water absorption chiller.

The heat drawn from the room to be cooled assists in the evaporation of the ammonia in the evaporator. The low-pressure ammonia steam leaving the evaporator enters the absorber where it is absorbed into an ammonia-diluted solution, which becomes concentrated. This process takes place at a slightly higher temperature than that of the environment to which it is necessary to give up heat.
The concentrated solution of ammonia leaving the absorber is pumped in liquid phase with the solution pump (20...30 We/kWch) to the high-pressure of the desorber. The heat originating from the hot source (exhaust gas, steam or direct heating by fuels) allows a large proportion of the ammonia in solution to be evaporated in the desorber. At the same time some water evaporates. A rectification column is used to separate water from ammonia. The ammonia leaving from the top of the column condenses in the condenser, and is partly led back to the rectification column. The rest of the ammonia flows through a refrigerant heat exchanger and a throttle valve back to the evaporator.

Four types of sorption chillers are currently available on the market:

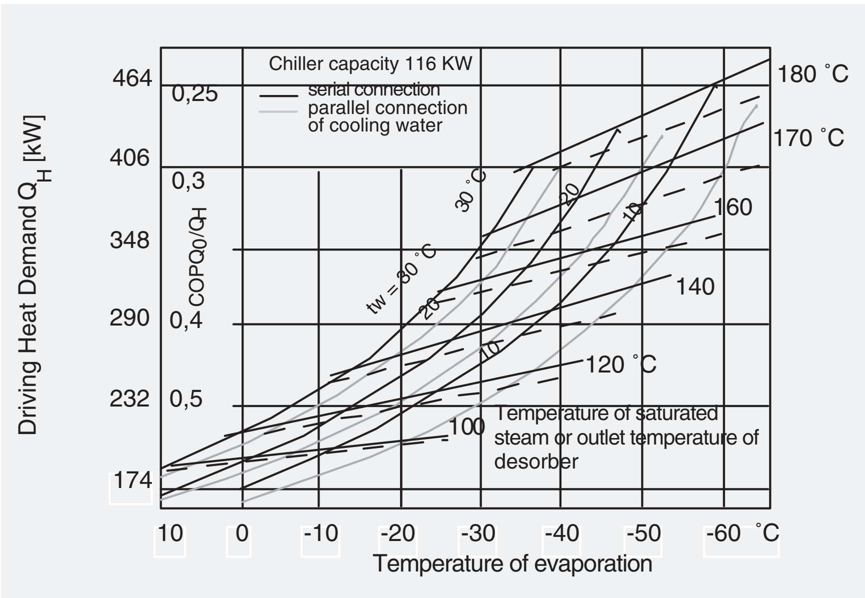
- One-stage (single-effect) absorption chillers
  Binary solutions: water/LiBr or ammonia/water
  Temperature range for thermal energy: 80...130°C
- Two-stage (double-effect) absorption chillers
  Binary solutions: water/LiBr or ammonia/water
  Temperature range for thermal energy: 170...180°C
- Directly fired two-stage (double-effect) absorption chillers
  Binary solutions: water/LiBr or ammonia/water
  Energy supplied by an integrated boiler (gas- or oil-fired)

- One-stage adsorption chillers with periodic operation
  Binary medium: water/solid silica gel granules
  Temperature range for thermal energy: 60...120°C

The main advantage of absorption machines is their low electrical consumption, due to compression in liquid phase as opposed to the mechanical vapour compression in gaseous phase. However, the size of necessary equipment is imposing compared with a vapour compression system and is only economically viable (if at all) where the quantity of heat supplied originates from a potentially lost source. The high investment cost of ammonia/water absorption systems (high pressurized systems) takes special conditions to be viable. Water/LiBr absorption systems operating at vacuum conditions are less expensive and therefore more viable in large district-cooling applications.

To estimate the specific energy demand of ammonia absorption chillers for cold generation, see Figure 36 below, which shows the thermal efficiency (COPth) as a function of the evaporation temperature with different driving hot water temperatures and cooling water temperatures. To reach the evaporation temperatures of -15…-20°C, a driving hot water temperature should be around 120…130°C. For evaporation temperatures around -5°C, which are interesting for air-conditioning applications in context with ice storage units (and ice slurry distribution) temperatures of 90…100°C for the driving hot water are sufficient.
In ammonia/water absorption chillers there is no danger of crystallization, which is a factor when considering water/LiBr absorption chillers.

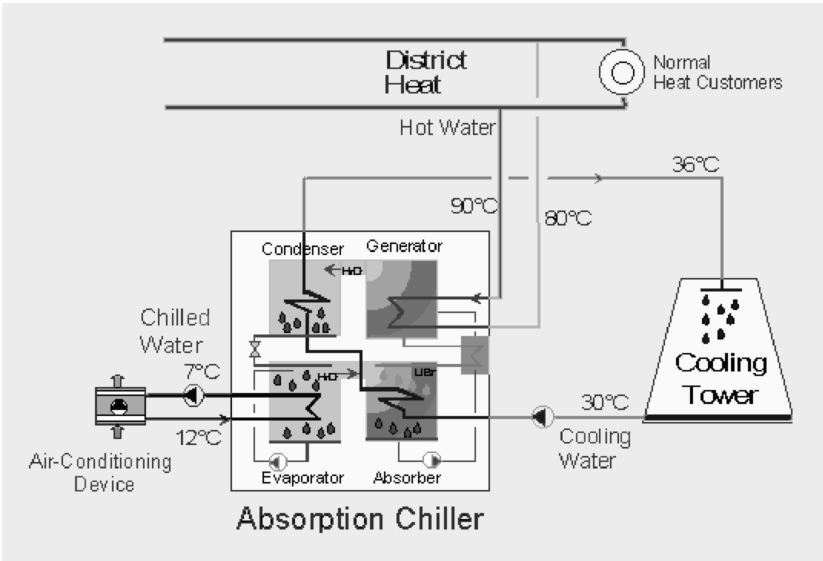
Figure 36
Performance characteristics of a single-effect ammonia/water absorption chiller



The specific capital cost of an ammonia/water absorption chiller (>1000kWch chillers without cooling towers, etc.) is about USD 600/kWch for nominal operating conditions and approximately USD 900/kWch for the cold supply with a district-heating network with temperatures 100/80°C. Due to very high capital costs, ammonia/water absorption chillers are rarely used for air-conditioning purposes. They are mainly used in the food and chemical industry, in which relatively high capital costs are not that significant because of high operation rates.
Disadvantages of this system are the high weight of the equipment and the higher filling quantity of toxic ammonia (approximately 3 kg of NH_3 per kWch) than that in the ammonia compression chiller.
The chiller can be operated continuously at partial loads down to 20% of full load similarly to the water/LiBr absorption chillers.

The water / lithium bromide combination (LiBr) is commonly used for cycles in absorption machines. In this case, the water plays the role of refrigerant. The operating temperatures of chilled water circuit are >3°C, which enables only water based thermal storage. At the moment, the lithium bromide-based machines are only commercially feasible absorption machines in air-conditioning.
The specific capital cost of a water/LiBr absorption chiller (>1000kWch chillers without cooling towers etc.) is about USD 100…120/kWch at nominal operation conditions and approximately USD 120…150/kWch in the cold supply with a district-heating network with temperatures of 100/80°C. The specific capital cost at the end of the normal operating range 80/70°C is above USD 200/kWch.
Commercially available powers vary from 10 to 100 kW for the ammonia-based machines and 100 kW to 6 MW for the lithium bromide-based machines.

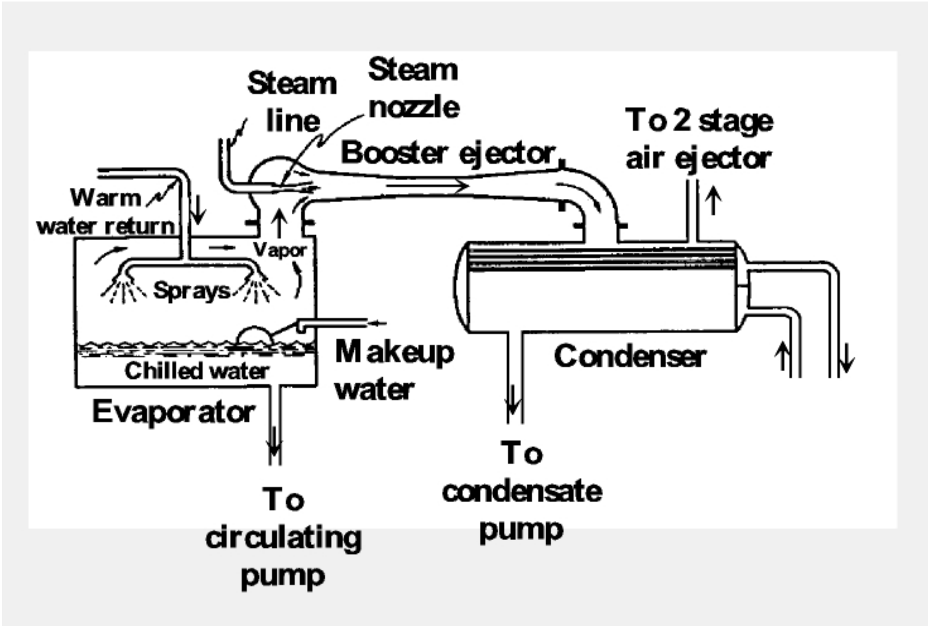
Figure 37
Connection of district-heating water operated Water/LiBr absorption chiller with typical operating temperatures.



Steam-jet Refrigeration

The steam-jet refrigeration cycle is quite similar to other vapour compression refrigeration cycles but instead of a compressor there is a steam ejector to compress the refrigerant to the condenser pressure. Water is used as a refrigerant. The cooling effect is produced by continuous vaporization of part of the water to be chilled (about 1% of the water has to be vaporized).
Ejectors are most commonly employed in applications in which direct vaporization is used for concentrating or drying food or chemicals, and the need for heat exchangers is eliminated. Usually, steam-jet refrigeration systems produce chilled water at temperatures of 2…21°C. This range would be suitable for air-conditioning. Steam-jet refrigeration has several advantages: simplicity of the design, no vibration, high reliability, low maintenance need and low costs. Steam-jet refrigeration systems are commercially available in capacities of 35...3500 kWch. It is not widely accepted in the field of air-conditioning mainly because of low efficiency (COP is typically 0.2…0.3) and rare sources of inexpensive drive steam. The scheme of the ejector is presented in the figure below.

Figure 38
Steam-jet refrigeration system with a surface condenser.



Summary of Different Cooling Technologies

Table 10
Overview of the most often used cold processes for district-cooling purposes

Process	Operating Conditions °C	COP	Electrical connection capacity incl. auxiliaries kWe/kWch	Electric demand kWhe/kWhch	Water demand m^3/kWhch
Compression chiller	6/12, 27/32 mechanical drive	4.2	0.28	0.35	2.7
Single-effect abs. chiller (water/LiBr)	6/12, 27/32 130...110 (90)	0.7	0.06	0.15	5.5
Single-effect abs. chiller (water/amm)	6/12, 27/32130/110	0.5	0.06	0.2	6.1

Capital costs

The process-specific capital costs as a function of capacity are illustrated in Figure 39 for the chilled water plants based on centrifugal chillers and water/LiBr absorption chillers. With low driving temperatures the heat-driven refrigerating processes are more expensive than the electric compression chillers. With driving temperatures of 130/110°C water/LiBr absorption chillers and compression chillers have nearly equivalent capital costs at a chiller capacity of 500 kWch and above. In any case, the total installed cost of an absorption chiller plant is higher because of need of higher cooling tower capacity. If low cost ‘waste heat’ is available the higher capital cost of absorption chillers may be justified by much lower energy costs.

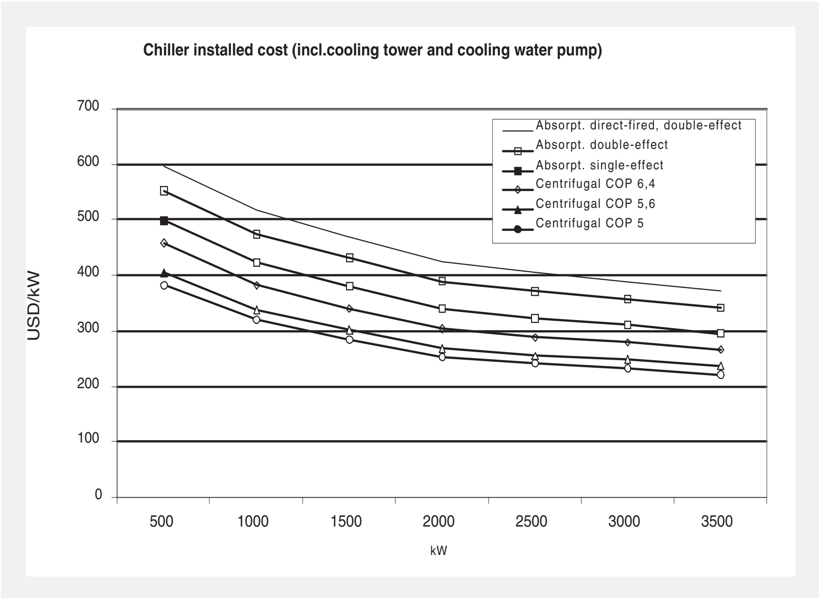


Figure 39
Specific capital costs of LiBr-Absorption cycles and compression chillers in the capacity range of 500-3500 kWc

Combi-connections of heat and mechanically driven chillers

In some cases the production of cooling can be shared between different production methods. The reasons might be the price of drive energy and/or the possibility to optimise the temperatures of different means of production. When absorption and compression chillers are connected in series, the total production cost could be optimised in a large system, which is connected to a CHP plant or some other source of low-cost heat. Typically the absorption chillers would be capable of meeting the base load in order to increase the hours for which they operate. The low output temperature of chilled water enables smaller chilled water storage and/or higher peak capacity while the storage peak time discharge temperature is lower than the chiller output temperature at peak load. The connection of vapour compression chillers makes it possible to utilise low-temperature properties of brines or other cool media in storage applications and in distribution in closed networks of small volume.

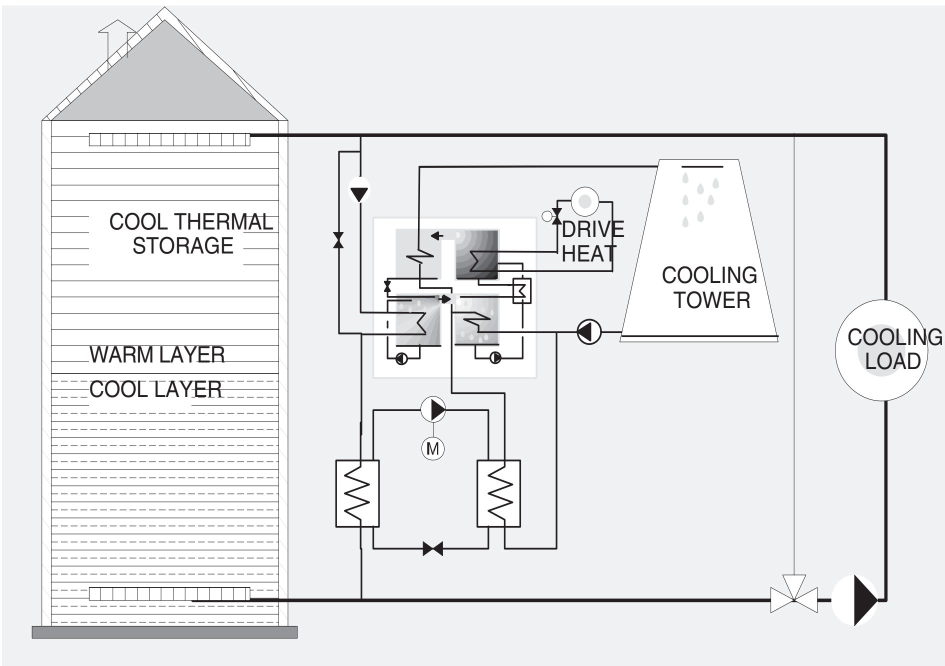


Figure 40
Combi-connection of single-effect Li/Br absorption chiller and vapour compression chiller with low-temperature cool storage

Ice Slurry Generation

Ice slurry is not yet commercially utilised in district-cooling systems. Apart from the benefit of the reduced size of the ice storage the high latent cooling capacity in ice slurry has the potential of significantly reduce the distribution cost. Because of great opportunities the ice slurry is opening for district cooling, ice slurry technology is described in greater detail here.

Generator Types

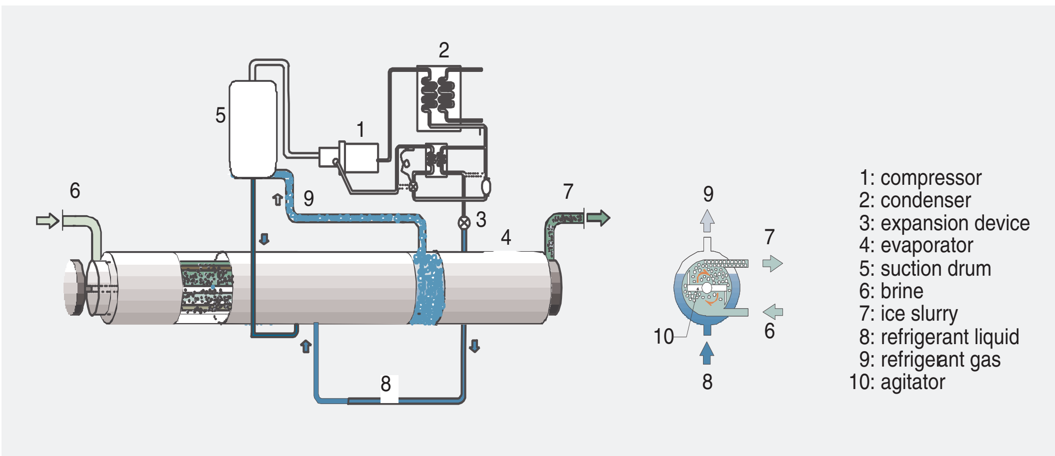
Currently, there are two ice-generating options that best fit into the design of an ice slurry based district-cooling system. One uses the principle of supercooling. The other uses the more traditional method of ice building on a refrigerated surface.

The “Superpac” by Sorenco Ltee of St. Foy, Quebec, Canada consists of an evaporator, which operates on the principle of supercooling of water. [14] A stream of water, when cooled slowly can be supercooled by several degrees without forming ice. After the water leaves the evaporator, ice crystals form when the flow of supercooled water is physically disturbed. The resulting ice fraction in the water depends on the supercooling of the liquid leaving the evaporator. The ice fraction increased by about 1.25% per degree of supercooling.

Sunwell Technologies Inc. of Woodbridge, Ontario, Canada manufactures ice slurry generators which produce ice in a brine solution at approximately -3 to -8^{o}C, depending on the brine concentration. As the liquid is cooled, ice crystals formed at the wall of the evaporator are moved into the bulk flow by an agitator system. The increase in ice fraction for each pass depends on the mass flow rate through the evaporator. Other successful methods of ice production include systems with direct contact between refrigerant and water as well as ice slurry produced in a vacuum. The direct contact method is an efficient method to produce ice. However, it requires some modifications in the redesign of system components and the use of a proper refrigerant due to the presence of water vapour. Ice generation by spraying water in a vacuum chamber requires large vapour compressors. This space requirement, as well as high costs associated with this technology, may restrict its application.

Of all of the types of ice slurry generation, the most developed and widely applied technology is the scraped surface type process developed by Sunwell Technologies of Canada. Figure 41 is a schematic of the ice generator.

Figure 41
Schematic of scraped surface type process ice slurry generator [15]



A typical COP for the generators producing ice is up to 2.4. The low COP is mainly due to the low temperature at the evaporator but also due to the fact that at present, only smaller chiller sizes normally with lower efficiency, are manufactured. The generators can adjust their operation based on the variation of cooling. They can also function as a water chiller when ice production is not required. Due to the small ice crystals produced by this system, high ice storage densities of 46.5 kWh/m^3 can be achieved. The ice slurry system provides an economic alternative to building air-conditioning. A life cycle cost evaluation over a 25-year period indicates that the operating cost, including electricity, maintenance, parts and depreciation, may be lower than that of other systems. [15]

The generator employs a typical vapour compression refrigeration cycle with a compressor, a condenser, an expansion system and an evaporator. In any chiller type, whether it is reciprocating, screw or centrifugal, the refrigerant evaporates in the outer jacket of the cylinder, cooling the binary solution in the inner section. Once the solution is below its freezing point, nucleation occurs in the coldest portion of the liquid. Rotating scrapers on the interior wall promote turbulence. They prevent the cohesion of ice particles and also prevent ice crystals from freezing on the walls of the cylinder. Ice crystals continue to grow until they exit the evaporator. It should be noted that there is a physical limit to the immediate growth of ice crystals. The crystals are made of pure water. As they freeze out of solution, the concentration of the freezing-point depressant at the ice-solution interface increases to the point at which further crystallization stops. The concentration of ice slurry varies according to operating conditions and concentration of freezing-point depressant. The exit ice fraction can vary between 0 and about 35%. Ice crystals have a typical size of about 100 microns in a binary solution with ethylene glycol concentrations in the range of 6% to 10%.

Conventional ice-making equipment involves ice building on a refrigerated surface as with plate type or ice-on-coil systems. These systems tend to occupy large space due to their low efficiencies. If the application is other than central storage, an additional mechanical conveyor system or a mixer process to generate slurry must be employed. An ice slurry system has advantages in energy efficiency, capacity and transport.

The main cost of an ice slurry thermal system is the generator itself. The cost of the storage tank is relatively minor. Current costs of ice generators in North America are about USD 160/kW. This cost is for the ice-generating equipment (evaporator) only. Additional costs of the refrigeration components need to be added to a complete price. Storage costs of such systems vary depending on the approach utilised. The cost of tanks in central storage systems is about 50% of that in distributed storage systems. [16] As the storage tank capacity rises, the average generator load reduces. Careful optimisation of the two can result in considerable cost savings.

Sizing of a storage tank is based on the design day cooling load. Once the average load is selected, the required maximum storage energy in the tank can be calculated. For every hour of the day, the energy requirement (percentage of peak load x 1 h) is subtracted from the generated energy (average load x 1h). This energy difference will be stored in the tank. This value will be positive for load requirements below the average load and negative for demands above the average load. When the load demand reaches the average load, the maximum stored energy will be in the tank. This value of stored energy is then used to calculate the size of the tank. Energy losses must be considered in the design. At this point, the ice slurry fraction may be approaching 40% as cold water is being withdrawn from the storage tank to generate more ice slurry during non-peak periods. Based on 40% ice fraction in the storage tank, at maximum storage capacity, the stored energy content would be about 50kWh/m^3.

Chiller Plant Operation with Cool Storage

An important point in selecting thermal storage and chiller equipment is to define an operating strategy for the chiller plant. Choices include “full storage” or “partial storage” alternatives. Partial storage strategies can be further characterized as power demand-limiting or load-levelling as described earlier.
Choice of operating strategy is dependent on the priorities of first cost, demand charges, and energy consumption charges in each case. An analysis of these factors must be performed on every project in order to make the optimum choice for each project.
When a new system is built in a new building area, the selection procedure of thermal cool storage starts from the analysis of energy prices and energy demand. If only chilled water (>7°C) is needed the existing absorption technology is useful in conditions in which heat (>90°C) or fuel (natural gas and/or biogas) is available at a very low price.

The use of absorption technology is compatible with a centralised chilled water storage system. Chilled water storage is large even though it is dimensioned on a partial capacity basis. Chilled water storage of full capacity or close to full capacity is possible in cases in which waste heat or very low-cost fuel is available only during a very limited off-peak period.

Chilled water storage of partial capacity operates in parallel with the chillers, and the operating temperatures are equal to chilled water distribution temperatures. During the night when the temperature of cooling water from cooling towers is at the lowest, it is possible to use lower charge temperature to maximise the storage capacity for the next peak load period. Low storage temperature is especially important in demand-limiting systems in which the storage supplies the cooling during the highest peak only. The freezing of water in the evaporator of the water/LiBr absorption machine also limits the nighttime charge temperatures around +4°C even if anti-ice substances are used in the storage. Thanks to off-peak electricity prices it is often feasible to operate electrically driven vapour compressor chillers to reach the lowest storage temperatures and to reduce the size of the storage system.

The Choice of Chillers for Cool Thermal Storage

When a thermal storage system is designed, several factors have to be considered when choosing the production plant. One of the main considerations is the type of compressor needed for the required temperatures and capacity. Table 9 shows the compressors and the capacity range. Technical and economic differences between various chiller compressors should always be evaluated on a project-specific basis.

A major consideration, especially, for ice systems is the leaving fluid temperature achievable by the chiller. Compressors for ice thermal storage have to operate at lower temperatures than normal. Ice storage systems operate at charging temperatures of -9 to -3°C, which are below the normal operating range of conventional cooling equipment for air-conditioning applications. The temperature of chilled water leaving the chiller is about 4 to 6°C. Reciprocating and screw chillers are adaptable to a wide range of leaving temperatures and can generally be applied to ice storage systems with small changes. Centrifugal chillers can generally be applied to ice making, but the selection must be made for the specific anticipated operating conditions. An existing centrifugal chiller originally selected to produce chilled water at 5 to 7°C will not produce the temperatures needed for ice making without modifications.

Table 11
Comparison of storage technologies [3]

**) Technology is not commonly used (high investment cost)*

	Chilled Water	**Low Temp. Chilled Water**	**External-melt Ice-on-coil**	**Internal-Melt Ice-on-coil**	**Ice Harvester**	**Encapsu-lated Ice**	**Eutectic Ice**	**Ice Slurry**
Chiller type	Standard water chillers	Low-temperature standard chillers	Low-temperature coolant or build-up refrigeration plant	Low-temperature standard chillers	Prepackaged or build-up ice making equipment	Low-temperature standard chillers	Standard water chillers	Low-temperature chillers
Chiller tech-nologies	Vapour Compress. Centrifugal Screw Absorption (LiBr)	Vapour Compress. Centrifugal Screw *Absorptio Ammonium	Vapour Compress. Centrifugal Screw Scroll Recipro-cating Rotating vane *Absorptio Ammonium	Vapour Compress. Centrifugal Screw Scroll Recipro-cating Rotating vane *Absorptio Ammonium	Vapour Compress. Screw Scroll Rotating vane	Vapour Compress. Centrifugal Screw Scroll Recipro-cating *Absorptio Ammonium	Vapour Compress. Screw Scroll Recipro-cating Rotating vane Absorption (LiBr)	Vapour Compress. Centrifugal Screw Steam Jet
Typical chiller cost	\$57 to \$85/kW	\$57 to \$142/kW	\$57 to \$142/kW	\$57 to \$142/kW	\$313 to \$427/kW	\$57 to \$142/kW	\$57 to \$85/kW	About \$160/kW
Tank volume	0.086 m^3/kWh	0.044 m^3/kWh	0.023 m^3/kWh	0.019 to 0.023 m^3/kWh	0.024 to 0.027 m^3/kWh	0.019 to 0.023 m^3/kWh	0.048 m^3/kWh	n.a
Storage installa-tion cost	13\$ to 30\$/kWh	12\$ to 30\$/kWh	\$14 to \$20/kWh	\$14 to \$20/kWh	\$5.70 to \$8.50/kWh	\$14 to \$20/kWh	\$28 to \$43/kWh	n.a.
Charge tempe-rature	4 to 6°C	- 6 to 1°C	- 9 to - 4°C	- 6 to - 3°C	- 9 to - 4°C	- 6 to - 3°C	4 to 6°C	- 9 to 0°C
Chiller COP	5.9 to 5.0	4.1 to 2.9	4.1 to 2.5	4.1 to 2.9	3.7 to 2.7	4.1 to 2.9	5.9 to 5.0	2.4
Tank inter-face	Open (or closed) tank (trap valve)	Open (or closed) tank (trap valve)	Open tank	Closed system	Open tank	Open or closed system	Open tank	Open or closed system

Technology, Chillers

Most cool storage applications use packaged chillers to generate cooling. Internal-melt ice-on-coil, encapsulated ice and normally external-melt ice-on-coil use packaged ice-making plants. Some external-melt ice-on-coil systems are also installed with built-up refrigeration systems, with the refrigerant as the charging fluid, and the ice-making pipes as the evaporator for the system. Ice harvester systems use packaged or built-up chillers with especially designed evaporator sections for making and harvesting ice.

Internal-melt ice-on-coil and encapsulated ice systems require chillers capable of producing charging temperatures of -6 to -3°C. External-melt ice-on-coil systems require chillers capable of generating charging temperatures of -7 to -3°C for an ice thickness of 40 mm and for systems that build ice up to 65 mm charging temperatures of -12 to -9°C. Reciprocating chillers or screw chillers, depending on the size of the chillers, are often used because of low charging temperature and the relative wide range of operating temperatures. Centrifugal compressors may be used provided they are properly selected for the intended range of evaporator temperatures.

The chilled water storage systems and the eutectic salt storage systems use conventional chillers to provide charging temperatures of 4 to 6°C. This system is often used in applications with high loads and typically uses centrifugal chillers. However, reciprocating and screw chillers can also be used. Chilled water and eutectic salt storage systems can often be added to an existing cooling plant simply by adding a storage tank and associated piping. In most cases, existing chillers require no modifications. The large size of a chilled water storage tank limits the places where it is applicable.

Absorption chillers could be used to charge chilled water and eutectic salt systems. However, the efficiency falls significantly at leaving temperatures below approximately 6°C.

Technology, Distribution

Cool Distribution Network Design in General

The design of network is strongly affected whether the distribution system is a closed network built only for a known number of customers and cooling loads or if it is open for new customers after the building of framework. Closed networks are typically building complexes like airports, campuses, hospitals and shopping centres. A city chilled water network is typically an open network. Open network philosophy also contains the individual freedom of not getting connected even though being in the middle of the distribution network.

Distribution of cooling energy can be accomplished through distributing chilled water or ice slurries from centralized production units. There are also combinations of these two methods. When choosing an approach in separate cases, there are several facts to be estimated to find the most economical solution. Centralising the cold production has the following advantages: the necessary peak power can be reduced because the connected building load will not peak at the same time. The production of cooling also becomes more planned for better total economy. The bigger unit size of the chiller or chillers reduces the total capital costs as the specific costs, USD/kWc, are reduced. On the other hand, constructing a new chilled water distribution network increases total capital costs. The thermal loss of the network increases operating costs. The wider the network and the farther off the consumers from each other or the cold plant, the more expensive the network. This fact is the main obstacle against the success of district cooling.

The investment cost of piping is high when the dimensioning has to meet the high and often short peak loads. The operating cost comprises the pumping cost and cool loss. The pumping cost is high on the small-dimension networks delivering high peak loads. A small-dimension network of lower investment helps also to reduce the cool loss.

There are no unified solutions for district-cooling pipes like in the field of district heating in which the pre-insulated pipe is the dominant technology. In cases of large-diameter district-cooling pipes buried in ground of low temperature no insulation may be necessary. The distribution pipes can be laid in channels or without channels as pre-insulated pipes and overhead pipes.
The plastic surface would be useful inside the steel pipe to reduce the friction but outside the steel pipe it effectively prevents corrosion. In both cases the plastic coating is also working as insulation. Pipes produced for natural gas transmission lines are high-quality steel pipes with outside PE coating. They are firm and usable also in large-diameter sizes (900 mm).
In smaller diameters and high ground temperatures the insulation of pipe material is more important and pre-insulated pipes are more feasible alternatives. Low supply temperatures (+1°C) require better insulation because there is a higher temperature difference between the piping and the surrounding ground and because the production cost of low-temperature cooling energy is also higher.

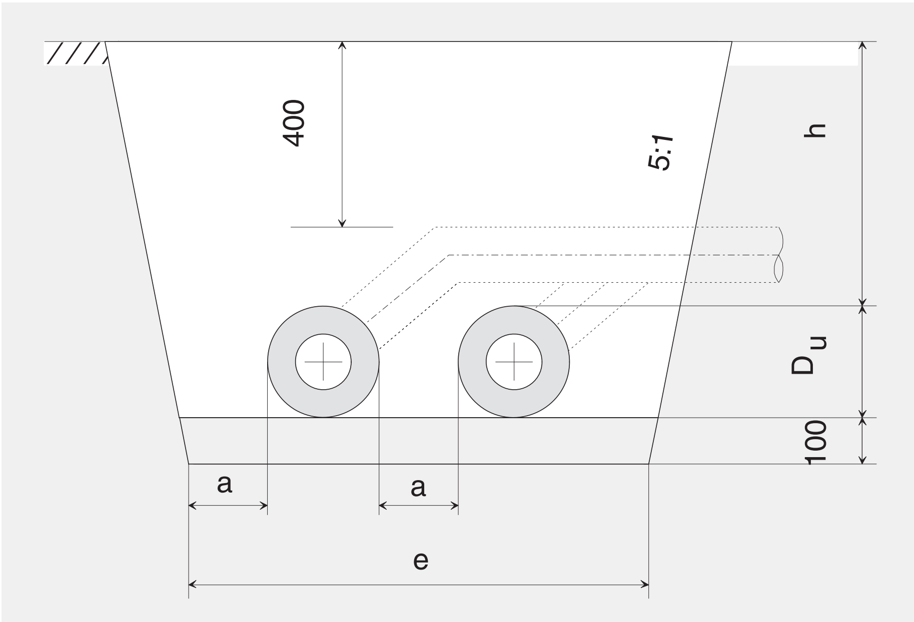
Different Transmission Media and the Demands for Piping

The following aspects have to be taken into account in district-cooling supply:
District-cooling systems are mostly used in urban areas, where a high connection density can be expected. The set-up of a district-cooling system under these conditions is often connected with complicated transferring conditions and the high building costs of the chilled water network.
Due to low temperature differences of chilled water distribution very high flows must be pumped to reach the peak cooling capacities needed. The diameters of flow pipes grow even though high flow speeds are used in smooth pipes (plastic). As a result, the distribution costs increase even though the insulation of distribution pipes is thinner than that of district-heating pipes.
At an individual cold supply without cooling base load the operating time of a cold supply system is 2000...3000 h/a (Central Europe). Cooling networks are in operation all year round, which causes additional cold losses in the net. Typical cold losses are between 2 and 10% of the annual cooling demand.

The earth- and duct-laid pipelines are used for the district energy systems. The pipeline systems of HDPE, jacket pipe (plastic jacket pipe) and steel pipes with corrosion prevention are used.

HDPE pipes, steels pipes and jacket pipe systems known from district-heating systems can be used for earth-laid pipe systems. There are also case-designed pipe systems made of PE and PVC.

Figure 42
Typical earth-laid pipe trench of pre-insulated (steel) pipes with plastic jacket pipes



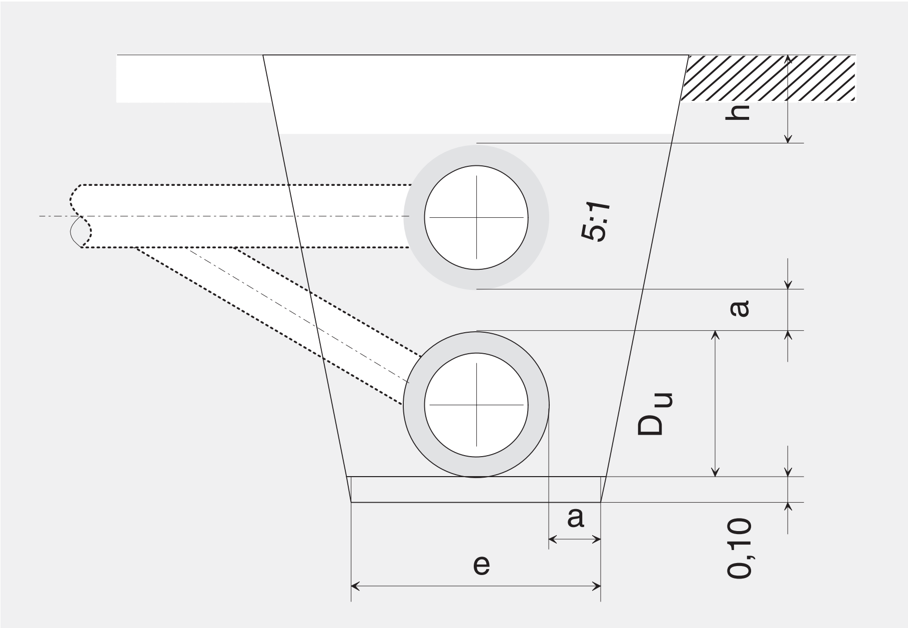
Earth-laying

The earth-laying of pre-insulated pipes is the most common method to build district energy networks. The construction has developed during the extensive building of district-heating networks and it is now the most technically advanced system, fast to build and with a long life. The total costs of constructing the network include excavation, installation, material and filling costs. Most of these costs depend on the time used. Thus, cutting the total time used for construction and installation is an effective way of reducing the total costs.

The temperature difference between the typical supply temperatures (6°C) and return temperatures (12...15°C) of chilled water and the soil is so low that it is possible to build a district-cooling network without cold insulation. This is worth considering, especially, for the return line. The cold losses are approximately 10 W/m of pipe for HDPE tubes and 15 W/m of pipe for steel pipes at earth-laying systems. The proportional cold losses are 1...20% in the range of inverse line density of 50...1000 m of pipe/MWth with an operating time of 8760 h/a. The increase of heat loss is often minimal when compared with the savings on insulation material and installation work. HDPE pipes for a nominal pressure of 6 bar are more favourable than steel pipes only up to a nominal diameter of 200 mm. The assembly and laying cost is a little lower for HDPE pipes. For steel pipes there are still additional costs of the corrosion prevention required (paint and plastic jacket). Both systems are comparable at larger nominal diameters.

The most common problem is to find sufficient installation room for large-diameter district-cooling pipes in city centre areas. The laying of supply and return pipes can be either side by side or piggy-back. Piggy-back laying means that the pipes are on the top of each other, which makes the pipe trench narrower.

Figure 43
Vertical "piggy-pack" installation of single pipes.



The saving on installation costs by using the piggy-back method is about 15%. The order of level of supply and return pipes must be defined in each case to minimise the heat loss but in the case of un-insulated pipes the cold supply pipe should be laid on the bottom of the trench.

Duct-laying Systems - Transfer in a Collector

For duct-laying systems, contrary to earth-laying systems, the insulation of HDPE tubes and steels pipes is necessary (at least insulation against condensing). Furthermore, to support the pipes a frame construction has to be built. Due to lower bearing distances of HDPE pipes, more support is needed. Thus, the effort of the sub-construction for this pipe system is larger. The cost advantages of HDPE for the pipe material will be compensated through this disadvantage. The duct-laying plastic jacket pipes and steel pipes, which have obtained additionally protective paint and cold insulation, are economically equal.

Brines and other media for low-temperature (<1°C) distribution

Brines make it possible to use lower temperatures (-1°C) in distribution and higher dT can reduce the flow rate needed. The viscosity of brines is high, which reduces the turbulence and pressure loss in piping but unfortunately this also takes place in heat exchangers.

Brines are not commonly used because of increased costs in large volume systems and also because of the corrosion caused by the formerly used brines. The recent development in the field of brines and other anti-freeze agents has been rapid and new low-cost and low-viscosity media have been introduced.
Potassium formate, which has lately been used for defrosting of aeroplanes, is a potential heat transmission medium of low cost and low viscosity. Corrosion inhibitors must be added to avoid corrosion by potassium formate.
Aspen Petroleum Ab presented brine-containing alkali salts of acetic acid and formic acid in 1999. This brine called Temper® has low viscosity and favourable corrosion properties.
Organic media like natural betaine (Thermera® by the Finnish Fortum Ltd.) enable an environmentally sound way of transmitting low-temperature cooling to the places where toxic anti-ice media cannot be used. The prices of alternative anti-ice media are in the beginning higher than those of glycols because of high investment in new production facilities and because of the positive market situation for environmentally sound alternatives. The waste treatment cost of ethylene glycol helps the marketing of natural anti-ices.

Distribution of Ice Slurry

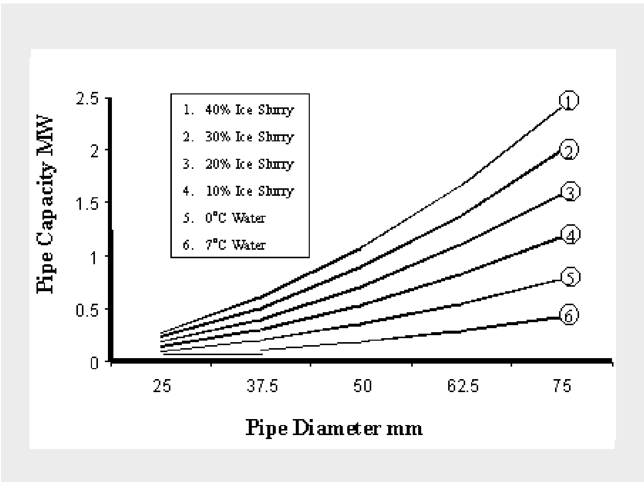
Benefits of Ice Slurry in Piping Networks

The distribution network in a district-heating or district-cooling system represents a significant part of the capital cost of the entire operation. To minimise the cost of the distribution system, it is necessary to reduce the pipe diameter as much as possible. The economic advantage gained from a reduction in the pipe diameter does not only result in lower capital costs but also in lower operating costs. The operating costs are reduced due to lower pumping energy requirements and the reduction in heat gains. Although an ice slurry based system can operate with reduced diameter distribution networks, the system should be designed to accommodate future expansion.

Maintaining a large temperature difference between the supply temperature and the return temperature in a district-cooling network will minimize the piping diameters needed. In a conventional chiller operation, supply temperatures are usually 7°C. With good design, the return temperature should be 14°C. The use of ice slurry in a system reduces this supply temperature to 0°C (or below with the aid of freezing-point depressants). This results in an increase in its cooling capacity per unit of mass flow.

The effect of ice fraction on the pipeline capacity relative to chilled water is shown in Figure 44. For an ice fraction of 20%, this graph shows an almost fourfold advantage of ice slurry over the conventional chilled water at 7°C. This improvement in "cool" carrying capacity means that the pipe cross-section can be reduced by a factor of approximately four. This is equivalent to a diameter reduction by a factor of two. The pipe capacity is defined as the maximum cooling load that a flow rate in a pipe can meet, given velocity limits. Since the cost of the pipes and their installation increase more or less linearly with diameter size, considerable savings on capital investments can be achieved by using ice slurry as the cooling media.

Figure 44
Effect of Ice Fraction on Pipe Capacity [17]



Cooling distribution costs vary and are affected by location, soil conditions and above all, the designed temperature difference between supply and return. The unit cost of installing cooling pipes in the ground (2001) can be estimated by the following:

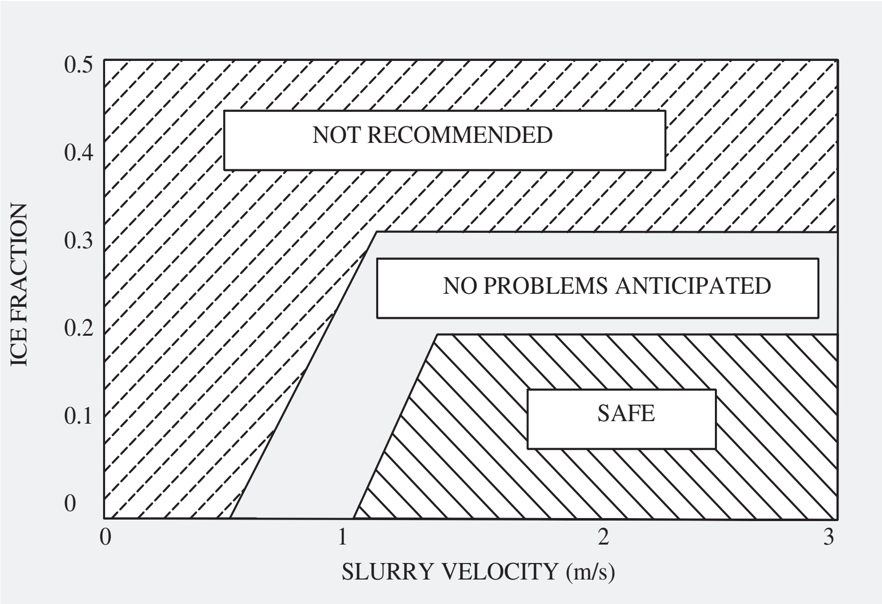
Cost USD/m = 2*Internal Diameter (mm) + 300 [18]

The above unit cost estimate describes the cost of pipes and valves as well as excavating and restoration, engineering and some contingency. This estimate assumes that the system is large enough so that the start-up costs of pipe laying is minimised. The estimate is also a mix of city as well as green field trenching. The impact of using ice slurry on the cost of cooling distribution is readily seen. Using 20% ice slurry in the cooling media, cost savings of about 8% could be realised for small service pipes (25 mm) and savings near 25% for pipes approaching 140 mm.

Pipeline Transport of Ice Slurry

The perceived concern associated with pumping ice slurry through pipelines is potential plugging due to ice agglomeration. However, the research conducted to date indicates that ice slurry can be safely pumped through pipelines within a certain envelope of ice fractions and velocities. The recommended parameters are shown in Figure 45. Ice fractions above 25% are not recommended. Velocities should be maintained above 0.5 m/s and preferably above 1 m/s. It is expected that further R&D will prove that higher ice fractions, up to 50%, can be safely used in district-cooling systems.

Figure 45
Recommended Ice Slurry Transport Conditions [17]



Re-establishing flow in a stagnant ice slurry district-cooling system is not considered to be a potential problem. In stagnant flow conditions, the ice particles will float to the top of the pipe. Local ice fractions, caused by buoyancy forces, should not exceed 40%. With initial conditions of 20% ice fraction, it is expected that the bottom half of the pipe will be available for the turbulent flow to re-establish pre-stagnant conditions.

There are several possibilities for the ice at the top of the pipe to be re-slurrified. The ice in the agglomerated ice pack can be re-distributed by the turbulence in the resulting flow. Also, the agglomerated ice pack can move as a unit and will break up at the first bend, valve or fitting. Gupta's stop/start experiments at the Canadian National Research Council (NRC), have shown that ice agglomeration in district-cooling systems based on freezing-point depressants is not a concern, even under repeated no-flow conditions. [19]

The distribution system of a district-cooling system must be able to deliver the ice slurry to various loads with a wide range of flow requirements. Therefore, the ability to split the ice slurry flow, while maintaining a uniform concentration, is important. Detailed flow splitting experiments have been carried out at the Canadian National Research Council. Tests using fine binary ice slurries have exhibited a strong homogeneous behaviour. No problems were encountered when ice slurry was split and transferred through two parallel sections of NRC's experimental loop. The fine crystals have negligible body force relative to the turbulent diffusion within the flow itself. Therefore, it is reasonable to expect that the ice will distribute uniformly in both legs as the flow is divided.

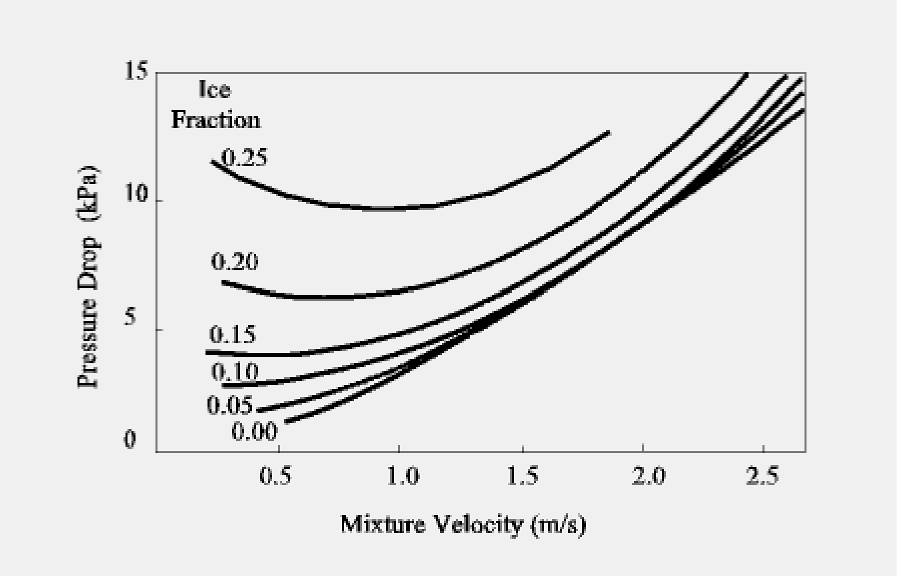
Variation of Pressure Gradient with Ice Fraction

Pressure drop in a pipeline is a function of friction factor, the length/diameter ratio, the velocity and the density of the water or ice slurry. The density of ice slurry decreases marginally with an increase in ice fraction. However, this decrease in density is not enough to make an observable difference in the pressure drop measurements. Depending on the physical shape and size of the ice particles, some authors claim that there is no difference at low fractions and a slight increase in pressure gradient at higher ice fractions. Others observe a marginal decrease in pressure drop. A decrease in pressure gradient is usually observed with coarse ice particles. This decrease in pressure gradient is caused by the reduction in turbulence. This reduction is due to the presence of ice in the water. Up to a certain ice fraction, these coarse ice particles act as a friction-reducing additive.

Experimental pressure drop measurements have been performed with ice slurry velocities above 0.30 m/s. The data indicates that there is essentially no difference or, in some instances, a small reduction in pressure gradient between water and ice slurry containing coarse particles below a threshold ice fraction. [20] This minimum threshold has not yet been established with experimental data.

At low velocities, the reduction of turbulence allows phase separation to occur with ice floating to the top of the channel. This will increase the pressure gradient. For a well functioning district-cooling system, the ice slurry should be maintained in a homogeneous state. This will be achieved by operating the distribution system above the minimum velocity. Snoek et al. [21] have indicated that there is no significant pressure gradient variation at ice fractions below 10%. Data was collected from a 23.8 metres test section of pipe with a diameter of 38 mm. The experiments were conducted at the Canadian National Research Council, Ottawa, Canada. At ice fractions above 10% and at lower velocities, the pressure gradient increased with ice fraction. This increase in pressure gradient was also attributed, in part, to the higher glycol concentration at the higher ice fractions. A minimum pressure drop velocity was established for ice fractions between 10% and 25%, as shown in Figure 46. Below this minimum velocity, the pressure gradient increased when the velocity was decreased. Phase separation, with ice floating to the top of the pipe, caused the effective liquid-flow cross-section to decrease. The slow moving or stationary ice mass effectively decreased the flow area and increased the liquid velocity.

Figure 46
Pressure Drop in 23.8 Metres Long Pipe with Varying Ice Fractions [21]



Pressure drop data was also used to develop a simple two-phase multiplier model to predict the change in friction factor due to the presence of ice particles in the water. The simple two-phase multiplier correlation was developed as a function of the ice fraction and the mixture velocity.

Figure 46 shows the relationship between the mixture velocity, the ice fraction and the pressure drop. It is clear that the "water-only" pressure drop is not much different from that of the 5% and 10% ice fraction mixture flows, especially at higher velocities. At higher ice fractions, the pressure drop starts deviating from the water-only pressure drop. As the ice fraction increases, the pressure drop of the two-phase mixture increases accordingly.

To compare the slurry friction factor "fs" and the water-only friction factor "f" their ratio was computed. Analogous to two-phase liquid-gas pressure drop theory, a two-phase multiplier (N^2lo) was defined as the ratio of the two-phase water-ice pressure drop to the pressure drop of water-only at the same conditions. To compare the pressure drop of the water- ice-glycol mixture to that of pure water, the effect of glycol on the viscosity of the mixture had to be excluded. The glycol content of the water varied with ice fraction. As ice was frozen out of the solution, the glycol formed a greater fraction of the remaining water.

$$N^2lo = \frac{\in P/[L/D) \cdot (V^2/2g)]\ ice\ slurry}{\in P/[L/D) \cdot (V^2/2g)]\ water\ only}$$

where:	L	=	length pipe, m
	$\in P$	=	pressure drop over L, kPa
	D	=	diameter of pipe, m
	V	=	velocity, m/s
	g	=	9.81 m/sec^2
	N^2lo	=	two-phase multiplier

Selection of Cool Distribution Pipe Materials

The suitability of the tube line system depends on the transferring system, the nominal diameter and also on the cold losses to be taken into account. Chilled water, glycol and brine fluids accept most different pipe material alternatives. The pipe dimensions, the size and the highest pressure level of the distribution network define the materials, which are technically possible in each network. When building an ice slurry distribution system the precise laying technique with smooth branches and bend configuration (no downward bends) shall be followed.

Fundamental advantages and disadvantages of the different systems are listed in Table 13 below.

Table 13
Advantages and disadvantages of different pipesystems
*** HDPE tubes are used without length compensation in the range of tap water supply systems.*

	Advantages	**Disadvantages**
HDPE pipe	No corrosion prevention required Low heat conductivity Low pressure loss (smooth surfaces) Use in water supply systems without effort of after-insulation Simple assembly ** Compared with steel pipes, up to nominal diameter of 200 mm more economical	Large thermal expansion Sensitive to water hammer Sensitive to external stress (risk of pipe damage) Short bearing distances and after-insulation required for duct-laying systems Branch line installation is difficult
Steel pipe (with PE coating)	Suitable also for higher operation pressures Large bearing distances for duct-laying systems Low thermal expansion Good mechanical endurance	Corrosion prevention required Comparatively large installation expenses After-insulation necessary for duct-laying systems
Jacket pipe systems	Insulation and corrosion prevention are not necessary at the construction side Of advantage for duct-laying (for example together with DH supply lines insulation is not necessary at earthlaying)	Expensive At short pipe pieces and many adapters quite large effort of after-insulation

Customer Connections

Indirect Chilled Water Connection:

Indirect connection via heat exchanger is the most common and safe way of connecting numerous consumers to a large cooling network. The design pressure of the house network can be much lower when the operating pressure and pressure shocks of the distribution network do not affect the house side. Indirect connection separates the flow controls at house networks and the central flow control, which helps to stabilize the system against pressure swings. Low design pressure brings savings on investment and maintenance. The risk of house network failure is also smaller. Heat exchangers must have a high heat transfer coefficient to reach the lowest possible temperature difference between the supply from the district-cooling network and the house network, which requires large and therefore expensive heat exchangers. In any case, the temperature level of the house network remains >1°C above the temperature of the district-cooling side.

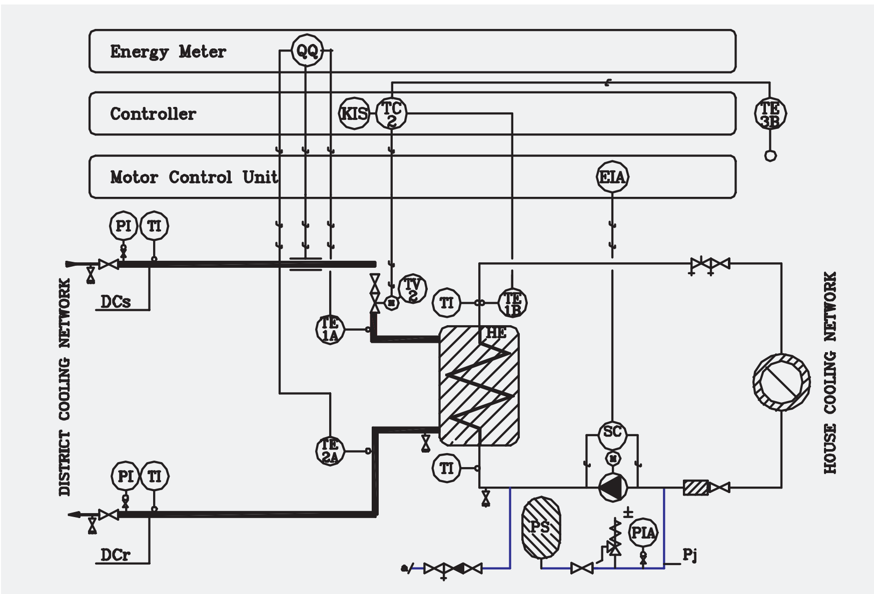


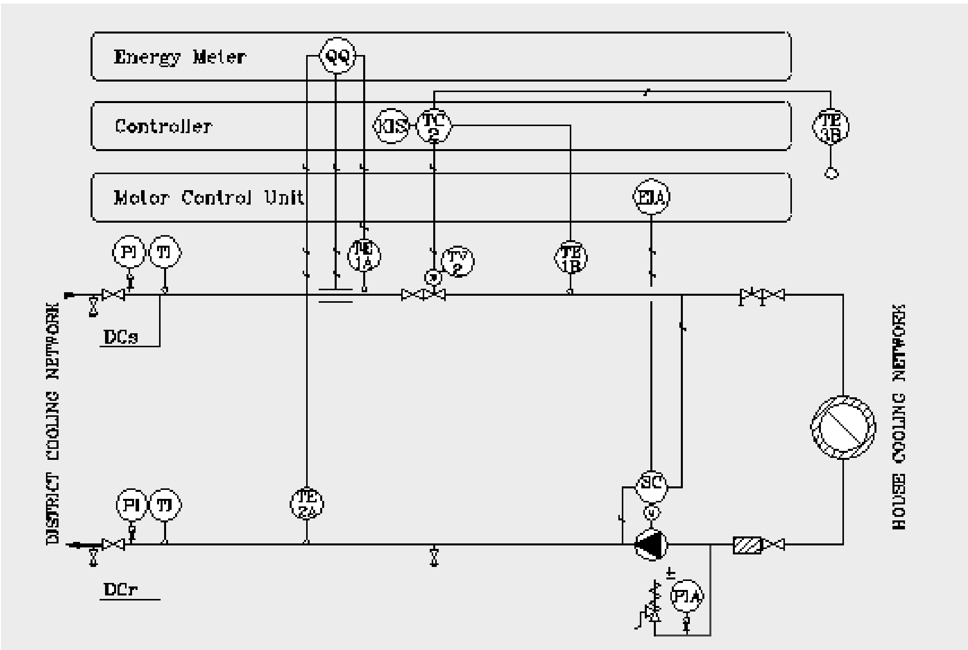
Figure 47
Indirect connection between chilled water DC network and consumer installation

Direct Chilled water Connection:

When pure water is used in district-cooling distribution, the direct connection to the house networks of customers is possible. Direct connection enables the full utilisation of district-cooling water dT. When using higher supply temperatures at a district-cooling plant (cool production by district heat operated absorption) it is more feasible to use direct connection.

Unclear ownership and potential problems with the quality of circulated water have reduced the use of otherwise functioning direct systems. When the same water or other liquid circulates all the way from chillers or cool storage to the coils of air-conditioning equipment, the possibility of leak affecting the whole system is much higher as when all clients are separated by heat exchangers. All client equipment is open to high pressures and possible pressure shocks in the DC network, which makes client network design more expensive but there is still higher risk of leak. To secure the network against leaks in the client networks the customer connection must be equipped with automatic cut-off valves. The pressure drop over air-conditioning coil is high, which often requires customer-located booster pumps, which must be considered when direct connection is planned. The thermal expansion of a chilled water system is relatively small but the responsibility for expansion volume must be defined. If the client is temporarily closed out of the network, the client's house network should have its own expansion tank or at least a relief valve for warming up the closed system.
Direct-connected systems can be found at university campuses and other centrally organised municipalities.

Figure 48
Direct connection between chilled water DC network and consumer installation



Customer Connections in Brine Systems:

The ice-slurry and brine distribution system uses indirect connection to customer house network. Ice-slurry melts when flowing through the heat exchanger at the customer. The flow control of the relatively small ice-slurry flow has to be precise to maximise the performance. Heat exchangers working directly with brines on the primary side and water on the secondary side are difficult to optimise. In heat exchangers of low-cost plate type the risk of water-side freezing is obvious, and high-viscosity brines require long heat exchanger surfaces to be optimally heated up.

Ice storage located at the customer between the brine and the house network helps to avoid the freezing of house network and effectively reduces the peak capacity demand of cool production and distribution.

The brine distribution makes it possible to use small-sized ice cool storage (ice-on-coil) at customer locations where large water tanks are seldom possible. In any case, the client must be relatively big and far from the chiller plant to make customer-located storage feasible. Customer storage makes brine distribution more attractive because of lower distribution network investment. Local storage is seldom used in the common water-based distribution systems while the advantages are often small compared with the extra cost of numerous local storage tanks.

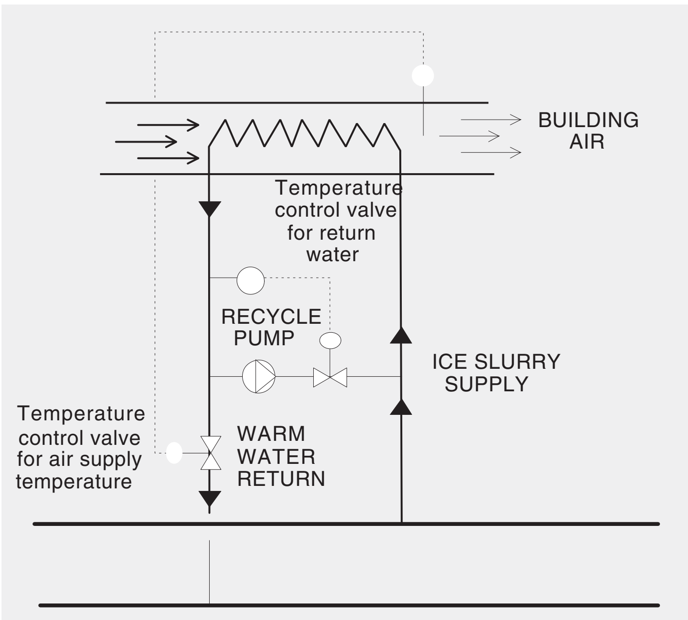
Ice Slurry in Customer Installations

The use of ice slurry in district-cooling networks through reduced flow rates can dramatically affect the distribution pipe diameters and pumping power requirements. However, using ice slurry directly in existing buildings designed for chilled water still requires additional consideration. The section below outlines three implementation strategies: direct ice slurry, distributed storage and warm water recycle.

Direct Ice Slurry

The direct ice slurry method involves sending the ice slurry through the distribution network to the cooling coils as shown in Figure 49. Ice slurry mixed with return water is supplied to the cooling coils where air near 25°C passes over the coils and exits near 18°C to meet the cooling demand. The coil return temperature controls the flow of re-circulated water. The building air temperature controls the flow of ice slurry. The return water temperature at design day conditions would be approximately 14°C and the supply temperature near zero degrees.

Figure 49
Direct Ice Slurry Utilization [21]



Initially, this may appear to be the simplest approach for implementing ice slurry at the end user. However, the technical problems associated with this concept may require further attention. Existing heat transfer equipment installed in large buildings has been designed for chilled water and not ice slurry. A direct conversion of existing chilled water equipment into ice slurry service would result in a lower internal heat transfer coefficient and an increased temperature difference. It would also create a potential for ice blockage within the heat exchanger. However, recent experiments with plate heat exchangers at the National Research Council in Ottawa, Canada, indicate that the potential for ice blockage in this kind of equipment is very limited. In multi-channel heat exchangers, blockages are not a great concern. A blocked passage would experience reduced mass transfer and its contents would warm up and melt the ice blockage.

The increased energy carrying capacity of ice slurry will substantially decrease the required mass flow to meet a given cooling load. Even a modest fraction of ice (20%) will reduce the flow rate by a factor of 5.2 in chilled water cooling coils originally designed for a temperature differential of 6°C. The reduced flow rate translates directly into a lower fluid velocity inside the exchanger.

The heat transfer coefficient (h) within the coil is proportional to the Reynolds number (Re) raised to the 0.8 power and the Prandtl number (Pr) raised to the 0.4 power or:

$$h = 0.023 \cdot Re^{0.8} \cdot Pr^{4}$$

Neglecting the slurry's effect on viscosity and molecular momentum, the Prandtl number can be taken as a constant and the change in the Reynolds number will be limited to velocity effects only. Under these assumptions, the internal heat transfer coefficient would be proportional to the reduction in velocity to the 0.8 power. Consequently, the internal heat transfer coefficient will be reduced by a factor of 3.7 for a chilled water coil originally designed for a 6°C temperature difference when operated with an ice fraction of 20%. This means that if the system is operated this way, the heat exchanger surface area may need to be extended.

The final impact of a lower internal heat transfer coefficient on the actual cooling capacity of the coil will depend on the combined effects of external and internal heat transfer coefficients on the overall heat transfer coefficient.

The temperature difference between the air and slurry will be greater than the original air to chilled water temperature difference. If a system originally designed for a supply of 6°C water will be used with ice slurry, it must be verified by an analysis. It should be verified that the magnitude of this increase is sufficient to offset the reduced internal heat transfer coefficient due to lower flow velocities.

Based on experiences in North America and Europe, cost estimates of direct connection heat exchangers including controls are summarised in Figure 50.

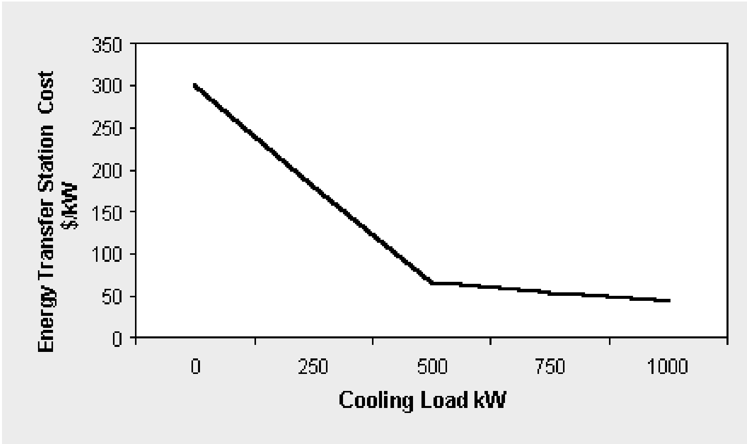


Figure 50
Energy Transfer Station Investment [18]

Distributed Ice Storage

One solution to overcome the potential problem associated with sending ice slurry directly into conventional energy transfer equipment is to implement distributed ice storage. The local ice storage is then used to separate ice from the carrier water. Ice-free water can then be pumped from the storage tank and delivered to the air-handling equipment.

The flow rate of the cold water can be selected to match design requirements of the existing chilled water equipment. Similarly, the design inlet temperature requirements can be matched by installing a mixing valve upstream of the circulation pump. One part of the warmed water leaving through the heat exchanger is returned to the ice storage tank for re-chilling as the water passes through the ice pack. The other part is re-circulated with the cold water entering from the tank. The concept is shown in Figure 51.

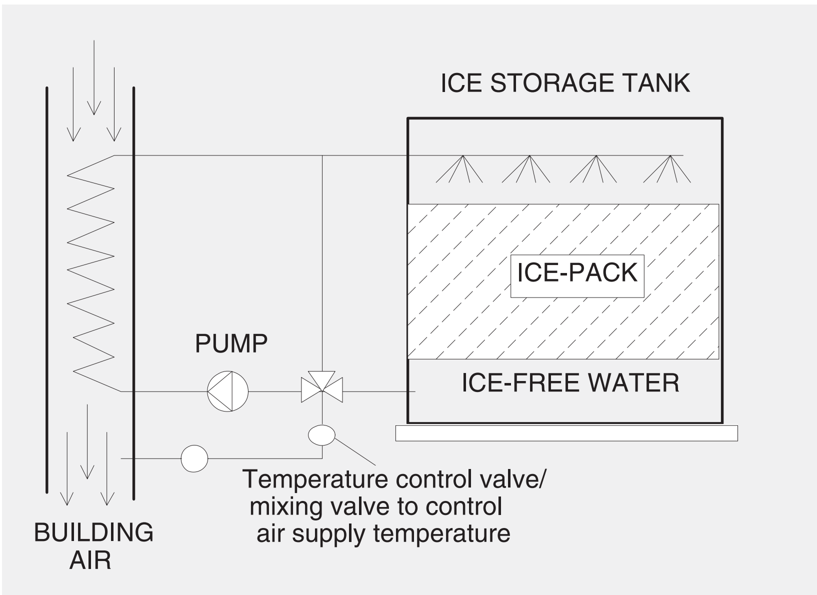


Figure 51
Distributed Ice Slurry Utilisation [21]

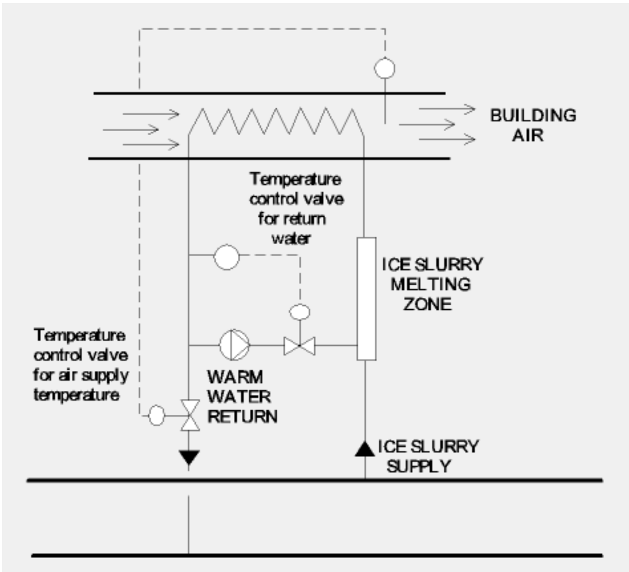
The distributed ice storage technique is a relatively simple way to integrate ice slurry without affecting the operation of existing heat transfer equipment. Distributed ice storage also provides a simple means of load levelling. This is accomplished by sizing the chiller to meet some average load rather than the designed peak load. The peaking load is met with the stored energy capacity. This stored energy is accumulated during off-peak times, mainly during the night-time. The stored energy is then dispensed during peak cooling times during the day.

Warm Water Recycle

The warm water recycle concept was conceived to facilitate the use of ice slurry without affecting the design conditions of existing heat transfer equipment and at the same time eliminate the need for local ice storage. The basis of the idea is to transform low-flow ice slurry of high cooling capacity to high-flow chilled water of lower cooling capacity.

The method involves the use of a warm water recycle stream from the discharge of the heat exchanger as shown in Figure 52. The ice-slurry melting zone may be a simple mixing chamber or even a length of pipe. The key design question is to melt all ice fractions with the recycle water. If a 20% ice fraction slurry were supplied, then the ratio of warm water recycle at 14°C to slurry would need to be about 1.2.

Figure 52
Warm Water Recycle [21]



The warm water is mixed with the incoming ice slurry to completely melt the ice fraction and produce the desired flow rate and design inlet temperature. In this way, the exact design conditions of both flow and temperature can be matched for each specific end user around the entire district-cooling network.

Both the “Distributed Ice Storage” and “Warm Water Recycle” concepts have been installed in a working district-cooling system at the Canmet Energy Technology Centre of the Department of Natural Resources Canada [19]. Two buildings with a peak cooling load of 140 kW are cooled by melting the ice slurry supplied from a chiller and storage system located 75 metres away. The ice slurry chiller and storage are located in one building. Cooling for this building is supplied by cold water from the storage tank thus simulating a distributed storage system. The other building is cooled with a 5 cm diameter pipeline connected directly to the ice generator. Ice slurry is pumped to this remote building where the heat of fusion of the ice is used to cool the secondary side water of the building air-conditioning system.

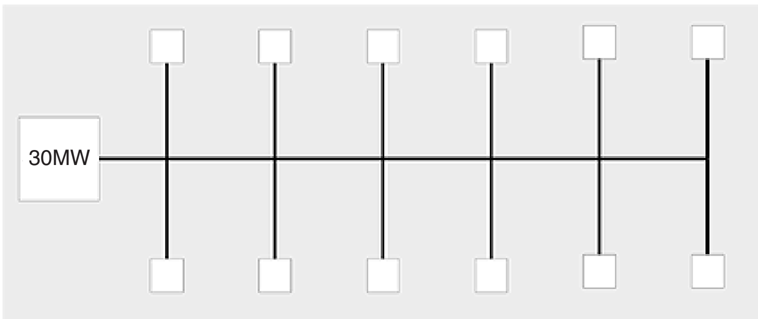
Case studies

Description of Cases

In this chapter comparative case studies are made between different storage technologies In case studies different cooling storage types are applied to the cooling system of a fictive district in two different kinds of climate conditions. The selected locations are cities in Northern Europe (NE) and Southern Europe (SE). The base design data gathered/obtained from Stockholm represent the conditions in a NE city, and data from Barcelona the conditions in a SE city.
Even though both locations are European locations they represent a colder climate and a warmer climate with a substantial spread in utilization hours, etc. Thus, conclusions from the case studies can also be drawn for other locations with other climate conditions.

The centralised cooling system will supply similar districts in both locations and in all cases studied in order to limit the number of variables in the calculations. The district is considered to be a large area consisting of new blocks of buildings and one central cooling plant. There are 12 buildings all alike with a total cooling demand of 30 MW. The chilled water from the plant is distributed through a pipe network to the buildings. The dimension of pipes varies with different supply temperatures. A simple layout of the district cooling system is shown in Figure 53.

Figure 53
District-cooling system for the case studies. 30 MW system capacity.



District-cooling Design Parameters

In theory, a district energy system can be designed according to detailed demand and energy use calculations of each individual building. However, the mathematical sum of each individual calculation is rarely equivalent to the district energy system in reality for the following reasons:

- Statistics of energy use and demand in existing buildings is commonly not available
- HVAC calculations tend to overestimate cooling demand and underestimate energy use
- The district energy rate structure affects the energy use pattern of the customer
- The energy use of a building changes with changing occupancy
- With many customers connected to the district energy system the peak load at the plant will be lower than the sum of each individual peak due to “diversification”

Therefore, the design of a district energy system is normally based on “rule-of-thumb” values for cooling demand, energy use, load patterns, etc. especially in the feasibility stage. Typical parameters are discussed in the following.

Ambient temperature

The ambient temperature is one of the used design parameters when evaluating the feasibility of a district-cooling system. High design ambient temperatures combined with low annual average temperatures indicate low utilization hours for the system and vice versa. Monthly maximum average ambient temperatures for Barcelona and Stockholm are shown in the diagram below and compared with the monthly ambient temperature in some other locations. The data below is gathered from the "International Station Meteorological Climate Summary", which is based on a maximum daily average temperature over a 20-year period. As can be seen in Figure 54, the temperature curve for Barcelona is "flatter" than the curves for Stockholm and Chicago indicating higher utilization hours for a cooling system in Barcelona.

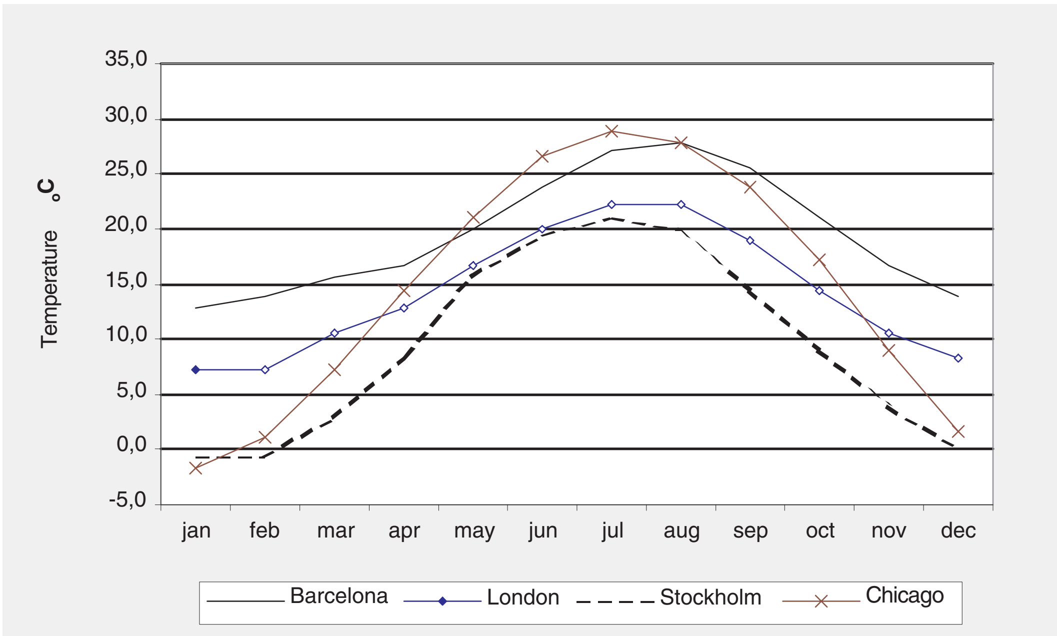


Figure 54
Maximum daily average ambient temperature for different locations. Source: "International Station Meteorological Climate Summary"

The design dry bulb ambient temperature is 26°C for Stockholm, 31°C for Barcelona and 34°C for Chicago according to ASHRAE. However, the wet bulb temperature should also be considered while the cooling demand for de-humidification purposes can be high when the wet bulb starts to climb above 20°C. Design wet bulb temperature is 18°C for Stockholm, 24°C for Barcelona and 25°C for Chicago.

Cooling Degree Days

The parameter Cooling Degree Days (CDD) for a building is defined as the difference between the daily ambient temperature, normally calculated as the average of the daily maximum and minimum temperature, and an indoor design temperature, normally set to 18°C. The annual CDD is the sum of the degree days per day.

CDD with 18°C base temperature considers only the cooling load from the ambient air and does not include internal heat load generated inside the building. The internal cooling load results from internal heat sources such as lighting, computers and people, etc. The internal cooling load can have a significant effect on the total cooling load and therefore must be considered. In countries with hot and humid climate the share of internal load of total cooling load is not very important, but in northern climates the internal cooling load can be significant.

In Barcelona the number of cooling degree days according to weather data is 471, based on the Celsius scale with a base temperature of 18°C. This can be compared with Stockholm that only has 40 cooling degree days. The bar chart in Figure 55 compares the cooling degree days for a selected number of locations.

Figure 55
Degree days in different locations based on Celsius scale.

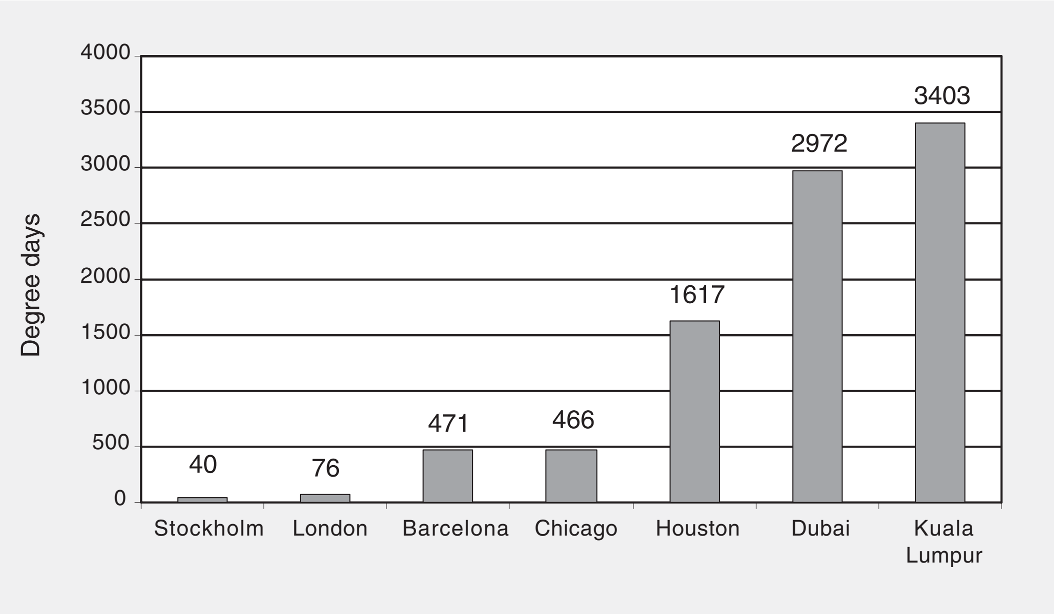
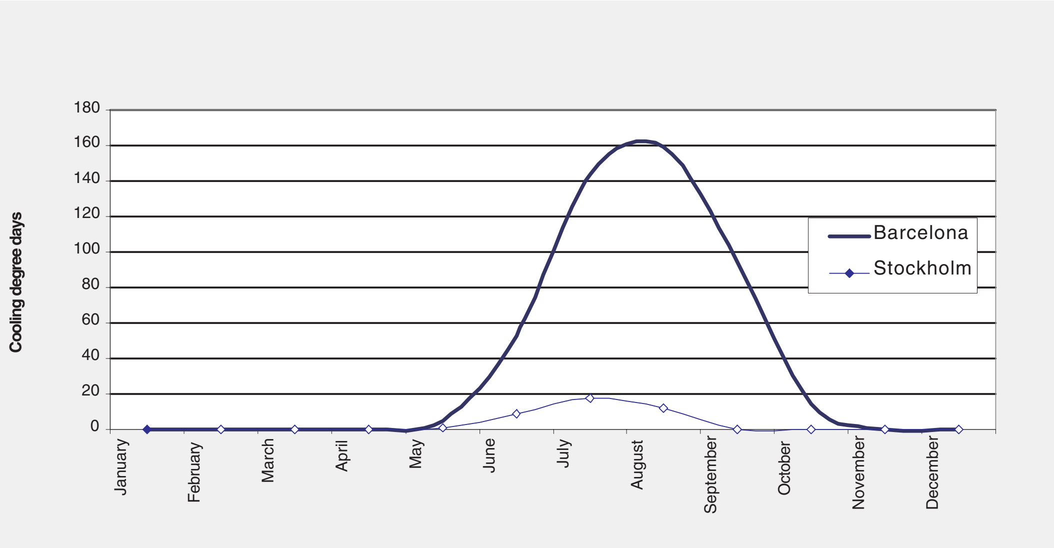


Figure 56
Degree days per month in Barcelona and Stockholm



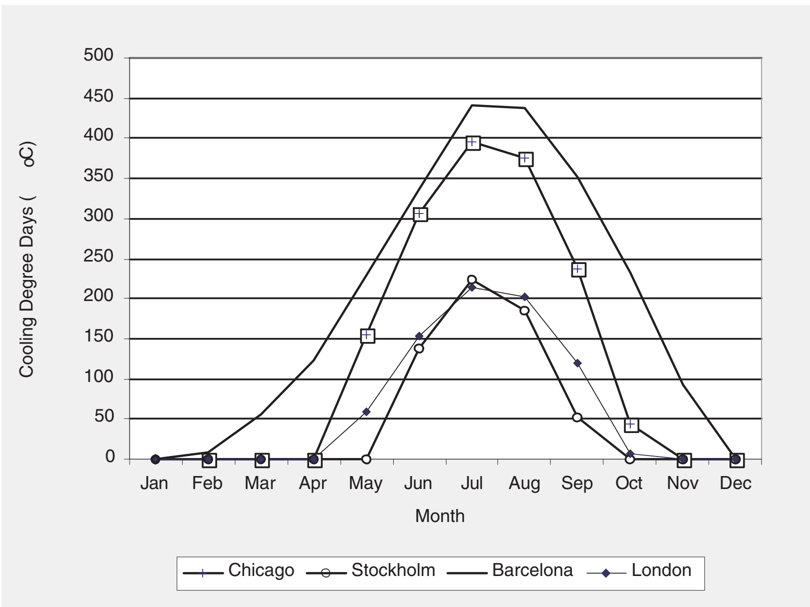
Utilization hours for Buildings

Equivalent full load utilization hours (EqFLH) per year represent the number of hours the system would sustain full load to generate the equivalent amount of energy that is generated over the course of a year. The utilization hours for an individual building are nowadays normally calculated with HVAC computer software. The calculations tend to overestimate the peak demand and underestimate the energy use compared to what is achieved when the building is connected to district cooling. Also, a detailed energy balance for each individual building can be difficult to perform for an initial feasibility study of a district-cooling system. A more generalised estimate of the utilization hours can be based on the cooling degree days and the temperature difference between the design ambient temperature Tmax in the location and the base temperature Tbase, which is normally 18oC, as shown below. With the addition of internal cooling load based on experience this will indicate the utilization hours for the area. However, when using the 18°C base temperature, the "internal cooling load" will be a substantial part of the total utilization hours, especially, for colder climates such as Stockholm. A lower base temperature of about 10°C, which is closer to the base temperature where modern office buildings start to need cooling, will add internal cooling load to the CDD.

$$\text{Utilization hours} = \frac{CDD \cdot 24}{T_{max} - T_{base}} + \text{internal load}$$

While the actual CDD with another base than 18°C is difficult to obtain, CDD based on average monthly temperatures and with a base temperature of 10°C is used in Figure 57. For some areas it is possible to obtain "growing degree days" with a variable base temperatures that can be used for this purpose. Based on the CDD with a base temperature of 10°C, equivalent full load utilization hours for Stockholm of slightly below 1000 hours can be calculated according to the formula above without the addition of an internal load factor. With the same calculations the EqFLH for Chicago is about 1500 hours and for Barcelona over 2000 hours.

Figure 57
Cooling degree days based on monthly average temperatures and 10°C base temperature



The utilization hours in Scandinavia are in approximately 1000 hours/year for normal office buildings without special demands for base-load cooling. In northern US the EqFLH is in the 1000-1500 hour range with a system in Chicago experiencing about 1400 hours. The full load utilization hours can increase substantially in case any kind of machinery with high power consumption, creating excess heat, such as computer servers, etc., is placed in the building. In some cases Swedish district-cooling systems have witnessed around 1500-2000 hours/year for individual buildings with high internal load and good insulation.

In this study, full load utilization hours of 2000 hours/year were used to estimate the cooling energy consumption in connected buildings in the case of Barcelona and 1000 hours in Stockholm.

Diversification factor

Diversification factors are commonly in the range of 0.90. In Chicago there have been cases with 0.85 and as low as 0.80. The factor tends to be lower when the central production plant serves a mix of office and residential buildings with peak demand occurring at different times during the day. Cases with higher diversification factors than 0.90 are also experienced when the district-cooling system customers are few and have much alike operation schedules.

As a system has to be designed for a "worst case" scenario, a diversification factor of 0.95 is estimated for district-cooling systems in the homogeneous office regions reviewed in this study. In other words, the maximum load that the central plant would experience is the sum of the peak loads of the connected building multiplied by 0.95.

Cooling load duration

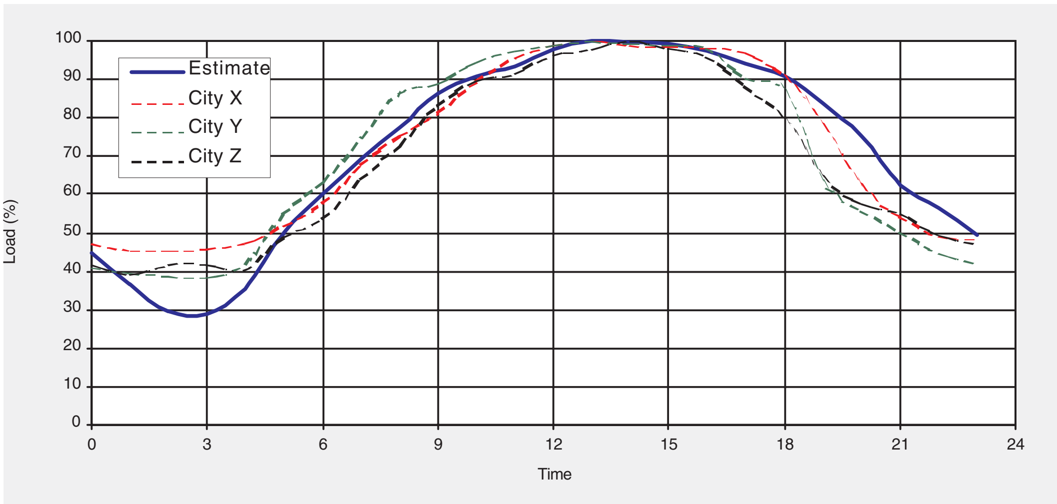
Daily load curve

A daily load curve is a very important tool to size the plant equipment and to estimate the relation between chiller capacity and thermal storage. The daily load curve is preferably based on the actual data on the district-cooling system where a storage tank will be installed. If the actual data is not available, the daily load curve can be based on the data on other similar systems or the ambient temperature data. In this case study the daily load curve shown in Figure 58 is calculated based on the hourly temperature data on Stockholm, Sweden. The curve shows that the load reaches 100% of the total cooling capacity at about 13.00 in the afternoon.

The curve is very similar to the curves from two cities in Sweden and one in North America. Even if the load pattern differs somewhat, the daily average load for all curves is about 70%. A daily load curve will never "be right" even for a specific system because it will always vary depending on the customer base at the moment, "worst case" scenarios for dry and wet bulb temperature variations over the day, etc. The "synthetic" curve in Figure 58 is therefore used for both case studies.

Figure 58
Daily load curve for a typical plant in Europe.

Figure 59
The daily load curve in comparison with actual curves from cities in Sweden and North America.

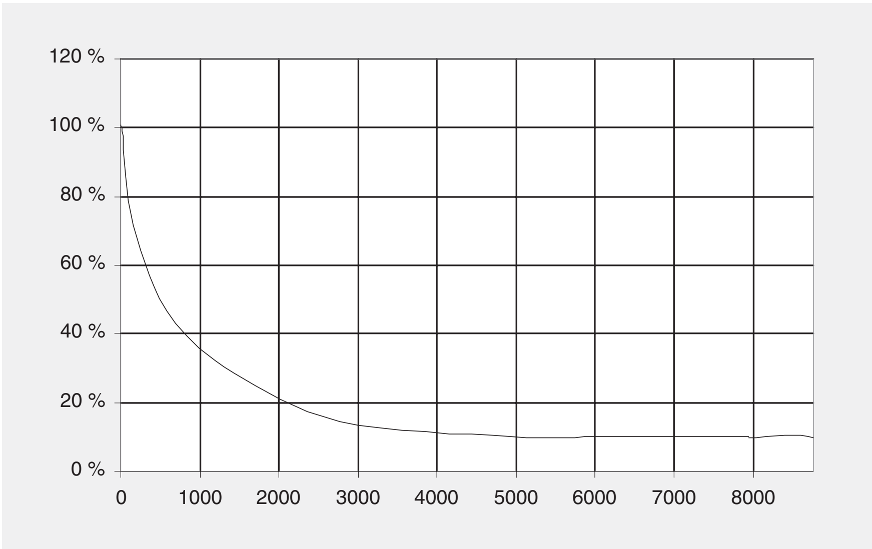


Annual Load Curve

The load duration curve graphically illustrates the relationship between the system load and the duration time of the load. The curve is a proven planning tool used for initial selection of the production plant equipment and facilities and sizing of the distribution system piping to minimize capital expenditure.

The load duration curves of the systems in the case study are based on weather data on Stockholm and Barcelona and experience gained from other district-cooling systems. The annual load duration curve of a system is displayed in Figure 60.

Figure 60
Annual cooling load curve



Utilization hours of the Central Production Plant

When using the selected diversification factor, the capacity of the central production plant will be lower than the sum of peak demands of the buildings. The same amount of cooling energy must still be produced in the production plant, thus giving a longer utilization time for the central plant than for the buildings themselves. The plant utilization hours are adjusted with the selected diversification factor to calculate total utilization hours for the central production plant in the case of SE/Barcelona (2000/0.95 = 2100 hours) and in the case of NE/Stockholm (1000/0.95 = 1050 hours).

Specific cooling loads

The specific cooling load for a building, given as W/m², depends on the amount of heat the building is exposed to above the preferred indoor temperature. The heat can either come from the outside through the ambient air as well as via direct sun radiation or from heat sources inside the building, such as people and appliances (computer equipment in offices etc.).

The specific cooling load and other design data for a district-cooling system such as cooling degree days and utilization hours vary depending on the local climate. Specific cooling loads for different systems and cities are shown in Table 14. In Stockholm the cooling load for office buildings is typically 40-50 W/m².

Table 14
Comparison of specific cooling loads and design criteria for different systems/cities.

System/City	Country	Type of building	Specific cooling load W/m²	Cooling degree days Base 18°C	Design dry bulb temp. °C	Utilization hours h
Stockholm	Sweden	Offices	40-50	40	26	1000
St Paul	USA	Offices	60	402	33	1200
Chicago	USA	Offices	70	466	34	1 400

In the case studies a specific cooling load of 50 W/m² is used for NE/Stockholm and 60 W/m² for SE/Barcelona with a design dry bulb temperature of 31°C.

Energy Consumption

The energy consumption of each individual building largely depends on how the building is utilized, as well as on the construction of the building, its location, the ambient air data and exposure to sun radiation. All these parameters vary from building to building and each building has an individual specific cooling consumption.
As the cooling load in a building depends on both external and internal sources, in some cases cooling load may be required during the whole year. In the Scandinavian countries where the climate is rather cold during part of the year the cooling demand is due to internal sources, which makes a district-cooling system feasible.

The buildings in this case study are assumed to be new, modern buildings, which means for Stockholm especially good insulation and a larger demand for cooling.

However, the above estimates of equivalent full load utilization hours (EqFLH) and specific cooling load can be used to calculate a generalized value of the cooling energy consumption for a building.

The total annual cooling energy can be calculated based on the equivalent full load utilization hours of the buildings and the cooling demand of total buildings. In the NE/Stockholm case the expected consumption is 30 MWc/0.95 x 1000 hours = 31;500 MWh of cooling and in the SE/Barcelona case the expected consumption is 30 MWc/0.95 x 2000 hours = 63,000 MWh of cooling.

Case studies: initial values

Location: Northern Europe (NE)

The first case is a north European country where Stockholm represents a typical city. The district-cooling system is considered to serve new buildings in a large area in a city. There is one central production plant and the chilled water is distributed to similar blocks in one area.

The following design data is used for the NE/Stockholm case:

Building cooling demand	31.5 MW
Specific cooling load	50 W/m^2
Building area	630 000 m^2
Building EqFLH	1 000 hours
Cooling degree days	40
Diversification factor	0.95
Annual cooling energy	31 500 MWh

Location: Southern Europe (SE)

The district-cooling system area in Barcelona in southern Europe (SE/Barcelona) is considered to be similar to that in Stockholm to make comparison possible. Barcelona represents a typical city and as in Stockholm the area consists of new buildings and one central production plant. However, the size of the blocks is larger in Stockholm than in Barcelona. In Stockholm the specific cooling load is smaller and for that reason, more pipes and a larger piping network are needed. This is taken into consideration in the capital cost estimates.

The following design data is used for the NE/Barcelona case:

Building cooling demand	31.5 MW
Specific cooling load	60 W/m^2
Building area	525 000 m^2
Building EqFLH	2 000 hours
Cooling degree days	471
Diversification factor	0.95
Annual cooling energy	63 000 MWh

Distribution networks

The assumptions are a new system with new buildings. There are 12 blocks of buildings with a cooling demand of 2.5 MW each. The area of a block is 6400 square metres and the buildings are 10 floors high.

Trench length per MW is a measure of the density in the network. The length of how far 1 MW reaches is different in networks depending on the outtake rate of energy to customer. In larger cities customers are located more densely and the outtake rate is larger per meter. In this case study the trench length per MW is assumed to be 120 m/MW in Stockholm and 100 in Barcelona. The total network length is then accordingly 3000 metres in Barcelona and 3600 metres in Stockholm.

Table 15
District-cooling pipe cost

Dimension (mm)	Total price (USD/m)
600	1270
500	1100
450	1015
400	930
350	845
300	760
250	675

Total transfer piping costs are calculated based on the above assumptions and are added to the investment cost estimates. For a system with chilled water storage and with a supply temperature of 7°C, the transfer piping cost is USD 3 261 300 in Stockholm and USD 2 717 750 in Barcelona. For a solution tank system with a supply temperature of 1°C, the transfer piping cost is USD 2 674 800 in Stockholm and USD 2 229 000 in Barcelona.

For a system with ice storage and with a supply temperature of 2°C, the transfer piping cost is USD 2 848 200 in Stockholm and USD 2 473 800 in Barcelona. For a chilled water system with a supply temperature of 5°C, the transfer piping cost is USD 2 975 700 in Stockholm and USD 2 580 900 in Barcelona.

Customer connection

In this case study the connection to the customer is indirect through heat exchangers. Energy transfer stations, ETSs, are installed in each building.

The investment cost of an ETS is limited to the heat exchangers, control valves, expansion tank and shut-off valves, based on Swedish prices. The price is USD 45/kW, which gives a total investment cost of USD 1 350 000.

Selected Case Storages

The effect of the storage on the total economy of the cooling system is studied by applying different sizes of storage to the system. The selected storage discharge capacities in different calculation cases are:

Case 0: Electrical chillers and no storage
Case 1: Storage discharge 20% of total load, i.e. 6 MW.
Case 2: Storage discharge 30% of total load, i.e. 9 MW.
Case 3: Storage discharge 40% of total load, i.e. 12 MW.

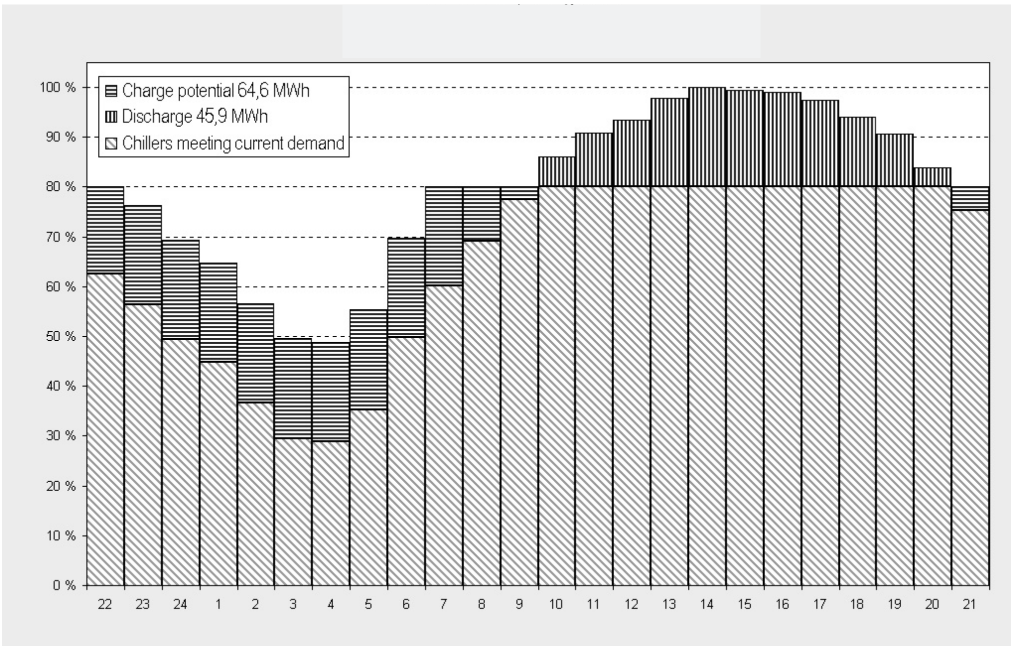
Storage dimensioning and operation strategy

The operation mode with chilled water storage is a combination of load levelling and demand limiting operation modes. The storage is dimensioned according to the load curve of the hottest day so that the total capacity of the system meets the cooling load peak and total demand of the day.

In Case 1 the storage discharge (and charge) capacity is 20% of the peak load, i.e. 6 MW. The peak load of the hottest day determines the required chiller capacity to 24 MW (80% of peak demand). With this dimensioning, the system is required to meet 45.9 MWh cooling demand with stored cooling energy (see). Thus, in Case 1 the cooling system design values are:

- Chillers, total 24 MW
- Storage charge/discharge capacity 6 MW, energy storage capacity 46 MWh.

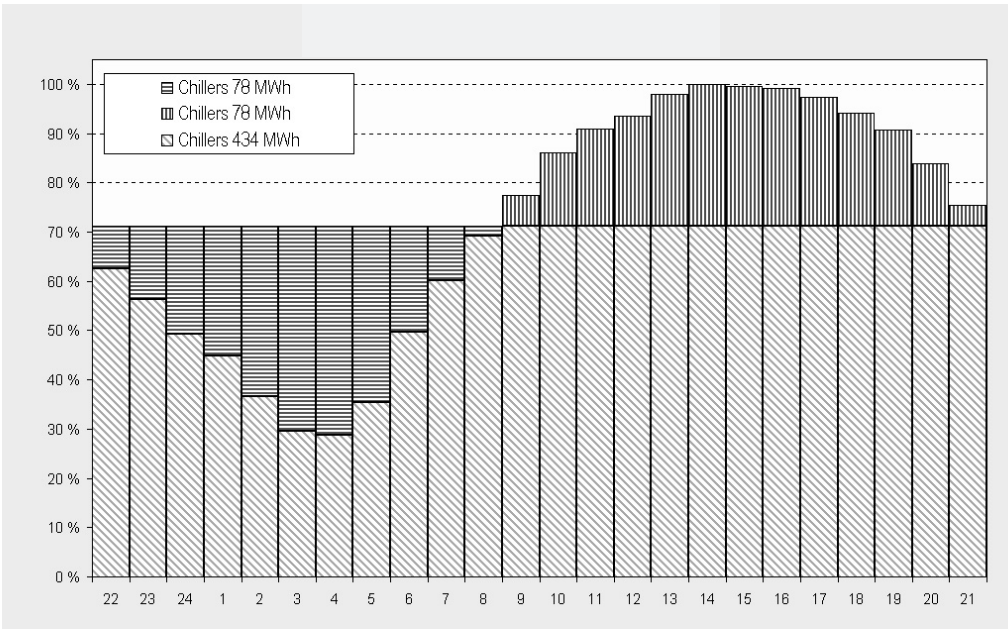
Figure 61
System dimensioning principle
with 20% storage.



In Case 2 the storage discharge capacity is 30% of the peak load, i.e. 9 MW. Charging capacity must be 12.7 MW in order to charge the storage full during the night of the hottest 24 hours. The chiller capacity is 70% of total load. With this dimensioning, the system is required to meet 78 MWh cooling demand with stored cooling energy (see Figure 62). Thus, in Case 2 the cooling system design values are:

- Chillers, total 21 MW
- Storage charge capacity 9 MW, discharge capacity 12.7 MW and energy storage capacity 78 MWh.

Figure 62
System dimensioning principle
with 30% storage.

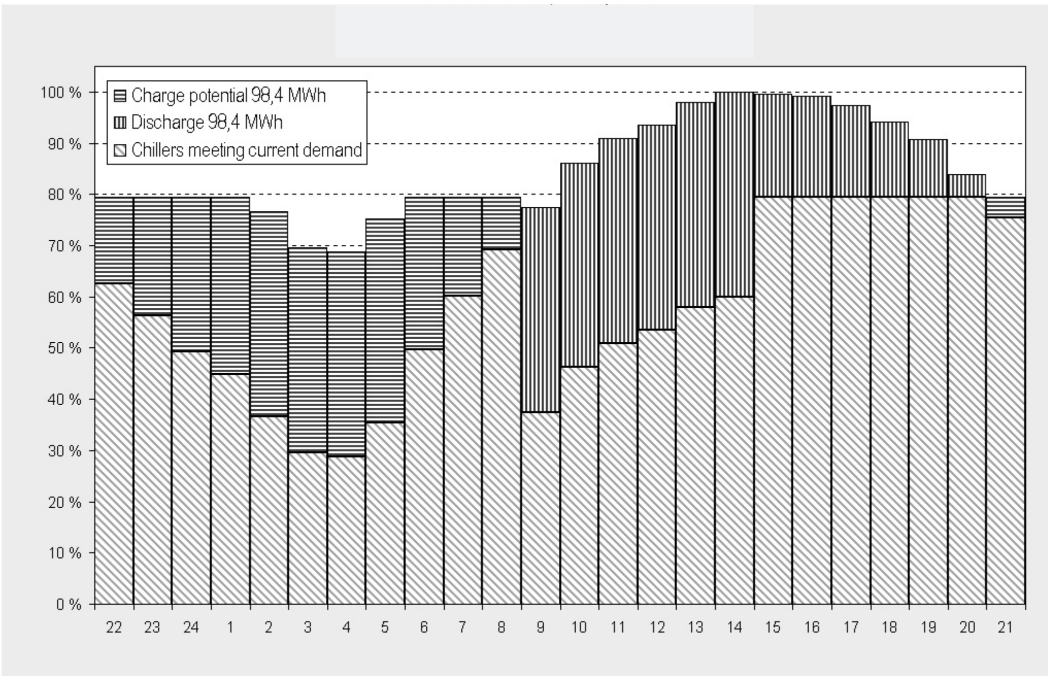


In Case 3 the storage discharge capacity is selected to be 40% of the peak load. Contrary to Case 1 and Case 2 the system cannot be dimensioned to have 100% capacity consisting of 40% storage and 60% chiller capacity. The limit condition is storage 30% and chiller capacity 30% (as in Case 2). If the chiller capacity is further decreased and storage capacity increased, the system cannot charge enough cooling energy into the storage during the low load to cover the high load cooling demand during the day. Therefore, the dimensioning principle of 40% storage case is different from Case 1.

In Case 3 the storage discharge (and charge) capacity is 12 MW. Storage is dimensioned according to demand limiting principle (see Figure 63). Storage is discharged with maximum capacity when electricity prices are high. To be able to store all the needed cooling energy, the system must have excess chiller capacity. Thus, the storage capacity is 40% (12 MW) and chiller capacity 80% (24 MW) resulting in a total system with 20% excess capacity. With this dimensioning, the system is required to meet 100 MWh cooling demand with stored cooling energy (see Figure 63). In Case 3 the cooling system design values are:

- Chillers, total 24 MW
- Storage charge/discharge capacity 12 MW, energy storage capacity 100 MWh.

Figure 63
System dimensioning principle with 40% storage.



The dimensioning method in Case 3 (demand limiting method) is usually feasible only in conditions with significant difference between the low and high electricity tariffs.

Chilled Water and Solution Tank Storage

Two types of chilled water storage are considered in the case studies:

1. Conventional chilled water storage with supply temperature of +7°C (return temperature is +14°C)
2. Low supply temperature (+1°C) storage charged with sodium nitrate /nitrite solution

The chilled water tank technology is widely used in several locations, while the sodium nitrate /nitrite solution represents more recent development with a couple of references since 1994. Low supply temperature is beneficial in many ways since it reduces the size of storage tank, pumps, distribution pipes and the heat exchanger equipment at consumer installations.

Production plants

To compare different storage options several scenarios have been prepared for a central production plant with different storage configurations, which are:

Scenario 1: Electrical chillers and no storage
Scenario 2: Chilled water storage, 40% storage capacity
Scenario 3: Chilled water storage, 30% storage capacity
Scenario 4: Chilled water storage, 20% storage capacity
Scenario 5: Solution tank storage, 40% storage capacity
Scenario 6: Solution tank storage, 30% storage capacity
Scenario 7: Solution tank storage, 20% storage capacity

Table 16
Production plant configurations, chilled water and brine ($NaNO_3$)

	Storage capacity	Water chillers		Ice chillers/ Ice generators	
		No	MW	No	MW
Full mechanical chilling	0%	6	30	-	-
Chilled water / Brine ($NaNO^3$)	40%	4	18	-	
	30%	4	21	-	
	20%	5	24	-	

The production plant cost is estimated including all necessary equipment such as chillers, heat rejection equipment, pumps, mechanical piping, storage tanks, building, ventilation, etc.

The investment costs of a production plant, distribution piping network and customer connections are estimated and a grand total is calculated for every storage type size.
The calculations for chilled water and solution tank storage show that moderate-sized storage alternatives require a lower investment than the alternative without storage.

The investment costs of the seven scenarios are presented in the tables below.

Table 17
Investment costs of water-based storage, NE

NE / Stockholm	Chilled Water 40 % Storage	Chilled Water 30 % Storage	Chilled Water 20 % Storage	Full Mech. 7/14°C Chilling	Solution Tank 40 % Storage	Solution Tank 30 % Storage	Solution Tank 20 % Storage	Full Mech. 1/14°C Chilling
	USD	USD	USD	USD	USD	USD	USD	USD
Chillers, heat exch. and cooling towers	1 758 173	1 563 079	1 758 173	2 092 787	2 278 181	2 021 886	2 278 181	2 847 727
Pumps, pipe and insul.	3 163 377	3 200 739	3 163 377	3 349 322	2 841 142	2 785 681	2 841 142	2 938 643
Storage tank/coils or slush gener…	1 543 653	1 218 242	844 653	104 566	1 195 504	948 060	664 504	149 380
Building and auxiliaries	1 344 000	1 192 800	1 344 000	1 680 000	1 712 941	1 594 353	1 712 941	1 976 471
Electrificat. and automation	1 736 225	1 602 775	1 736 225	2 032 781	2 032 781	1 865 968	2 032 781	2 668 258
Others	1 525 170	1 525 170	1 525 170	1 397 263	1 525 170	1 525 170	1 525 170	1 404 758
Total plant cost	11 070 598	10 302 804	10 371 598	10 656 719	11 585 719	10 741 118	11 054 719	11 985 236
Transfer Piping	3 261 300	3 261 300	3 261 300	3 261 300	2 674 800	2 674 800	2 674 800	2 674 800
ETS cost	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000
Total Distribution cost	4 611 300	4 611 300	4 611 300	4 611 300	4 024 800	4 024 800	4 024 800	4 024 800
Contingency 10 %	1 568 190	1 491 410	1 498 290	1 526 802	1 561 052	1 476 592	1 507 952	1 601 004
Total USD	17 250 088	16 405 515	16 481 188	16 794 821	17 171 571	16 242 510	16 587 471	17 611 040

Table 18
Investment costs of water-based storage, SE

SE / Barcelona	Chilled Water 40 % Storage	Chilled Water 30 % Storage	Chilled Water 20 % Storage	Full Mech. 7/14°C Chilling	Solution Tank 40 % Storage	Solution Tank 30 % Storage	Solution Tank 20 % Storage	Full Mech. 1/14°C Chilling
	USD	USD	USD	USD	USD	USD	USD	USD
Chillers, heat exch. and cooling towers	1 790 610	1 591 866	1 790 610	2 133 333	2 312 564	2 052 401	2 312 564	2 890 706
Pumps, pipe and insul.	3 165 445	3 196 259	3 165 445	3 352 201	2 871 322	2 815 231	2 871 322	2 969 569
Storage tank/coils or slush gener…	1 543 653	1 218 242	844 653	104 566	1 195 504	948 060	664 504	149 380
Building and auxiliaries	1 344 000	1 192 800	1 344 000	1 680 000	1 712 941	1 594 353	1 712 941	1 976 471
Electrificat. and automation	1 736 225	1 602 775	1 736 225	2 032 781	2 032 781	1 865 968	2 032 781	2 668 258
Others	1 530 598	1 530 598	1 530 598	1 404 758	1 530 598	1 530 598	1 530 598	1 404 758
Total plant	**11 110 531**	**10 332 540**	**10 411 531**	**10 707 639**	**11 655 711**	**10 806 610**	**11 124 711**	**12 059 141**
Transfer Piping	2 717 750	2 717 750	2 717 750	2 717 750	2 229 000	2 229 000	2 229 000	2 229 000
ETS cost	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000
Total Distribution	**4 067 750**	**4 067 750**	**4 067 750**	**4 067 750**	**3 579 000**	**3 579 000**	**3 579 000**	**3 579 000**
Contingency	1 517 828	1 440 029	1 447 928	1 477 539	1 523 471	1 438 561	1 470 371	1 563 814
Total USD	**16 696 109**	**15 840 319**	**15 927 209**	**16 252 928**	**16 758 182**	**15 824 172**	**16 174 082**	**17 201 955**

Ice Storage (ice-on-coil, ice slurry)

The capital cost is calculated for production plants utilising two types of thermal ice storage. The storage types are external-melt ice-on-coil and ice slurry. The external-melt ice-on-coil storage is commercially available and used in several locations while ice slurry is under development with a promising potential for reducing distribution costs.

The dimensioning and operation principles of ice storage systems differ from the chilled water systems because the charging of ice storage takes lower temperatures than the distribution of cooling. It is feasible to use different types of chillers for charging the storage and for producing chilled water for instant use. In the following different chiller and storage combinations to meet the cooling demand are presented.

Production plants

To compare different storage options several scenarios have been prepared for a central production plant with different storage configurations, which are:

Scenario 1: Electrical chillers and no storage

Scenario 2: Ice-on-coil storage, 40% storage capacity

Scenario 3: Ice-on-coil storage, 30% storage capacity

Scenario 4: Ice-on-coil storage, 20% storage capacity

Scenario 5: Ice slurry storage, 40% storage capacity

Scenario 6: Ice slurry storage, 30% storage capacity

Scenario 7: Ice slurry storage, 20% storage capacity

In Scenario 1 the production plant consists of six electrical centrifugal chillers with a capacity of 5 MW each. The chillers are package type and use R134a as refrigerant.

The other scenarios include different numbers of ice chillers generating ice to the storage and the additional load is from base chillers.

In Scenario 2, there are three base chillers with a capacity of 13.8 MW operating continuously and three ice chillers with a capacity of 11.8 MW generating ice for the ice-on-coil storage or directly to the load. In Scenario 3, there are three chillers with a capacity of 15 MW and two ice chillers with a capacity of 9.5 MW. In Scenario 4 with 20% storage four chillers with a capacity of 19.2 MW operate and two ice chillers with a capacity of 6 MW generate the ice.

Table 19
Production plant configurations, chilled water and brine ($NaNO_3$)

	Storage capacity	Water chilling capapcity		Ice chilling capacity	
		No	MW	No	MW
Full mechanical chilling	0%	6	30	-	-
Ice-on-coil	40%	3	13.8	3	11.8
	30%	3	15	2	9.5
	20%	4	19.2	2	6
Ice slurry	40%	3	14.2	3	11.5
	30%	3	18.7	3	7.1
	20%	4	22.1	2	3.9

The production plant cost is estimated including all necessary equipment such as chillers, heat rejection equipment, pumps, mechanical piping, storage tanks, building, ventilation, etc.

The investment costs of a production plant, distribution piping network and customer connections are estimated and a grand total is calculated for every storage type size.
The calculations for ice storage alternatives show a higher investment cost than the calculation for the scenario without storage. For the storage options the investment in storage equipment is determining but even the chiller equipment such as the chiller for producing ice has to work at lower temperatures and extra, special chillers for ice generating are needed.

The investment costs of the seven scenarios are presented in the tables below.

Table 20
Investment costs of ice-based storage, NE

NE / Stockholm	Ice on coil 40 % Storage	Ice on coil 30 % Storage	Ice on coil 20 % Storage	Ice Slurry 40 % Storage	Ice Slurry 30 % Storage	Ice Slurry 20 % Storage	Full Mechanical Chilling
	USD	USD	USD	USD	USD	USD	USD
Chillers, heat exch. and cooling towers	2 457 795	2 356 200	2 077 630	2 681 584	2 351 166	2 162 544	2 092 787
Pumps, pipe and insul.	3 986 781	3 705 224	3 559 466	3 848 193	3 634 462	3 490 389	2 879 090
Storage tank/coils or slush gener…	1 269 274	968 811	734 188	1 643 803	1 088 213	692 364	104 566
Building and auxiliaries	1 901 366	1 894 799	1 898 648	1 912 750	1 904 407	1 902 229	1 919 458
Electrificat. and automation	1 916 943	1 880 284	1 902 961	1 982 785	1 933 837	1 921 499	2 032 781
Others	2 219 989	1 988 476	1 875 148	2 382 986	2 087 917	1 881 761	1 628 037
Total plant cost	13 752 148	12 793 794	12 048 041	14 452 101	13 000 002	12 050 786	10 656 719
Transfer Piping	2 848 200	2 848 200	2 848 200	2 848 200	2 848 200	2 848 200	2 975 700
ETS cost	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000
Total Distribution cost	4 198 200	4 198 200	4 198 200	4 198 200	4 198 200	4 198 200	4 325 700
Contingency 10 %	1 795 035	1 699 199	1 624 624	1 865 030	1 719 820	1 624 899	1 498 242
Total USD	19 745 383	18 691 193	17 870 865	20 515 331	18 918 022	17 873 885	16 480 661

Table 21
Investment costs of ice-based storage, SE

SE / Barcelona	Ice on coil 40 % Storage	Ice on coil 30 % Storage	Ice on coil 20 % Storage	Ice Slurry 40 % Storage	Ice Slurry 30 % Storage	Ice Slurry 20 % Storage	Full Mechanical Chilling
	USD	USD	USD	USD	USD	USD	USD
Chillers, heat exch. and cooling towers	2 493 120	2 356 200	2 114 110	2 715 202	2 351 166	2 197 570	2 133 333
Pumps, pipe and insul.	3 986 781	3 705 224	3 559 466	3 848 193	3 634 462	3 490 389	2 879 090
Storage tank/coils or slush gener…	1 269 274	1 066 761	734 188	1 643 803	1 088 213	692 364	104 566
Building and auxiliaries	1 901 366	1 904 407	1 898 648	1 912 750	1 898 648	1 902 229	1 919 458
Electrificat.,auto mation	1 916 943	1 880 284	1 902 961	1 982 785	1 933 837	1 921 499	2 032 781
Others	2 229 028	2 042 476	1 884 483	2 391 844	2 087 971	1 890 724	1 638 411
Total plant cost	**13 796 512**	**12 955 352**	**12 093 856**	**14 494 577**	**12 994 297**	**12 094 775**	**10 707 639**
Transfer Piping	2 473 800	2 473 800	2 473 800	2 473 800	2 473 800	2 473 800	2 580 900
ETS cost	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000	1 350 000
Total Distribution	**3 823 800**	**3 823 800**	**3 823 800**	**3 823 800**	**3 823 800**	**3 823 800**	**3 930 900**
Contingency 10	1 762 031	1 677 915	1 591 766	1 831 838	1 681 810	1 591 858	1 463 854
Total USD	**19 382 343**	**18 457 067**	**17 509 422**	**20 150 215**	**18 499 907**	**17 510 433**	**16 102 393**

Operating cost

The water price and electric tariffs from the two locations are used to calculate the operating cost. The water price and electricity tariffs are shown in Table 22 and the operating cost in Table 24… Table 27.

Table 22
Prices of electricity and water

		NE/Stockholm	**SE/Barcelona**
Electricity price high load	USD/kWh	0.080	0.110
Electricity price low load	USD/kWh	0.050	0.050
Water price	USD/m3	1.0	1.2

All the cases are dimensioned so that the peak capacity of the chillers is cut. This results in savings in operating costs due to optimised operation: increasing operation when electricity prices are low and decreasing operation when electricity prices are high. There are four different frame conditions that determine the optimised operation:

Table 23
Operation principle by electricity tariff

Electricitytariff	**Cooling load Chiller capacity**	**Operating principle**
HIGH	> 1	Storage is discharged as much as possible
LOW	> 1	Storage is discharged as much as neces sary (load - capacity)
HIGH	≤ 1	Storage is discharged as much as possible, if there is excess energy available in storage
LOW	≤ 1	Charge storage as much as possible

The operating costs include the annual electricity cost, annual water and chemical cost, annual plant, piping and ETS maintenance cost and the cost of operators and administrators. The maintenance cost of the plant is about 3% of the plant investment. For the piping it is 0.5% and for the ETS 1% of the investment. It is estimated that three persons could be employed to a cost of USD 36 000/year and person.

The non-storage, low supply temperature scenario with full mechanical chilling has the highest operating cost and the chilled water tank storage scenario has the lowest cost. The main reason is the electrical cost.
As the electricity tariffs are different in Stockholm and Barcelona the result between the two cases is different, in Stockholm the chilled water scenario with 30% storage has the lowest operating cost but in Barcelona the chilled water scenario with 40% storage has the lowest operating cost. The operating costs are shown in the tables below.

Table 24
Operating cost of waterbased cases, NE

NE / Stockholm	Chilled Water 40 % Storage	Chilled Water 30 % Storage	Chilled Water 20 % Storage	Full Mech. 7/14°C Chilling	Solution Tank 40 % Storage	Solution Tank 30 % Storage	Solution Tank 20 % Storage	Full Mech. 1/14°C Chilling
	USD	USD	USD		USD	USD	USD	USD
Electrical Cost	379599	381038	391123	416613	420 499	423 976	439 051	471 805
Water & Chemical cost	87 070	87 070	87 070	87 070	89 691	89 688	89 691	89 688
Plant Maintenance & Repair Cost	202 000	193 000	202 000	227 000	218 000	203 000	218 000	258 000
Piping system Maintenance	13 589	13 589	13 589	13 589	11 145	11 145	11 145	11 145
ETS systems Maintenance	13 500	13 500	13 500	13 500	13 500	13 500	13 500	13 500
Cost for operators and admin.	108 000	108 000	108 000	108 000	108 000	108 000	108 000	108 000
Total Plant Operating Cost	**803 758**	**796 197**	**815 282**	**865 772**	**860 835**	**849 309**	**879 387**	**952 138**

Table 25
Operating cost of ice-based case, NE

NE/ Stockholm	**Ice on coil** 40 % **Storage**	**Ice on coil** 30 % **Storage**	**Ice on coil** 20 % **Storage**	**Ice Slurry** 40 % **Storage**	**Ice Slurry** 30 % **Storage**	**Ice Slurry** 20 % **Storage**	**Full Mechanical Chilling**
	USD	USD	USD	USD	USD	USD	USD
Electrical cost	409 183	402 302	415 845	424 455	414 462	421 481	437 887
Water & chemical cost	87 243	87 243	87 243	87 243	87 243	87 243	87 243
Plant maintenance & repair Cost	212 000	209 000	207 00	202 000	202 000	204 000	230 000
Piping system maintenance cost	14 241	14 241	14 241	14 241	14 241	14 241	14 241
ETS systems maintenance cost	13 500	13 500	13 500	13 500	13 500	13 500	13 500
Cost of operators and admin.	108 000	108 000	108 000	108 000	108 000	108 000	108 000
Total plant operating cost	844 167	834 286	845 829	847 439	839 446	848 465	891 509

Table 26
Operating cost of water-based cases, SE

SE / Barcelona	Chilled Water 40 % Storage	Chilled Water 30 % Storage	Chilled Water 20 % Storage	Full Mech. 7/14°C Chilling	Solution Tank 40 % Storage	Solution Tank 30 % Storage	Solution Tank 20 % Storage	Full Mech. 1/14°C Chilling
	USD	USD	USD		USD	USD	USD	USD
Electrical Cost	863 350	878 685	917 427	1017452	949 027	971 323	1 026 406	1 151 676
Water & Chemical cost	219 852	219 852	219 852	219 852	226 470	226 470	226 470	226 470
Plant Maintenance &	200 000	190 000	200 000	230 000	220 000	210 000	220 000	260 000
Piping system Maintenance	13 589	13 589	13 589	13 589	11 145	11 145	11 145	11 145
ETS systems Maintenance	13 500	13 500	13 500	13 500	13 500	13 500	13 500	13 500
Cost for operators and	108 000	108 000	108 000	108 000	108 000	108 000	108 000	108 000
Total Plant Operating	**1 418 291**	**1 423 626**	**1 472 368**	**1 602 393**	**1 528 142**	**1 540 438**	**1 605 521**	**1 770 791**

Table 27
Operating cost of ice-based cases, SE

SE / Barcelona	Ice on coil 40 % Storage	Ice on coil 30 % Storage	Ice on coil 20 % Storage	Ice Slurry 40 % Storage	Ice Slurry 30 % Storage	Ice Slurry 20 % Storage	Full Mechanical Chilling
	USD	USD	USD	USD	USD	USD	USD
Electrical cost	874 538	905 815	934 595	917 513	926 034	973 617	1 056 878
Water & chemical cost	219 852	219 852	219 852	219 852	219 852	219 852	219 852
Plant maintenance & repair cost	212 000	209 000	207 00	200 000	202 000	204 000	230 000
Piping system maintenance cost	12 369	12 369	12 369	12 369	12 369	12 369	12 905
ETS systems maintenance cost	13 500	13 500	13 500	13 500	13 500	13 500	13 500
Cost of operators and admin.	108 000	108 000	108 000	108 000	108 000	108 000	108 000
Total plant operating Cost	**1 440 259**	**1 468 536**	**1 495 316**	**1 471 234**	**1 481 755**	**1 531 338**	**1 641 135**

Economic evaluation

Background to Economic Evaluation

Discounted pro-forma cash flows are used as the bases for economic evaluation of the two case studies. They include:

1) Operating cash flows
2) Investment cash flows
3) Financing cash flows

In order to make various technical solutions comparable, financing cash flows are simplified and assumed to be the same for all the alternatives evaluated, i.e. 25% of the required capital investments are financed by equity whilst the The remaining 75% are financed by ordinary senior debt. The price of debt is assumed to be 5% p.a. with a maturity of 25 years, which is also the total lifetime of the pro-forma project.

All the investment cash flows are assumed to take place at the beginning of the project, assuming a 100% system build-up before the operations start. The major capital investment components consist of production plant equipment, a cooling storage facility, a distribution system and energy transfer stations.

Operating cash flows include revenues from district-cooling sales, as well as all operating costs of the system, i.e. electricity costs, water and chemical costs, plant maintenance and repairs, piping system maintenance, energy transfer station maintenance, as well as costs of operations and administration.

As the operations are assumed to be financed by a mix of equity and debt, there should be two different returns distinguished, i.e. returns to equity holders (after debt amortisation and debt service), and returns to investment (the combined net cash flows to equity holders and debt holders). The providers of debt are normally secured in some way against default (by assets pledged, for example) and they will be paid before the investors of equity. The equity holders are, on the other hand, the last claimants on the operations, and it is therefore the cash flow return to equity (ROE) that will determine the ultimate profitability of a cooling installation.

Cooling Tariffs and Revenues

There are several criteria to be considered when pricing district cooling. The tariff should cover capital costs and operating costs of energy production and distribution. There should be necessary funds provided for replacement and re-investments in facilities. Cost of district cooling to customers should be competitive with other cooling alternatives, in order to sign up and retain customers. Tariffs should be clear and easy to understand.

Table 28
The general criteria for developing a tariff structure

Criteria for public utility tariffs	Best practice
Cost coverage	Tariffs should cover the variable and fixed operating and capital costs of production and distribution.
Capital expenditure coverage	Tariffs should enable the company to finance the necessary replacement of plants, networks and equipment, as well as future extension.
Competitiveness	District-cooling tariffs should be competitive with alternative cooling methods.
Stability	Permanent substantial changes in costs (e.g. power prices) should be transferred to tariffs. Temporary changes in costs should not affect tariffs.
Transparency	Tariffs should be transparent and easy to understand.

Required Rate of Return on Equity

In order to compare cooling system alternatives with various storage technologies and sizes, there has been a 12% rate of return on equity (ROE) requirement imposed over the project's lifetime of 25 years. A cooling tariff (in US cents per kWh) that allows meeting this ROE requirement has been calculated for each case, thus providing basis for comparing the various cooling storage sizes and technologies. The logic of comparison is thus simple: the storage technology that allows setting the lowest cooling tariff while providing the same ROE (12%) is financially the most preferential.

All financial analysis is done in today's prices, i.e. no price escalation due to inflation is included. Therefore, the required ROE of 12% shall be seen as a real-terms rate.

The exclusion of inflation and tax considerations for improved comparison between different storage technologies, storage sizes and geographical locations also raises a methodological issue of the calculated cooling tariffs. It shall be noted that those are not full nominal tariffs that one would observe in a real life situation. Rather, those tariffs provide a transparent indication and relative comparison between cooling methods and their regional application.

Electricity Costs

The assumed electricity costs of the Northern Europe (Stockholm) and the Southern Europe (Barcelona) base cases, including VAT, are:

	NE/Stockholm	SE/Barcelona
Base case	Low load tariff of USD 0.050/kWh High load tariff of USD 0.080/kWh	Low load tariff of USD 0.050/kWh High load tariff of USD 0.110/kWh

For sensitivity analysis two different variations of electricity tariffs are tested:

	NE/Stockholm	SE/Barcelona
Sensitivity 1	Flat electricity tariff of USD 0.050/kWh at all times	Flat electricity tariff of 0.050 USD/kWh at all times
Sensitivity 2	Low load tariff of USD 0.050/kWh High load tariff of USD 0.110/kWh	Low load tariff of USD 0.050/kWh High load tariff of USD 0.170/kWh

Taxes

The corporate profit taxes are explicitly excluded from the cash flow calculations. The electricity tariffs used are inclusive of value added tax (VAT).

Economic Results of Case Studies

The economic results of the case studies are presented in form of graphs where the cooling tariff for each technology scenario is plotted against the size of the cooling storage. Thus there are four observations for each type of cooling storage in each location, i.e. 0% storage (full-sized electrical chillers), 20% storage, 30% storage and 40% storage.

In order to see an approximate minimum tariff point, i.e. the cooling storage size with which the lowest tariff can be achieved, a polynomial trend line could be fitted to the 4-point scatter plot. However, in most cases it is better to use fraction line to show the level of cooling price in each case.

Chilled Water (7/14°C) and Solution Tank Storage (1/14°C)

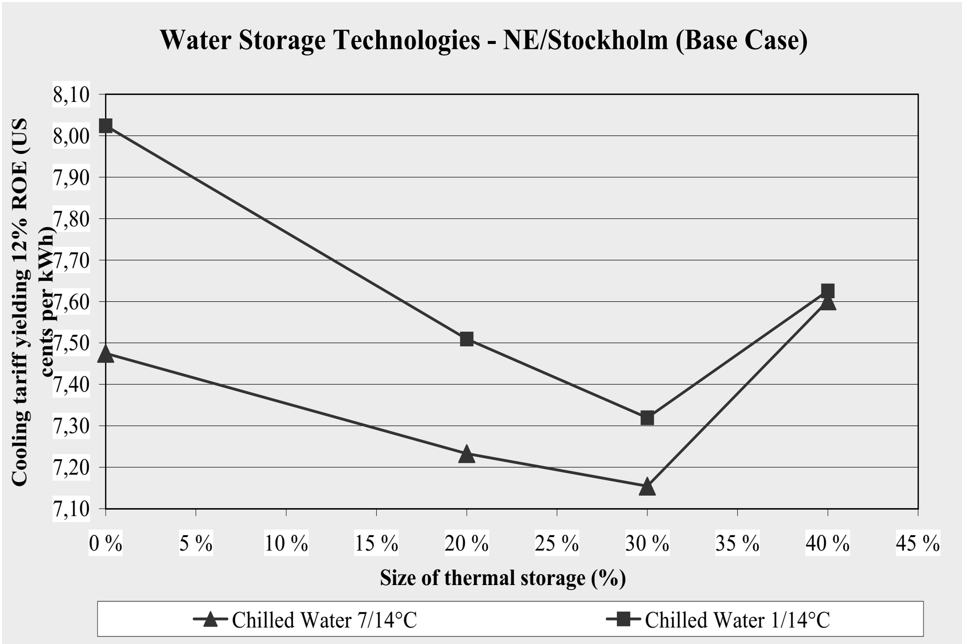
Location 1: NE/Stockholm

Base case

In the NE/Stockholm case the cooling price for a system with no storage and with supply/return temperatures of +7/14°C (base case two-time tariff 8 / 5 ¢/kWhe) is 7.48 ¢/kWh. About 60% of this price is due to investment cost repayment, cash flow to equity and interest charges. The remaining 40% of the cooling price consists of operating costs, i.e. variable operating costs like electricity consumption of the chillers, pumps and tower fans, and fixed operating costs like maintenance and repair costs.

The cooling price drops when part of the chiller capacity is replaced with chilled water storage capacity. With 20% storage the cooling price falls by 3% to 7.23 ¢/kWh. With 30% storage the cooling price is further diminished to 7.16 ¢/kWh. With 40% storage the cooling price (7.60 ¢/kWh) is 2% higher than in the base case. All the results of the case study with chilled water storage are presented in Figure 64 and Table 29. The system with 30% storage results in the lowest cooling price when compared with other cases. The reduction in the cooling price is 4% compared with the base case. About two thirds of the total savings consist of savings in operating costs, one third from savings in investment costs.
When replacing part of the chiller capacity with chilled water storage, the total investment cost decreases slightly up to 30% storage capacity. With 40% storage the investment costs increase by 9% compared with the base case because of “excess” capacity. The system has 24 MW of chiller capacity and 12 MW of storage capacity.
Owing to a two-time tariff, savings are achieved in operating costs. With 20% (6 MW, 46 MWh) chilled water storage, cooling production on a high load tariff reduces 48% compared with the system with no storage. This leads to annual savings of USD 25000, which represents one third of total savings. Also, the maintenance and repair costs decrease due to a smaller chiller plant. With 30% (9 MW, 78 MWh) chilled water storage, cooling production on a high load tariff reduces 68% compared with the base case with no storage. With 40% storage (12 MW, 100 MWh) the reduction in cooling production on a high load tariff is 81% compared with the base case.

Figure 64
Cooling price with chilled water storage (7/14°C) and solution storage (1/14°C.Case NE/Stockholm. Base case:Electricity tariff 8 / 5 ¢/kWhe



In the NE/Stockholm case the cooling price of a system with no storage and with supply/return temperatures of +1/14°C (base case) is 8.02 ¢/kWh. About 60% of this price is due to investment cost repayment, cash flow to equity and interest charges. The remaining 40% of the cooling price consists of variable and fixed operating costs.

The cooling price falls when part of the chiller capacity is replaced with sodium nitrite solution storage capacity. With 20% storage the cooling price reduces by 6% to 7.51 ¢/kWh. With 30% storage the cooling price is further diminished to 7.32 ¢/kWh. With 40% storage the cooling price (7.63 ¢/kWh) is 5% lower than in the base case. All the results of the case study with sodium nitrite solution storage are presented in Figure 64 and Table 29. The system with 30% storage results in the lowest cooling price when compared with other cases. The reduction in the cooling price is 9% compared with the base case. About 45% of the total savings consist of savings in operating costs, 55% of savings in investment costs.

Owing to a two-time tariff, savings are achieved in operating costs. With 20% (6 MW, 46 MWh) chilled water storage, cooling production on a high load tariff decreases 48% compared with the system with no storage. This leads to annual savings of USD 42000, which represents 44% of total savings. Also, the maintenance and repair costs decrease due to a smaller chiller plant.

When the sodium nitrite solution storage system is compared with the chilled water system, it is clear that choosing the lower supply temperature raises the total costs. A base case with a supply temperature of 1°C has 7% higher total costs than the a base case with a supply temperature of 7°C. But if there is lack of space and chilled water storage cannot be applied, low temperature storage can be the solution. The cooling price of the low temperature system with 30% storage is 7.32 ¢/kWh, which is 0.16 ¢/kWh (2%) lower than the cooling price with 7°C supply temperature and no storage. As a conclusion of the above: if low supply temperature is required, the costs are reduced if a storage system is applied.

Table 29
Comparison of different size of water storage and solution storage

Cooling storage technology	Size of thermal storage	Cooling tariff yielding 12% ROE (¢/kWh)	Comparison within a given storage technology	Comparison across water storage technologies
	0%	7,47	+/-0%	
Chilled water 7/14°C	20%	7,23	-3,24 %	+/-0%
Chilled water 7/14°C	30%	7,15	-4,28 %	+/-0%
Chilled water 7/14°C	40%	7,60	1,70 %	+/-0%
	0%	8,02	+/-0%	
Solution tank 1/14°C	20%	7,51	-6,41 %	7,35 %
Solution tank 1/14°C	30%	7,32	-8,78 %	0,46 %
Solution tank 1/14°C	40%	7,63	-4,96 %	-2,08 %

Sensitivity study 1: Flat electricity tariff

When the electricity tariff is flat, there is no significance of when the cooling production takes place. Therefore, the variable production costs are nearly unchanged between cases with and without storage. Only the pumping costs slightly increase with increasing storage capacity due to increasing pumping demand in discharging. The electricity price has no effect on the savings in the investment costs. As much as 45% of the total savings consist of savings in variable and fixed operating costs in the case with 30% storage.

The cooling price in the system with no storage is 7.52 ¢/kWh. The system with 30% storage has the lowest cooling price, 7.05 ¢/kWh. This is 2.8% lower than in the base case with a flat tariff.

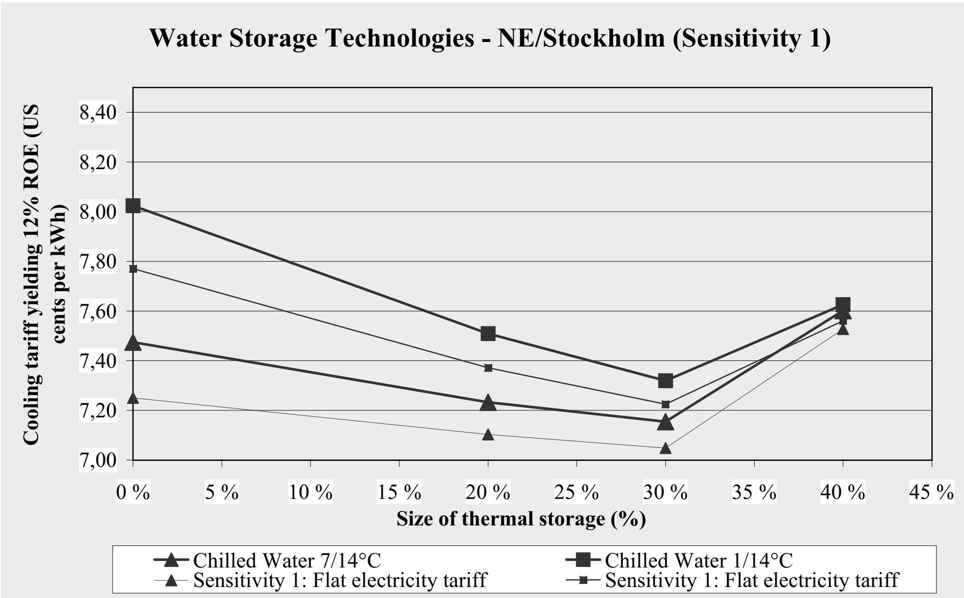


Figure 65
Cooling price with chilled water storage (7/14°C) and solution storage (1/14°C). Case NE/Stockholm. Sensitivity 1:Flat electricity tariff, 5 ¢/kWhe

The solution storage is equal to the chilled water storage; there is no significance when the cooling production takes place when the electricity tariff is flat. Therefore, the variable production costs are nearly unchanged in the cases with and without storage. Some 30% of the total savings consist of savings in variable and fixed operating costs in the case with 30% storage.
The cooling price in the system with no storage is 7.77 ¢/kWh. The system with 30% storage has the lowest cooling price, 7.23 ¢/kWh. This is 7% lower than in the base case with a flat tariff.

Sensitivity study 2: Doubled difference between high and low electricity tariff

When the difference between the high and low tariffs is doubled compared with the base case, savings achieved by transferring the production from a high tariff period to a low tariff period are more significant. As much as 78% of the total savings consist of savings in variable and fixed operating costs in the case with 30% storage.

The cooling price in the base case with a doubled tariff is 7.70 ¢/kWh. The system with 30% storage has the lowest cooling price, 7.20 ¢/kWh. This is 6.5% lower than in the base case with a doubled tariff.

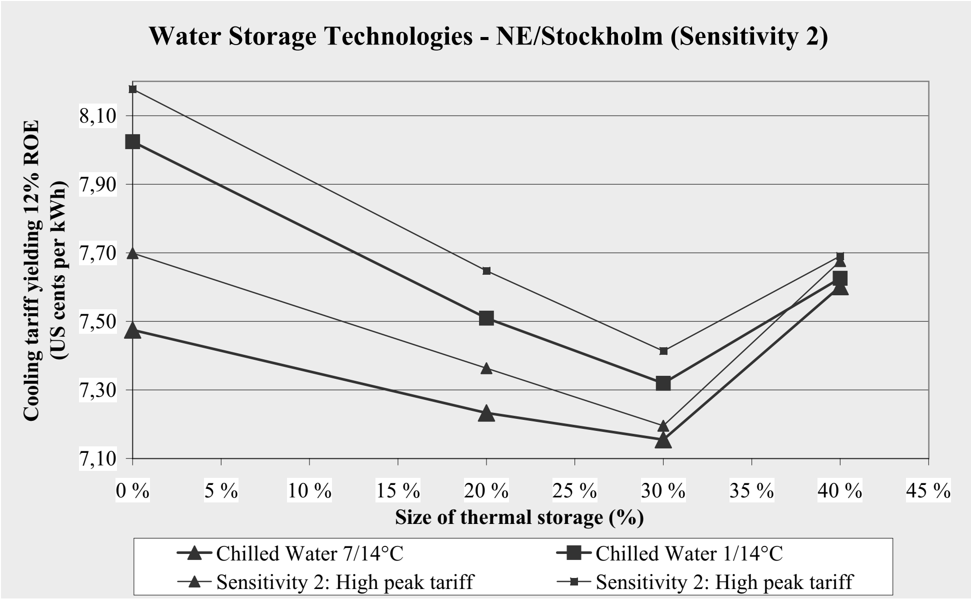


Figure 66
Cooling price with chilled water storage (7/14°C) and solution storage (1/14°C). Case NE/Stockholm. Sensitivity 2:Doubled difference in two-time electricity tariff, 11 / 5 ¢/kWhe

When the difference between the high and low tariffs is doubled compared with the base case, the cooling price of the low temperature supply system in the base case is 8.18 ¢/kWh. Some 49% of the total savings consist of savings in variable and fixed operating costs in the case with 30% storage. The system with 30% storage has the lowest cooling price, 7.41 ¢/kWh. This is 9% lower than in the base case with a doubled tariff.

Location 2: SE/Barcelona

Base Case

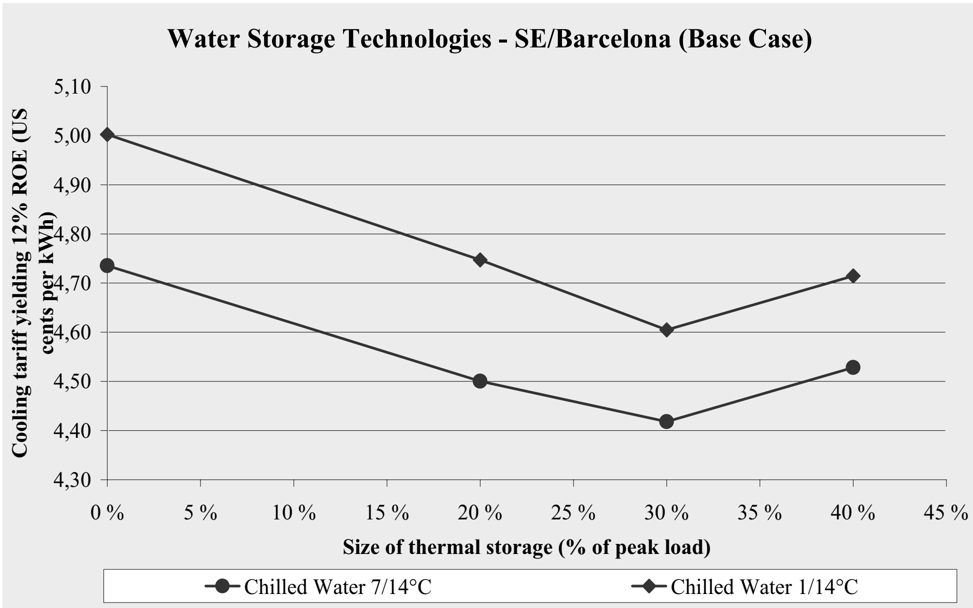
In the SE/Barcelona case the cooling price of a system with no storage and with supply/return temperatures of +7/14°C (base case) is 4.74 ¢/kWh. About 46% of this price is due to investment cost repayment, cash flow to equity and interest charges. The remaining 54% of the cooling price consists of operating costs, i.e. variable operating costs like electricity consumption of the chillers, pumps and tower fans, and fixed operating costs like maintenance and repair costs.

The cooling price drops when part of the chiller capacity is replaced with chilled water storage capacity. In the base case (with no storage and two-time tariff 11 /5 ¢/kWhe) the total cooling price is 4.74 ¢/kWh. With 20% storage the cooling price falls by 5% to 4.50 ¢/kWh. With 30% storage the cooling price is further diminished to 4.42 ¢/kWh. With 40% storage the cooling price (4.53 ¢/kWh) is 4% higher than in the base case. All the results of the case study with chilled water storage are presented in Figure 67. The system with 30% storage results in the lowest cooling price when compared with other cases. The reduction in the cooling price is 7% compared with the base case. About two thirds of the total savings consist of savings in operating costs, one third from savings in investment costs.

When replacing part of the chiller capacity with chilled water storage, the total investment cost decreases slightly up to 30% storage capacity. With 40% storage, the investment costs increase by 3% compared with the base case because of “excess” capacity.

Owing to a two-time tariff, savings are achieved in operating costs. With 20% (6 MW, 46 MWh) chilled water storage cooling production on a high load tariff decreases by 43% compared with the system with no storage. This leads to annual savings of USD 90000, which represents 64% of total savings. Also, the maintenance and repair costs fall due to a smaller chiller plant. With 30% (9 MW, 78 MWh) chilled water storage, cooling production on a high load tariff decreases by 60% compared with the base case with no storage. With 40% storage (12 MW, 100 MWh) the reduction in cooling production on a high load tariff is 73% compared with the base case.

Figure 67
Cooling price with chilled water storage (7/14°C) and solution storage (1/14°C). Case SE/Barcelona. Base case: Electricity tariff 11 / 5 ¢/kWhe



In the case with low (+1°C) supply temperature the cooling price in the system with no storage and with supply/return temperatures of +1/14°C (base case) is 5.00 ¢/kWh in Barcelona. About 44% of this price is due to investment cost repayment, cash flow to equity and interest charges. The remaining 54% of the cooling price consists of variable and fixed operating costs.

The cooling price decreases when part of the chiller capacity is replaced with sodium nitrite solution storage capacity. With 20% storage the cooling price falls by 8% to 4.75 ¢/kWh. With 30% storage the cooling price is further diminished to 4.60 ¢/kWh. With 40% storage the cooling price (4.72 ¢/kWh) is 6% lower than in the base case. The system with 30% storage results in the lowest cooling price when compared with other cases. The reduction in the cooling price is 9% compared with the base case. About 85% of the total savings consist of savings in operating costs, 15% of savings in investment costs.

Owing to a two-time tariff, savings are achieved in operating costs. With 20% (6 MW, 46 MWh) chilled water storage, cooling production on a high load tariff decreases by 44% compared with the system with no storage. This leads to annual savings of USD 42000, which represents 70% of total savings. Also, the maintenance and repair costs decrease due to a smaller chiller plant.

When a sodium nitrite/nitrate solution storage system is compared with a chilled water system, it is clear that choosing the lower supply temperature increases the total costs. A base case with a supply temperature of 1°C has 6% higher total costs than the base case with a supply temperature of 7°C. In case of lack of space chilled water storage cannot be applied, and low temperature storage can then be the solution. The cooling price of a low temperature system with 30% storage is 4.60 ¢/kWh, which is 0.13 ¢/kWh (3%) lower than the cooling price with 7°C supply temperature and no storage.

Table 30
Comparison of different size of water storage and solution storage

Cooling storage technology	Size of thermal storage	Cooling tariff yielding 12% ROE (¢/kWh)	Comparison within a given storage technology	Comparison across water storage technologies
	0 %	**4,74**	+/-0%	
Chilled Water 7/14°C	20 %	**4,50**	-4,96 %	+/-0%
Chilled Water 7/14°C	30 %	**4,42**	-6,70 %	+/-0%
Chilled Water 7/14°C	40 %	**4,53**	-4,37 %	+/-0%
	0 %	5,00	+/-0%	
Chilled Water 1/14°C	20 %	4,75	-5,10 %	5,64 %
Chilled Water 1/14°C	30 %	4,60	-7,95 %	0,25 %
Chilled Water 1/14°C	40 %	4,71	-5,75 %	-2,76 %

Sensitivity study 1: Flat electricity tariff

When the electricity tariff is flat, there is no significance of when the cooling production takes place. Therefore, the variable production costs are nearly unchanged between the cases with and without storage. Only the pumping costs slightly increase with increasing storage capacity due to increasing pumping demand in discharging. The electricity price has no effect on the savings in the investment costs. Some 38% of the total savings consist of savings in variable and fixed operating costs in the case with 30% storage.

The cooling price in the system with no storage is 4.29 ¢/kWh. The system with 30% storage has the lowest cooling price, 4.20 ¢/kWh. This is 2.0% lower than the base case with a flat tariff.

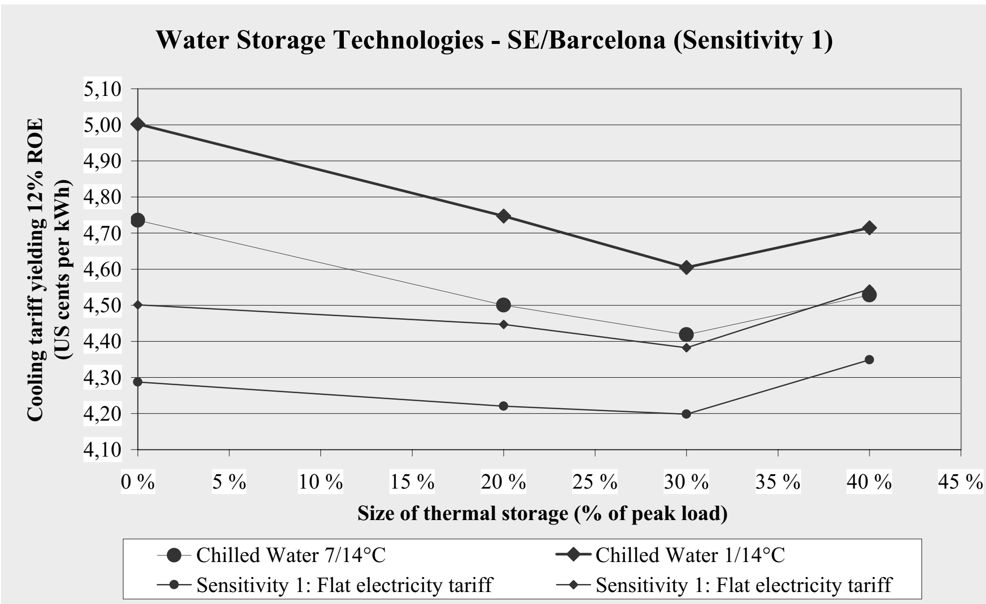


Figure 68
Cooling price with chilled water storage (7/14°C) and solution storage (1/14°C). Case SE/Barcelona. Sensitivity 1: Flat electricity tariff, 5 ¢/kWhe

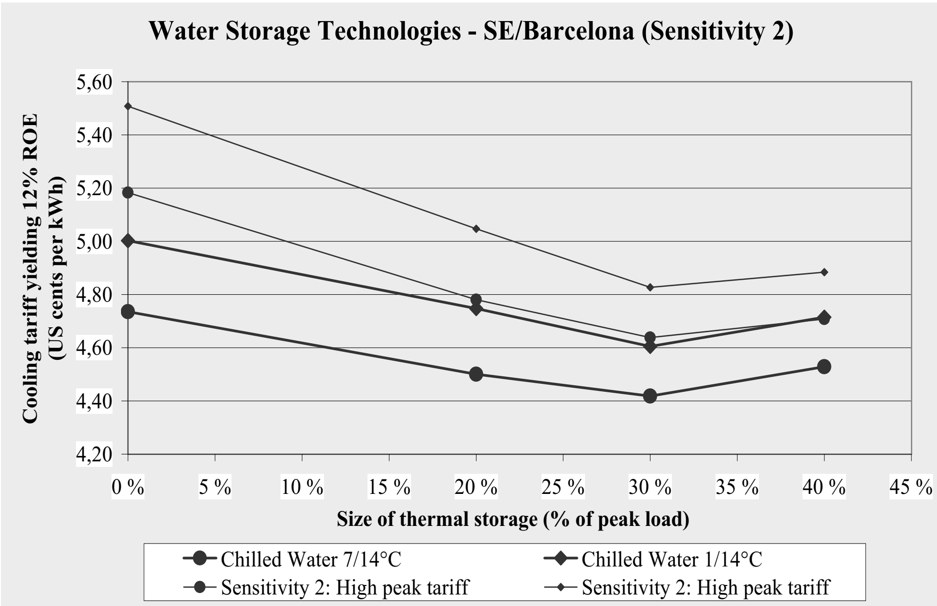
The solution storage is equal to the chilled water storage; there is no significance when the cooling production takes place when the electricity tariff is flat. Therefore, the variable production costs are nearly unchanged between cases with and without storage. As much as 51% of the total savings consist of savings in variable and fixed operating costs in the case with 30% storage.

The cooling price in the system with no storage is 4.50 ¢/kWh. The system with 30% storage has the lowest cooling price, 4.38 ¢/kWh. This is 2.6% lower than in the base case with a flat tariff.

Sensitivity study 2: Doubled difference between high and low electricity tariff

When the difference between the high and low tariffs is doubled compared with the base case, savings achieved by transferring the production from a high tariff period to a low tariff period are more significant. As much as 90% of the total savings consist of savings in variable and fixed operating costs in case with 30% storage.

Figure 69
Cooling price with chilled water storage (7/14°C) and solution storage (1/14°C). Case SE/Barcelona. Sensitivity 2: Doubled difference in two-time electricity tariff 17 / 5 ¢/kWhe



The cooling price in the base case of a low supply temperature system with a doubled tariff is 5.18 ¢/kWh. The system with 30% storage has the lowest cooling price, 4.64 ¢/kWh. This is 10.5% lower than in the base case with a doubled tariff.

The feasibility of the solution storage is the same as that of the chilled water storage. When the difference between high and low tariffs is doubled compared with the base case, the cooling price in the base case is 5.51 ¢/kWh. As much as 92% of the total savings consist of savings in variable and fixed operating costs in the case with 30% storage. The system with 30% storage has the lowest cooling price, 4.83 ¢/kWh. This is 12% lower than in the base case with a doubled tariff.

Absorption cooling

In addition to previous calculations, technically potential absorption chiller technique was also briefly considered. Two cases are calculated:

- Case 1: 100% of compressor chiller capacity is replaced with absorption chiller capacity
- Case 2: 30% of compressor chiller capacity is replaced with absorption capacity. Absorption capacity is operated as base capacity.

The approach is that the price of cooling energy produced with absorption technique must be at the same or lower level than the cooling price with conventional compressor chillers to favor absorption chillers. The limit for the heat price was calculated taking the following matters into consideration:

- Absorption chillers have a lower COP than compressor chillers. This leads to higher water consumption in heat rejection
- The difference in investment costs between absorption and compressor chillers.

In Stockholm the electricity tariff is USD 0.05/0.08/kWh and the average price of electricity is USD 0.059/kWh in the cooling production. In Barcelona the electricity tariff is USD 0.05/0.11/kWh and the average price of electricity is USD 0.069/kWh in the cooling production.

Table 31 presents the limit heat prices that would make absorption cooling favourable. The first figure excludes the difference in investments, i.e. it shows the limit heat price with which the variable production costs are equal in both systems. This is only a theoretical limit, because the investment costs also need to be taken into account. The second limit heat price takes the additional investment into account and shows the heat price that covers the extra water costs and investment costs. The same interest rate, equity and calculation period are used in this calculation as in previous calculations.

Table 31
Limit heat prices for absorption chillers.

Case description	Absorption capacity of total	Average electricity price [USD/MWh]	Cooling energy produced with absorption [MWh]	Heat price 1 (* [USD/MWh]	Heat price 2 (** [USD/MWh]
Stockholm	100%	59.5	30000	5.4	-1.3
Stockholm	30%	58.2	23560	5.3	2.2
Barcelona	100%	69.0	63000	6.2	5.8
Barcelona	30%	66.0	44940	3.1	4.2

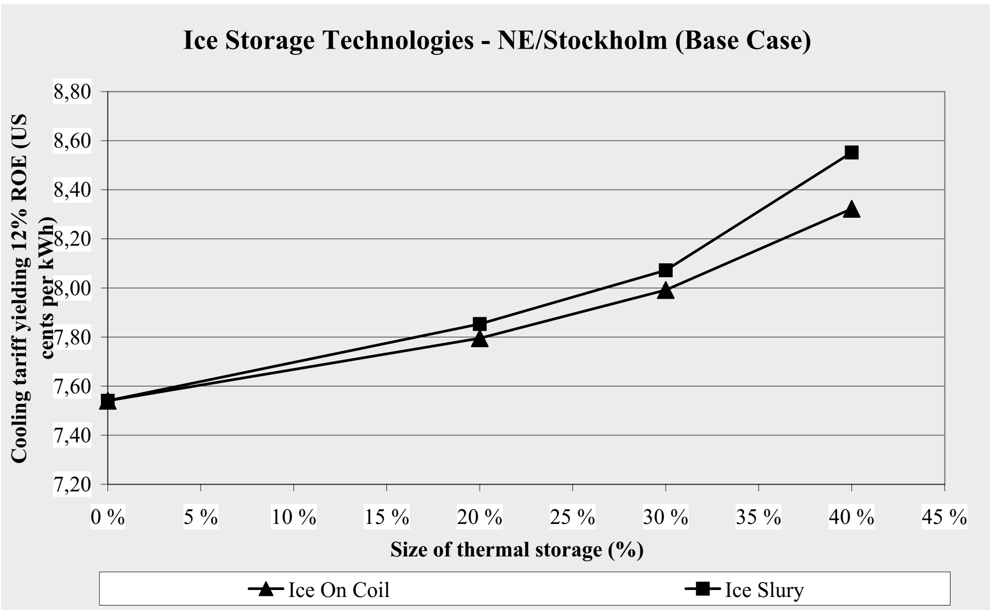
** Covers additional variable operating cost*
*** Covers additional variable operating cost and additional investment*

Ice-On-Coil and Ice Slurry Storage

Location 1: NE/Stockholm

In the case of NE/Stockholm the assumed electricity tariffs (under the base case scenario) do not show that using any type of ice storage can yield a lower cooling tariff for customers. The figure below indicates that the lowest cooling tariff can be achieved when using electrical centrifugal chillers for covering full cooling demand in the system.

Figure 70
Cooling price with ice-on-coil storage and ice slurry storage. Case NE/Stockholm. Base case:Electricity tariff 8 / 5 ¢/kWhe



If 20% storage is introduced in the cooling system, the tariff increases by approximately 3% in the case of ice-on-coil and by approximately 4% in the case of ice slurry technology. The table below shows how further increasing the storage size affects the cooling tariff to be set in order to reach the same return on equity.

Table 32
Comparison of different sizes of ice-on-coil storage and ice slurry storage

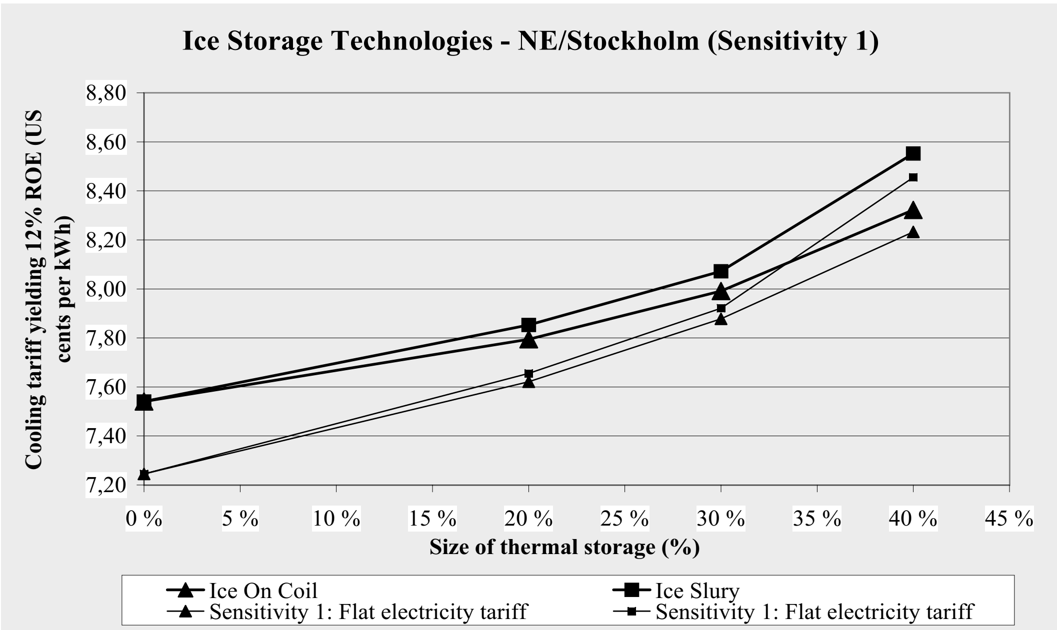
Cooling storage technology	Size of thermal storage	Cooling tariff yielding 12% ROE (¢/kWh)	Comparison within a given storage technology	Comparison between storage technologies
	0%	7.54	+/- 0%	
Ice-on-coil	20%	7.79	+3.3%	+/- 0%
Ice-on-coil	30%	7.99	+6.0%	+/- 0%
Ice-on-coil	40%	8.32	+10.3%	+/- 0%
	0%	7.54	+/- 0%	
Ice slurry	20%	7.85	+4.1%	+0.8%
Ice slurry	30%	8.07	+7.0%	+1.0%
Ice slurry	40%	8.55	+13.4%	+2.8%

The ice-on-coil technology is a very slightly cheaper solution than the ice slurry storage. The comparison between these two ice storage technologies shows that this difference yields approximately 1%-3% increase in the cooling tariff.

Sensitivity study 1 - Flat electricity tariff

Using a uniform electricity purchase price that equals the low load tariff under the base case assumptions, the results of the economic analysis confirm the expectation, i.e. there is no financial rationale for using storage.

Figure 71
. Cooling price with ice-on-coil storage and ice slurry storage. Case NE/Stockholm. Sensitivity 1: Sensitivity 1: Flat electricity tariff, 5 ¢/kWhe

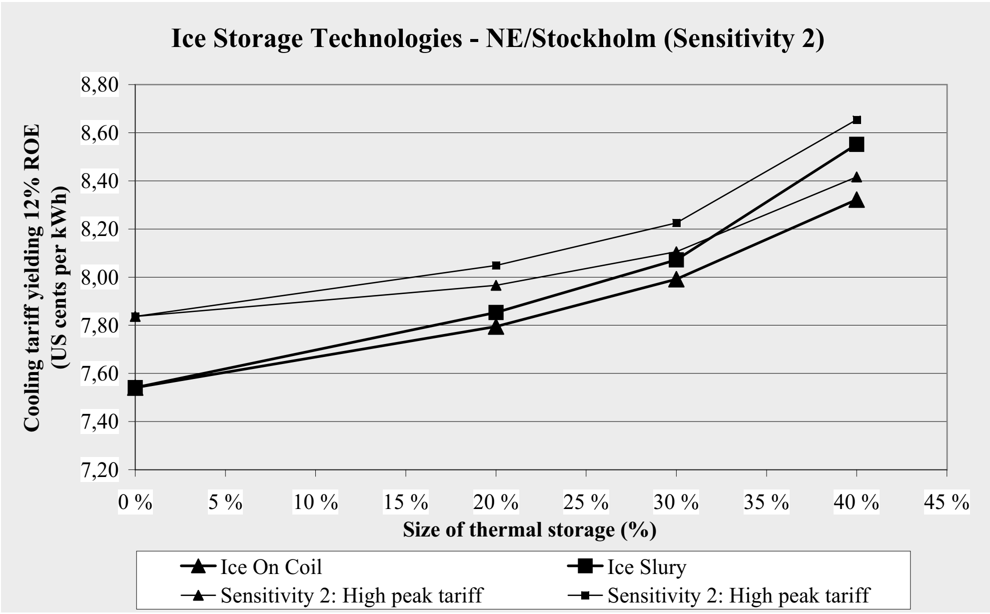


Furthermore, by eliminating the peak electricity tariff of 8 ¢/kWhe and purchasing all-day-round electricity at 5 ¢/kWhe, the price of cooling produced by electric centrifugal chillers drops from 7.54 ¢/kWh to 7.25 ¢/kWh, i.e. by approximately 3.8%.

Sensitivity study 2 - Doubled difference between high and low electricity tariff

If in the case of NE/Stockholm the peak electricity tariff is increased from the base case assumption of 8 ¢/kWhe to 11 ¢/kWhe (+37.5%) and everything else held constant, the cooling price without any storage goes up from 7.54 ¢/kWh to 7.84 ¢/kWh (approx. +4%) and the trend line between 0% and 20% storage shows that there might be a rationale for approx 5% storage capacity, as shown in the figure below.

Figure 72
Cooling price with ice-on-coil storage and ice slurry storage. Case NE/Stockholm. Sensitivity 2: Doubled difference in two-time electricity tariff, 11 / 5 ¢/kWhe



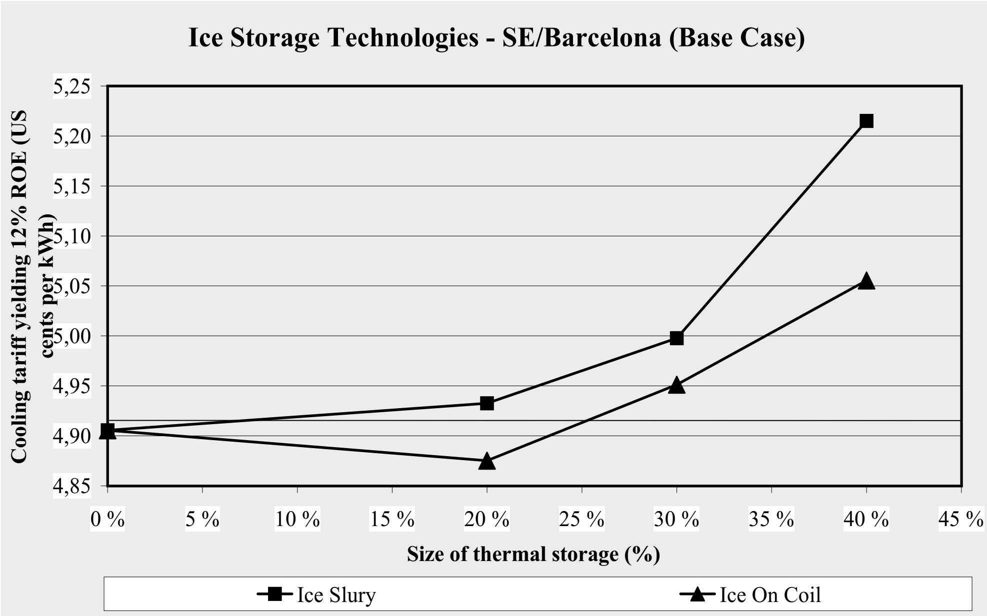
This conclusion, however, is to be validated by a separate investment cost re-calculation for such small cooling storage as lies relatively far away from the existing observation point of 20% storage.

Location 2: SE/Barcelona

Base case

When analysing the results for SE/Barcelona (base case assumptions of electricity tariffs), it can be seen that the lowest cooling price can be achieved with ice storage sized between 0 and 20% of the cooling demand. In particular, the trend line for ice-on-coil storage reaches its minimum between 10 and 15%, whilst the trend line for ice slurry storage at approximately 10%, as shows in the figure below.

Figure 73
Cooling price with ice-on-coil storage and ice slurry storage. Case SE/Barcelona. Base Case: Electricity tariff, 11 / 5 ¢/kWhe



As to the comparison of the resulting cooling price for ice-on-coil vs. ice slurry technologies, the ice slurry technology gives on average 1% to 3% higher tariff than the ice-on-coil. The table below provides comparison within a given storage type, as well as between the ice-on-coil and ice slurry technologies.

Table 33
Comparison of different size of ice-on-coil storage and ice slurry storage

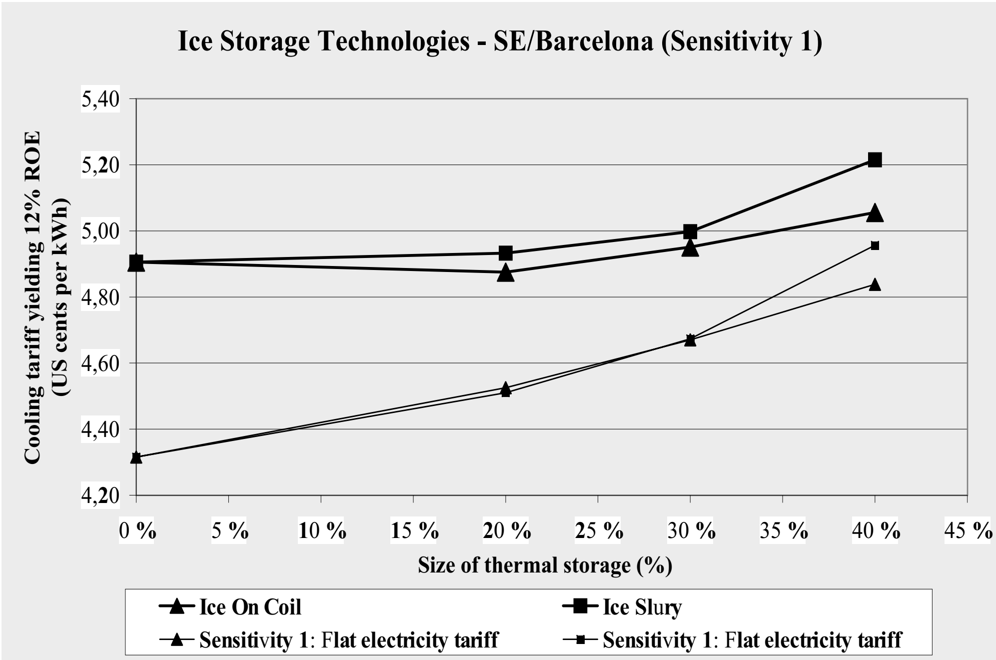
Cooling storage technology	Size of thermal storage	Cooling tariff yielding 12% ROE (¢/kWh)	Comparison within a given storage technology	Comparison between storage technologies
	0%	4.91	+/-0%	
Ice On Coil	20%	4.88	-0.6%	+/-0%
Ice On Coil	30%	4.95	+0.8%	+/-0%
Ice On Coil	40%	5.06	+3.1%	+/-0%
	0%	4.91	+/-0%	
Ice Slurry	20%	4.93	+0.4%	+1.0%
Ice Slurry	30%	5.00	+1.8%	+1.0%
Ice Slurry	40%	5.22	+6.3%	+3.0%

Sensitivity study 1 - Flat electricity tariff

A flat electricity tariff of 5 ¢/kWhe does not financially justify any cooling storage under SE/Barcelona assumptions (see the figure below).

The cooling price of all-electric chiller generation under this sensitivity test falls from 4.91 ¢/kWh to 4.32 ¢/kWh, i.e. by 12%.

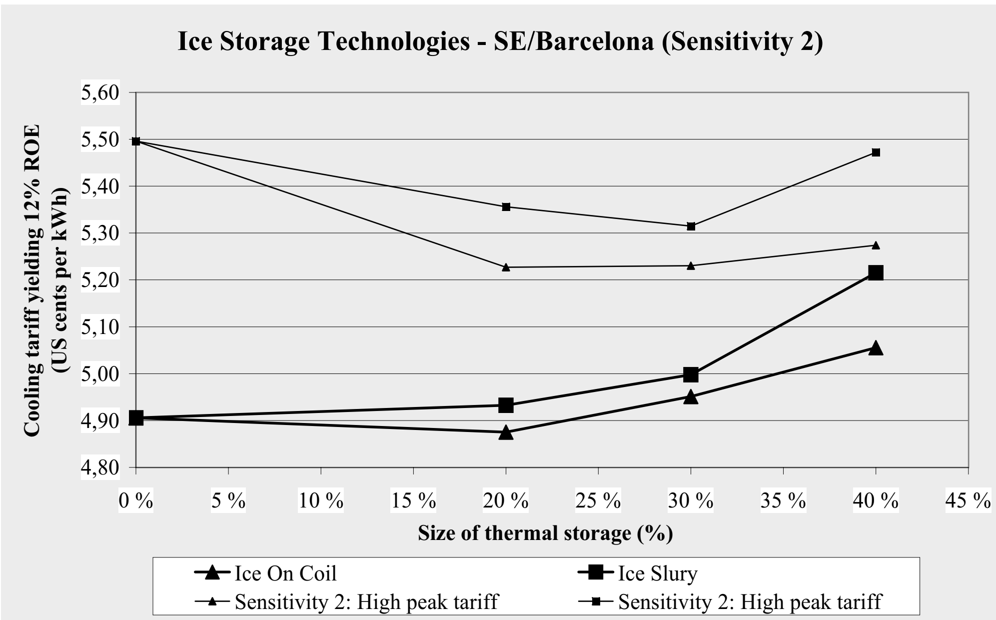
Figure 74
Cooling price with ice-on-coil storage and ice slurry storage. Case SE/Barcelona. Sensitivity1: Flat electricity tariff, 5 ¢/kWhe



Sensitivity study 2 - Doubled difference between high and low electricity tariff

The most pronounced conclusion regarding the optimal ice cooling storage size can be drawn if under the SE/Barcelona conditions the peak electricity price is increased from 11 ¢/kWhe to 17 ¢/kWhe (+54.5%), all other assumptions held constant. In this situation the optimal ice-on-coil and ice slurry storage shifts from 10%-15% to 22%-27%, as indicated by the trend lines in the figure below.

Figure 75
Cooling price with ice-on-coil storage and ice slurry storage. Case SE/Barcelona. Sensitivity 2: Doubled difference in two-time electricity tariff, 17 / 5 ¢/kWhe



The cooling price of all-electric chiller generation under this sensitivity test falls from 4.91 ¢/kWh Slightly larger ice-on-coil storage (approx. 27%) gives better feasibility than ice slurry storage (approx. 22%).

Simultaneously, the overall cooling price level increases. The increase is 12% for all-electric chiller generation (from 4.91¢/kWh to 5.50 ¢/kWh) and it becomes smaller as more and more ice storage capacity is introduced into the cooling system.

Conclusions

When comparing the locations of NE/Stockholm and SE/Barcelona, it can be said that only under the Southern Europe assumptions any significant ice-based cooling storage can be financially substantiated. In Stockholm ice-based storage is not feasible, but water-based storage with low investment and operating costs are economically well grounded.

With a uniform profitability requirement for the system owner (12% ROE) the overall cooling price levels charged in SE/Barcelona (from 4.99 ¢/kWh to 5.22 ¢/kWh) are 37% lower than those charged in NE/Stockholm (from 7.54 ¢/kWh to 8.55 ¢/kWh) despite the fact that the electricity prices are overall lower in NE/Stockholm. This can be explained by the substantial difference in the number of cooling degree days, i.e. 40 in Stockholm and 471 in Barcelona, which lead to twice as much cooling energy sold in the southern location, thus giving a substantially larger annual revenue base to contribute to covering the fixed system investment costs.

Figure 76 shows how the cooling price is built up. In Stockholm 61% of the cooling price consists of investment costs (USD 0.046/kWh). In Barcelona the investment costs account for 46% of total costs (USD 0.022 /kWh). The more cooling energy can be produced with the same cooling system, the lower the fixed investment cost per kWh decreases. (See Figure 24). In Stockholm the variable production costs are USD 0.017/kWh, in Barcelona USD 0.020/kWh due to higher electricity prices and water costs in Barcelona.

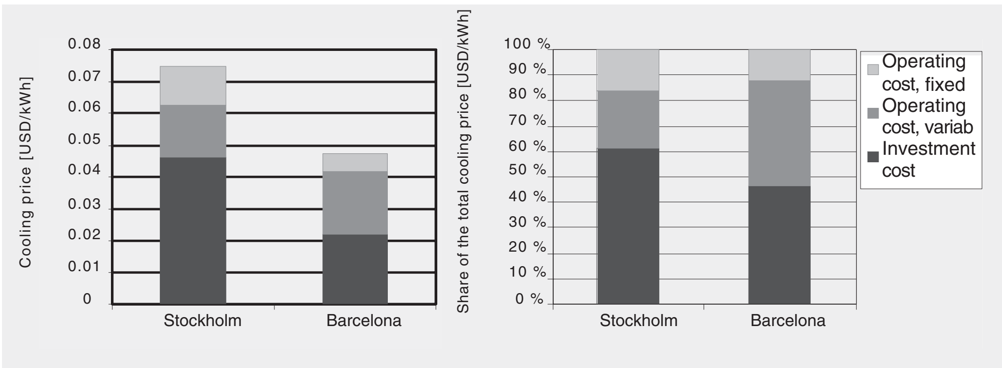


Figure 76
Price formation of the cooling price in Stockholm and Barcelona. Annual cooling load is 30,000 MWh/year in Stockholm and 63,000 MWh/year in Barcelona.

Chilled water storage leads to minimum investment cost when no special value is given for building space. Chilled water storage is feasible up to the technical capacity limit of the system while the specific investment in storage discharge capacity is lower than the investment in instantaneous chilling capacity.

Because of the assumptions of zero inflation and absence of tax considerations, however, the calculated prices for cooling should be viewed only as indicative and used for relative comparison between the two geographical locations and the various storage technologies analysed. The real prices for district cooling would be determined by the market competitive forces (options for locally generated cooling vs. district cooling), the consumers' mentalities and their willingness-to-pay for the advantages provided by district cooling vis-à-vis locally generated cooling. The lowest price that a financially profitable district-cooling operator would be able to charge is the one that would cover the system owner's full capital and operating costs, whilst the ceiling price would always be the one determined by other alternatives that the consumers are able and willing to consider. Storage technologies shall be viewed as additional space for the system operator to widen the profitable and still competitive price interval.

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Load Calculation and Simulation Software of Buildings

There are plenty of software products for calculation of heating and cooling loads of a building. The uses and applicability of the software varies a lot. Here is a list of some software available. Freeware is marked by *).

BESTEST*

Software test BESTEST (Building Energy Simulation TEST) by National Renewable Energy Laboratory (1617 Cole Boulevard, Colorado, USA (303) 384-7520) is a method for testing and diagnosing the simulation capabilities of the exterior envelope portions of building energy simulation programs. It reveals the strengths and weaknesses of a given software package in a methodical way. BESTEST evaluates design and analysis tools relative to their ability to adequately model the envelope dynamics of buildings.

TRACE Load 700

TRACE™ Load 700 software is the building and load design modules of TRACE™ 700, Trane Air Conditioning Economics - to evaluate the effect of building orientation, size, shape, and mass based on hourly weather data and the resulting heat-transfer characteristics of air and moisture.
The program uses algorithms recommended by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE). Choose from seven different ASHRAE cooling and heating methodologies, including the Exact Transfer Function. The program encourages "what if" analyses, letting you enter construction details in any order and then easily change the resulting building model as the design progresses. Multiple project views and "drag-and-drop" assignment of coil loads simplify the modelling process and help you identify optimal zoning.
Templates provide an easy way to analyse the effects of changes in building loads such as airflows, thermostat settings, occupancy, and construction. An extensive library of construction materials, schedules, shading types, load-generating sources, and weather profiles (nearly 500 locations) simplify data entry and allow greater modelling accuracy.

Input: Building design parameters, air distribution characteristics

Output: Print or export any of 40 monthly/yearly summary reports and hourly analyses, including system "checksums," psychrometric state points, peak cooling/heating loads, building envelope loads, and ASHRAE 90 analysis

Computer Platform: Personal computer with a 486/33 or higher processor, Microsoft Windows 95/98/2000/NT operating system, 32 megabytes (MB) of RAM, 25 MB of free hard disk space, Super VGA display, CD-ROM drive, and a Microsoft-compatible pointing device

Programming Language: Microsoft Visual C++ development system

Strengths: Uses techniques recommended by or consistent with ASHRAE recommendations, and models more than 25 types of air distribution systems. Flexible data entry, multiple views and "drag-and-drop" load assignments simplify the modelling process. Customizable libraries and templates streamline data entry and enable greater modelling accuracy. Experienced HVAC engineers and support specialists provide free technical support.

Weaknesses: Requires TRACE 700 to perform energy and cost analyses.
Formal training is recommended for new users.
Contact: Trane C.D.S. Support Center
The Trane Company
3600 Pammel Creek Road
La Crosse, Wisconsin 54601-7599, USA

telephone (608) 787-3926
facsimile (608) 787-3005
e-mail cdshelp@trane.com
web http://www.trane.com/commercial/software
Availability: $695 USD for a standard single license; $1,042.50 USD for a site/LAN license

System Analyzer

Software for load calculation and energy and economic comparative analysis. System Analyzer permits a quick evaluation of virtually any building, system, and equipment combination. Thus, it can be used either as a scoping tool to decide what systems may be appropriate for an initial design, or to get a general feeling of how one system/equipment combination may perform over another. If a certain combination seems especially promising, further analysis can be done by exporting inputs into TRACE 600.

Input: Building design parameters, system configurations and utility rates.

Output: Print any of the 30 design and analysis reports and graphs such as building loads, equipment energy consumption, economic analysis, yearly cash flows and monthly building load profiles for comparisons or presentations.

Computer Platform: PC-compatible 486 or higher (Pentium recommended), Windows 3.1 or higher, 12 MB RAM (16 MB recommended); 13 MB free hard disk space.

Programming Language: CA-Realizer

Strengths: System Analyzer is a powerful, interactive presentation tool and it's graphical interface allows even a beginner with minimal HVAC experience to get a complete energy and economic analysis in as little as 10 minutes. The graphs, when printed on a color printer, provide powerful visual proof for proposals to justify better HVAC systems.

Weaknesses: The program lacks some of the extensive details of load and energy components available in the TRACE suite.

Contact: See the contact information of TRACE

SOLAR-5 *

Displays 3-D plots of hourly energy performance for the whole building or for 9 schemes and any of 40 different components. SOLAR-5 also plots heat flow into/out of thermal mass, and indoor air temperature, daylighting, output of the HVAC system, cost of electricity and heating fuel, and the corresponding amount of air pollution. It uses hour-by-hour weather data. It contains an expert system to design an initial base case building for any climate and any building type, that an architect can copy and redesign. Contains a variety of decision-making aids, including combination and comparison options, color overlays, and bar charts that show for any hour exactly where the energy flows.

Input: From only four pieces of data initially required (floor area, number of stories, location, and building type), the expert system designs a basic building, filling in hundreds of items of data; user can make subsequent revisions, usually beginning with overall building dimensions, window sizes, etc.

Output: Dozens of different kinds of 3-D plots, tables, and reports. For example, displays heat gain/loss for over a dozen different building components; shows heat flow into and out of the thermal mass of the building, as well as the output of the heating and air conditioning systems; displays air temperatures (outdoors or indoors) and air change rates; predicts the cost of heating fuel and electricity; calculates the building's air pollution 'footprint' for six gasses including carbon dioxide; (the above output plot of heat gain shows why South Windows are such good winter passive solar collectors).

Computer Platform: All Windows platforms and emulators; needs 2 megabytes of RAM

Programming Language: Visual Fortran

Strengths: Intended for use at the very earliest stages of the design process (when most of the decisions that determine a building's energy performance are made); user friendly; rapid, calculating all 8760 hours of the year (SOLAR-5 has been validated against Doe-2 and Blast using the BESTEST procedure)

Weaknesses: Not intended for complex mechanical system design or equipment sizing.

Contact: Professor Murray Milne
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e-mail Milne@ucla.edu
web http://www.aud.ucla.edu/energy-design-tools
Availability: SOLAR 5 was mostly recently updated in June 2000 and is free. It can be downloaded from the above site; it is not copy protected; sharing is encouraged.

RIUSKA

Part of an integrated simulation system for the building services design and facilities management purposes to be used in everyday design process. Programs cover thermal simulation needs of the whole building life cycle from the preliminary design to renovations. The main components of the simulation system are RIUSKA - the interface for simulation database, result module and calculation engine, and SMOG - the building geometry modeller.

With RIUSKA user can add building envelope materials, internal loads and HVAC-system into the created 3D-model of the building and perform thermal calculations. RIUSKA can be used for space simulations to dimension cooling or heating equipments, or for energy calculations of the whole building. As a calculation engine RIUSKA is presently using DOE-2.1E. RIUSKA & SMOG offer a number of different output reports for different needs during the design process.

Input: In RIUSKA user needs to give input data for materials, internal loads, schedules and HVAC-system or he/she can choose from predefined library values. Inputs are entered, or chosen from library values in windows-like environment.

Output: Number of different spreadsheet-type reports can be produced, presenting inputs used and hourly/yearly values for cooling and heating energy consumptions, temperatures and cooling loads.

Computer Platform: PC-compatible, Windows 95/98/NT.
Programming Language: Visual Basic

Strengths: Simple interface structure and possibility to use default library values enables quick access to space or building simulations.

Weaknesses: Only the predefined HVAC-systems are available to be used.

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Availability: Contact Olof Granlund for availability and pricing.

SMOG

Part of an integrated simulation system for the building services design and facilities management purposes to be used in everyday design process. Programs cover thermal simulation needs of the whole building life cycle from the preliminary design to renovations. The main components of the simulation system are RIUSKA - the interface for simulation database, result module and calculation engine, and SMOG - the building geometry modeller.

With SMOG, which runs on top of AutoCad user can create a 3D-model of a building and perform steady-state heat loss calculations. In SMOG user can draw the building him/herself or import IFC 1.5.1 compatible building geometry data. RIUSKA & SMOG offer a number of different output reports for different needs during the design process.

Input: User needs to input the building geometry data into SMOG either by modelling it or by importing IFC 1.5.1 compatible building data. SMOG offers efficient tools for the modelling work.

Output: Number of different spreadsheet-type reports can be produced, presenting inputs used and hourly/yearly values for cooling and heating energy consumptions, temperatures and cooling loads.

Computer Platform: PC-compatible, Windows 95/98/NT.

Programming Language: C++

Strengths: Simple interface structure and possibility to use default library values enables quick access to space or building simulations.

Weaknesses: Only the predefined HVAC-systems are available to be used.

Contact: See the contact information of RIUSKA

CL4M Commercial Cooling and Heating Loads

Uses ASHRAE methods and algorithms to calculate cooling loads, heating loads and air requirements for each space, and coil specifications, for commercial buildings. CLTDs, SHGFs, CLFs and almost all other factors in the ASHRAE load calculations for each surface and space are calculated and displayed for the engineer's inspection. Latitude and longitude of building location may be specified to the degree, altitude to the foot, and calculations are made for any range of days of the year, and range of hours desired. Building may be rotated or reflected and construction types easily changed for studies.

Handles variations in sky clarity, ground reflectivity, building shading, humidity and altitude. Almost unlimited flexibility in wall, roof and glass descriptions, load types and profiles, A/C set points, partial or full return air plenums in each zone. Worldwide application; weather tapes not required. Psychrometric calculations are altitude corrected.

Uses algorithmic calculations rather than disk database lookup for CLTD's, SHGF's, CLF's and psychrometric calculations, for greater speed and accuracy. This is a substantial difference from most similar programs. Set up for large, multi-storey buildings, but sufficiently convenient to be used for smaller buildings and residences.

Expertise Required: Familiarity with ASHRAE commercial load calculation theory and methods; HVAC design experience is desirable. Computer experience not necessary.
Users: Over 500.

Audience: HVAC engineers and commercial HVAC contractors.

Input: Overall design characteristics of the building such as wall, roof and glass specifications and dimensions for each zone. Internal load characteristics must be known. Trials may be run to compare results.

Output: HVAC cooling, heating loads, air requirements for each zone and air handler, as well as coil specifications are output. Details of calculations for each surface in every zone may be displayed and studied; zone and air handler load details are printed out, together with building totals.

Computer Platform: PC-compatible, 286 or higher, running any flavor of Windows or DOS.
Programming Language: Fully compiled with portions in assembler language.
Strengths: Uses ASHRAE methods, in most respects the same as those of National Bureau of Standards publication No. 69. Program runs in all DOS and Windows environments. In wide commercial use for over 10 years. Uses algorithmic calculations rather than table lookup. Believed to be fastest program in commercial use on microcomputers. Free technical assistance by telephone, fax or email.
Weaknesses: This is a DOS program, which installs and runs itself through any version of Windows by clicking its Windows icon.

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MC2 Engineering Software
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facsimile (305) 860-9146
e-mail mc2engsoft@aol.com
web http://www.mc2engineeringsoftware.com
Availability: Site license for most powerful 255-zone version - $995.00, 40-zone version $450.00

BLAST

Performs hourly simulations of buildings, air handling systems, and central plant equipment in order to provide mechanical, energy and architectural engineers with accurate estimates of a building's energy needs. The zone models of BLAST (Building Loads Analysis and System Thermodynamics), which are based on the fundamental heat balance method, are the industry standard for heating and cooling load calculations. BLAST output may be utilized in conjunction with the LCCID (Life Cycle Cost in Design) program to perform an economic analysis of the building/system/plant design.

Keywords: energy performance, design, retrofit, research, residential and commercial buildings

Expertise Required: High level of computer literacy not required; engineering background helpful for analysis of air handling systems.

Users: Over 500.

Audience: Mechanical, energy, and architectural engineers working for architect/engineer firms, consulting firms, utilities, federal agencies, research universities, and research laboratories.

Input: Building geometry, thermal characteristics, internal loads and schedules, heating and cooling equipment and system characteristics. Readable, structured input file may be generated by HBLC (Windows) or the BTEXT program.

Output: More than 50 user-selected, formatted reports printed directly by BLAST; also the REPORT WRITER program can generate tables or spreadsheet-ready files for over one hundred BLAST variables.

Computer Platform: PC-compatible, 386 or higher; HP/Apollo. Source code is available and has been successfully compiled on most UNIX workstations.

Programming Language: FORTRAN

Strengths: PC Format has Windows interface as well as structured text interface; detailed heat balance algorithms allow for analysis of thermal comfort, passive solar structures, high and low intensity radiant heat, moisture, and variable heat transfer coefficients -- none of which can be analyzed in programs with less rigorous zone models.

Weaknesses: High level of expertise required to develop custom system and plant models.

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web http://www.bso.uiuc.edu

Availability: Software prices range from $450 for an upgrade package to $1500 for new installations.

BLAST, HBLC

Software tool for calculating heating and cooling loads for buildings. Allows the user to access complex heat-balance algorithms using a Windows interface. Geometric inputs are entered graphically using intuitive click-and-drag mouse functions and allows the user to visualize the building model as it is developed. HBLC (Heat Balance Loads Calculator) creates an input file for and runs the BLAST simulation program. After simulating, HBLC retrieves results from the simulation and can present these results in a graphical presentation

Keywords: heating and cooling loads, heat balance, energy performance, design, retrofit, residential and commercial buildings

Expertise Required: High level of computer literacy not required; engineering background helpful for analysis portions.

Users: Over 500.

Audience: Mechanical, energy, and architectural engineers working for architect/engineer firms, consulting firms, utilities, federal agencies, research universities, and research laboratories.

Input: Interactive program in Windows environment.

Output: Can access most of BLAST's features. Presents graphs, for example, individual zone loads, load splits, etc.

Computer Platform: Windows 95 or NT preferred - may be able to use Windows 3.1 environment.

Programming Language: Visual Basic

Strengths: Input allows for easy detailing of geometric building model. Access to complex, accurate BLAST models as well as simple presentation of results. Access to all the BLAST libraries and these can be customized to user needs. Customization of necessary Fan System and Plant parameters for the described facility. Context sensitive help for HBLC features. Access to all the BLAST Family of Programs through the HBLC interface. Access to the BLAST Manual (Help file) from within HBLC.

Weaknesses: Some features of BLAST's geometry are not available through this interface.

Contact: See the contact information of BLAST

Availability: Software available only as part of BLAST package. Software prices range from $450 for an upgrade package to $1500 for new installations.

APACHE

Software tool for thermal design & energy simulation related to buildings. In design mode, APACHE covers the calculation of heating, cooling and latent room loads, the sizing of room units, internal comfort analysis and codes/standards checks. In simulation mode, APACHE performs a dynamic thermal simulation using hourly weather data. Linked modules deal with the performance of HVAC plant and natural ventilation. APACHE is a component of the IES Virtual Environment, an integrated computing environment encompassing a wide range of tasks in building design.

Applications:
Thermal design (heating, cooling & latent load calculations), Equipment sizing , Codes & standards checks

Dynamic building thermal performance analysis, Systems and controls performance, Energy use
Modules:
Geometrical modelling, building data input & visualisation
Management of data relating to materials, occupancy, plant operation and climate.
Shading analysis, Heat Gain calculations, Heat Loss calculations, Dynamic thermal simulation
Natural ventilation & indoor air quality analysis, HVAC system simulation
Results presentation & analysis

Keywords: thermal design, thernal analysis, energy simulation, dynamic simulation, system simulation

Expertise Required: 2 days training is recommended for the basic modules, with additional courses available for specific applications. Available in UK and other countries by arrangement.

Users: Many throughout Europe.

Audience: mechanical building services engineers, local government, building managers & landlords, building design consultants, architects, and university research and teaching departments.

Input: Geometrical building data may be imported from a range of CAD systems via customised links or DXF files. Geometrical models may alternatively be entered using facilities within the Virtual Environment. Input of data relating to materials, occupancy, internal gains, climate, air movement and systems is managed via graphical interfaces and supported by extensive databases.

Output: APACHE presents a wide range of outputs in tabular and graphical form. Outputs may be exported in a variety of common formats.

Computer Platform: PC running Windows 95, 98 or NT (3.51 or higher). 100 MB Ram or paging disk. 100 MB disk space. CD-Rom drive.

Programming Language: Visual Basic, C++, Fortran 77

Strengths: Operates within an integrated computing environment covering a range of building analysis functions. Strong links with CAD. Undergoing rapid development, with continuing input from research and engineering practice. Supported by in-house expertise. Rigorous analysis and visualisation of shading and solar penetration. Flexible & versatile system HVAC and controls modelling. Integrated simulation of building and HVAC systems.

Weaknesses: Certain energy systems not covered currently, eg phase-change materials, roof ponds.

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web http://www.ies4d.com

Availability: Contact IES at the above address or visit the IES4D web site.

BSim2000

Software package for evaluating the indoor climate and energy conditions as well as the designing of the heating, cooling and ventilation plants. The BSim2000 package comprise the programs: SimView (user interface and graphic model editor), tsbi5 (building simulation tool), XSun (dynamic solar and shadow simulation), SimLight (daylight calculation tool), Bv98 (compliance checker) and SimDXF (CAD import facility).

BSim2000 permits calculation on complex buildings with several (in principle indefinitely many) thermal zones and rooms simultaneously. BSim2000 utilises data from all structures in the thermal evaluation. BSim2000 interact directly with other applications for compliance with building regulations, dynamic - with animated results - solar and shadow distribution, CAD import for model making and daylight calculations. Results from BSim2000 can be exported as boundary conditions for CFD programs. Building models can be exported as input files to Radiance for detailed light analyses.

Keywords: building simulation, energy, daylight, thermal analysis, indoor climate

Expertise Required: Users must have some general knowledge on building design and how buildings behave thermally in order create the building model. Courses are offered.

Users: Approximately 50 licences, of which most are in Denmark. The majority of the existing tsbi3 license holders (200) are expected to shift to BSim2000 within year 2000.

Audience: Engineers, researchers and students.

Input: When using the BSim2000 the building is divided into rooms, some in thermal zones. Only rooms in thermal zones will be simulated dynamically, the rest can be used in other applications. The following groups of information are needed: materials, building component, equipment and systems.

Input interface

Input is given as properties of the individual objects or directly in the graphical representation of the model.

Output: The user can chose any of the calculated parameters for each thermal zone plus data from ambient climate, as output on hourly, weekly, monthly or periodical basis, in either tabular or graphic form. The variables can also be presented in “sum” graphs or tables. Finally the energy balances for each zone or the whole building can be shown. Outputs can be copied (graphics or numbers) for presentation in other programs.

Computer Platform: PC equipped with an Intel Pentium II processor (min. 200 MHz) or compatible. Operating system MS-Windows 9x, NT/2000.

Programming Language: Visual C++

Strengths: Analysis of the indoor thermal climate in complex buildings or buildings with special requirements for the indoor climate. Intuitive graphic user interface.

Weaknesses: Simple models for airflow, i.e. no zone model taking into account the airflow caused by wind pressure on the facades.

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web http://www.by-og-byg.dk/english/publishing/software/bsim2000/index.htm

Availability: Check web site for price and ordering information.

MARKETMANAGER (free evaluation version available)

Models any type of residential, commercial or industrial facility under single or multiple scenarios. With MARKETMANAGER, users can instantly select from libraries of predetermined heating and cooling equipment, HVAC, motors, lighting systems, appliances, pools, process equipment, and other building equipment. It simulates integrated building performance including load, system, and plant calculations modeled on an hourly basis. Standard ASHRAE algorithms are used in heating and cooling load calculations for local weather conditions. Model and evaluate the impacts of different rate structures including TOU, demand charges, load factors, or ratchets. Also compare "what if" scenarios to test performance measures, varying equipment, or alternate rate structures. Then generate reports based on your calculations.

Keywords: performance simulation, load calculation, energy analysis, measures, scenarios, rate tariffs

Expertise Required: Energy engineering, energy auditing, or energy management background helpful, regional and private training available.

Users: 2000+

Audience: Energy engineers, performance contractors, energy auditors, and sales engineers.

Input: Enter information into a Windows-based interface (Building layout - HVAC, envelope and systems setup, performance measures, rate tariff structures, weather data).

Output: Projected load calculations after applying local weather conditions, utility performance measures, internal gains and other factors. Numerous pre-set and customizable reports and graphs comparing load energy and cost calculations in different scenarios.

Computer Platform: System Requirements - Microsoft Windows (version 3.1, Windows for Workgroups, Windows 95, Windows NT) IBM-compatible 386 or higher.

Programming Language: Borland Pascal/Delphi, Crystal Reports, Graphics Server SDK.

Strengths: Windows graphic interface. Ability to show the affects of an entire building on load calculations and the interaction between lighting, heating, cooling and HVAC equipment. Evaluate "what if" questions by creating and customizing different scenarios. Completely automated data import/export. Internal quality control. Easily compare alternative rate tariffs.

Weaknesses: High level of expertise required to model and evaluate complex buildings and facilities.

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Availability: Software prices range from $995 to $4,495 depending on the type of license purchased. See web site to download an evaluation version.

HAP v4.0

A versatile system design tool and a powerful energy simulation tool in one package. HAP (Hourly Analysis Program) v4.0 for Windows also provides the ease of use of a Windows-based graphical user interface, and the computing power of Windows 32-bit software.

HAP's design module uses a system-based approach which tailors sizing procedures and reports to the specific type of system being considered. Central AHUs, packaged rooftop units, split systems, fan coils and PTACs can easily be designed, as can CAV, VAV, single and multiple-zone systems. The ASHRAE-endorsed Transfer Function Method is used to calculate building heat flow.

HAP's energy simulation module performs a true 8760 hour energy simulation of building heat flow and equipment performance. It uses TMY weather data and the Transfer Function Method. Many types of air handling systems, packaged equipment, and plant equipment can be simulated. Costs can be computed using complex utility rates. Extensive, easy to read reports and graphs document hourly, daily, monthly and annual energy and cost performance.

Keywords: energy performance, load calculation, energy simulation, HVAC equipment sizing

Expertise Required: General knowledge of HVAC engineering principles.

Users: 5000 worldwide.

Audience: Practicing engineers involved in the design, specification and analysis of commercial HVAC systems/equipment. Design/build contractors, HVAC contractors, facility engineers and other professionals involved in the design and analysis of commercial building HVAC systems. It can be used for new design, retrofit and energy conservation work.

Input: Building geometry, envelope assemblies, internal heat gains and their schedules; equipment components, configurations, controls and efficiencies; utility rates.

Output: 48 design and analysis reports available to view or print. Design reports provide system sizing information, check figures, component loads, and building temperatures. Simulation reports provide hourly, daily, monthly and annual performance data. Users control the content and format of all graphical reports.

Computer Platform: Windows 95/98/NT compatible PC, Pentium or higher, minimum 32MB RAM, minimum 20 MB hard disk space.

Programming Language: Software is compiled. Source code is not available.

Strengths: HAP balances ease of use with technical sophistication. Technical features are comparable to DOE 2.1; comparison studies with DOE 2.1 have yielded good correlation. The Windows graphical user interface, report features, data management features, on-line help system and printed documentation combine to provide an efficient, easy to use tool.

Weaknesses: HAP is not an effective tool for the research scientist. Because it is designed for the practicing engineer, HAP does not permit modification of source code to model one-of-a-kind equipment configurations and control schemes often studied in research situations.

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e-mail software.systems@carrier.utc.com
web http://www.carrier-commercial.com

Availability: In the USA first year license fee for HAP v4.0 is $1195; renewal fee in subsequent years is $240.

DOE-2
Hourly, whole-building energy analysis program calculating energy performance and life-cycle cost of operation. Can be used to analyze energy efficiency of given designs or efficiency of new technologies. Other uses include utility demand-side management and rebate programs, development and implementation of energy efficiency standards and compliance certification, and training new corps of energy-efficiency conscious building professionals in architecture and engineering schools.

Keywords: energy performance, design, retrofit, research, residential and commercial buildings

Expertise Required: Recommend 3 days of formal training in basic and advanced DOE-2 use.

Users: 800 user organizations in U.S., 200 user organizations internationally; user organizations consist of 1 to 20 or more individuals.

Audience: Architects, engineers in private A-E firms, energy consultants, building technology researchers, utility companies, state and federal agencies, university schools of architecture and engineering.

Input: Hourly weather file plus Building Description Language input describing geographic location and building orientation, building materials and envelope components (walls, windows, shading surfaces, etc.), operating schedules, HVAC equipment and controls, utility rate schedule, building component costs. Available with a range of user interfaces, from text-based to interactive/graphical windows-based environments.

Output: 20 user-selectable input verification reports; 50 user-selectable monthly/annual summary reports; user-configurable hourly reports of 700 different building energy variables.

Computer Platform: PC-compatible; Sun; DEC-VAX; DECstation; IBM RS 6000; NeXT; 4 megabytes of RAM; math coprocessor; compatible with Windows, UNIX, DOS, VMS.

Programming Language: FORTRAN 77

Strengths: Detailed, hourly, whole-building energy analysis of multiple zones in buildings of complex design; widely recognized as the industry standard.

Weaknesses: High level of user knowledge.

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facsimile (510) 486-4089
e-mail FCWinkelmann@lbl.gov
web http://simulationresearch.lbl.gov

Availability: Cost $300 to $2000, depending upon hardware platform and software vendor.
To keep up-to-date on DOE-2 and other simulation programs, request most recent copy and free subscription to Building Energy Simulation User News from Kathy Ellington, fax (510) 486-4089.

ASEAM *

Evaluation of high-potential, cost effective energy efficiency projects in existing Federal buildings; calculates results that are within 4-5% of DOE-2 annual energy results; using quick input routines, permits evaluation of a 10,000 ft2 building in about ten minutes. ASEAM (A Simplified Energy Analysis Method) Version 5.0 automatically creates DOE-2 input files. The FEMP Architects and Engineers Guide to Energy Conservation in Existing Buildings (published November 1990) uses ASEAM as a primary example of how software can be used in over 180 retrofit projects.

Keywords: energy performance, existing buildings, commercial buildings

Expertise Required: Designed to be used by non-engineers with minimal training.

Users: Several hundred.

Audience: Federal energy personnel.

Input: Building type and location, outside dimensions, percent glazing, usage patterns, number of floors, central systems and plant.

Output: Average monthly and annual energy savings from retrofits, taking into account all interactive effects using parametric analysis for optimization.

Computer Platform: PC-compatible, 286 minimum, with math coprocessor preferred.

Programming Language: C

Strengths: Currently allows an engineer to easily perform very sophisticated whole building energy analysis (calibrates to utility data using Lotus macros, does parametric analysis on dozens of energy conservation opportunities).

Weaknesses: Should have the same analytical process fully automated for less sophisticated users.

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web http://www.fishbaugher.com

Availability: Free from the web site. DOE no longer provides technical support or training for ASEAM.

EnergyPlus *

EnergyPlus is a new generation building energy simulation program designed for modelling buildings with associated heating, cooling, lighting, ventilating, and other energy flows. EnergyPlus builds on the most popular features and capabilities of BLAST and DOE-2 but includes many innovative simulation capabilities including time steps of less than an hour and modular systems simulation modules that are integrated with a heat balance-based zone simulation. Other planned simulation capabilities include solar thermal, multizone air flow, and electric power simulation including photovoltaic systems and fuel cells.

EnergyPlus is a simulation engine which reads input and writes output as text files. EnergyPlus input and output data structures were designed to facilitate third party interface development. Most users will use graphical user interfaces to develop the input files when these tools become available.
Realistic system controls
Moisture adsorption and desorption in building elements
Interzone air flow
Low temp radiant heating/cooling
Interior surface convection
Thermal comfort modelling options
Evaporative cooler models
Steam absorption chiller
Air flow sizing based on zone requirements
Accurate sky illumination model for daylighting calculations
Ability to read multiple interval per hour weather data files
Plenums
Enhanced calculation of return air heat gain from lights
Flat plate exhaust air heat recovery

Automated creation of EnergyPlus geometry input from CAD files
Example heating, ventilating, and air-conditioning system and equipment input templates
User-customizable reports
The EnergyPlus simulation program reads and writes output as text files. Its input and output data structure is designed to allow easy development of third-party interfaces--such as the 15 already available for DOE-2.

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TRNSYS

TRNSYS is a transient systems simulation program with a modular structure. It recognizes a system description language in which the user specifies the components that constitute the system and the manner in which they are connected. The TRNSYS library includes many of the components commonly found in thermal energy systems, as well as component routines to handle input of weather data or other time-dependent forcing functions and output of simulation results. The modular nature of TRNSYS gives the program tremendous flexibility, and facilitates the addition to the program of mathematical models not included in the standard TRNSYS library. TRNSYS is well suited to detailed analyses of systems whose behaviour is dependent on the passage of time.

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Software for Chiller Plant Optimization

QuikChill *

Chiller upgrade analysis software tool designed for both early screening and detailed design analysis of chiller systems. It performs economic and energy analyses of potential upgrade scenarios involving complex chiller plants with minimal plant and building data. QuikChill handles consolidation of existing chillers, integration of new chillers, and retrofits of existing chillers. QuikChill estimates loads using DOE-2-generated curves, which plot the relationship between cooling load and outdoor temperature. These curves reasonably predict annual cooling system requirements when used with local hourly temperatures and the peak load met by the existing system. Hourly temperature data is available for over 240 locations and users can supply peak information.

QuikChill's combination of simplified inputs, investment orientation, and unique approach to hourly cooling load estimation help fill an analytical void for the post-CFC chiller industry.

Keywords: chillers, CFC, retrofits, building loads

Expertise Required: General knowledge of chilled water systems is helpful. No computer programming experience required.

Users: Over 50.

Audience: Building owners, facility managers, energy managers.

Input: Existing and upgrade chiller information, building load profile, cooling tower and chilled water operating conditions, staging for multi-chiller scenarios, operating schedule, utility rate schedule, hardware and other additional costs.

Output: Effects of potential upgrades on hours of operation, kW, and kWh. Energy and economic savings expected. Overall project reports and reports for individual chillers.

Computer Platform: PC-compatible, 386 or higher, Windows 3.1 or higher required.

Programming Language: Visual Basic

Strengths: Provides a quick, easy, and accurate method for analyzing potential chiller system upgrades. Graphical interface allows for visual feedback. The potential effects of other building system upgrades are considered.

Weaknesses: Current version is limited to analyzing water-cooled centrifugal chillers.

Contact: Energy Star Buildings
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MC 6202J
Washington, DC 20460, USA
telephone 1-888-STAR-YES
facsimile
e-mail
web http://www.epa.gov/buildings

Availability: Available free of charge to all Energy Star Buildings Partners through the Energy Star Buildings web site.

Dimensioning of Ice Storage

EZDOE

An easy to use IBM PC version of DOE-2. EZDOE calculates the hourly energy use of a building and its life-cycle cost of operation given information on the building's location, construction, operation, and heating and air conditioning system. Using hourly weather data and algorithms developed by Lawrence Berkeley National Laboratory, EZDOE takes into account complex thermal storage effects of various building materials. In addition, it can also accurately simulate the operation of all types of heating and cooling plants including ice water thermal storage and cogeneration systems. Up to 22 different air handling systems each with multiple control options are supported. The types of heating and cooling plants allowed is nearly infinite as thousands of combinations of chillers, boilers, furnaces, pumps, and cooling towers are allowed. There is even provision for user defined plants and performance curves. The economic analysis capabilities of EZDOE allow for complex utility rate structures, fuel costs, initial equipment costs, replacement costs, and annual costs of non-plant items and baseline data for comparative runs. A large library of over 230 hourly weather data files is available for EZDOE. One weather data file of your choice is supplied with EZDOE while others are available at additional cost.

Keywords: energy performance, design, retrofit, research, residential and commercial buildings

Expertise Required: Basic familiarity with building geometry and HVAC systems is desirable but not absolutely necessary.

Users: Unknown.

Audience: Architects and engineers involved in new and retrofit building projects, researchers, equipment and utility marketers.

Input: Features full screen editing with simple

Output: Offers all of the standard reports as the workstation version of DOE-2. These reports can be viewed on the screen, stored in a disk file, or printed.

Computer Platform: Requires an 80486 or higher IBM PC compatible.

Programming Language: N/A

Strengths: Implements DOE-2 in an easy-to-use full screen editing environment with dynamic error checking. All input data is checked at the time of entry so that no improper data can be entered. If you have a question about what the program is requesting, you can press the “?” or F10 key to obtain additional help explanations. All data is saved to disk as it is entered. Four major types of data are requested: Loads, Systems, Plants, and Economics. Load data contains the building and space dimensions, wall and glass orientations, construction materials, people, lighting, equipment, and much more. The Systems data involves all information concerning air handling and heat delivery systems. VAV, constant volume, PTAC, dual duct, two/four pipe fan coils, and radiators are just a small sampling of the many system types supported by EZDOE. The Plant data concerns the cooling and heating equipment such as chillers, boilers, cooling towers and pumps. The Economic section considers initial, annual, cyclical, replacement, and operating costs.

Weaknesses: Limited to capabilities within the DOE-2 program.

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e-mail info@elitesoft.com
web http://www.elitesoft.com

Availability: Demonstration copy available for download from the web site, or order a copy, with complete documentation. Demonstration copies retain all the functionality of the full programs, they are just limited on the size of the project data that can be entered.

Dimensioning and simulation of chilled water network

STANET

STANET® is an integrated application for network analysis. Besides calculation, graphic input, output, a database browser is included. The browser may be displayed together with the network map. STANET® may be used as a network information system because it uses standard database files, which may be extended by the user. Because graphics and database are integrated, data exchange with other applications is simple.

Simple creation and modification of networkmodels:
Powerful input capabilities via keyboard, mouse and digitizer tablet, including import of background pictures (pixel format: TIFF, BMP, etc. or vector format: DXF format).
Flexible import export interface: ODBC, ASCII-Text, dBase-III, MapInfo, ArcView, SICAD, GEOGRAT, Ganesi, etc.
Analysis and simulation
Short simulation time, e.g. 2 seconds for 10.000 nodes on PentiumPro 180 MHz
Extended period simulation, e. g. over a day
Diameter and routing optimization
Free definition of mathematical and hydraulical settings
Automated proof of topology
Automated creation of subnetworks from closed values and regulators
Calculation of additional values like temperature radiation into the ground and quality tracking
Contact: Fischer-Uhrig Engineering
Wuerttembergallee 27

D - 14052 Berlin, Germany
e-Mail: info@stafu.de
Telephon: +49 - 30 - 300 993 90
Fax: +49 - 30 - 304 43 05

FLOWRA

Flowra 32 has a graphical user interface and it dimensions and simulates a network in graphic form. Flowra 32 can carry out all district energy network planning operations.
The accurate and efficient Flowra 32 is used in different parts of the world.

Contact: Komartek Oy,
Laserkatu 6, 53850 Lappeenranta, Finland
Tel 0201 42 3800, Fax 0201 42 3820

AFT Fathom

AFT Fathom is a general-purpose pipe network analysis tool that can be used to simulate any incompressible fluid flow problem. Fathom is widely used to design chilled water and hot water systems, fire suppression systems, water distribution, chemical process plants, and HVAC duct systems. Fathom includes extensive built-in databases of fluids (liquids and gases), pipe materials, pipe fittings, and insulation materials. Fathom includes integrated heat transfer capabilities to model piping heat transfer and heat exchangers.
Users: Estimated 8,000 -10,000 current users.
Audience: Mechanical, energy, chemical, architectural engineers working for manufacturing, architectural, consulting, and utilities companies and research universities and research laboratories.
Input: Pipe or duct specifications and components (fittings, pumps, control valves, etc.) are drawn and entered into the graphical user interface.
Output: Tabular or graphical output of over 100 calculated variables.

Computer Platform: All IBM PC computers running Windows 95, 98, ME, NT, or 2000.
Programming Language: Visual Basic
Strengths: Easy to use graphical interface allows users to set up problems and get results quickly. Full featured pipe network analysis tool can solve networks of any size and has integrated heat transfer analysis with variable fluid properties. Complete representation of pump behavior with efficiency and NPSHA allows users to size pumps and investigate energy efficient pumping designs with variable speed pump drives.
Weaknesses: Software only models steady-state flows. Quasi-steady time-varying flow conditions must be modeled as different scenarios for each time period of interest.

Contact: Applied Flow Technology
PO Box 6358
Woodland Park, CO 80866, USA
telephone 800-589-4943
facsimile 719-686-1001
e-mail support@aft.com
web http://www.aft.com

Availability: Full license for stand-alone or network installation for Fathom 5.0 is $1495.

AFT Mercury

AFT Mercury is a general-purpose pipe network optimization tool. AFT Mercury combines a proven pipe network analysis tool (AFT Fathom) with state-of-the-art optimization tools to allow designers to optimize pipe and component selection for lowest initial cost or lowest Life Cycle

Cost (LCC) for a pipe or duct systems.

Keywords: optimization, pipe optimization, pump selection, duct design, duct sizing, chilled water systems, hot water systems

Expertise Required: Engineering background in piping or duct design required. No special knowledge of computers required.

Users: New product in 2001

Audience: Mechanical, energy, chemical, architectural engineers working for manufacturing, architectural, consulting, and utilities companies and research universities and research laboratories.

Input: Pipe or duct specifications and components (fittings, pumps, control valves, etc.) are drawn and entered into the graphical user interface. Pipe, duct, pumps (fans) and power cost data required to obtain useful life cycle cost optimization.

Output: Tabular or graphical output of over 100 calculated variables.

Computer Platform: All IBM PC computers running Windows 95, 98, ME, NT, or 2000.

Programming Language: Visual Basic

Strengths: Only software available to optimize pipe networks. Gradient based and Genetic Algorithm optimization methods can optimize pipe or duct networks with hundreds of design variables with both continuous optimization (any pipe size allowed) and discrete optimization (best combination of standard pipe sizes) to meet a set of design criteria (minimum flow requirements, maximum pressures, etc.). Users can optimize over multiple operating scenarios to minimize the total energy use and cost.

Weaknesses: More data input is required to optimize for life cycle costs.

Contact: See AFT Fatholm

Availability: AFT Mercury 5.0 is currently available. Software purchase includes training class on pipe/duct system optimization.

Optimization Softwares

GenOpt *

A generic multi-parameter optimization program for system optimization. It automatically determines the values of user-selected design parameters that lead to the best operation of a given system. Optimizes a user-selected objective function, such as a building's calculated annual energy use. It also offers an interface for easily implementing your own optimization algorithms into its library.

GenOpt has an open interface on both the simulation program side and the optimization algorithm side. By modifying a configuration file, it allows users to easily couple any external program (like DOE-2, SPARK, BLAST, EnergyPlus, TRACE, TRNSYS, etc., or any user-written program). GenOpt is written entirely in Java so that it is platform independent. An interface for coupling external simulation programs and adding custom optimization algorithms is available. If the simulation input files are available, the time to set up an optimization problem is typically less than one hour.

See example screen images

Keywords: system optimization, parameter identification, nonlinear programming, optimization methods, HVAC systems

Expertise Required: Computer literacy required. No programming skills required for using the standard package. Basics of Optimization Theory for non-experienced users explained in program documentation. Knowledge of Optimization Theory and basic skills in Java required for adding your own optimization algorithms to the GenOpt library.

Users: 100
Audience: Engineers, researchers.

Input: The following input is read from ASCII files: (a) Specification of parameters to be optimized and their minimum and maximum values, (b) settings for the optimization algorithm, (c) configuration file of simulation program (i.e., how to start simulation program, where to read simulation output), (d) input template file (used for generating simulation program input file).

Output: Online plotting of optimization progress (objective function and free parameters). Result of optimization in text file.

Computer Platform: Platform independent (written in Java).

Programming Language: Java

Strengths: Utility classes such as for linear algebra, optimality check, or doing line-search can be used for user's own implementation of optimization algorithms.

Weaknesses: GenOpt is not aimed at solving optimization problems when the properties of the objective function (e.g., gradient, Hessian) are analytically available. It will work on those problems, but more efficient programs exist for that purpose.

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web http://gundog.lbl.gov/GO/index.html

Availability: GenOpt 1.1 is available free of charge.

IEA District Heating and Cooling

Optimization of Cool Thermal Storage and Distribution

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