

**A COMPARISON OF WATER-ETHANOL,
PURE WATER AND ICE AS STORAGE MEDIA FOR
BUILDING THERMAL STORAGE APPLICATIONS**

by

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Abstract

Cold storage can be used to shift the electricity demand associated with cooling from on-peak to off-peak times and thereby save primary energy and utility charges. Common storage media include pure liquid water and (pure) ice. Recently, an alternative to these media has been promoted to perform superior over these media: a mixture of water and a freezing point depressing additive, e.g. an alcohol. The advantages over conventional water or ice storage were that the medium could be stored at lower temperature, thereby allowing for a higher design temperature difference, increased energy storage density and lower total cost, relative to pure water or ice storage. However, possible disadvantages include cost for the additive, and higher energy cost for providing the medium at a lower temperature.

This study gives an introductory overview over common storage media and cold storage operating strategies, then important thermophysical and transport properties of a binary mixture, water and ethanol, are derived using fundamental thermodynamic relations, mixing rules and curve fits of measured data. Three operating strategies (full storage, partial storage with chiller priority and with tank priority) are simulated with three different storage media (pure water, water-ethanol, and ice) each, plus a conventional system without storage, totalling 10 different system setups. Two physical setups are modeled with the simulation software TRNSYS, one for the liquid storage systems and direct chilling, and one for dynamic ice storage systems. The building load is obtained

from a data file provided by the Energy Center of Wisconsin, featuring detailed data obtained from an office building. The individual equipment sizes for each system are determined by processing the design day data repeatedly to create a steady state periodic solution. The main simulation includes the months May through September. The results are presented together with an economic analysis over a study period of 20 years, including investment and operating expenses and accounting for the time value of money.

Using water-ethanol as the thermal storage medium does not generate Life Cycle Savings over ordinary water due to higher operating expenses (if stored at lower temperature than feasible with water) and higher investment costs (caused by ethanol price and not offset by smaller equipment). Ice storage using an ice harvester is economically undesirable due to very high initial cost, though other ice generating devices may generate Life Cycle Savings over liquid storage systems. On-peak demand charges must be avoided for they are the main contributor to operating expenses. This can be achieved in a full storage operating mode. When non-cooling building energy demand is present, sophisticated partial storage control strategy might generate more savings than full storage.

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Chapter 1

Introduction: Cold Storage

This chapter outlines the state-of-the-art in cold storage. Storage media requirements, commonly used substances and the selection of water-ethanol are discussed as well as common operating concepts.

1.1 Required Characteristics of Storage Media

There are certain requirements that a possible storage medium has to meet in order to be suitable for (cold) storage. Some are obvious and some are less obvious but equally important. The following compilation lists some of these requirements.

- Data: the fluid's thermophysical and transport properties have to be *known* over the range of conditions (temperature, pressure, if applicable: concentration) intended for use in the storage application. An entire chapter of this thesis is dedicated to estimation of fluid properties for water-ethanol, since this data was not entirely available.

- Density: the storage system's efficiency can be significantly improved by using a stratified tank in contrast to a completely mixed tank¹. Therefore the storage medium's density needs to be a monotonic function of temperature over the temperature range of the storage application.
- Freezing point: one key design parameter is the temperature difference between chiller supply and coil return. The higher this difference, the lower are (theoretically) the mass flow rates and the storage volume. The freezing point represents one physical lower boundary for the chiller supply temperature². Therefore it is desirable to have a liquid with a low freezing point.
- Heat capacity: media with high specific heat are favorable, because the specific heat determines the amount of energy that can be stored in a given volume (assuming a fixed temperature difference). The tank size has a major influence on system first cost. If the storage process also involves a phase change, the latent heat of fusion is also of interest for the same reason.
- Phase separations: to simplify the system, the medium should show only one liquid phase under operating conditions.
- Viscosity: the medium should have a low viscosity. A high viscosity causes high friction in the tubes. High friction will increase the pump sizes, pump energy demand, and possibly the tube diameter. Also, since a fraction of the pump's energy will be converted into heat, a highly viscous medium will require more chiller capacity.
- Thermal conductivity: though this is only of minor importance, the thermal conductivity should also be low, because this can increase the capability of the tank to

¹The concept of stratified fluid tanks is explained in section 5.1.

²For water, the temperature at which it has its maximum density is another lower boundary, which will be explained in greater detail later.

maintain a thin thermocline (this topic will be discussed later).

- **Cost:** the amount of storage capacity needed implies that the storage medium has to be inexpensive in order not to raise installation cost significantly.
- **Toxicity:** the medium should not be harmful to humans or the environment. However, security measures can be taken to prevent leakage or harmful concentrations of the medium in the surrounding environment. Of course some extreme measures will be expensive and need to be included in the cost calculation.
- **Flammability:** the medium should not be flammable under any conditions that could arise during the process. This applies to both its liquid and its vapor phase. If the medium is a mixture, there might be certain concentrations of one substance where the mixture or one of its phases are inflammable, and these concentration windows have to be investigated. The same thoughts about security issues as under toxicity apply here, too.

1.2 Commonly Used Storage Media

The commonly used cold storage systems can be distinguished by the storage medium used and by whether storage uses latent heat only or takes advantage of the phase change energy of the storage medium.

1.2.1 Water

Pure water is the most commonly used storage medium, since it has many advantages over other substances:

- low price and (practically) infinite availability

- physical and chemical stability
- non-toxic and non flammable
- high specific heat (4.19 kJ/kg K)
- well known property data

For some applications, including cold storage for building air conditioning, some of the characteristics of water³ are not always desirable, for example

- **the occurrence of a density maximum at (ca.) 4°C.** This limits the use of water in a stratified fluid storage tank operating at temperatures below this limit. If water were cooled below 4°C, it would be lighter than the water at 4°C and could therefore not be stored *below* the warmer water in the tank, instead, it would rise, mix with warmer water of the same density, and prevent the stratification effect.
- **the freezing point of 0°C.** Since pure water does not exist in equilibrium in a liquid state at temperatures below 0°C, it is not possible to use water to cool another fluid down to lower temperatures than (ideally) 0°C in a heat exchanger.

In building air conditioning, however, the air is supplied at about 10 to 15°C, which does not require these low storage temperatures. Even cold air distribution (supply air temperatures in the range of 4 to 10°C) is possible, which reduces the necessary air volume flow rate and thereby fanpower. On the other hand, the performance of cold air distribution systems or other applications requiring low storage temperatures might be improved if the water supply temperature were below 0°C.

³at atmospheric pressure of 1 atm = 1.01325 bar

1.2.2 Ice

Storing water in solid form takes advantage of the enthalpy of fusion of water, which is $h_{fg} = 334.944 \text{ kJ/kg}$ at atmospheric pressure and 0°C . Assuming a reasonable temperature difference of 15 K between the water supply and return temperature for a liquid water storage system as well as for an ice storage system, it is obvious that ice storage needs considerably less storage volume to store the same amount of energy:

$$\begin{aligned} \frac{Q_{ice}}{Q_{water}} = 1 &= \frac{V_{ice} \rho_{ice} (c_p \Delta T + h_{fg})}{V_{water} \rho_{water} c_p \Delta T} \\ &= \frac{V_{ice} * 970 * (4.19 * 15 + 334.944)}{V_{water} * 1000 * (4.19 * 15)} \\ \Leftrightarrow V_{water} &= 6.1 * V_{ice} \end{aligned} \quad (1.1)$$

Practically, the necessary volume for ice storage is about 1/5 to 1/8 of the volume of a comparable water storage system (EPRI, 1985).

Static and Dynamic Ice Storage Systems

The generation of ice may take place in different ways. In a *static* system, the ice is generated in the storage tank itself. The tank is usually fully sealed and insulated. It is penetrated by narrowly (2...5 cm) spaced coils of the chiller evaporator, and ice is built up in layers on these coils. The water in the tank is agitated to ensure its uniform temperature and a uniform ice generation throughout the tank. The supply water is taken from the top, whereas the warmer return water enters at the bottom, and this creates further movement of the water in the tank.

In a *dynamic* ice storage system the ice is formed outside the tank in plate, chunk, crushed or slurry form by means of an ice harvester or an ice slurry generator. In the case of an ice harvester, the ice is generated in cycles: water is circulated over a plate evaporator to

create a layer of ice. Once the layer reaches a critical thickness, the evaporator receives hot refrigerant from the compressor discharge to force the layer to fall down. The warm refrigerant in the evaporator causes the ice to release from the external surface. The ice falls into a storage tank below creating a large porous plug heat exchanger. The density of the ice created in this manner is usually only 1/2 the value that is achieved in a static system (highly porous). The cyclic process is more complicated and requires more controls, and energy gains from the environment can be higher for the open tank. These are a few reasons why static ice storage systems are more popular than dynamic systems.

Ice Storage Tank Effectiveness

One of the characterizing quantities of an ice storage tank is its effectiveness ε , which can be expressed as the dimensionless water supply temperature:

$$\varepsilon = \frac{T_{ret} - T_{sup}}{T_{ret} - T_{sup,ideal}} \quad 0 \leq \varepsilon \leq 1 \quad (1.2)$$

Stewart⁴ showed that while discharging a dynamic ice storage tank the leaving water temperature stays relatively constant for a long time (meaning $\varepsilon = \text{constant}$), and that only towards the end this temperature will start to approach the water inlet temperature ($\varepsilon \rightarrow 1$).

Especially for an ice storage system, the tank size strongly depends on the tank performance curve, which is illustrated in the following example:

1. The chiller and the tank have been sized, for example with an appropriate rough-cut equation.
2. Over the course of the (design) day, the tank can be discharged to meet (part of) the load and then be recharged with the chiller. When the peak load occurs, ε will

⁴(Stewart and Gute, 1995)

have a certain value ε_{peak} , depending on what the tank's maximum capacity is, how much ice has already been melted and how much ice has been build up with the chiller.

3. The maximum load that can be met with the tank at peak time is

$$\dot{Q}_{max} = \varepsilon_{peak} \dot{m}_{max} \Delta T_{des} \quad (1.3)$$

where \dot{m}_{max} is the design water supply mass flow rate to the coil. If the value of \dot{Q}_{max} is lower than the actual peak demand, the process of sizing chiller and tank must be continued iteratively.

1.2.3 Other Liquids using Sensible Heat

To avoid initiating crystallization of water at 0°C, water is often mixed with other liquids or solids (which then dissolve, e.g. salts) to depress the freezing point to lower temperatures. Also, substances other than water (or mixtures not containing water) can be used. However, there are trade-offs that have to be taken into account, which is illustrated by considering some examples:

- **Salts** dissolved in water usually form bases and acids that destroy the system components (pumps, coils, pipes, valves, and the tank) by increasing corrosion rates. They might also be harmful to people and the environment.
- **Glycols** (e.g. ethylene-glycol) are expensive and require a considerably higher amount of pumping power, due to their high viscosity. Toxicity and environmental impacts are also a concern. This is also valid for **glycol-water** mixtures.
- **Alcohols** are flammable if they evaporate from the solution and reach a certain air-alcohol concentration window; the upper and lower concentration limits depend

on the actual substance. They might be costly, too, depending on the tank volume, the sort of alcohol and its desired concentration.

- **Ammonia**, in its pure state, is highly toxic, and the dangers resulting from leakage of an ammonia storage tank are too high, or, in other words, as of today an ammonia storage system is economically not feasible in cases where occupants might be exposed to it, considering the necessary high amount of safety measures. **Ammonia-water**, as a mixture, might be a feasible and less dangerous alternative, but this matter will not be investigated on in this thesis.
- Most **refrigerants** have to be produced in complicated and therefore expensive processes. Refrigerants containing CFCs could be produced for less, but due to their impact on the ozone layer they are officially not produced any more. Less harmful alternatives do still require sealed systems and safety measures. Consequently, refrigerants are not applicable for cold storage.

1.2.4 Other Substances using the Latent Heat of Fusion

Again, the most promising substances are mixtures of water with alcohols or some glycols, due to the advantage in price, health and security. Other materials like eutectic salts have been evaluated mainly for their use in heat storage, and in general the melting (or freezing) point of these substances lies at temperatures not suitable for cold storage (well above comfortable room temperatures). Water-alcohol and water-glycol are used in a slurry form, with the advantage that the slurry is liquid and pumpable even below 0°C. Currently, research is being conducted on the thermal and transport properties of ice slurries and water-ethanol slurries by several institutions. Future cold storage applications can be expected to feature these new media.

1.2.5 Water-Ethanol

Out of the many alternative storage media mentioned previously, I chose to compare a binary water-ethanol mixture with the common storage media pure water in its liquid form (“water”) as well as in its solid form (“ice”). Ethanol is the second most simple alcohol, according to its chemical structure C_2H_5OH , only Methanol contains less carbon atoms. Ethanol is probably the most widely used alcohol (only to mention the beverage industry), and its properties (in its pure state) are well known. Being a product of fertilization it is environmentally friendly, bio-degradable, and non-toxic, if not consumed in excess.

There might be safety concerns due to ethanol’s flammability, and appropriate measures will have to be taken, should ethanol be used in a large-scale application. The CRC Handbook of Tables for Applied Engineering Science (CRC, 1972)⁵ lists the flammability windows for ethane (C_2H_6) as 3.0...12.5% of volume and for ethylene (C_2H_4) as 2.75...28.6% of volume. No values for any alcohols are given, but it can be assumed that the flammability limits will be similar, starting from very low concentrations. However, the normal boiling point of pure ethanol is 173°F or 78°C, a temperature which is well above the temperature range intended for this application.

The results of this project, which are based on ethanol, should be transferable to other simple alcohols.

1.3 Economics and Energy Savings

Building air-conditioning systems without thermal storage operate in a way that the cooling device draws electricity⁶ from the utility supply line such that the momentary demand can be satisfied with the cooling equipment employed. The accumulated energy

⁵page 388

⁶Most air conditioning operate on electric power, absorption chillers are used mostly in large scale industrial applications due to their size and their first cost.

demand of many customers therefore shows a peak during certain times of the day, since the customers all are exposed to roughly the same ambient conditions and will have similar occupation profiles (due to similar business hours).

The peak usually occurs in the later afternoon on regular work days. The utility has to provide the necessary peak capacity to satisfy its customers, and during most of the day the equipment will operate at a very low fraction of its capacity (part load ratio) because less than peak capacity is required. The utility's average power generation effectiveness (the fraction of electric energy produced over the amount of primary energy used) will therefore be lower than if there was less capacity and a higher daily part load ratio. This, in turn, means higher cost of electric energy.

1.3.1 On-peak and Off-peak Charging

In an effort to penalize those customers who require the utility to maintain this high capacity and to encourage the use of electric energy at times where the utility equipment operates only at low part load ratio, regular work days were divided up into an on-peak period and an off-peak period, with a high price for electricity consumed on-peak and a low price off-peak. Many utilities in the United States offer these Time-Of-Use (TOU) charging options to major customers. If major customers shift their demand from on-peak hours to off-peak hours, the utility might even be able to avoid having to purchase new generation equipment to increase its capacity.

One of the main purposes of cold storage is to shift the use of electricity from the on-peak period to the off-peak period where electricity is cheaper. Other purposes include lower first cost due to smaller equipment, and less total energy use due to better utilization of the equipment (better average COP), which is possible even though additional heat gains occur in a storage tank. Depending on the utility's TOU rates and system operational

strategy, and assuming that the storage system is well designed for the customer's needs, the customer's energy bill can drop significantly, paying off the investment for the storage system within few years.

1.3.2 Demand Charging

With major customers, a utility measures the customers' maximum demand in order to calculate how much capacity the utility needs to provide under worst conditions. Then the utility charges its customers according to this peak demand, even if the customer has only required this energy rate once in a long timespan, e.g. a month. These rates will remain effective for a longer period, and it is possible that a single maximum peak is followed by an entire year of higher rates (in the case of ratcheted rate structures). There is definitely a motivation for customers to avoid high demand peaks, and there are operating strategies aiming to lower peak demand, which will be explained further below.

1.3.3 Energy Savings

The casual observer might conclude that the savings in cost do not seem to be accompanied by reduced energy consumption for a thermal energy storage system. The storage tank, which is usually exposed to ambient conditions, loses availability to the environment (it actually gains energy, but this undesired gain has to be compensated by additional chiller activity). Due to more piping, more power is required to operate pumps than in a non-storage system.

However, cold storage can allow (depending on the operation strategy) the refrigeration equipment to be operated almost exclusively under full load where the COP is highest, and turned off otherwise. The savings in electric power are likely to be higher than the power needed to compensate undesired gains. Additionally, by flattening the utility's

total demand curve, the utility's average energy efficiency can rise, which results in the consumption of fewer primary energy units, e.g. fuel. If the cold storage system is replacing an old non-storage cooling system, the savings must be attributed to both the newer (more efficient) equipment and the fact that it is being operated under better load conditions, and distinction between those is difficult, but the overall energy consumption will definitely be reduced.

Other features that can lead to reduced total energy consumption include the fact that at night the ambient temperature is lower, and the condenser can reject heat from the evaporator more efficiently, which increases the actual chiller performance. Plus, the availability of very cold water, e.g. from an ice storage tank, allows the air to be distributed at a lower than usual temperature (cold air distribution), which in turn requires less air flow and thereby less fan power.

1.4 Common Operating Strategies

The following section gives a brief overview over the different operating concepts in cold storage. The concepts are simplified to bring out the fundamental differences between them. In practice, systems will be much more complicated, the concepts themselves may be combined or altered, or certain restrictions may apply to a specific system.

It is important to use the following equations with the actual load of the coil (\dot{Q}_{coil}), and not only the building load (\dot{Q}_{bldg}) due to occupants, appliances, ambient conditions, etc. The difference between these two loads is that the coil has to meet \dot{Q}_{bldg} and the ventilation load \dot{Q}_{vent} , which will in most cases increase \dot{Q}_{coil} significantly over \dot{Q}_{bldg} . In this simulation, the building load is read from a data file, and the ventilation load is calculated for the conditions present at every discrete timestep.

1.4.1 Full Storage

In full storage operation the on-peak load is met by the tank alone, with the chiller not operating at all during this time. During the off-peak period the chiller recharges the tank and meets the off-peak load. This kind of system is easy to control, but it results in a comparatively large tank and chiller.

The following equations, which translate the above described way of operation into energy balances, can be used to roughly size tank and chiller. The on- and off-peak periods are defined by t_{on} and t_{off} , the respective hours of the day at which each period starts (and the other one ends). $\dot{Q}_{chiller}$ is the design chiller capacity (kW), Q_{day} is the cooling energy demand on the design day (kJ), Q_{on} is the demand during the on-peak period on the design day⁷, and c_p , ρ , T_{ret} and T_{sup} are the (design) values for the storage medium.

$$Q_{day} = \int_0^{24 \text{ hr}} \dot{Q}_{coil} dt \quad (1.4)$$

$$Q_{on} = \int_{t_{on}}^{t_{off}} \dot{Q}_{coil} dt \quad (1.5)$$

$$\Delta t_{on} = t_{off} - t_{on} \quad \Delta t_{off} = 24 \text{ hr} - \Delta t_{on} \quad (1.6)$$

$$V_{tank} = \frac{Q_{on}}{c_p \rho (T_{ret} - T_{sup})} \quad (1.7)$$

$$\dot{Q}_{chiller} = \frac{Q_{day}}{\Delta t_{off}} \quad (1.8)$$

Equation 1.7 assumes that the tank will be completely discharged on the design day, but usually a safety factor must be included. Heat gains from the pumps and the environment are also not included. Similar restrictions apply to the following equations.

⁷The concept of design days is explained in section 3.2.1

1.4.2 Partial Storage

In partial storage, the chiller may also run during on-peak hours and the tank does not need to be as large as for the full storage system.

Partial Storage with Chiller Priority

In partial storage with chiller priority, the chiller is designed to run 24 hours on the design day, and the tank meets that part of the peak load which cannot be met by the chiller. The size of the chiller can be estimated by

$$\dot{Q}_{chil,des} = \frac{Q_{day}}{24hr} \quad (1.9)$$

At every discrete timestep, the actual cooling load is either higher or lower than $\dot{Q}_{chil,des}$. The case of the coil load \dot{Q}_{coil} being exactly equal to $\dot{Q}_{chil,des}$ can be included in either alternative. If \dot{Q}_{coil} is higher, additional water supply from the tank is needed, and if it is lower, the chiller has available capacity to charge the tank.

The tank can be roughly sized using the following equations:

$$Q_{tank,des} = \int_0^{24hr} (\dot{Q}_{coil} - \dot{Q}_{chil,des})^+ dt \quad (1.10)$$

$$V_{tank} = \frac{Q_{tank,des}}{c_p \rho (T_{ret} - T_{sup})} \quad (1.11)$$

The expression $(X)^+$ is an abbreviation for $\max(0, X)$, meaning that this term is either positive or zero, and similarly $(X)^-$ stands for $\min(0, X)$, meaning that this term is either negative or zero. So the integral in equation 1.10 represents the energy that has to be provided by the tank. It is also useful to verify that the chiller has enough capacity to

recharge the tank:

$$-\int_0^{24\text{ hr}} (\dot{Q}_{coil} - \dot{Q}_{chiller})^- dt \text{ must be } \geq Q_{tank} \quad (1.12)$$

Figure 1.1 illustrates partial storage with chiller priority and the meaning of the integrals in equations 1.10 and 1.12. For an ideal tank (perfect stratification, no energy gains), the two integrals are equal. In practice, the chiller has to be larger to compensate non-idealities, and the necessary tank size decreases.

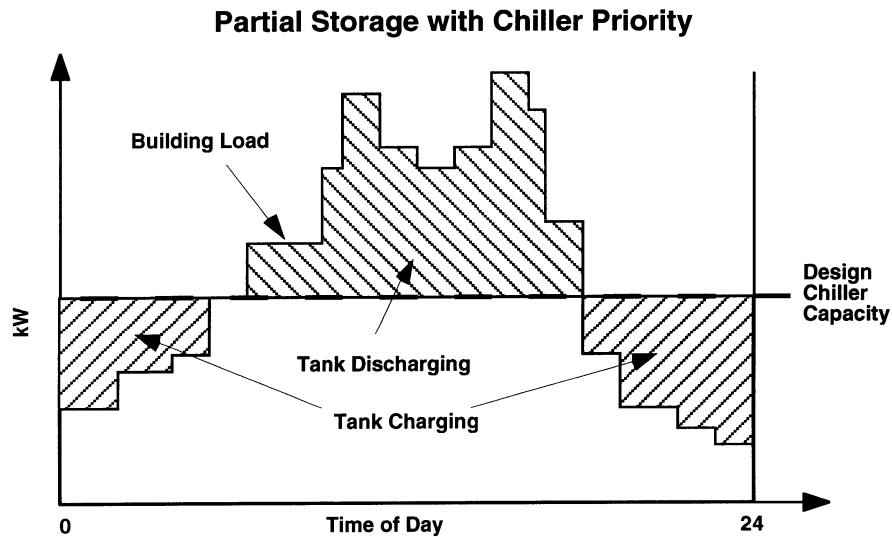


Figure 1.1: Partial storage with chiller priority. The areas *Tank Charging* and *Tank Discharging* represent the integrals in equations 1.10 and 1.12. The load is assumed constant over each discrete timestep. In reality the chiller capacity is not constant (the design capacity), because even if the mass flow rate is held at its design value, the temperature of the medium entering the chiller will vary over the day.

Partial Storage with Tank Priority

Partial storage with tank priority requires the anticipation of the daily load profile, because the purpose is to operate the chiller during on-peak hours only until the remaining daily

load can be met with the remaining tank capacity.

The load can be anticipated in many ways. A very simple approach is to assume that on every day the shape of the load profile will resemble the last day's profile, which must then be recorded. Comparing each timestep's load value with the same timestep on the last day allows to scale the profile up or down. This is the technique I used in my controller Type 99 in section 5.7.

Other algorithms estimate occupancy profiles and the day's ambient condition (weather) and combine these factors with recorded data and other factors to a load profile prediction where the shape does not necessarily depend on the previous day. These methods are very sophisticated and require a great deal of effort.

It is difficult to size the equipment for partial storage with tank priority, and every selection will have to prove its feasibility in a simulation. Often this operation strategy is combined with demand leveling, which will be explained later. A good start is to use the equations for partial storage with chiller priority, which I did to size the equipment for this simulation.

Of particular importance is how a tank priority system is controlled. Questions that need to be answered at every timestep include

- What will be the coil load until the end of the on-peak period?
- Will the tank be able to meet this load alone, or should the chiller operate on-peak?
 - For a stratified storage tank, will the supply water be cold enough, or is the thermocline already located at the tank bottom?
 - Under demand leveling control: at what fraction of design capacity should the chiller operate, should the demand level be increased?
 - For how long does the chiller have to operate on-peak?

- Will the chiller be able to recharge the tank entirely until the begin on the next day's on-peak period (and is this at all necessary)?

In fact, I experienced that all tank priority systems had to be resized every time more than just minor changes were applied to my tank priority control strategy.

Load Leveling and Demand Leveling

Two other often used expressions are load leveling and demand leveling. *Load leveling* means to minimize the chiller capacity by operating it throughout the entire (design) day, and this strategy corresponds to partial storage with chiller priority.

Demand leveling means to minimize the maximum energy per time that the utility has to provide (see demand charging, section 1.3.2). Assuming some non-cooling load shape, the idea is not to exceed the occurring maximum demand caused by the non-cooling load, but instead run the chiller with the remaining capacity ("filling the shoulders"). This means that the chiller is operating at part load conditions more frequently than under the load leveling strategy.

A fictitious building load profile is shown in figure 1.2. The second subfigure illustrates full storage operation, whereas the last two show the principle of partial storage operation with demand leveling and with load leveling. The total building load (units of power) is plotted over the hours of the design day, cooling and non-cooling load are accumulated. The areas represent the energy consumption on the design day⁸. The cooling load occurs in the way shown in the first subfigure. The other three subfigures illustrate when the energy needed to cool the storage medium is actually taken from the utility.

⁸The areas for cooling load and demand should be equal in all four examples, however, figure 1.2 is not an equal-area projection.

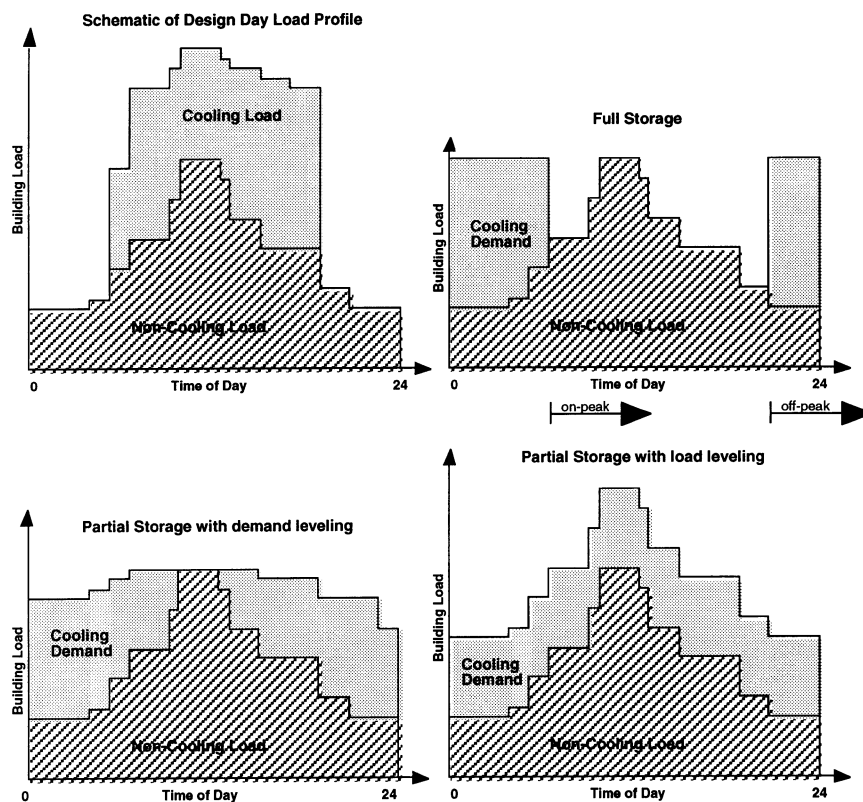


Figure 1.2: Schematic (fictitious) building load for a design day. This Figure illustrates the concepts of full storage and partial storage with demand leveling and load leveling (EPRI, 1985).

Chapter 2

Thermophysical Properties of a Binary Water-Ethanol Mixture

In order to use water-ethanol as a storage medium, its thermophysical and transport properties need to be known. Especially density, specific heat, (dynamic) viscosity and freezing point are of great interest. Other mixture properties that need to be understood include:

- what happens to the density characteristic if pure water has ethanol added
- if both fluids are miscible under the intended operation conditions
- how the freezing point changes with concentration

The intended application determined the range for the properties: the following chapter looks at a temperature range of 253 K to 353 K ($-20 \dots 80^\circ\text{C}$), a pressure of 1 *atm* and all possible concentrations of ethanol in water ($x_{\text{ethanol}} = 0 \dots 1$).

The symbols used in this chapter can be found on page 52 at the end of this chapter.

2.1 Equations to Determine Mixture Properties

For an arbitrary extensive quantity Y , which could be any one out of the set (U, H, F, G, S, V, C_p) ¹, the following mixing rule is valid:

$$Y^M = Y - \sum_{i=1}^{comp} N_i y_i^0, \quad (2.1)$$

Y^M is the (absolute) function of mixing and represents the change in property Y if a solution is formed out of pure components. y_i^0 is the molar value of Y for the pure component i . The partial molar function of mixing, y_i^M , is the derivative of Y^M with respect to the number of moles N_i of a component i in the solution:

$$y_i^M = \left(\frac{\partial Y^M}{\partial N_i} \right)_{T,P,n_{j \neq i}} \quad (2.2)$$

and reflects the influence of component i on the mixing function. With equation 2.1, it follows

$$y_i^M = \left(\frac{\partial Y}{\partial N_i} \right)_{T,P,n_{j \neq i}} - y_i^0 = y_i - y_i^0 \quad (2.3)$$

The partial molar mixing functions add up to the absolute mixing function:

$$Y^M = \sum_{i=1}^{comp} N_i y_i^M \quad (2.4)$$

and dividing by the total number of moles in the solution, N , leads to

$$\frac{Y^M}{N} = y^M = \sum_{i=1}^{comp} x_i y_i^M \quad (2.5)$$

¹(Belousov and Panov, 1994), page 133-135

The mixing functions are connected to ideal solution properties by

$$Y^M = Y_{id}^M + Y^E \quad (2.6)$$

For an ideal solution, most mixing functions are equal to zero:

$$U_{id}^M = H_{id}^M = H_{id}^M = V_{id}^M = C_{p,id}^M = 0 \quad (2.7)$$

but G_{id}^M , F_{id}^M and S_{id}^M are not, because $\mu_{id}^M - \mu_i^0 = RT \ln x_i$ causes

$$G_{id}^M = RT \sum_i x_i \ln x_i \quad (2.8)$$

$$\text{and } S_{id}^M = - \left(\frac{\partial G_{id}^M}{\partial T} \right)_{P,N} = -T \sum_i x_i \ln x_i \quad (2.9)$$

Equations 2.7, 2.8 and 2.9 are based on the assumption that the excess functions Y^E are defined relative to a value of Y_{id}^M in the sense of Raoult's Law (Prausnitz et al., 1986)². All empirical equations that were used in this paper define excess functions in the same sense. Summarizing, a mixture property Y can be determined from

$$Y = \sum_{i=1}^{comp} N_i y_i^0 + Y_{id}^M + Y^E \quad (2.10)$$

where the first term refers to pure component properties, the second to ideal solution properties, and the third to an excess function.

²page 247

2.2 Heat Capacity

2.2.1 Definition

The heat capacity at constant pressure is defined as

$$c_p = T \left(\frac{\partial s}{\partial T} \right)_{P,N} \quad (2.11)$$

The entropy can be expressed with the Gibbs energy using the fundamental relation for the Gibbs energy

$$dg = -s dT + v dP + \sum_{i=1}^{comp.} \mu_i dn_i, \quad (2.12)$$

which is written for a two component (binary) system as

$$dg = -s dT + v dP + \mu_1 dn_1 + \mu_2 dn_2 \quad (2.13)$$

It is obvious that for constant P and constant composition the last three terms vanish, and rearrangement leads to

$$-s = \frac{dg}{dT}. \quad (2.14)$$

Inserting of equation 2.14 into equation 2.11 leads to the working equation that was used to determine c_p :

$$c_p = -T \left(\frac{\partial^2 g}{\partial T^2} \right)_{P,N} \quad (2.15)$$

Rearranging equations 2.1 and 2.6, the molar Gibbs energy for a mixture is expressed as

$$g = x_1 g_{1,pure} + x_2 g_{2,pure} + g^E + RT (x_1 \ln x_1 + x_2 \ln x_2). \quad (2.16)$$

The second partial derivative of g with respect to T is needed in equation 2.15, and it can be expressed as the sum of the partial derivatives of the terms g consists of. Note that the term $RT(\dots)$ vanishes during derivation.

$$\left(\frac{\partial^2 g}{\partial T^2}\right)_{P,N} = x_1 \left(\frac{\partial^2 g_{1,pure}}{\partial T^2}\right)_{P,N} + x_2 \left(\frac{\partial^2 g_{2,pure}}{\partial T^2}\right)_{P,N} + \left(\frac{\partial^2 g^E}{\partial T^2}\right)_{P,N} \quad (2.17)$$

Inserting equation 2.16 into equation 2.15, it turns out that c_p can be expressed using the pure component values of c_p and adding a correction that encounters for the molar excess gibbs energy. The correction term can therefore be understood as the molar excess heat capacity.

$$c_p = -x_1 T \left(\frac{\partial^2 g_{1,pure}}{\partial T^2}\right)_{P,N} - x_2 T \left(\frac{\partial^2 g_{2,pure}}{\partial T^2}\right)_{P,N} - T \left(\frac{\partial^2 g^E}{\partial T^2}\right)_{P,N} \quad (2.18)$$

$$= x_1 c_{p,1,pure} + x_2 c_{p,2,pure} + c_p^E \quad (2.19)$$

with

$$c_p^E = -T \left(\frac{\partial^2 g^E}{\partial T^2}\right)_{P,N} \quad (2.20)$$

2.2.2 Reference Data

Several literature sources were searched for measured data of $c_p = f(T, x)$, but only one source provided this data³. The measurements documented by Synowietz⁴ were taken in the years from 1880 to 1932, and tables are given that rely on journal articles from 1907 and 1935. Belousov and Panov⁵ gave a summary on systems for which the heat capacities of binary solutions have been experimentally determined, and besides Synowietz they

³Another set of 9 data points for density and 8 for specific heat was found in the Handbook of Tables for Applied Engineering Science (CRC, 1972), page 96, several months after this part of the project had been finished, and it could therefore not be incorporated

⁴(Lacmann and Synowietz, 1977), page 309-311

⁵(Belousov and Panov, 1994), page 142-143

mention four newer sources. Two of these turned out to contain no data on water-ethanol mixtures, and the other two were not available for checking the calculated values.

Synowietz lists 48 measurement results, and the pairs of (T, x) that are included can be seen in figure 2.1:

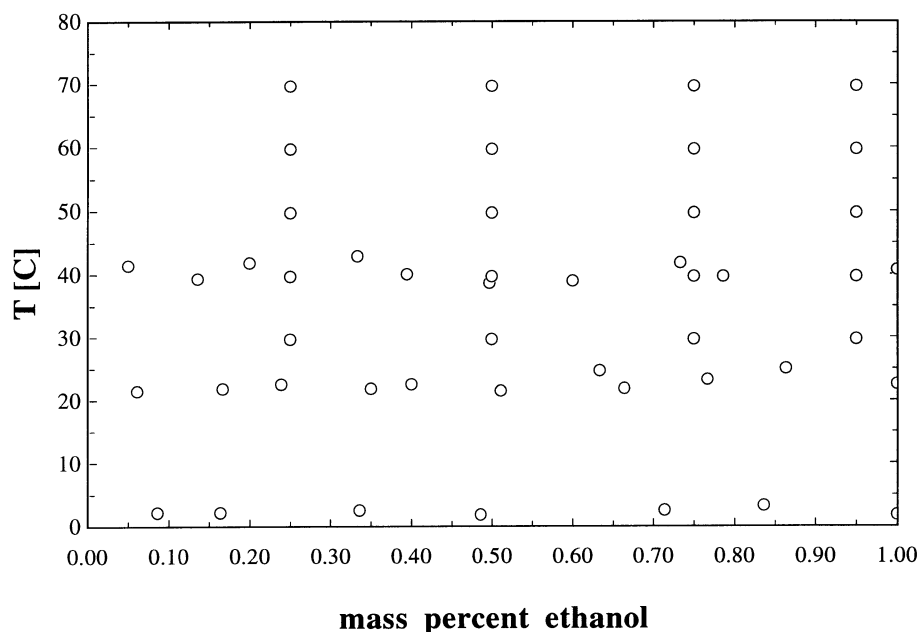


Figure 2.1: Distribution of measurements, taken from Synowietz

2.2.3 Pure Component Heat Capacity Values

Values for the heat capacity of pure water were taken from EES (Klein and Alvarado, 1996). EES provides data for a fluid called *Steam NBS*. Steam NBS gives data for water as a real fluid, based on data from the National Institute of Standards and Technology (NIST, formerly the National Bureau of Standards, NBS).

Values for the heat capacity of pure ethanol were taken from Yaws (Yaws, 1992), who gives c_p as a function of temperature at fixed pressure of $P = 1 \text{ bar}$. The function is shown

	water	ethanol
C_1	-308295.6	-294513
C_2	434.7139	612.5647
C_3	-75.35	-139.751
C_4	0	$302.9907 * 10^{-3}$
C_6	0	$-286.814 * 10^{-6}$

Table 2.1: Coefficients of the Gibbs function used by the TAPP database

below.

$$c_{p,ethanol} = 100.92 - 111.839 * 10^{-3} T/K + 498.54 * 10^{-6} (T/K)^2 \frac{kJ}{kmolK} \quad (2.21)$$

As another source of pure component values the *Thermochemical And Physical Properties* database (TAPP, 1995) was considered. Here, the pure component molar Gibbs Energy is modeled as a temperature dependent function at constant pressure of $P = 1 \text{ bar}$:

$$g_{i,pure} = h_{ref} + C_{1,i} + C_{2,i} T + C_{3,i} T \ln T + C_{4,i} T^2 + C_{6,i} T^3 \quad (2.22)$$

h_{ref} is the enthalpy of the substance at $T = 298 \text{ K}$ and $P = 1 \text{ bar}$. The coefficients are⁶ listed in table 2.1. The heat capacity is given as

$$c_p = -C_{3,i} - 2 C_{4,i} T - 6 C_{6,i} T^2, \quad (2.23)$$

which is consistent with equation 2.22, but with the coefficients C_4 and C_6 being zero for water, the heat capacity of water is independent of temperature, which does not represent the real behavior. Therefore the TAPP functions were not chosen.

⁶No uncertainty values were provided by the authors

2.2.4 Excess Heat Capacity Values

The molar excess Gibbs energy was calculated with the UNIQUAC equation. The UNIQUAC equation approaches g^E using a combinatorial and a residual term, and its parameters r , q and q^* have been empirically adjusted to obtain better agreement especially for water-alcohol mixtures. The NRTL equation is of simpler structure, and according to Prausnitz⁷ it is more suitable for determination of h^E than for determination of g^E .

Another applicable approximation to g^E is the Wilson equation. The Wilson equation was not chosen because it demands knowledge of the molar volume of both components as a function of temperature (and pressure, if the Wilson coefficients are available for pressures different from 1 bar, which is the most commonly used value). Water shows the behavior of having its maximum density at $T \approx 4\text{ }^\circ\text{C}$, this behavior is different from any other liquid. The available functions of $\rho_{\text{water}}(T)$ did not represent this anomaly but modeled the density with a monotonic function. To predict the density of the mixture with the Wilson equation over a temperature range where pure water would freeze (and therefore no measured density values are available) plus taking into account the singular behavior of the density of water in that range, an extrapolation of this density function would be very uncertain.

The UNIQUAC approach of the molar excess Gibbs function consists of the following set of equations⁸:

$$g^E = g_{\text{combinatorial}}^E + g_{\text{residual}}^E = g_{\text{comb}}^E + g_{\text{res}}^E \quad (2.24)$$

For a binary mixture,

$$\frac{g_{\text{comb}}^E}{RT} = x_1 \ln \frac{\Phi_1^*}{x_1} + x_2 \ln \frac{\Phi_2^*}{x_2} + \frac{z}{2} \left(q_1 x_1 \ln \frac{\Theta_1}{\Phi_1^*} + q_2 x_2 \ln \frac{\Theta_2}{\Phi_2^*} \right) \quad (2.25)$$

⁷(Prausnitz et al., 1986), page 238

⁸(Prausnitz et al., 1986), page 238-244

	r	q	q^*
water	0.92	1.40	1.00
ethanol	2.11	1.97	0.92

Table 2.2: Coefficients for the UNIQUAC approach of the molar Gibbs function

$$\frac{g_{res}^E}{RT} = -q_1^* x_1 \ln(\Theta_1^* + \Theta_2^* \tau_{21}) - q_2^* x_2 \ln(\Theta_2^* + \Theta_1^* \tau_{12}) \quad (2.26)$$

The coordination number z is set to 10 and the segment fraction Φ^* and the area fractions Θ and Θ^* are given by

$$\Phi_i^* = \frac{x_i r_i}{x_1 r_1 + x_2 r_2} \quad (2.27)$$

$$\Theta_i = \frac{x_i q_i}{x_1 q_1 + x_2 q_2} \quad (2.28)$$

$$\Theta_i^* = \frac{x_i q_i^*}{x_1 q_1^* + x_2 q_2^*} \quad (2.29)$$

with $i = 1$ for water and $i = 2$ for ethanol. Values for r , q , and q^* were taken from Prausnitz⁹: The adjustable parameters τ_{12} and τ_{21} are given as

$$\tau_{12} = \exp\left(-\frac{A_{12}}{R_{cal}T}\right) \quad (2.30)$$

$$\tau_{21} = \exp\left(-\frac{A_{21}}{R_{cal}T}\right), \quad (2.31)$$

and values for A_{12} and A_{21} were obtained from the VLE Data Collection¹⁰:

$$A_{12} = -31.629 + 0.4759 T = a_{0,12} + a_{1,12} T \quad (2.32)$$

$$A_{21} = -96.473 + 0.6843 T = a_{0,21} + a_{1,21} T \quad (2.33)$$

⁹(Prausnitz et al., 1986), page 240

¹⁰(Gmehling, 1988), page 114

These coefficients are recommended values based on consistent measurements. They are calculated for $R = 1.98721 \frac{\text{cal}}{\text{mol K}}$. The symbol R_{cal} was introduced to indicate this wherever R has to be inserted in these certain units, and inserting R in different units will lead to other results. A former issue of the same series of 1981¹¹ gave different recommended values for the coefficients:

$$A_{12} = -34.884 + 0.5558 T \quad (2.34)$$

$$A_{21} = -43.693 + 0.4523 T \quad (2.35)$$

The newer data were chosen for the calculations in this paper, because it was assumed that the authors obtained better fits with the new values, otherwise they would not have changed them.

For the second partial derivative of g^E with respect to T , the first and second derivative of τ_{12} and τ_{21} will be used:

$$\tau = \exp\left(-\frac{a_0 + a_1 T}{R_{cal} T}\right) = \exp\left(-\frac{a_0}{R_{cal} T} - \frac{a_1}{R_{cal}}\right) \quad (2.36)$$

$$\frac{d\tau}{dT} = \exp\left(-\frac{a_0}{R_{cal} T} - \frac{a_1}{R_{cal}}\right) \left(-\frac{a_0}{R_{cal} T^2}\right) \quad (2.37)$$

$$= \tau \left(-\frac{a_0}{R_{cal} T^2}\right) \quad (2.38)$$

$$\frac{d^2\tau}{dT^2} = \exp\left(-\frac{a_0 + a_1 T}{R_{cal} T}\right) \left(\frac{a_0^2}{R_{cal}^2 T^4} - 2\frac{a_0}{R_{cal} T^3}\right) \quad (2.39)$$

$$= \tau \left(\frac{a_0^2}{R_{cal}^2 T^4} - 2\frac{a_0}{R_{cal} T^3}\right), \quad (2.40)$$

valid for both τ_{12} and τ_{21} (indices were omitted due to readability). Φ , Θ , and Θ^* are constant because r , q , and q^* are constant.

The first and second derivative of the combinatorial part of g^E are zero, because g_{comb}^E

¹¹(Gmehling et al., 1981), page 157

does not depend on T:

$$\left(\frac{\partial g_{comb}^E}{\partial T} \right)_{P, N_i} = \left(\frac{\partial^2 g_{comb}^E}{\partial T^2} \right)_{P, N_i} = 0 \quad (2.41)$$

The residual part of g^E consists of the factor RT and another factor whose two parts depend on τ_{12} and τ_{21} . τ_{12} and τ_{21} are functions of temperature only, and their derivatives have already been determined above. Therefore it is useful to express g_{res}^E as

$$g_{res}^E = RT * Z(\tau_{12}, \tau_{21}) \quad (2.42)$$

with

$$\begin{aligned} Z(\tau_{12}, \tau_{21}) &= -q_1^* x_1 \ln(\Theta_1^* + \Theta_2^* \tau_{21}) - q_2^* x_2 \ln(\Theta_2^* + \Theta_1^* \tau_{12}) \\ &= Z_{21} + Z_{12} \end{aligned}$$

from equation 2.26. The function Z has no other use but to simplify the presentation of the derivatives. As the two terms in Z depend either on τ_{12} or τ_{21} but not on both at the same time, Z can be split up into two terms Z_{12} and Z_{21} , both of which depend only on one variable. This simplifies the calculation a lot. Now the first and second derivative of g_{res}^E can be expressed as

$$\frac{dg_{res}^E}{dT} = R Z + RT \frac{dZ}{dT} \quad (2.43)$$

$$= R Z + RT \left[\frac{dZ_{21}}{dT} + \frac{dZ_{12}}{dT} \right] \quad (2.44)$$

$$= R Z + RT \left[\left(\frac{\partial Z_{21}}{\partial \tau_{21}} \right) \left(\frac{\partial \tau_{21}}{\partial T} \right) + \left(\frac{\partial Z_{12}}{\partial \tau_{12}} \right) \left(\frac{\partial \tau_{12}}{\partial T} \right) \right] \quad (2.45)$$

$$= R Z + RT \left[\frac{dZ_{21}}{d\tau_{21}} \frac{d\tau_{21}}{dT} + \frac{dZ_{12}}{d\tau_{12}} \frac{d\tau_{12}}{dT} \right] \quad (2.46)$$

and

$$\begin{aligned}\frac{d^2 g_{res}^E}{dT^2} &= R \frac{dZ}{dT} + R \frac{dZ}{dT} + RT \frac{d^2 Z}{dT^2} \\ &= 2R \left[\frac{dZ_{21}}{d\tau_{21}} \frac{d\tau_{21}}{dT} + \frac{dZ_{12}}{d\tau_{12}} \frac{d\tau_{12}}{dT} \right] + RT \left[\frac{d^2 Z_{21}}{d\tau_{21}^2} \frac{d^2 \tau_{21}}{dT^2} + \frac{d^2 Z_{12}}{d\tau_{12}^2} \frac{d^2 \tau_{12}}{dT^2} \right]\end{aligned}$$

The derivatives of τ_{12} and τ_{21} have already been determined, and the derivatives of $Z = Z_{21} + Z_{12}$ with respect to τ_{21} and τ_{12} , respectively, are

$$\frac{Z_{21}}{d\tau_{21}} = \frac{-q_1^* x_1 \Theta_2^*}{\Theta_1^* + \Theta_2^* \tau_{21}} \quad (2.47)$$

$$\frac{dZ_{12}}{d\tau_{12}} = \frac{-q_2^* x_2 \Theta_1^*}{\Theta_2^* + \Theta_1^* \tau_{12}} \quad (2.48)$$

$$\frac{d^2 Z_{21}}{d\tau_{21}^2} = \frac{q_1^* x_1 \Theta_2^{*2}}{(\Theta_1^* + \Theta_2^* \tau_{21})^2} \quad (2.49)$$

$$\frac{d^2 Z_{12}}{d\tau_{12}^2} = \frac{q_2^* x_2 \Theta_1^{*2}}{(\Theta_2^* + \Theta_1^* \tau_{12})^2} \quad (2.50)$$

The second partial derivative of g^E (equation 2.17) can now be combined from the previously calculated subderivatives:

$$\begin{aligned}c_p^E &= -2RT \left(\frac{dZ_{21}}{d\tau_{21}} \frac{d\tau_{21}}{dT} + \frac{dZ_{12}}{d\tau_{12}} \frac{d\tau_{12}}{dT} \right) \\ &\quad - RT^2 \left(\frac{d^2 Z_{21}}{d\tau_{21}^2} \frac{d^2 \tau_{21}}{dT^2} + \frac{d^2 Z_{12}}{d\tau_{12}^2} \frac{d^2 \tau_{12}}{dT^2} \right),\end{aligned} \quad (2.51)$$

and by inserting into equation 2.19, the mixtures's heat capacity c_p can be approximated.

2.2.5 Results

According to the calculation, the excess heat capacity is only a small percentage of the absolute value, see figure 2.2. The resulting values for c_p are shown in figure 2.3.

The tabulated data from Synowietz shows only rough agreement with the function used to

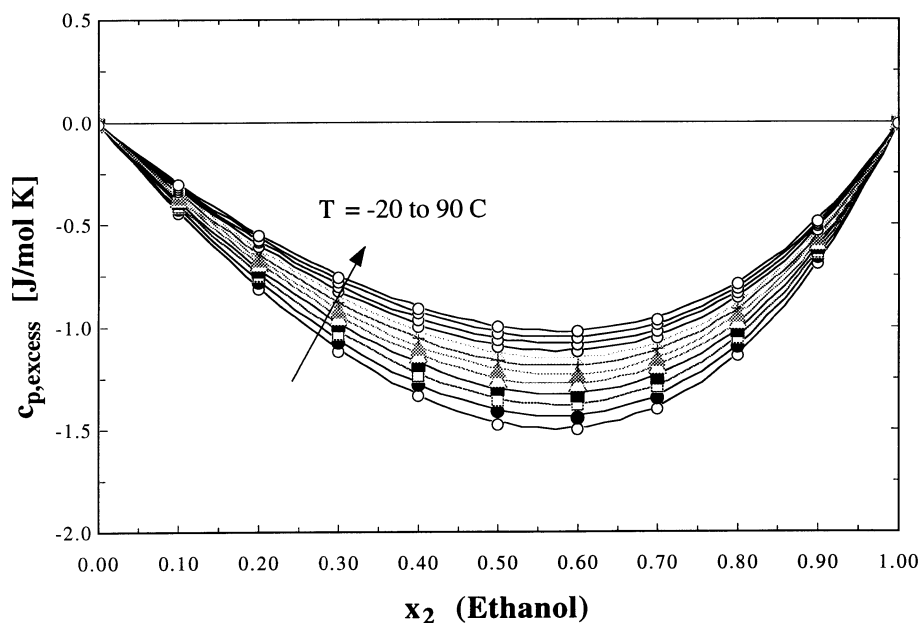


Figure 2.2: Calculated Excess Heat Capacity for -20 to 90 C as a function of concentration

determine the heat capacity of pure ethanol (equation 2.21). The measurements that have been performed for 95 mass-% of ethanol lead to far higher values than those calculated with equation 2.21. Curve fitting this data in order to extrapolate to lower temperatures can not be recommended, because there are only a few measurements in the vicinity of 0 C. However, the measurements predict a maximum heat capacity in the vicinity of $\xi = 20\%$, which is not predicted by the approximation with the UNIQUAC functions, simply because the applied curve fit equations cannot capture this behavior. Using higher order equations for the curve fit, in the other hand, would have made extrapolations even less reliable.

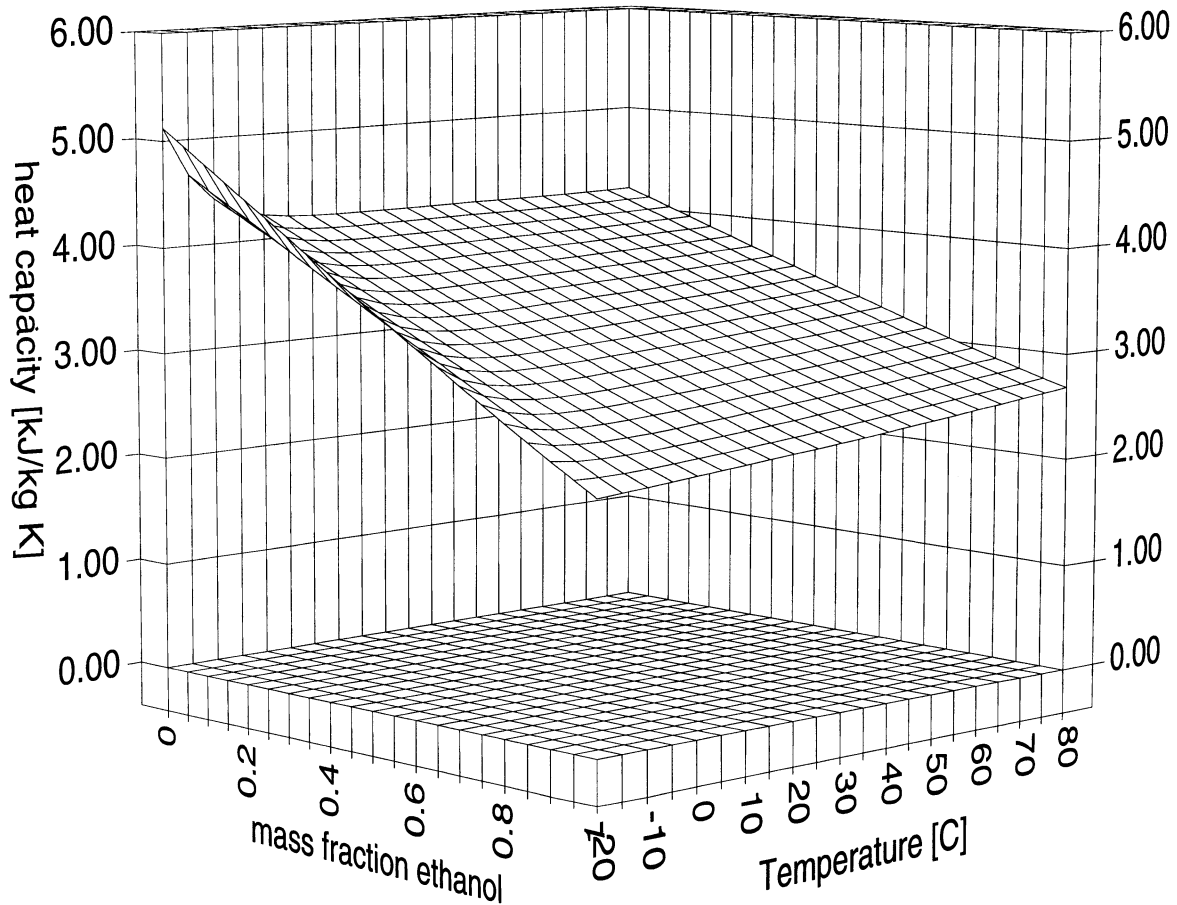


Figure 2.3: c_p as function of T and ξ

2.3 Density

2.3.1 Method of Approximation

Calculation of Excess Properties

The thermodynamically “correct” way of determining the density of a liquid mixture corresponds to the proceeding for determining c_p . It was shown in section 2.1 that the volume of a binary mixture can be calculated from the weighted pure component values and a quantity that accounts for mixing effects. In the case of volume, this mixing quantity

is the excess volume (see equation 2.7), and thus the equation for the molar volume of a mixture reads

$$v = x_1 v_{1,pure} + x_2 v_{2,pure} + v^E \quad (2.52)$$

The density then is simply the reciprocal of the volume:

$$\rho = \frac{1}{v} \quad (2.53)$$

The density could easily be calculated from volumetric data if this data was available. Recalling equation 2.13 for the gibbs function for a two component system,

$$dg = -sdT + vdP + \mu_1 dn_1 + \mu_2 dn_2, \quad (2.54)$$

the volume can be expressed as

$$v = \left(\frac{\partial g}{\partial P} \right)_{T,N} \quad (2.55)$$

and the same relationship holds for the excess volume ¹²:

$$v^E = \left(\frac{\partial g^E}{\partial P} \right)_{T,N} \quad (2.56)$$

Curve-Fitting Measured Data

At the time of this examination, neither data on the molar excess volume nor sufficient data on the pressure dependence of the molar excess gibbs energy or its UNIQUAC coefficients were available. Thus, a different way of estimating v^E has been selected. Measured data of the density of water-ethanol have been taken from Synowietz¹³. The data has been checked on consistency with the pure component values for $x_e \rightarrow 0$ and

¹²(Prausnitz et al., 1986), page 196

¹³(Lacmann and Synowietz, 1977), page 129

$x_e \rightarrow 1$. The results will be shown below. A suitable mixing rule was chosen for this application. Several Mixing rules were examined (Nielsen, 1978). With the selected mixing rule, the apparent specific (mass basis) excess volume was calculated and curve fitted in order to allow extrapolation to lower temperatures that were not within the range of the measurements.

2.3.2 Density of the Pure Components

Values for the density of pure water were taken from EES. Again, the EES fluid *Steam NBS* was used. The corresponding density function shows the real behavior of water of having a maximum density at $T \approx 4\text{ }^\circ\text{C}$. It is absolutely necessary to include this phenomenon of water if reliable predictions on the mixture behavior in this temperature area shall be made. Doublechecking with data from the NBS Steam Tables¹⁴ shows very good agreement, and the EES function allows extrapolation to temperatures below the freezing point of water, because the function is (mathematically) steady in the temperature range of interest.

For the density of pure ethanol a modified Rackett-Equation was used (Yaws, 1992):

$$\rho_{ethanol} = A * B^{-(1-\frac{T}{T_c})^{(2/7)}} \quad (2.57)$$

with the coefficients

$$\begin{aligned} A &= 0.2567 \frac{g}{ml} = 0.2567 \frac{g}{cm^3} \\ B &= 0.240 \\ T_c &= 515.8\text{ }K \end{aligned}$$

¹⁴(NBS, 1984), page 9

2.3.3 Measured Data

The data from Synowietz, page 129, are a compilation of measurements from the years 1905-1969, taken from 25 different sources. The densities are given as tabular values for concentrations of 2, 5, 10, \dots , 95, 100 mass-% and for $T = 0, 20, \dots, 100^\circ\text{C}$ and for a constant pressure of $1\text{ atm} = 1.01325\text{ bar}$. The data for $\xi = 1$, ξ being the mass fraction of ethanol, matched very well with the prediction of equation 2.57. The table values for $\xi = 0.02$ also approached the NBS values implemented into EES. The comparison is shown in figure 2.4. The plot for the density of water does not show a maximum density as the plot was created in steps of 20 K. However, due to the 20 K steps in the measured data, no maximum density will ever show up in the measured data.

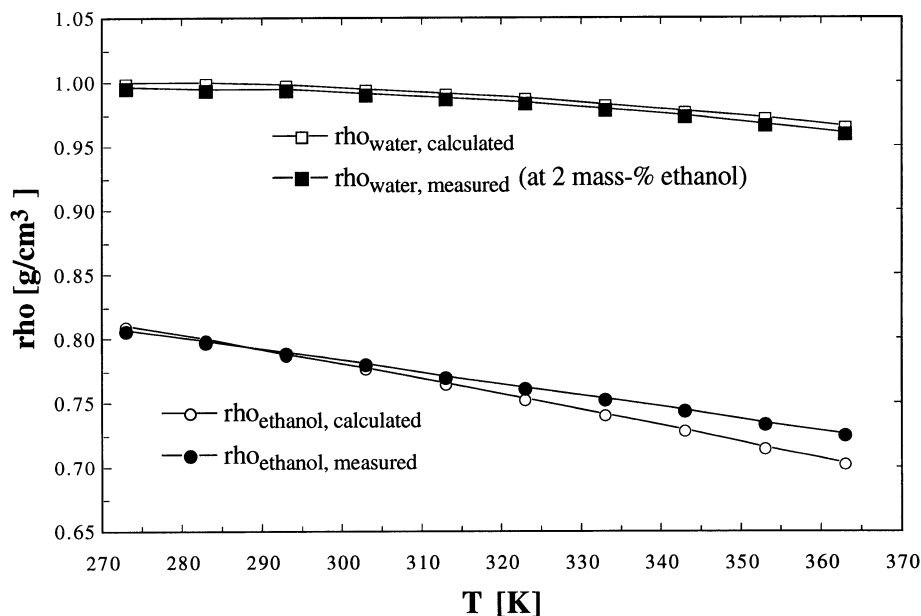


Figure 2.4: Consistency of measured density data for ethanol and water with functions used for pure component values

2.3.4 Mixing Rule

The simplest possible mixing rule is to add weighted pure component values in order to get the value for mixture. I chose to work with the reciprocal of density, the specific volume. As the densities are given in mass per volume, it seems useful to choose the mass fraction ξ as weighting parameter.

$$\frac{1}{\rho} = \sum_{i=1}^{comp} \frac{\xi_i}{\rho_i} \quad (2.58)$$

Mass fraction ξ and mole fraction x are related in the following way:

$$\xi_i = \frac{MW_i}{MW} x_i \quad (2.59)$$

with the MW_i molar weight of component i and MW as the molar weight of the mixture, which can be calculated from

$$MW = \sum_{i=1}^{comp.} MW_i \xi_i \quad (2.60)$$

The mean deviation d_{mean} for all n available data points (except for $T = 100\text{ C}$, because of the phase change of pure water that will influence the density values of pure water) is $d_{mean} = 2.357\%$. The mean deviation is calculated

$$d_{mean} = \sum_{i=1}^n \frac{\rho - \rho_{meas}}{\rho_{meas}} \quad (2.61)$$

Better agreement with the measurements showed the following rule, which seems to be identical with equation 2.52, but in fact it is not the molar volume but the specific volume (on a mass basis) that is added (indicated by a prime to distinguish it from molar volume):

$$v' = \sum_{i=1}^{comp.} \xi_i v'_i + v'^E \quad (2.62)$$

$$\frac{1}{\rho} = \sum_{i=1}^{comp.} \frac{\xi_i}{\rho_i} + v'^E \quad (2.63)$$

This mixing rule was chosen so that the specific excess volume could be determined by subtracting the calculated value for a specific value (T, ξ_2) (index 2 for ethanol) from the measured value.

The available measured data was used to set up a function

$$v'^E = f(T, \xi_2) \quad (2.64)$$

To obey the boundary conditions of $v'^E = 0$ for both $\xi_2 = 0$ and $\xi_2 = 1$, the following structure for v'^E was chosen:

$$v'^E = \xi_2(1 - \xi_2)(a_0 + a_1\xi_2 + a_2\xi_2^2 + a_3\xi_2^3) \quad (2.65)$$

The first two factors guarantee that the boundary conditions will be fulfilled, and the third factor was chosen to be of low order to allow later extrapolation. Unfortunately, the available version of EES was not capable of fitting the measured data directly into the structure of equation 2.65, therefore two steps had to be taken. First, the coefficients a_0 to a_3 were determined for all available temperatures by setting ρ from equation 2.63 equal to the measured values for all concentrations $\xi_2 = 0 \dots 1$. Then, these coefficients were individually curve-fit with the temperature as independent variable. Here, second order polynomials were considered sufficient because of good agreement of curve fit and measurements.

The resulting functions for the coefficients a_0 to a_3 are

$$a_0 = 1.411 - 9.296 * 10^{-3} T + 1.453 * 10^{-5} T^2 \quad (2.66)$$

$$a_1 = -13.98 + 9.108 * 10^{-2} T - 1.547 * 10^{-4} T^2 \quad (2.67)$$

$$a_2 = 23.36 - 1.613 * 10^{-1} T + 2.919 * 10^{-4} T^2 \quad (2.68)$$

$$a_3 = -13.45 - 0.1015 T - 1.987 * 10^{-4} T^2 \quad (2.69)$$

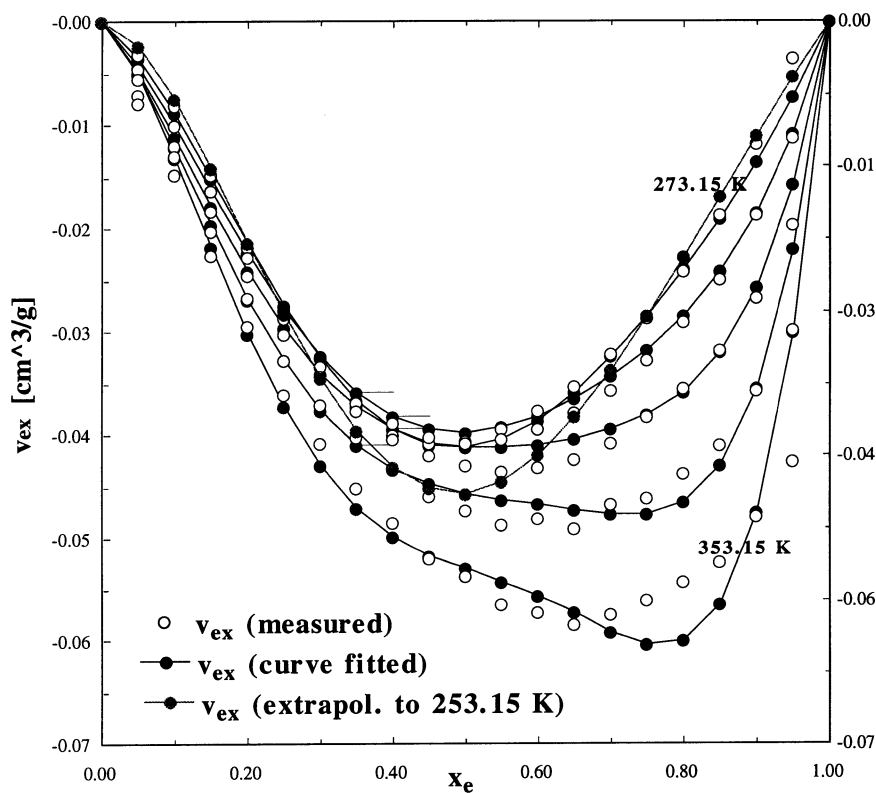


Figure 2.5: Excess Volume: measured and curve-fitted (273.15 K to 353.15 K) and extrapolated at 253.15 K

Figure 2.5 demonstrates the measured as well as the excess volume curve-fitted into equation 2.65. The excess volume has a comparatively small order of magnitude, but it is needed to determine if and where a point of maximum density occurs. The curve-fit shows its greatest deviation from the measured data at a temperature of 313.15 K or 80°C at $x_e \approx 0.80$. This difference is of low importance since the purpose of the fit is to predict the behavior of water-ethanol at low temperatures, where good agreement with

the measured data is obvious.

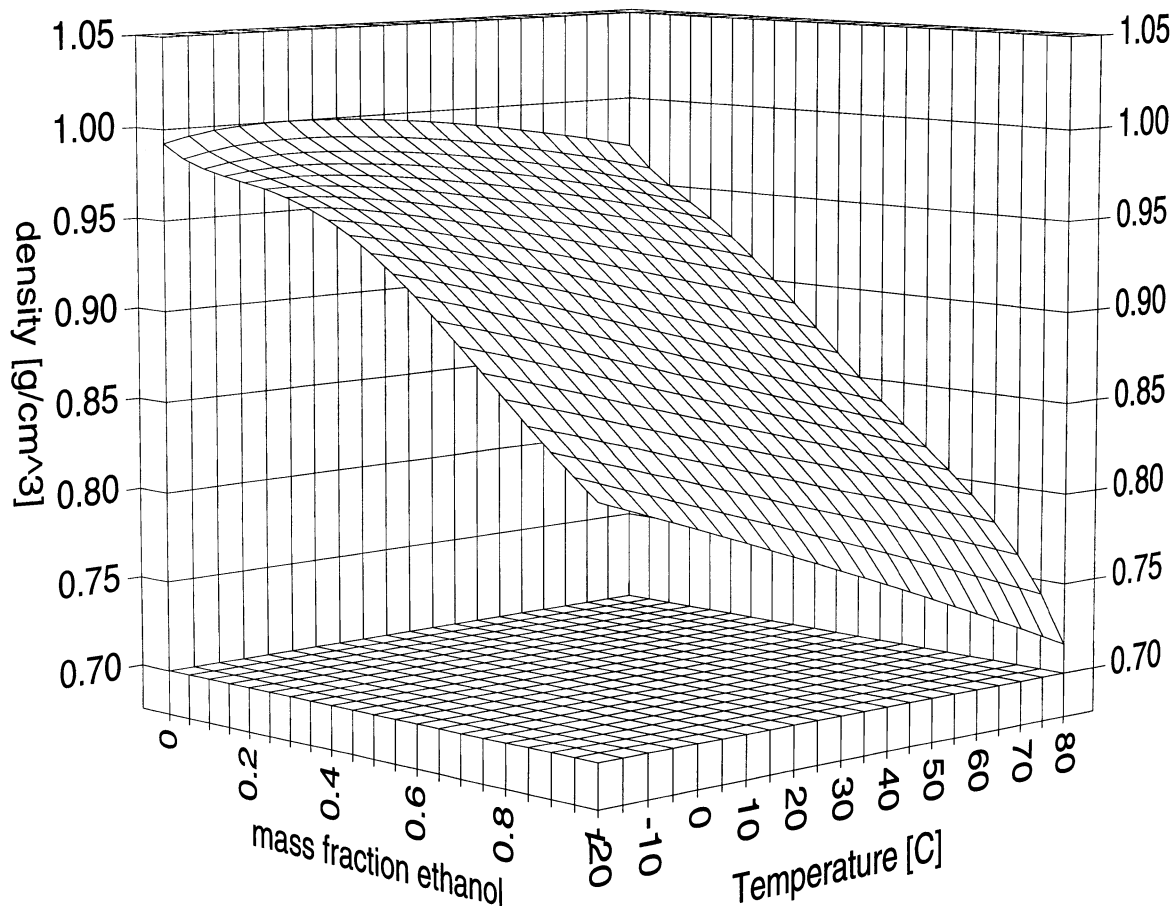


Figure 2.6: Density of water-ethanol. The temperature range is $T = -20 \dots 80^\circ\text{C}$, and the concentration range is $0 \dots 1$.

The result of the extrapolation is best visualized in a 3-dimensional chart showing the calculated density over the entire range of temperatures and concentrations (figure 2.6). For $x_2 = 0$ (pure water), the point of maximum density is visible, and for $x_2 = 1$ (pure ethanol) the density changes linearly with temperature, as was already shown in figure 2.4. The maximum gets flatter with increasing ethanol concentration. At $T = -20^\circ\text{C}$, the influence of the negative excess volume can be seen well. If water-ethanol were an ideal mixture, the excess volume would be zero and the mixture density was on the direct

connection between the pure component values. In this prediction a negative excess volume causes an increase in density, which is higher for lower temperatures.

2.4 Dynamic Viscosity

2.4.1 Method of Approximation

Unlike c_p and ρ , the dynamic viscosity η (a transport property) is not easily related to fundamental thermodynamic properties. Instead, it will be approached with a mixing rule that combines pure component values, mole fractions and empirical constants to the desired quantity.

For liquid water-alcohol mixtures, a well-fitting mixing rule is described by Stephan¹⁵:

$$\ln \eta = x_1 \ln \eta_1 + x_2 \ln \eta_2 + \ln \left(1 + \frac{x_1 x_2}{A + B x_2^2} \right) \quad (2.70)$$

The coefficients A and B are temperature dependent, and since they are given¹⁶ for a range of $275 \text{ K} \leq T \leq 350 \text{ K}$, they have to be extrapolated to the low temperatures in question. The maximum error is estimated by the authors to be 3.25% for the given temperature range. (If this mixing rule is applied with fixed, temperature independent values for A and B , the maximum error is stated to be 84.58%.) The values for A and B are given in table 2.3.

A curve fit for this data was calculated with EES:

$$\begin{aligned} A &= 0.12256 - 2.2608 * 10^{-3} T + 7.0860 * 10^{-6} T^2 \\ B &= 4.1227 - 3.1217 * 10^{-2} T + 6.5974 * 10^{-5} T^2 \end{aligned} \quad (2.71)$$

¹⁵(Stephan and Heckenberger, 1988), page LIII

¹⁶page 314

$T [K]$	A	B
275	0.037669325	0.520552418
280	0.045289338	0.552897986
285	0.053509488	0.586615935
290	0.062299237	0.622009077
295	0.071626618	0.659377806
300	0.081458183	0.699020617
305	0.091757421	0.741233021
310	0.102486975	0.786311948
315	0.113605678	0.834552244
320	0.125069589	0.886248632
325	0.136831606	0.941693516
330	0.148841222	1.001176722
335	0.161044503	1.064981636
340	0.173384509	1.133381208
345	0.185800630	1.206636092
350	0.198230661	1.284986869

Table 2.3: Coefficients for the mixing rule for viscosity

and the result is shown in figure 2.7, including values for extrapolation of A and B . Accuracy data for the curve fit is given in the following table (EES results): R^2 is the

	A	B
R^2	99.99%	99.98%
RMS	$5.9868 * 10^{-4}$	$3.9032 * 10^{-3}$
Bias	$-6.9389 * 10^{-18}$	$4.5797 * 10^{-16}$

Table 2.4: Accuracy of curve-fit of viscosity mixing rule coefficients

ratio of the sum of squares due to regression to the sum of squares about the mean of the data expressed as a percentage. RMS is the root mean square error. The Bias is the average difference between the curve-fitted values and the observations.

With equations 2.70 and 2.71, the dynamic viscosity of water-ethanol can be estimated.

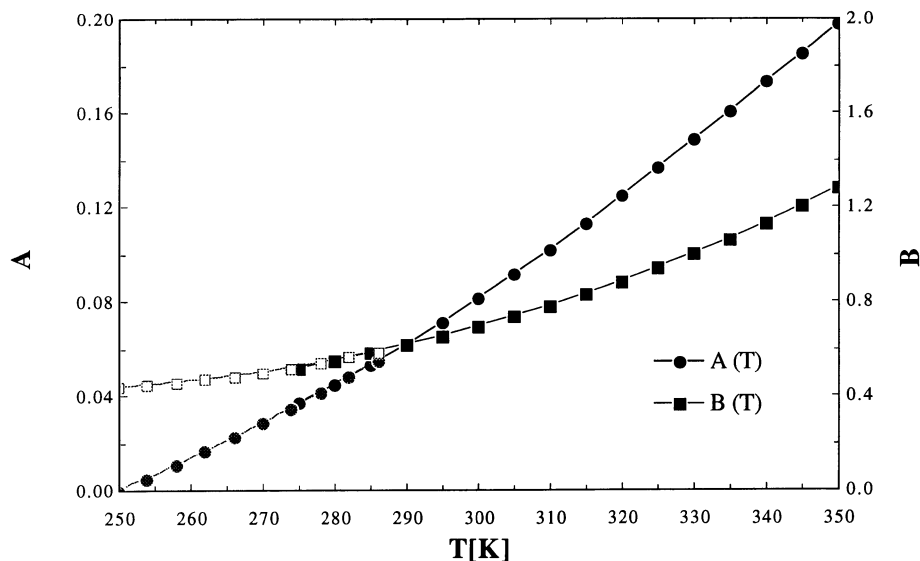


Figure 2.7: Curve fit of recommended coefficients A and B with extrapolation to lower temperatures

2.5 Freezing Point

2.5.1 Approximation for Dilute Solutions

For an ideal solution in the sense of Raoult's Law the freezing point depression is a function only of the number of particles present. The absolute value of the lowering of the freezing point ΔT_f can be expressed as¹⁷

$$\Delta T_f = E_f \bar{m} \quad (2.72)$$

Here, $\Delta T_f = T_f - T_{f,m}$, the difference between the freezing point of the pure solvent and the freezing point of the mixture¹⁸. \bar{m} is the molality (moles of solute per kg of solvent), and E_t is the Cryoscopic Constant, which is a property of the solvent only and does not

¹⁷(CRC, 9596), page 15-21

¹⁸Note that for a freezing point *depression*, ΔT_f will be positive

depend on the solute. The equation to determine the Cryoscopic Constant is

$$E_f = \frac{R T_f^2 MW_1}{\Delta h_{fus}} \quad (2.73)$$

where T_f is the freezing temperature, MW_1 the molar weight of component 1 (the solvent), and Δh_{fus} its molar enthalpy of fusion. The following equation calculates the Cryoscopic Constant for water:

$$E_f = \frac{8.31451 \frac{J}{molK} (273.15 K)^2 18.02 \frac{g}{mol}}{6000} = 1.86 \frac{kg K}{mol} \quad (2.74)$$

The molality \bar{m}_2 is related to the molar fraction of the solute by

$$\bar{m} = \frac{\text{moles of 2}}{\text{mass of 1}} = \frac{N_2}{m_1} = \frac{\frac{m_2}{MW_2}}{m_1} = \frac{x_2}{x_1 MW_2} = \frac{x_2}{(1 - x_2) MW_2} \quad (2.75)$$

The unit of the molality is mol/kg , and as the unit of the molar weight is either g/mol or $kg/kmol$, a unit conversion by multiplication with 1000 will be necessary. Combining equations 2.72, 2.73 and 2.75 lead to an equation for the freezing point depression:

$$\Delta T_f = \frac{R T_f^2 x_2}{(1 - x_2) \Delta h_{fus}} \frac{MW_1}{MW_2} \quad (2.76)$$

It is obvious that for $x_2 \rightarrow 1$, ΔT_f approaches infinity, and that therefore equation 2.72 can only serve as an estimate for low values of x_2 .

2.5.2 Approximation with van't Hoff's Law

At its freezing point, liquid water is in equilibrium with its solid phase, and the fugacities of the two phases have to be equal¹⁹:

$$f_{1,sol} = f_{1,liq} \quad (2.77)$$

A solute that is added to these two phases will almost always dissolve in the liquid phase, and thereby the fugacity of the solvent will be decreased. To reestablish equilibrium, the solid phase fugacity has to decrease, too, and the system will react by lowering the temperature of the two phases. In a quasistatic process, the system will always remain in equilibrium, the fugacities will be equal at any time, and therefore the change in fugacity has to be equal:

$$df_{1,sol} = df_{1,liq} \quad (2.78)$$

$$d \ln f_{1,sol} = d \ln f_{1,liq} \quad (2.79)$$

The natural log may be introduced as it is a monotonic function. At constant pressure, the fugacity of the solid solvent depends only on temperature, and the fugacity of the liquid mixture depends on temperature and concentration. The total derivative of equation 2.79 is written as

$$\left(\frac{\partial \ln f_{1,sol}}{\partial T} \right)_P dT = \left(\frac{\partial \ln f_{1,liq}}{\partial T} \right)_{P,x_2} dT + \left(\frac{\partial \ln f_{1,liq}}{\partial x_2} \right)_{P,T} dx_2 \quad (2.80)$$

¹⁹the following derivation was taken mainly from Klotz and Rosenberg, page 317-318

The first and second partial derivatives can be eliminated using a relation that follows from the Gibbs-Helmholtz equation²⁰:

$$\left(\frac{\partial \ln f_i}{\partial T}\right)_{P,x} = \frac{\bar{h}_i - h_i^*}{RT^2} \quad (2.81)$$

where \bar{h}_i is the partial molar enthalpy of component i and h_i^* is the molar enthalpy of i in the ideal gas state (very low pressure). Inserting into equation 2.80, the ideal gas enthalpy vanishes, and rearrangement leads to the relation

$$\frac{\bar{h}_{1,sol} - h_1^*}{RT^2} dT = \frac{\bar{h}_{1,liq} - h_1^*}{RT^2} dT + dx_2 \quad (2.82)$$

$$dT = -\frac{RT^2}{\bar{h}_{1,liq} - \bar{h}_{1,sol}} dx_2 \quad (2.83)$$

The solid phase is pure, so $\bar{h}_{1,sol}$ can be written $h_{1,sol}$. Now equation 2.83 is integrated from the state of pure solvent at its freezing point, $(T_f, x_2 = 0)$, to a state where some solute has been added and therefore the freezing point has been depressed in order to reestablish equilibrium, $(T_{f,m}, x_2 = x_{2,m})$:

$$\frac{\bar{h}_{1,liq} - h_{1,sol}}{R} \left(\frac{1}{T_f} - \frac{1}{T_{f,m}} \right) = -x_{2,m} \quad (2.84)$$

$$\frac{\bar{h}_{1,liq} - h_{1,sol}}{R} \left(\frac{T_{f,m} - T_f}{T_{f,m} T_f} \right) = -x_{2,m} \quad (2.85)$$

For this integration it is assumed that $\bar{h}_{1,liq} - h_{1,sol}$ remains constant over the small temperature range. Further simplifications can be made that lead to an expression for the freezing point depression ΔT_f which is similar but different from equation 2.76: First, one could assume that $T_{f,m} \simeq T_f$ and the denominator can therefore be approximated with T_f^2 . The numerator is $-\Delta T_f$ (compare with its definition below equation 2.72). Second,

²⁰compare Klotz and Rosenberg, page 239, and Prausnitz, page 195

one could assume that the partial molar enthalpy of pure component 1 (the solvent) approximates the molar enthalpy of pure component 1, $\bar{h}_1 = h_1$, which is the case when the solvent obeys Raoult's Law. A more detailed examination on this simplification will be given below. Now the enthalpy difference can be understood as the enthalpy of fusion of pure component 1:

$$h_{1,liq} - h_{1,sol} = \Delta h_{fus} \quad (2.86)$$

Those two assumptions allow to calculate a freezing point depression ΔT_f from equation 2.85 as

$$\Delta T_f = \frac{RT_{f,m}^2}{\Delta h_{fus}} x_{2,m} \quad (2.87)$$

or, more accurately, as this equation is valid for dilute solutions,

$$\lim_{x_{2,m} \rightarrow 0} \frac{\Delta T_f}{x_{2,m}} = \frac{RT_{f,m}^2}{\Delta h_{fus}} \quad (2.88)$$

which is known as van't Hoff's Law of freezing point depression. The sign of equations 2.87 and 2.88 depends on the definition of ΔT_f .

2.5.3 Approximation without Ideal Solution Assumptions

The result presented in the preceding section is based on the assumptions that

1. $T_{f,m} \simeq T_f$ and therefore $T_{f,m}T_f \simeq T_f^2$
2. $\bar{h}_1 = h_1$
3. $\bar{h}_{1,liq} - h_{1,sol}$ remains constant over the small temperature range during integration (equation 2.85)

The **first** simplification can be dropped without causing any complication, because using EES or a comparable equation solver, it is not necessary to have explicit equations.

The **second** assumption $\bar{h}_1 = h_1$ implies that the partial molar excess enthalpy \bar{h}^E equals zero, because \bar{h}^E is defined as²¹

$$\bar{h}_i^E = h_{pure,i} - \bar{h}_i \quad (2.89)$$

To avoid the error introduced by this assumption, the enthalpy of pure component i has to be corrected by the partial molar excess enthalpy for i , which can be calculated from the activity coefficient of component i , according to Prausnitz²²:

$$\bar{h}_i^E = -RT^2 \left(\frac{\partial \ln \gamma_i}{\partial T} \right)_{P,x} \quad (2.90)$$

In a two component system ($i = [1,2]$), the activity coefficient γ_1 of component 1 is expressed as

$$\begin{aligned} \ln \gamma_1 = & \ln \frac{\Phi_1^*}{x_1} + \frac{z}{2} q_1 \ln \frac{\Theta_1}{\Phi_1^*} + \Phi_2^* \left(l_1 - \frac{r_1}{r_2} \right) \\ & - q_1^* \ln(\Theta_1^* + \Theta_2^* \tau_{21}) + \Theta_2^* \left(\frac{\tau_{21}}{\Theta_1^* + \Theta_2^* \tau_{21}} - \frac{\tau_{12}}{\Theta_2^* + \Theta_1^* \tau_{12}} \right) \end{aligned} \quad (2.91)$$

with

$$l_1 = \frac{z}{2} (r_1 - q_1) - r_1 + 1 \quad (2.92)$$

$$l_2 = \frac{z}{2} (r_2 - q_2) - r_2 + 1 \quad (2.93)$$

and all constants as derived in section 2.2.4. It has already been shown on page 28 that Φ , Θ , and Θ^* are constant because r , q , and q^* are constant. Applying the same procedure as in the earlier section and using the same expressions for the derivatives of τ , the partial

²¹(Prausnitz et al., 1986), page 198

²²page 198

molar excess enthalpy of component 1 can be expressed as

$$\begin{aligned} \bar{h}_i^E = & RT^2 \Theta_2^* q_1^* \left[\left(\frac{\Theta_2^* \tau_{21}}{(\Theta_1^* + \Theta_2^* \tau_{21})^2} \right) \frac{d\tau_{21}}{dT} \right. \\ & \left. + \left(\frac{1}{\Theta_2^* + \Theta_1^* \tau_{12}} - \frac{\Theta_1^* \tau_{12}}{(\Theta_2^* + \Theta_1^* \tau_{12})^2} \right) \frac{d\tau_{12}}{dT} \right] \end{aligned} \quad (2.94)$$

The **third** assumption that $\bar{h}_{1,liq} - h_{1,sol}$ remains constant over the small temperature range during integration is kept, because no more accurate expression could be found. This third assumption might be the objective of further investigation.

2.5.4 Reference Data

Reference data was taken from the CRC Handbook of Chemistry and Physics²³. The data ranged from 0 to 68 mass-% ethanol in water, which is not quite 50 mole-%.

2.5.5 Results

The resulting freezing point depressions are given graphically in figure 2.8. The predictions with the simple dilute solution approach, with van't Hoff's Law and without the simplifying assumptions 1 and 2 are lower than the measured depression. Since all predictions show significant deviations from the measurements, a curve-fit of the measured data appears more reliable to use. EES delivers the following third order polynomial which shows good agreement with the measured data (RMS of 0.41253):

$$\Delta T_f = -0.1765786 + 101.5153 * x_2 + 323.1374 * x_2^2 - 709.7395 * x_2^3 \quad (2.95)$$

²³(CRC, 7879), page D-274

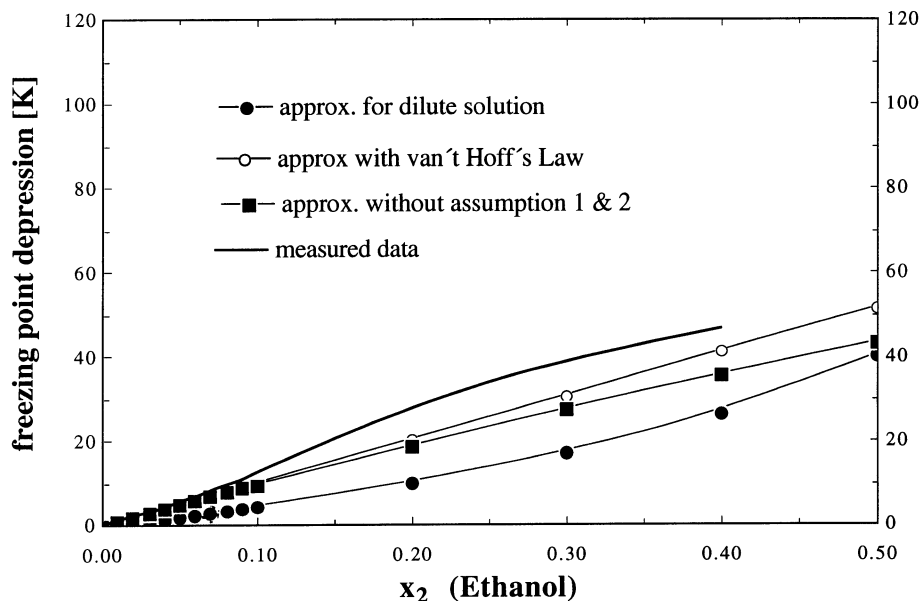


Figure 2.8: Freezing point depression for water-ethanol. The figure shows estimates with different approaches and measured data.

2.6 Applicability of Water-Ethanol for Cold Storage

The binary water-ethanol mixture appears to meet the main requirements that were posed on possible cold storage media:

2.6.1 Specific Heat

Figure 2.3 shows that c_p is almost a linear function of both T and x over nearly the entire range of (T, x) that was investigated. The graph does not predict an unfavorable drop in specific heat for any condition. The increase of c_p for low temperatures and low ethanol concentrations is insignificant, because the pairs of (T, x) in question will most probably lie in the solid (frozen) phase of the mixture.

2.6.2 Density

Figure 2.6 shows that there is no density maximum in the temperature range shown for ethanol concentrations above ca. 30%. Figure 2.9 is an extremely helpful illustration to determine the possible temperature and concentration range of any cold storage application. It presents the freezing point of water-ethanol as a function of concentration, plotting several freezing point approximations as well as the measured data. Included are lines of constant density and, as a dark line, the temperature at which the mixture reaches the point of maximum density²⁴.

Any application using water-ethanol with the intention to maintain the liquid state should operate in a temperature-concentration range that is located above any of the temperature lines shown in figure 2.9. This assumes that the mixture is not frozen and that the density is always increasing with a decrease in temperature.

As a consequence, I chose a mixture of 25 mass-% ethanol and 75 mass-% water as storage medium for this thesis. The freezing point should then be well below -10°C. Since the TRNSYS components require (and allow) only constant values of ρ and c_p , I averaged those values to $\rho = 975 \text{ kg/m}^3$ and $c_p = 4.00 \text{ kJ/kgK}$. The viscosity was not required as a parameter, since friction in pipes has been neglected.

²⁴calculated with EES, using the curve-fit data presented in figure 2.6

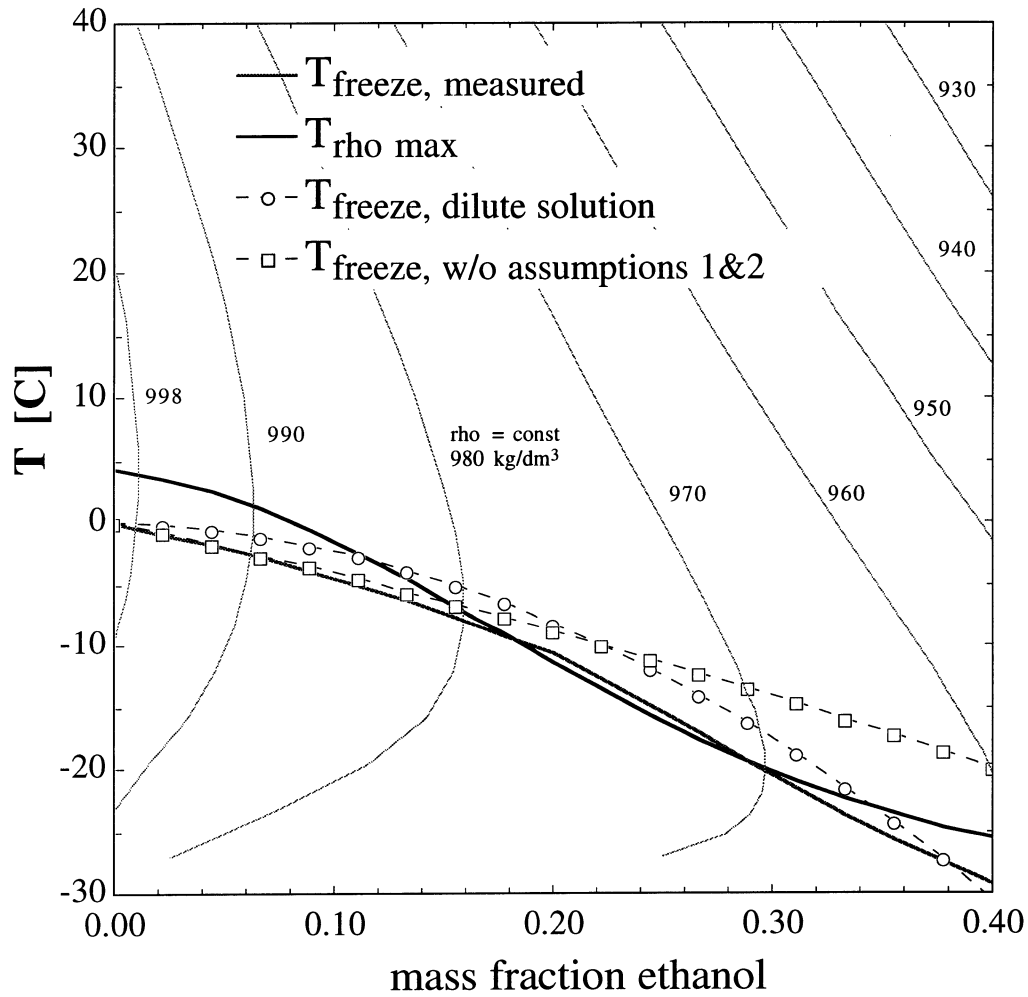


Figure 2.9: Freezing point and temperature of maximum density as functions of concentration. This illustrates feasible combinations of temperature and concentration for applications. The lines of constant density of the mixture are calculated with the curve-fit density function. Those parts of the lines which lie *below* the freezing point line do not exist because the mixture (or at least its water phase) would be frozen at these conditions.

2.7 Nomenclature used in chapter 2

Roman Symbols

A	coefficient
A_{ij}	coefficient in UNIQUAC approach relating substance i with substance j
B	coefficient
$Bias$	see page 41
$C_{1...6}$	coefficients
C_p	(absolute) specific heat
E_f	cryoscopic constant of water
F	(absolute) Helmholtz energy
G	(absolute) Gibbs energy
H	(absolute) enthalpy
MW	molar weight
N	total number of molecules, usually measured in moles
P	pressure
RMS	root mean square error
R^2	see page 41
R_{cal}	universal gas constant on a calorie unit basis
R	universal gas constant
S	(absolute) entropy
T	temperature
U	(absolute) internal energy
V	(absolute) volume
Y	(absolute) arbitrary extensive quantity
Z	introduced to increase readability (no physical meaning)
$a_{0...1,ij}$	coefficients to calculate A_{ij}
$a_{0...3}$	coefficients
d	deviation
f	fugacity
g	(specific) Gibbs energy
h^*	molar enthalpy in the ideal gas state
\bar{h}	partial molar enthalpy
m	mass
\bar{m}	molality
q^*	coefficient in UNIQUAC approach
q	coefficient in UNIQUAC approach
r	coefficient in UNIQUAC approach
s	(specific) entropy
v	(specific) volume on molar basis

v'	(specific) volume on mass basis
x	mole fraction
y	(specific) arbitrary extensive quantity

Greek Symbols

ΔT_f	freezing point depression
Δh_{fus}	(molar) enthalpy of fusion = $h_{liq} - h_{sol}$
γ	activity component
μ	chemical potential
η	dynamic viscosity
Φ^*	segment fraction (coefficient in UNIQUAC approach)
τ_{ij}	coefficient in UNIQUAC approach relating substance i with substance j
Θ^*	area fraction (coefficient in UNIQUAC approach)
Θ	area fraction (coefficient in UNIQUAC approach)
ξ	mass fraction

Exponents

E	refers to an excess value
M	refers to mixing
0	refers to a pure component value

Indices

$comb$	combinatorial (used in UNIQUAC approach)
c	critical
f	freezing
id	refers to the model of an ideal solution
liq	liquid
m	equilibrium
ref	reference state
res	residual (used in UNIQUAC approach)
sol	solid

Chapter 3

The Simulated Systems

This chapter describes previously introduced storage media and operating strategies that have been selected for comparison with a water-ethanol based system. The chapter also contains a description of the building load data that served as input to the simulations.

3.1 Selected System Configurations

Out of the many options mentioned in section 1.2, systems based on the following storage media were selected:

- pure water
- a water-ethanol mixture with 25 Vol.-% ethanol
- ice (dynamic)

The reason to investigate a dynamic ice storage system instead of a more popular static system (compare section 1.2.2) was simply that there was a TRNSYS representation of a simple open ice storage tank with an ice harvester available, whereas the static system representation would have had to be written and tested.

Each of the three storage media are used with the following three operating strategies:

- full storage,
- partial storage with chiller priority, and
- partial storage with tank priority,

For comparison, results for a conventional system were generated by using a pure water system with a tank size of zero, thus the performance of 10 individual systems is investigated.

3.2 Building Load Data

The building load data are a primary input to the simulation. The system components are sized in order to meet the load during the entire simulation period, based on the storage medium used and the operating strategy selected.

3.2.1 Data Source

The building load data were provided by the Energy Center of Wisconsin (ECW, 1996) where a database of building load data is maintained. Site #55 is a large office building with $390,000 ft^2$ of cooled floor space. Built in 1987, it is equipped with a 907 ton chiller and another 200 ton chiller with a total nominal COP of 4.09. The building is operated 64 hours/week and with a peak occupancy of 1150 employees.

The ECW data file covers the year 1990 and, among other data, gives the total building energy use (kWh), the energy used for cooling (kWh), the cooling capacity (tons), the ambient temperature (F) and the relative ambient humidity (%).

The characteristics of site #55 and the data for 1990 can be summarized as

- design capacity 3874 kW (1107 tons) for 390,000 ft^2 of conditioned floor space, that is 352 ft^2/ton
- peak demand occurred on September 4 and was 2500 kW (711 tons), that is 64.5% of the design value
- peak integrated daily cooling energy demand also occurred on September 4 and was 38,926 kWh/day (11,069 ton-hr/day)
- design COP of 4.09. For 1990, the maximum hourly COP was 5.04, the minimum was 1.23, and the average hourly COP was 2.39

The day on which the peak load occurs can be defined as the **design day**, or one could choose the day on which the maximum integrated daily load occurs. The choice is easy when both maxima occur on the same day, as it is the case for this data (September 4, 1990). The design day represents the worst cooling conditions the system is required to meet and has to be designed for. All systems presented in this study were sized to meet the design day load found in the ECW data.

3.2.2 Measuring Building Loads

The coil load can be understood as the instantaneous amount of energy removed from the building to keep the building space at constant internal energy. The building load is the instantaneous amount of energy added to the room by occupants, internal gains, external gains, etc. The available data represents the sum of all zones in site #55 and does not distinguish between zones of different load. Therefore, the building is modeled in TRNSYS as a single room using TRNSYS Type 92 (described in section 5.6). The obtained results will only reflect the differences between the compared systems, the absolute results should be regarded with caution because of this (and other) simplifications.

The load itself cannot be measured, because it is the sum of a large number of small contributions. A typical (summer) building load consists of the following time-dependent contributions:

- environmental gains through walls, roof, floor, windows, doors, cracks, etc. due to solar radiation, convection and infiltration (sensible and latent)
- internal gains due to occupants (both latent and sensible)
- appliances, such as (for office buildings) computers, printers, copiers, small-scale kitchen appliances (mainly sensible)
- lighting (sensible)

Examples for building load calculation and details on occupancy loads can be found in the ASHRAE Handbook of Fundamentals (ASHRAE, 1993).

The cooling capacity in the ECW data file is the amount of cooling provided to the entire building for each 15 minute timestep (it comes in kWh, an energy unit). It has been obtained from measurements of actual electricity consumption and weather data which was transformed into the chiller capacity using engineering models of chillers and DX units. The data is explicitly intended to facilitate the comparison of different cooling systems by eliminating the need to run a building simulation.

Assuming that the building maintains its average temperature and humidity level and that there is only a negligible amount of thermal/moisture storage, which is both justified for an office building, the 15 minute cooling capacity is the sum of all above mentioned contributions to the load.

In this simulation the cooling capacity data are regarded as if it were the entire building load *without the ventilation load*. This is no major downside, because the simulated building is fictitious anyway (major simplifying assumptions). What is important is that

the structure and distribution of the building load profile match the reality, which is still given even if site #55's ventilation load is not subtracted from the cooling capacity value. Future references to the “building load” shall always be interpreted as the “real” sum of all contributions to the load except for the ventilation load.

The cooling capacity data found in the ECW data file is printed over the entire year 1990 in figure 3.1 and over the period from July 1st to October 1st in figure 3.2.

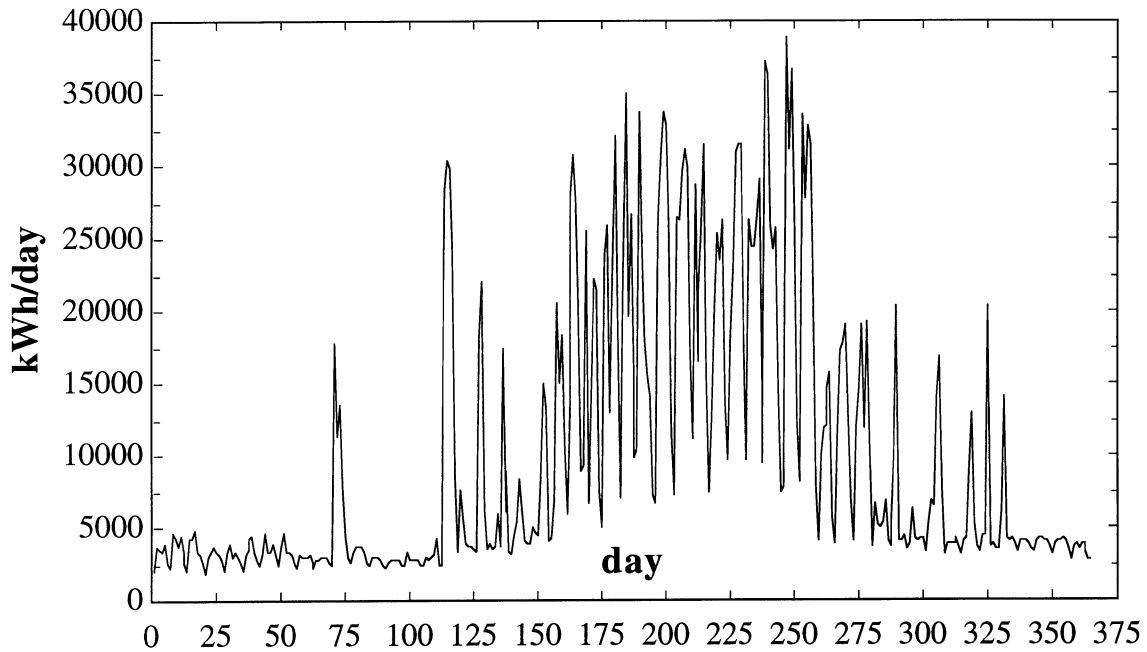


Figure 3.1: Daily building-total electrical energy use of office building site #55, entire year 1990

3.2.3 Sensible Heat Ratio

The total building load \dot{Q}_{bldg} is the sum of the sensible load (energy input to the room causing an increase in room temperature that can be *sensed*) and the latent load (energy entering in the form of water vapor, causing an increase in the room's moisture content).

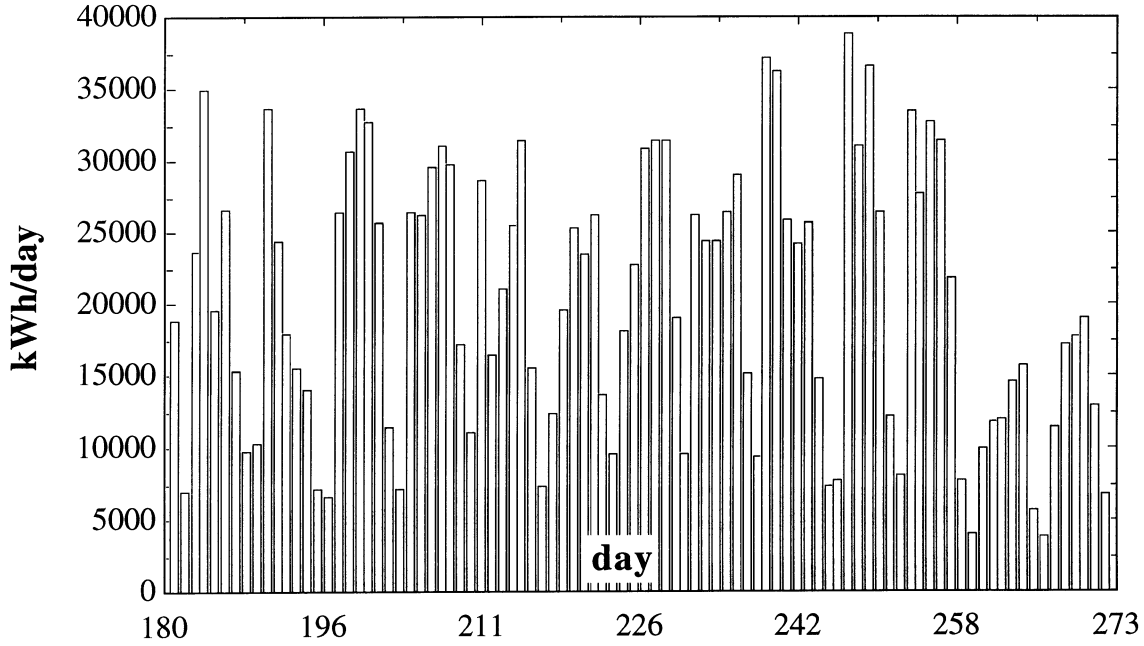


Figure 3.2: Daily building-total electrical energy use of office building site #55, July 1st to October 1st, 1990

For the purposes of this simulation the sensible heat ratio is defined as

$$SHR = \frac{\dot{Q}_{sens}}{\dot{Q}_{bldg}} = \frac{\dot{Q}_{sens}}{\dot{Q}_{sens} + \dot{Q}_{lat}} = 0.8 \quad (3.1)$$

to be constant under all conditions. This assumption is a simplification, as illustrated with the following example: The major source of moisture is the occupants, followed by the ventilation load. During occupancy hours, the latent load will be higher than during night time. Still, considering the fact that some of the moisture will be stored in walls and furniture and then released during unoccupied periods, the assumption of $SHR = 0.8$ is reasonable for an office building and for the purpose of comparing several systems.

A more detailed way of handling the SHR would be to use a building model and calculate \dot{Q}_{sens} and \dot{Q}_{lat} from the ambient conditions, a measured or assumed occupancy profile, the estimated heat and moisture rejection of the occupants and appliances, and the calculated

ventilation load.

3.3 System Setup

Figures 3.3 and 3.4 show the system setup for the water and water-ethanol based storage systems and the ice storage system.

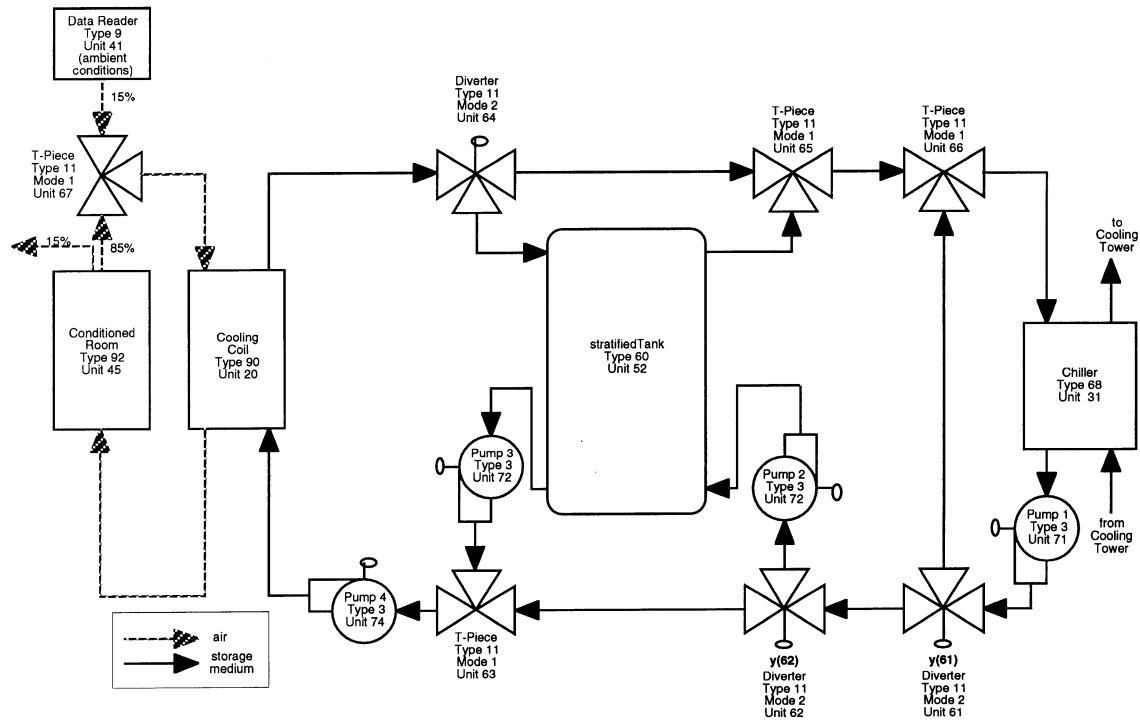


Figure 3.3: Flow Diagram representing the Liquid Storage System in TRNSYS, used for Water and Water-Ethanol. A more detailed version can be found on page 161 in Appendix D.

Each individual component represents a physical device like a chiller, cooling coil or storage tank as well as a TRNSYS module, a “Type”. The lines symbolize both physical pipe, duct or power connection between the individual components as well as connections in TRNSYS, which is why these diagrams are very helpful to set up the system and understand the simulation output. The unit number shown under each type number is important only in TRNSYS, it is used as a unique identification number for every system

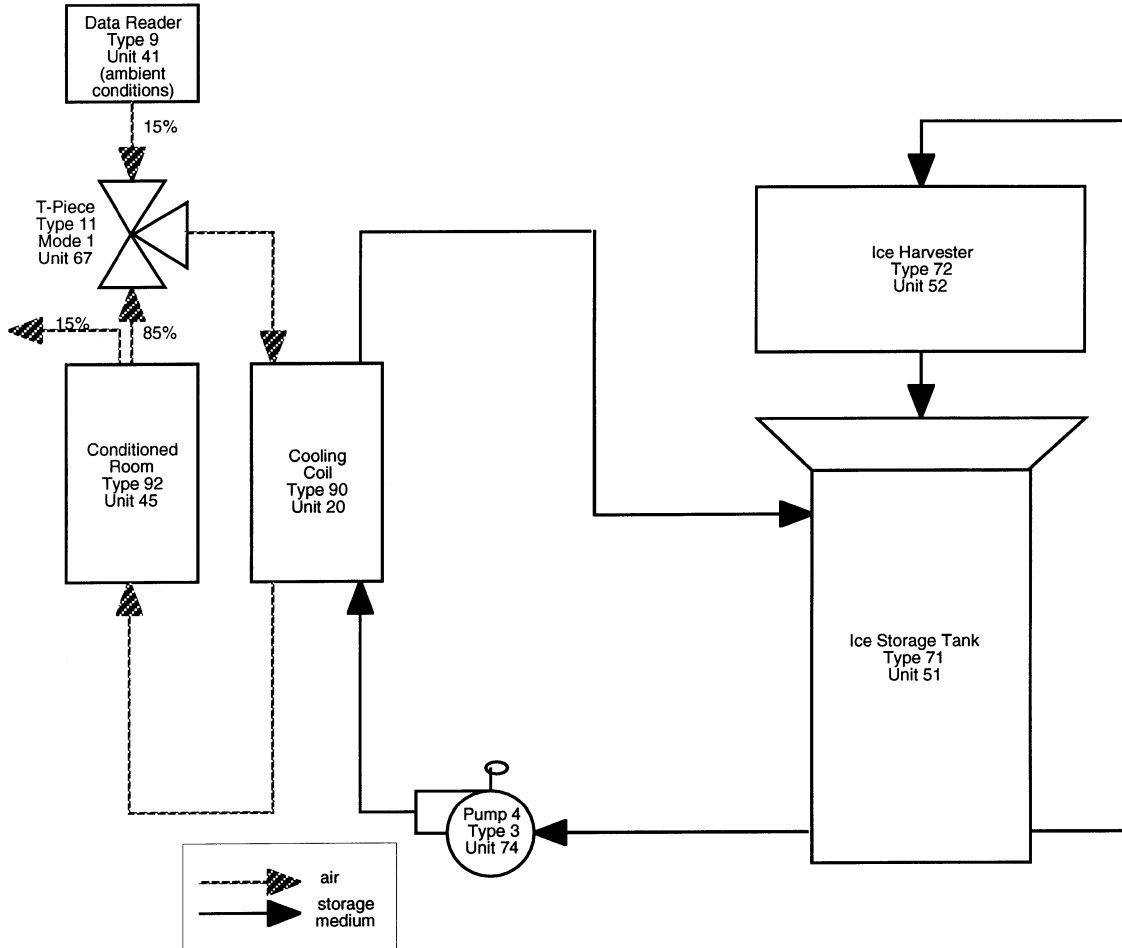


Figure 3.4: Flow Diagram representing the Ice Storage System in TRNSYS. A more detailed version can be found on page 162 in Appendix D.

component (there can be multiple occurrences of the same type of module, but each unit number occurs only once). Refer to the TRNSYS manual for further information.

3.3.1 Common Features in the Liquid Storage and Ice Storage Systems

In both system types, the coil delivers air at a supply temperature of $T_{a,sup} = 12^{\circ}\text{C}$ to the room. Of the return air, 15% are exhausted and replaced by outdoor air. This ratio

is maintained constant. The ambient conditions are read from the ECW data file as is the load, to which the room is exposed. The model for the room, Type 92, calculates the mass flow rate of air that needs to be supplied in order to maintain constant room temperature, and this information is passed to the coil. In turn, the coil model iterates the necessary mass flow rate of water (given the water supply temperature which depends on the chosen system) and passes this information on to the controller, Type 99, which sets the appropriate pumps and valves to provide this water mass flow rate.

The coil load is the sum of the building load and the ventilation load at each discrete timestep, this is being verified in the simulation for every timestep. In energy terms, the relevant equation, which is used in the TRNSYS deck, reads

$$0 = (\dot{Q}_{bldg} - \dot{Q}_{coil} - \dot{Q}_{vent}) \Delta t - \Delta U_{room} \quad (3.2)$$

This equation is depicted in figure 3.5.

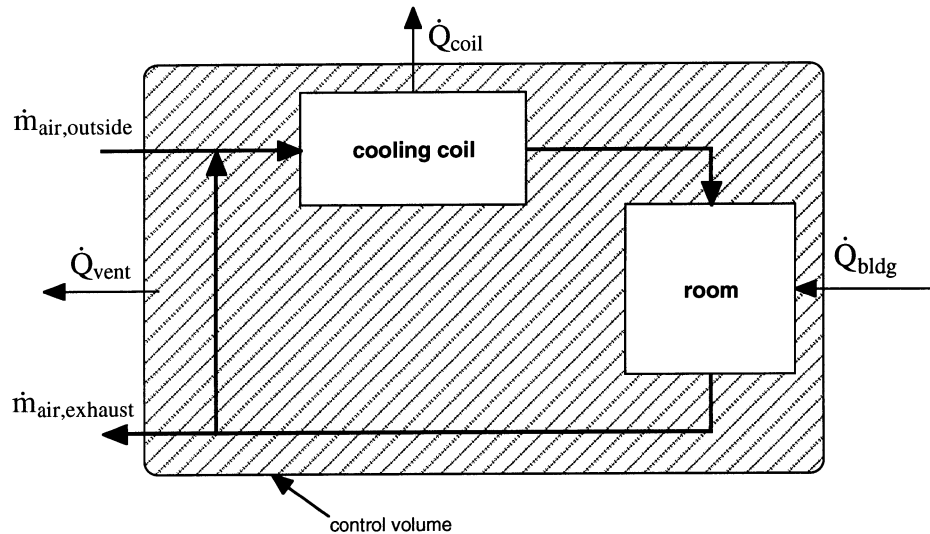


Figure 3.5: Illustration of the energy balance on the air side

3.3.2 The Liquid Storage System

The liquid system, used for the storage media water and water-ethanol, uses a liquid storage tank and a liquid chiller. The piping is set up such that the chiller can charge the tank as well as meet the coil load (or both at a time).

The chiller cools the storage medium returning from either the tank, the coil or both down to the chiller setpoint temperature, which is also the storage temperature that has been selected for each storage medium. Depending on several factors (e.g. load, selected operating strategy, time of day, tank charge level) the chiller meets the entire coil load or part of it and/or charges the tank.

The tank acts as the buffer to decouple the production of chilled liquid from the demand. The water cycle's energy balance, which is defined in the next equation and illustrated in figure 3.6, is checked for every timestep:

$$0 = \left(\dot{Q}_{chil} - \dot{Q}_{tank,amb} - \dot{Q}_{coil} - \sum_{i=1}^4 \dot{Q}_{pump,i} \right) \Delta t + \Delta U_{tank} \quad \text{where} \quad (3.3)$$

$$\sum_{i=1}^4 \dot{Q}_{pump,i} = (P_{pump,1} + \dots + P_{pump,4}) (1 - \eta_{pumps}) \quad (3.4)$$

The change in internal energy of the tank has to equal the energy removed from the water by the chiller less the sum of all energy gains due to the coil, the pumps and the tank itself.

3.3.3 The Ice Storage System

The ice storage system has a simpler setup in TRNSYS, as illustrated in figure 3.4. All supply water is taken from the tank, thus just the pump serving the cooling coil is needed. A pump between ice storage tank and harvester is not needed in TRNSYS (Cross, 1994).

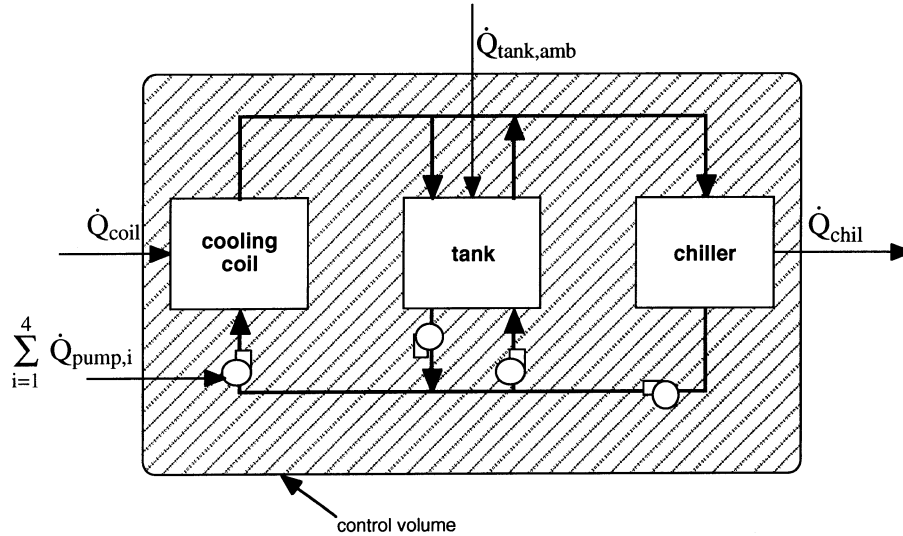


Figure 3.6: Illustration of the energy balance on the water side. Each pump, if operating, creates an increase in temperature at its individual location, but for the overall energy balance their contributions may be taken as one single energy flow rate.

As in the liquid systems, the operating status of the harvester depends on e.g. load, selected operating strategy, time of day, and tank charge level. Since there is less equipment (resulting in fewer TRNSYS units) in this system and since the ice storage tank is a rather simple TRNSYS model requiring very little calculations, computation time for the ice storage systems is significantly shorter than for the liquid systems.

3.3.4 Differences between the TRNSYS system and the physical system

There are some details where the system is set up differently in TRNSYS as it would be in reality¹.

For example, a real chiller would not always supply water at the desired temperature, because just after starting it there would be some liquid in the pipes and the chiller itself

¹There are also differences between the system setup in “reality” and a system that one would actually build, due to assumptions and simplifications, but this is not the issue in this section.

that has to be cooled down first. To avoid supplying water that is not at the desired temperature the chiller usually is connected to a recirculation loop with a temperature-controlled valve that recirculates a certain amount of water such that the chiller supplies at the design temperature. Apparently this recirculation loop is missing in the TRNSYS setup.

The reason is that the TRNSYS chiller model can be assumed as a black box including the chiller itself *and* the necessary loops, valves, etc. The underlying curve-fit takes this into account. In fact, today chilling equipment is manufactured as self-contained units where a local chiller controller assures that the chilled liquid is supplied at a user-specified temperature. Even more complicated tasks can be assigned to this local controller, such as limiting the chiller's demand to a certain fraction of its design power, an option that was unfortunately not available for any TRNSYS chiller model.

Chapter 4

Equipment Sizing

The air setpoint temperature $T_{a,set}$, at which air is supplied to the room, was selected to be 12°C in all cases. In order to deliver a sufficient amount of cold air at $T_{a,set}$ to maintain the room temperature under all conditions during the simulation, the equipment needs to be sized properly. Required key values include:

- the cooling coil's UA values: $(UA)_a$ in W/K and a factor for $\frac{(UA)_w}{(UA)_a}$
- the storage tank capacity, in terms of volume V in m^3 as well as deliverable energy in kWh
- the chiller (liquid systems) and harvester (ice system) capacity in kW
- the pump mass flow rates

Assuming large initial values for the above quantities allows one to gain some general experience with the system and the simulation software, but as soon as, for example, a partial storage system is investigated the equipment needs to be of reasonable size: if in a tank priority system the tank is too large, the chiller will never be needed to supplement the tank, and the system behaves like a full storage system, and if the chiller is too large

in a chiller priority system, the tank will never be used at all and the system behaves like a non-storage system.

The equipment is sized using rough cut calculations with subsequent adjustments of the parameters, exposing it to design day conditions. This means that a simulation is run for several days using only one day's (the design day's) data repeatedly until the simulation output for the design day does not change any more. This will be the case after a number of repetitions that depends on the complexity of the system, and this kind of solution is called a *steady state periodic solution*. In this simulation, the liquid-based systems reached a steady-periodic state after 6 days, and the (simpler) ice-based systems after even fewer repetitions. Therefore, 6 repetitions of the design day data were processed to determine the equipment size for all systems.

4.1 Selection of General System Parameters

There is no general procedure to exactly size equipment, because every system will be designed to behave slightly different. There are, however, some design issues that will be similar in every HVAC design procedure. For example, a decision has to be made on the system type, in terms of either constant or variable air volume (CAV or VAV), and for the coil the design entering water and leaving air temperature as well as a design room temperature and humidity level need to be specified. This simulation features a VAV system with

- $T_{w,set} = 5^{\circ}C$ ($0^{\circ}C$ for ice system)
- $T_{a,set} = 12^{\circ}C$
- $T_{room,set} = 24^{\circ}C$ and $RH_{room,set} = 50\%$

Also, the load was assumed to be 80% sensible and 20% latent ($SHR = 0.8$) for the entire simulation, as explained in section 3.2.3.

4.2 Calculation of the Air Mass Flow Rate for a VAV System

One way to calculate the necessary air supply rate for every timestep involved is given in the following equation:

$$\dot{Q}_{bldg} * SHR = \dot{m}_a * c_{p,a} * (T_{room,set} - T_{a,set}) \quad (4.1)$$

This assumes that the air leaving the room is actually the room setpoint temperature, $T_{room} = T_{room,set}$. A refinement is to use the room temperature itself or, if the room is not at homogeneous temperature, the temperature of the leaving air.

In practice, the load is unknown and cannot be read from a data file, and other methods to determine \dot{m}_a have to be found. Some involve comparison of measured room conditions (temperature, humidity, carbon dioxide concentration) to setpoint values, and some try to calculate the building load using occupancy data and other sources.

Even with the building load known, using equation 4.1 did not produce satisfying results in the sense that the room temperature would not remain constant when the calculated \dot{m}_a was actually supplied. The reason lies probably in how the room is modeled (see section 5.6), assuming no thermal and humidity storage capacity except the room air itself. This is why Type 92 was set up to iterate \dot{m}_a and pass it back to the cooling coil, which will in turn calculate the necessary water mass flow rate.

The room humidity does not immediately influence the value of \dot{m}_a . It is common practice to size \dot{m}_a by the sensible load only (using temperature differences) and to add (de-)

humidification as needed.

4.3 Measures of System Performance

The following quantities are used to determine the system performance. During the process of sizing the equipment, these quantities are calculated for every repetition of the design day. A steady-periodic state was said to be achieved when these values did not change significantly from one repetition to the next.

4.3.1 Maximum (Liquid) Supply Temperature

$T_{sup,max}$ is the maximum temperature at which the storage medium was supplied to the coil. If $T_{sup,max}$ exceeded the design value $T_{sup,des}$ by more than 1 K the system was considered to be undersized. Since the chiller supplies at $T_{sup,des}$, it is the tank supply temperature that is too high, which can be overcome by increasing the tank size or the chiller size (to better recharge the tank) or both.

4.3.2 Tank Level

The “tank level” is defined to estimate the tank’s state of charge. A tank level of 100% means that the tank is fully recharged, and a tank level of 0% means that the tank is fully discharged. The tank level is defined on an energy basis as

$$\lambda_{tank} = \frac{Q_{tank}}{Q_{tank,des}} \quad (4.2)$$

where the tank design capacity $Q_{tank,des}$ equals $V_{tank} \rho_{liq} c_p \Delta T_{des}$ for the liquid system and $V_{tank} \rho_{ice} (c_p \Delta T_{des} + \Delta h_{sub})$ for the ice system. $\Delta h_{sub} = 334.994 \frac{kJ}{kg}$ is the enthalpy of sublimation for water.

For the liquid system, the actual tank capacity Q_{tank} is calculated as the design capacity minus the change in internal energy of the tank since the beginning of the simulation, a quantity that is provided by the tank model. This requires that the tank be fully charged at the beginning of the simulation.

For the ice system, the actual tank capacity is calculated from the amount of ice currently present in the tank, the value of which is provided by the ice storage tank model.

If the liquid storage tank is discharged, the water supply temperature rises moderately, but as soon as the thermocline reaches the tank bottom it increases rapidly, and this situation needs to be avoided under any load condition. Supplying water at a temperature well above the design supply temperature prevents the coil from meeting the load and it will request the maximum flow rate allowed by the pumps and still not be able to cool the air down to $T_{a,set}$. To avoid this, a minimum tank level of 20% is defined below which the tank should not be discharged.

4.3.3 Tank Charge/Discharge Balance

Monitoring the tank's in- and outlets allows to calculate the change in the tank internal energy attributed to charging and discharging, \dot{Q}_{chg} and \dot{Q}_{dischg} (both are counted positive). Integrating the two energy flow rates and subtracting the energies should show that more energy was removed from the tank by charging it than was added to the tank by discharging it.

$$\int_0^{24\text{ hr}} (\dot{Q}_{chg} - \dot{Q}_{dischg}) dt > 0 \quad (4.3)$$

The tank gains additional energy from the environment, making up for the difference. If the tank is discharged more than it is charged, steady-periodic state is not reached, which should also show up in the values of the maximum tank level over the days. This can be overcome by increasing the tank charge and discharge pump capacity (pump 2/3),

increasing the chiller size, or reducing the tank volume.

4.3.4 Chiller Part Load Ratio

Once an initial chiller size has been roughly sized, it is then adjusted by changing the mass flow rate $\dot{m}_{1,max}$, using the equation

$$\dot{Q}_{chil,des} = \dot{m}_{1,max} c_p \Delta T_{des} \quad (4.4)$$

and the respective design electric capacity with the equation

$$P_{chil,des} = \frac{\dot{Q}_{chil,des}}{COP_{des}} \quad (4.5)$$

For every timestep the actual values \dot{Q}_{chil} and P_{chil} are recorded, and two more helpful system performance indicators are the chiller load ratios can be derived:

$$PLR = \frac{P_{chil}}{P_{chil,des}} \quad \text{and} \quad QLR = \frac{\dot{Q}_{chil}}{\dot{Q}_{chil,des}} \quad (4.6)$$

The maxima of these values show if the design chiller capacity is really needed or if the chiller is too small.

$\dot{m}_{1,max}$ cannot be exceeded, c_p is a constant during the simulation, and the chiller supplying at $T_{w,set}$, thus every occurrence of $QLR > 100\%$ is due to increases in the water return temperature to the chiller over its design value:

$$T_{w,ret} - T_{w,set} > \Delta T_{des} \quad (4.7)$$

Very high load ratios of $QLR = 120\%$ occurred only for single timesteps and were therefore accepted, as long as in general the chiller would not exceed its design capacity.

4.4 Pump Sizing

The pump sizes, though not the most important design parameters, need to be set to reasonable initial values. Pump 2/3 (tank charge/discharge) and pump 4 (coil) can be given high initial values and some later adjustment. However, pump 1 (chiller) requires special attention, because it is directly related to the chiller size by equation 4.4.

4.5 Coil Sizing

The first step is to find suitable UA values for the coil in the steady state periodic solution. This is accomplished by first finding the largest air mass flow rate (roughly $\dot{m}_{a,max} = 590,200 \text{ kg/hr}$) with very large UAs and after that varying $(UA)_a$ and the UA factor in a way such that

- the corresponding largest water(-ethanol) mass flow rate $\dot{m}_{w,max}$ showed a reasonable ratio to $\dot{m}_{a,max}$ (roughly $\frac{\dot{m}_{a,max}}{\dot{m}_{w,max}} = 3$)
- the air supply temperature was always the desired 12°C : if $(UA)_a$ is too low, the supply air will not reach 12°C , or the iteration within Type 90 will fail to converge, respectively.

It is important to allow for some added coil capacity, even if the system seems to work for the repeated design day. If the UA values (the coil size) are too low, the following situation might arise:

- The water mass flow rate \dot{m}_w is usually controlled such that the leaving air has the desired temperature: $T_{a,sup} = T_{a,set}$. If \dot{m}_w is larger, $T_{a,sup}$ will drop etc.
- If the coil is too small, $T_{a,sup}$ will always be higher than $T_{a,set}$, and the controller will set \dot{m}_w to its maximum value to get as close to $T_{a,set}$ as possible.

- Due to the high \dot{m}_w , the return water temperature $T_{w,ret}$ will drop significantly
- Depending on the operation strategy, the return water might be added to the tank, and most likely $T_{w,ret}$ will be half way in between the temperatures at the top and the bottom of the storage tank. This will enlarge the thermocline and prevent the stratification, due to increased fluid mixing in the tank.
- Since the tank is being discharged much faster now (the tank discharge pump will operate at its design value), the tank will have to be supported by the chiller. This will most likely force the chiller to run on-peak, using expensive on-peak energy and maybe even increasing the maximum demand for the relevant charging period, which can cause additional expenses.

As final coil size $(UA)_a = 500,000$ W/K and $\frac{(UA)_w}{(UA)_a} = 5$ were selected. Pump 4 was then designed such that it could provide the maximum flow rate of storage medium, 180,000 kg/hr for the water system, 135,000 kg/hr for the water-ethanol system, and 130,000 kg/hr for the ice storage system. The slightly higher values for the respective tank priority systems result from a slightly higher supply temperature at peak load, and the same argument explains the slightly lower values for the ice storage system.

4.6 Tank and Chiller/Harvester Sizing

Once the UA values are known and it has been verified that the load can be met over the *entire* simulation period¹, the tank and the cooling device can be sized. Other than the coil size, the size of these two components strongly depends on the selected storage mode, and also of each other, which is why they have to be sized together.

¹This was important during the simulation because for some timesteps the inlet air conditions were worse (e.g. very humid) than on the design day, combined with a fairly high load, and $T_{a,set}$ could not be reached

Basic equations to calculate initial values for the sizing process are given in section 1.4. With these values as a first guess, the systems are simulated with the repeated design day data, and adjustments are made to assure that each system reaches a steady-periodic state, even if the equipment is still oversized. Then the tank and chiller sizes can be reduced in small steps, changing only one value at a time, which allows to study their individual impact on the performance and of interdependencies. The quantities listed in section 4.3 are used as performance indicators.

4.6.1 Full Storage Mode

The full storage system is relatively easy to size, because the tank size does not depend on the chiller size. The tank has to meet the on-peak load and can be sized according to that, and the chiller can be sized afterwards to recharge the tank completely and simultaneously meet the off-peak load.

4.6.2 Partial Storage with Chiller Priority Mode

In this mode it is intended to keep the chiller as small as possible. To reach steady-periodic state, the minimum chiller size is the integrated coil load (see equation 1.9) plus the integrated heat gains due to tank and pumps, all on the design day. Using an initially oversized chiller and tank a steady-periodic solution must be reached, and both values can then be diminished in subsequent simulations to a point where the chiller runs at full capacity over the entire design day and where the tank is fully charged and discharged to the specified minimum tank level on every (design) day.

On a non-design day the chiller also operates every hour of the day where there is a coil load to meet. These loads will feature lower peaks and the integrated load will be lower as well, thus the tank will not be discharged as much, does not require as much recharging,

and the tank will be recharged earlier. When the tank is completely recharged the chiller will operate on part load.

4.6.3 Partial Storage with Tank Priority Mode

In a tank priority system, other than under chiller priority, the chiller should run as little as possible during the on-peak period on regular days. However, on the design day the chiller operates throughout the entire on-peak period. As a consequence, the tank is much smaller than for a full storage system.

If demand leveling is desired, the chiller will not operate with its full capacity during the on-peak period to avoid the high demand charge that is due under the chiller priority mode. In turn, the chiller must be sized a little larger than in chiller priority, because it has to make up for its limited on-peak operation during the following off-peak period.

If the system is not set up for demand leveling, tank and chiller may be equally sized for chiller priority and for tank priority. The implementation of tank priority requires the system engineer to provide answers to all those questions listed in section 1.4.2, page 15. EPRI also indicates that a “sophisticated” control strategy must be implemented. For example, since the bottom line of every air-conditioning application is that the conditioned space be comfortable at all times, it might become necessary to operate the chiller at maximum capacity during on-peak because the load has been underestimated and now cannot be met with the tank alone or even with the chiller operating at a lower level. One of these instances is sufficient to increase the peak demand value and the demand charge associated with it to what it would be if chiller priority were used.

The following list describes how demand leveling is realized in the controller Type 99. This algorithm is executed at the first call of each timestep that lies within an on-peak period.

1. The coil load is recorded for the last 24 hours ($\dot{Q}_{coil,i}$, $i = 1 \dots 96^2$).
2. Before overwriting the oldest saved $\dot{Q}_{coil,i}$ in the array (the coil load that occurred precisely 24 hours ago) with the present hour's coil load \dot{Q}_{coil} , the factor $scale = \frac{\dot{Q}_{coil}}{\dot{Q}_{coil,i}}$ is calculated.
3. The integrated coil load for the period until the end of this day's off-peak period is estimated with the $\dot{Q}_{coil,i}$. This value is then scaled up or down with the $scale$ factor.

$$Q_{est,1} = \left(\sum_{i=1}^{96} \dot{Q}_{coil} \right) * \Delta t * scale \quad (4.8)$$

4. The load estimate $Q_{est,1}$ is then diminished by the capacity that can be provided by the tank if it is discharged to the user-specified minimum level $\lambda_{tank,min}$ (20%, because of the rapid increase in supply temperature due to the movement of the thermocline)

$$Q_{est,2} = Q_{est,1} - Q_{tank,des}(\lambda_{tank} - \lambda_{tank,min}) \quad (4.9)$$

λ_{tank} is the actual tank level.

5. The current demand level $QLR_{max} = \frac{\dot{Q}_{chil,max,on}}{\dot{Q}_{chil,des}}$ is reset to a user-specified minimum value QLR_{min} at the beginning of a user-specified period, e.g. the beginning of a new month. $\dot{Q}_{chil,max,on}$ is the maximum on-peak chiller capacity for the month. The demand level is defined on the chiller capacity \dot{Q}_{chil} rather than on its power consumption, so even if a certain demand level is not exceeded the power consumed by the chiller may vary (due to various factors that influence the COP). In order to use a power-based demand level PLR_{max} , the controller would have to be able to calculate the maximum allowable chiller capacity for a certain demand level, which

²24 hours in 15 minute steps

would have required a feedback of other data to the controller. This might be the next step in a future improvement of this very simple algorithm.

6. For the liquid systems:

- The chiller pump is designed to operate without flow modulation, thus to level the demand means to restrict the amount of liquid being supplied to the load, $\dot{m}_{chil,sup}$, even if the full design flow rate $\dot{m}_{chil,des}$ circulates through the chiller. This will lower the temperature of the liquid entering the evaporator. The more water is being recirculated, the smaller is the difference ΔT between entering and leaving evaporator water temperature, and it can be approximated that

$$\frac{\dot{m}_{chil,sup}}{\dot{m}_{chil,max}} \approx \frac{\Delta T}{\Delta T_{des}} \quad (4.10)$$

Using this approximation, the maximum allowable supply mass flow rate that can be contributed to the load while not exceeding the demand level QLR_{max} is

$$\begin{aligned} QLR_{max} &= \frac{\dot{Q}_{chil,max,on}}{\dot{Q}_{chil,des}} = \frac{\dot{m}_{chil,max} c_{p,w} \Delta T}{\dot{m}_{chil,max} c_{p,w} \Delta T_{des}} \\ \Leftrightarrow \dot{m}_{chil,max} &= \dot{m}_{chil,des} * QLR_{max} \end{aligned} \quad (4.11)$$

- The tank makes up for the difference between \dot{m}_{coil} and $\dot{m}_{chil,max}$, and the latter had to be calculated to determine the necessary tank pump setting.
7. For the ice system: to make the harvester suitable for demand leveling, the harvester control function is set to the value of QLR_{max} during the on-peak period and to 1 or 0 during the off-peak period, depending on if the tank is fully charged or not.
8. The chiller operates during the entire on-peak period at the specified fraction of its

design capacity. This is not always needed, since on non-design days the tank could meet the entire on-peak load without the chiller, and this is also economically not favourable, since on-peak electricity is more expensive. However, this was the only way identified to allow the implementation of demand-leveling strategy. A great deal of time was spent on the implementation of other algorithms, but none of them produced the desired system behavior. Alternatives included

- other ways to predict the coil load (integrated load of last 24 hours, integrated load until next on-peak period starts, inclusion of tank charging, etc.)
- calculation of a time at which the chiller, operating at its current demand level, would have to be started in order to meet the estimated coil load

The need to size tank and chiller simultaneously for this control strategy can be illustrated with the following example which occurred during the sizing process. Other than in a full storage or chiller priority system, decreasing the tank size does *not* lead to the tank being discharged to a lower tank level, as one would assume. Instead, the minimum tank discharge fraction *increases*. The mechanism behind this phenomenon is that if the tank is smaller (than in a previous simulation), the controller calculates a higher demand level. Thus, the tank is used less than before, and the minimum tank level that is recorded can have a higher, lower or equal value than before (depending on other parameters).

Several tank/chiller combinations are suitable to meet the load, they differ in their maximum demand and their minimum tank level. I aimed for a tank/chiller combination that had sizes half way between the full storage and the chiller priority equipment. Another basis for tank/chiller sizing could have been to minimize operation cost.

4.7 Safety Factors in Equipment Sizing

Even if a system has been sized for a design load profile, it might not work properly every other day that has a lower peak load and a lower integrated daily load. The reason is that these two quantities are not the only important parameters in the sizing procedure. The system must also be designed for unfavorable load profiles, profile meaning the distribution of loads over the day.

A favorable profile is e.g. one where the peak loads occur early in the day when the tank is fully charged and the water supply temperature is very close to the design value. Here, the maximum water mass flow rate will be lower than if the peak load occurred late in the afternoon where most of the very cold water has already been used and the water supply temperature is a little higher. For example, if the design temperature difference is 20 K, and the supply temperature rises only 0.5 K over the course of the day (a very reasonable value), the maximum flow rate in the late afternoon will be $\frac{20}{20-0.5} = 1.0256$ times, or 2.56%, higher than for the same load early in the day.

Most of the other factors that had an influence on the equipment sizing were rather peculiar to the conditions of this special system.

I did not use safety factors when sizing the equipment, because the design day load was known in advance, which is not given in reality. Hence in reality one would oversize the components.

4.8 Equipment Sizing with Detailed Data

If detailed load profiles are available, the equipment can be sized much more accurately, just by simulating the entire period of interest over and over again until an optimum set of equipment sizes has been determined. This will take much more time, but might

very well lead to significant savings in life cycle cost by sizing the equipment as small as possible.

Detailed and accurate load data may not always be available. In some cases, data recorded for one building might be used for another building that is considered comparable. Additionally, recorded data is already the result of calculations which involve assumptions, for example a load “measurement” that is really a measurement of mass flow rates, temperature and moisture differences.

If a building simulation system is used, the load can be calculated or at least estimated for every timestep, which allows accounting for the history, i.e. the conditions in previous timesteps. This approach requires a huge amount of data characterizing the building structure and location, shading effects, occupancy profile, ambient conditions and much more. TRNSYS is capable of this kind of simulation, but for the sake of time this approach was not used for this simulation.

In practice, it is cheaper to oversize equipment in order to prevent scenarios like in section 4.5, rather than not be able to maintain the desired air quality. Designing engineers will often rely on their experience and thumb rules like “x tons per square foot” to size equipment rather than on time consuming simulations.

Chapter 5

System Components and their TRNSYS Equivalents

This section describes the TRNSYS components that were used to create a model for the three kinds of systems investigated in this simulation. TRNSYS is a *Transient System Simulation* Software developed by the Solar Energy Laboratory (Klein et al., 1996), and it was designed especially to simulate the performance of systems involving solar applications, heating, cooling, energy storage, hydronics and controlling devices. TRNSYS uses a modular approach, connecting individual modules (“Types”) just as they would be connected in a real system. A huge library of TRNSYS components is readily available, and many of the components described below belong to the standard TRNSYS library.

5.1 Liquid Stratified Storage Tank - Type 60

The liquid storage tank is represented by TRNSYS type 60, a stratified fluid storage tank. The tank volume is divided into equally sized nodes, of which the tank allows a maximum of 100. The chilled liquid enters right above the tank bottom through a velocity reducing

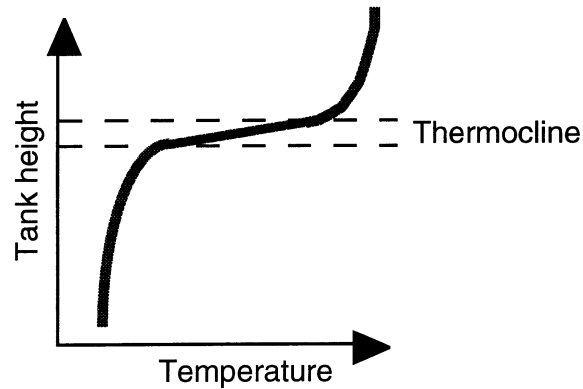


Figure 5.1: Schematic of the temperature profile in a stratified fluid storage tank

diffuser whose location is fixed. The tank model then adds the water to the node which is closest in temperature to the entering water. The high resolution of 100 is necessary to model the thermocline, which is the layer of water where the temperature rises from the chiller supply temperature (tank bottom) to the load return temperature (tank top). This thermocline is usually $0.3 \dots 0.45 \text{ m}$ thick, and since each node is assumed to have uniform temperature, the thickness of the nodes should be much less. With a maximum tank volume¹ of about 2000 m^3 and a height/diameter ratio of 1.5, the tank height $h_{\text{tank}} = 17.9 \text{ m}$, the node thickness becomes $h_{\text{node}} = 0.179 \text{ m}$.

Modeling the thermocline with only two nodes is not accurate. The consequence is that, due to the uniform temperature within each node and the fact that the thermocline moves up and down over a wide fraction of the tank volume, the thermocline will probably be larger than in a real tank. This in turn will cause the simulated tank to be larger than for a real system to supply water below a certain temperature. Unfortunately Type 60 could not be modified to allow for more nodes since the source code is not well documented. However, even in the largest tank a thermocline is developed², and for lower tank volumes the number of nodes in the thermocline is higher.

¹simulation result

²the tank outputs the temperature of each node

The tank is designed as a vertical cylinder made out of $12'' = 0.3048 \text{ m}$ of sealed concrete with a $4'' = 0.1016 \text{ m}$ polystyrene insulation and an overall heat loss coefficient of $U_{tank} = 0.0248 \frac{W}{m^2 K}$. The same value was assumed for the bottom and the sealed top. Since tank heat gains from the ambient play only a minor role in the system performance, due to the size of the tank and the large volume to surface area ratio, this simplification is reasonable.

5.2 Chiller - Type 68

The chiller employed, Type 68 (Cross, 1994), is a modification of the TRNSYS Type 53, and the main improvement is that Type 68 does not need a user-supplied external data file. Instead, the actual chiller power demand is calculated from the equation

$$\frac{P_{chil}}{P_{chil,des}} = \left[0.140 + 0.544 \left(\frac{Q_{chil}}{Q_{chil,des}} \right) + 0.316 \left(\frac{Q_{chil}}{Q_{chil,des}} \right)^2 \right] * [1 + 0.012 (T_{c,in} - T_{c,in,des}) - 0.015 (T_{e,out} - T_{e,out,des})] \quad (5.1)$$

where $T_{c,in}$ and $T_{c,in,des}$ are actual and design entering condenser water temperature, and $T_{e,in}$ and $T_{e,in,des}$ are actual and design leaving evaporator water temperature, all in $^{\circ}F$, with design values of $T_{c,in,des} = 85 \text{ }^{\circ}F (= 29.4 \text{ }^{\circ}C)$ and $T_{e,out,des} = 44 \text{ }^{\circ}F (= 6.7 \text{ }^{\circ}C)$, which were taken from ARI standard 550 (ARI, 1992). All temperature differences must be inserted in Fahrenheit, but the other variables may be in any unit system. It shall be mentioned that the ratios of actual over design power and capacity, respectively, have been assigned the variables PLR and QLR in section 4.3.4, LR standing for *load ratio*. The condenser water mass flow rate is set such that for the design capacity there will be a 6.0 K temperature rise of the condenser fluid (always water). In practice the condenser would be connected to a cooling tower, but to simplify and speed up the simulations no

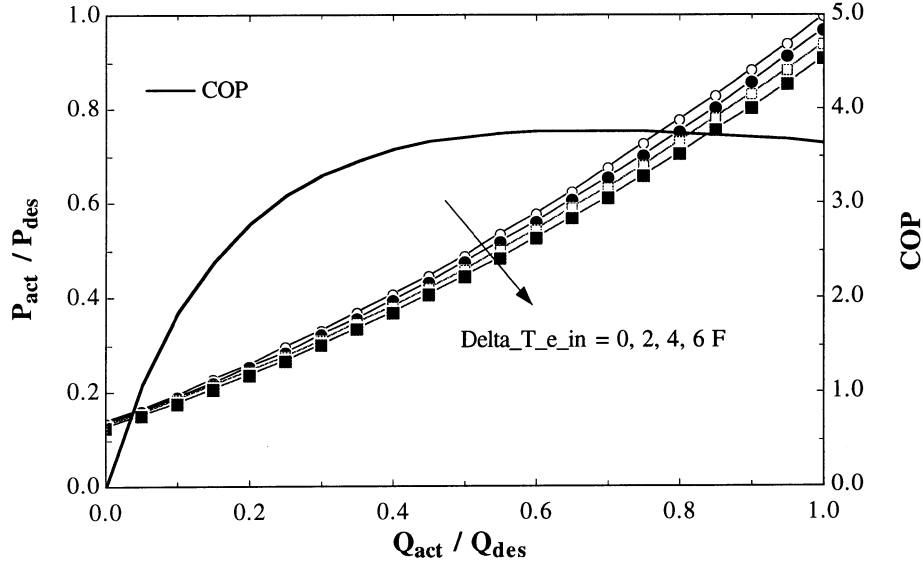


Figure 5.2: Chiller performance at ARI design conditions

such TRNSYS type was employed. Instead, the condenser water entering temperature $T_{c,in}$ was modeled with the ambient wet bulb temperature, a fixed approach of 4.0°C , and a fixed minimum for $T_{c,in}$ of 16.0°C .

$$\dot{m}_c = \frac{\dot{Q}_{chil,des}}{c_{p,w} * 6.0 K} \quad (5.2)$$

$$T_{c,in} = \max(T_{amb,wb} + 4.0 K, 16.0^{\circ}\text{C}) \quad (5.3)$$

The design chiller COP can be chosen in terms of $\dot{Q}_{chil,des}$ and $P_{chil,des}$. In order to compare the liquid systems with the ice system, where the harvester COP cannot be adjusted, the design chiller COP (under ARI conditions) was selected 3.976 (compare next section).

Figure 5.2 shows basic characteristics for such a chiller at design conditions.

5.3 Ice Storage Tank - Type 71

Type 71 was also written by Kevin Cross and features a simple effectiveness model for an ice storage tank. As in every storage component, the tank model keeps track of the tank content over time. Ice (created by an external ice harvester, see below) is produced by freezing water taken from the bottom of the tank and dumped into the opening on top. Coil supply water is taken from the bottom, and the return water enters at the top, which cools it down to 0°C and melts some fraction of the stored ice. The crucial output variable is the supply water temperature, which is calculated under the assumption of a simple tank effectiveness model: if the tank is charged by more than 20%, the supply temperature is 0°C, if it is charged by 20...0%, the supply temperature increases linearly with this fraction to the return water temperature (compare to section 1.2.2).

5.4 Ice Harvester - Type 72

The ice harvester by Cross freezes the water to ice which is then dumped into the open ice storage tank. Its performance has been determined in a detailed calculation (Cross, 1994) which was subsequently curve fit with one parameter being the ambient wet bulb temperature and the other being the design capacity. ARI design conditions are assumed to have been the basis for this curve fit like for the Type 68 chiller. No additional pump (Type 3) needs to be connected to it in the TRNSYS deck, following the example in Cross' thesis, the harvester input is directly connected to the ice storage tank.

The underlying curve-fit equation used to determine the actual harvester capacity \dot{Q}_{harv} is

$$\dot{Q}_{harv} = 0.0125 + (0.9035 - 0.0004658 * T_{wb} + 0.0005408 * T_{wb,des}) * \dot{Q}_{harv,des} \quad (5.4)$$

and for the actual harvester energy consumption P_{harv}

$$P_{harv} = 0.716 + (0.9469 + 0.009320 * T_{wb} - 0.01074 * T_{wb,des}) * \dot{Q}_{harv,des} - 0.0001389 * T_{wb,des}^2 \quad (5.5)$$

Both equations were taken from Cross' thesis. Units are Q in ton, P in kW, and T in °F. A design power consumption cannot be specified, and so the design COP cannot be altered. With the (final) sizes of the ice storage system (2200...3400 kW), the “design” COP is approximately 3.805.

Cross designed the harvester such that it could be turned either on or off, and limiting the harvester power consumption seemed not possible, which makes it unsuitable for a demand-leveling system.

One can imagine an ice harvester whose capacity can be decreased by limiting the number of heat exchanger plates the chilling fluid passes. This turned out to be easily implementable into Type 72. The (0...1) range of the control variable is now divided into 10 equal steps, and the control function that is input to the harvester is rounded up to the next common multiple of 1/10th. Each (now discrete) increase in the control variable can be associated with one more heat exchanger plate being added to the set of plates that is currently used for ice making. This way, the demand can be modulated.

The ice generation rate and power consumption are multiplied with the rounded-up control function. This assumes that the harvester COP is constant under varying load. which is not true, but a more sophisticated algorithm would have meant to derive a new curve fit. Additionally, if the harvester was not one single unit but a set of equal independent harvesters filling the same tank, they would behave exactly like this.

5.5 Cooling Coil - Type 90

The cooling coil is a new TRNSYS component that is based on the effectiveness-NTU method (Braun et al., 1989). The method had to be extended to include a wider range of conditions than the original development allowed. The method, its limitations and the applied extensions are described in the following section.

5.5.1 effectiveness-NTU Method

In this method a coil effectiveness is determined analogously to a simple heat exchanger by calculating an effectiveness from the number of transfer units (NTU) for the water side (NTU_w) and a comparable NTU for the air-water vapor side (NTU_a).

$$NTU_w = \frac{(UA)_w}{\dot{m}_w c_{p,w}} \quad \text{and} \quad NTU_a = \frac{(UA)_a}{\dot{m}_a c_s} \quad (5.6)$$

$$\text{with } (UA)_w = U_w * A_a \quad \text{and} \quad (UA)_a = U_a * A_a \quad (5.7)$$

Here, $(UA)_a$ is the product of the heat transfer coefficient U_a on the air side and the surface area A_a , and similarly $(UA)_w$ on the water side, but using the same area A_a .

The difference between the coil model and a simple heat exchanger lies in the definition of m^* . Instead of using the specific heat of air $c_{p,a}$, a saturation specific heat c_s is introduced:

$$m^* = \frac{\dot{m}_a c_s}{\dot{m}_w c_{p,w}} \quad (5.8)$$

$$\text{with } c_s = \left[\frac{dh_{a,sat}(T)}{dT} \right]_{T=T_w} \quad (5.9)$$

The saturation specific heat may be evaluated numerically, using the water inlet and outlet temperatures:

$$c_{p,s} \approx \left(\frac{h_{a,sat}(T_{w,out}) - h_{a,sat}(T_{w,in})}{T_{w,out} - T_{w,in}} \right) \quad (5.10)$$

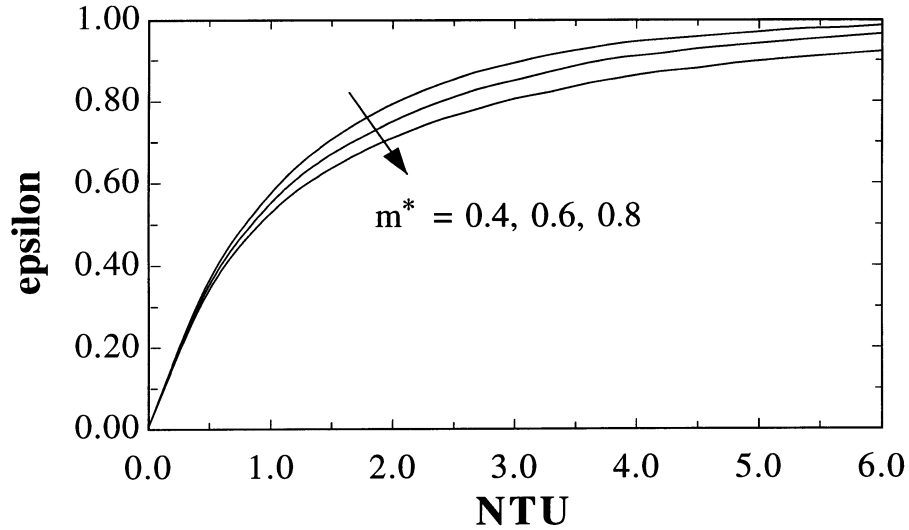


Figure 5.3: Coil effectiveness ε as function of NTU (for counterflow heat exchanger, using equation 5.12)

The saturation enthalpy is the enthalpy of saturated air-water vapor ($RH = 100\%$) at the corresponding temperature given in the brackets.

The two $NTUs$ are then combined to an overall NTU_{wet} , and using the effectiveness relations that can be found for all types of heat exchangers, the effectiveness ε can be determined³

$$NTU_{wet} = \frac{NTU_a}{1 + m^* \frac{NTU_a}{NTU_w}} \quad (5.11)$$

$$\varepsilon = \frac{1 - \exp(-NTU_{wet}(1 - m^*))}{1 - m^* \exp(-NTU_{wet}(1 - m^*))} \quad (5.12)$$

With ε calculated, the energy transfer rate in the coil can be expressed as a fraction of a certain maximum energy transfer rate:

$$\dot{Q} = \varepsilon \dot{Q}_{max} = \varepsilon \dot{m}_a (h_{a,in} - h_{a,sat}(T_{w,in})) \quad (5.13)$$

³In this simulation, the relation for a counterflow heat exchanger is used, which is reasonable for a coil with more than 6 rows.

The enthalpy $h_{a,sat}(T_{w,in})$ is the enthalpy of saturated air-water vapor ($RH = 100\%$) at water inlet temperature, which is the state that the air-water vapor can reach, at best.

5.5.2 Limitations of the effectiveness-NTU Method

The originally proposed effectiveness-NTU method for a coil (Braun et al., 1989) has some limitations which need to be removed in order to use this method for the current simulation. The limitations are described below:

Capacitance Ratio

In a simple heat exchanger, the variable corresponding to m^* is commonly called C^* , the capacitance ratio. It is defined as

$$C^* = \frac{C_{min}}{C_{max}} \quad (5.14)$$

where C_{min} and C_{max} are the smaller and the larger, respectively, capacitance rates of the two fluids in the heat exchanger. The capacitance rate is the product of mass flow rate and specific heat. Whichever product is smaller will be termed C_{min} , and the greater C_{max} .

In the effectiveness-NTU method for a coil, however, the air-water vapor is assumed to have the smaller capacitance ratio at all times, which is why m^* is generally less than unity and can be written as in equation 5.8. This requirement that m^* be less than unity turned out to be not fulfilled under all conditions that occurred during this simulation. For relatively low \dot{m}_a (compared to the design value), m^* would be above unity, thereby invalidating the effectiveness-NTU method. The following two paragraphs illustrate the states in the coil that may lead to $m^* > 1$:

In a variable volume system, a coil will be designed such that the heat transfer capacity on both the air and the water side, represented by the UA values, will be sufficient to cool

an air stream down to a certain setpoint temperature $T_{a,set}$. This will lead to a design capacitance ratio $m_{des}^* < 1$.

Under part-load conditions, a smaller mass flow rate of air is required to meet the given cooling load. The water flow rate is adjusted to maintain a fixed exiting air temperature $T_{a,set}$. If both \dot{m}_a and \dot{m}_w changed proportionally, m^* would not change and the requirement of $m^* < 1$ would always be met, but in fact, \dot{m}_w drops progressively with \dot{m}_a . To illustrate this, let us assume both \dot{m}_a and \dot{m}_w were reduced to half their previous value. Then, both corresponding NTUs and NTU_{wet} will double (equations 5.6 and 5.11). ε will rise (5.3) and cause $T_{a,out}$ to approach $T_{w,in}$ closer than before. If previously $T_{a,out}$ was *equal* to $T_{a,set}$, it will now be *below* it, and this means that less than half the previous \dot{m}_w is necessary to maintain $T_{a,out} = T_{a,set}$. This is the mechanism that causes m^* to rise under part load conditions, and as already stated, no settings for the UA values could be found to satisfy $m^* \leq 1$ under all operation conditions. The practical consequence of $m^* > 1$ can be that the water outlet temperature $T_{w,out}$ exceeds the air inlet temperature $T_{a,in}$ and thereby violates the second law of thermodynamics.

Artificial Humidification

The original paper by Braun does not refer to the humidity of the exiting air. The model provides information on how to calculate the outlet air temperature and enthalpy, and these two quantities plus the system pressure can be used to determine both the outlet air humidity ratio $w_{a,out}$ and relative humidity RH_{out} . If the inlet air is sufficiently dry, the original equations lead to a value for $w_{a,out}$ that is above the corresponding inlet humidity ratio $w_{a,in}$. In a stationary flow process, this would mean that additional water would have to be supplied to moisten the air, since no other source of humidity is available.

The reason for this kind of behavior probably has its roots in another assumption that was made in Braun's paper. The original equations were derived assuming a completely

wet coil, and most likely this is not given when artificial humidification occurs. In so far the model is not suitable for part load conditions, because a real coil will be partially dry and partially wet then.

5.5.3 Extended effectiveness-NTU Method

It is obvious that the model has to be extended to be suitable for these conditions.

Capacitance Ratio

The first step is to use the original definition of the capacitance ratio C^* (equation 5.14) instead of m^* to determine the effectiveness:

$$C^* = \min \left(\frac{\dot{m}_a c_s}{\dot{m}_w c_{p,w}}, \frac{\dot{m}_w c_{p,w}}{\dot{m}_a c_s} \right) = \min \left(m^*, \frac{1}{m^*} \right) \quad (5.15)$$

The NTU_{wet} , which is defined in equation 5.11 as

$$NTU_{wet} = \frac{NTU_a}{1 + m^* \frac{NTU_a}{NTU_w}} \quad (5.16)$$

is the overall NTU for a completely wet coil, and from the concept of NTUs it may be interpreted as the ratio of the overall heat transfer capacity over the minimum capacitance:

$$NTU_{wet} = \frac{(UA)_{overall}}{C_{min}} \quad (5.17)$$

For $C_{min} = \dot{m}_a c_s$, the NTU_{wet} can be calculated from equation 5.11. For the extended case of $C_{min} = \dot{m}_w c_{p,w}$, the NTU_{wet} must be corrected for the fact that the minimum capacitance rate is now the water side, and the correction factor is $\frac{\dot{m}_a c_s}{\dot{m}_w c_{p,w}}$, or simply m^* . Note that the m^* inside the formula is *not* replaced by C^* , as it results simply from the

algebra in the original derivation of the effectiveness-NTU method⁴. With little effort, equation 5.19 can be transformed into equation 5.20, which is more convenient to use.

$$CorrFac = \begin{pmatrix} 1 & \text{for } C_{min} = \dot{m}_a c_s \\ m^* & \text{for } C_{min} = \dot{m}_w c_{p,w} \end{pmatrix} \quad (5.18)$$

$$NTU_{wet} = \frac{NTU_a}{1 + m^* \frac{NTU_a}{NTU_w}} * CorrFac \quad (5.19)$$

$$= \frac{NTU_a}{1 + m^* \frac{NTU_a}{NTU_w}} * \frac{C_a}{C_{min}} \quad (5.20)$$

$$\varepsilon = \frac{1 - \exp(-NTU_{wet}(1 - C^*))}{1 - C^* \exp(-NTU_{wet}(1 - C^*))} \quad (5.21)$$

Also, equation 5.13 has to be modified, because \dot{Q}_{max} depends on which fluid has the minimum capacitance. After modification it reads

$$\dot{Q}_{max,a} = \dot{m}_a (h_{a,in} - h_{a,sat}(T_{w,in})) \quad (5.22)$$

$$\dot{Q}_{max,w} = \dot{m}_w c_{p,w} (T_{a,in} - T_{w,in}) \quad (5.23)$$

$$\dot{Q} = \varepsilon \dot{Q}_{max} = \varepsilon \min(\dot{Q}_{max,a}, \dot{Q}_{max,w}) \quad (5.24)$$

Artificial Humidification

To make the model's water mass balance consistent, the following corrections were applied to the equations for the case that $w_{a,out} > w_{a,in}$

$$w_{a,out}^* = w_{a,in} \quad (5.25)$$

$$T_{a,out}^* = T(w_{a,out}^*, Pressure, h_{a,out}) \quad (5.26)$$

⁴see equations 23 to 26 in (Braun et al., 1989)

Though disregarding the model prediction for $T_{a,out}$, the energy and mass balances are still satisfied. Another option would have been to simply apply equation 5.25 without equation 5.26, but then the energy balance on the air side would have been incorrect.

This correction is, however, just a patch and has no real physical foundation, and the model should be reworked to be suitable for (relatively) dry air inlet conditions and part loads.

The source code can be found in the appendix on page 134.

5.6 Air Conditioned Room - Type 92

This new TRNSYS type estimates the impact of a supply air stream on a room which is also exposed to some sensible and latent load. The room is characterized by its volume V , its average temperature T_r and humidity ratio ω_r . The air stream \dot{m}_a enters the room at supply conditions T_{sup} and w_{sup} , and the return air is assumed to be at the average room conditions. Also, it is assumed that the sensible and the latent load, \dot{Q}_{sens} and \dot{Q}_{lat} , of the room are known over time.

Type 92 keeps track of the room's energy and water balance over the entire simulation period. It also contains the option to iterate the air flow rate necessary to maintain a certain room temperature (under a known sensible and latent load and air supply conditions).

5.6.1 Simplifications and Assumptions

The room is modeled as one single volume of space filled with air, assuming no interior walls, furniture, occupants, etc. It is assumed to have thermal and humidity storage only due to the air that is filling its volume. The room air be always at uniform conditions and shall not interact with the walls, so there is no need to specify a geometry for the room.

The room model gives only a rough estimate (compared to a “real” building) of the state of the air in the volume, since many important real world impacts are neglected. It does, however, respond to a stream of cold supply air in a way very similar to a real room, which is why the model is useful for this simulation.

5.6.2 Derivation of Key Equations

The general first law energy balance for a room with one mass flow entering at some (T, ω) conditions and another one leaving the room at different conditions⁵ is

$$\frac{dU}{dt} = \dot{Q}_{sens} + \dot{Q}_{lat} + \dot{m}_{a,in}h_{a,in} - \dot{m}_{a,out}h_{a,out} \quad (5.27)$$

The above equation can be simplified, because in the case studied here the amount of air entering the room equals the amount of air leaving the room, $\dot{m}_{a,in} = \dot{m}_{a,out} = \dot{m}_a$, and the derivative of the room air mass m_a w.r.t. time, dm_a , is zero. Also, the air enters at supply conditions ($h_{a,in} = h_{sup}$) and exits at room conditions ($h_{a,out} = h_{room}$).

If the room conditions do not change over time (i.e. temperature and humidity ratio are constant), then the change in the room’s internal energy dU is zero. However, if the room conditions do change, this will affect the room’s internal energy, and $dU = m_a dh_{room} + h_{room} dm_a$ must be investigated. As shown before, dm_a vanishes in this case, and equation 5.27 becomes

$$m_a \frac{dh_{room}}{dt} = \dot{Q}_{sens} + \dot{Q}_{lat} + \dot{m}_a(h_{sup} - h_{room}) \quad (5.28)$$

For simulation purposes, this first order differential equation is now expressed with finite differences instead of differentials. The differential dt is replaced by Δt , which is $(t_{new} -$

⁵The room obviously does not change its height and velocity, thus there is no need to consider changes in potential and kinetic energies. The room volume is assumed constant, $dV = 0$, and changes in pressure are assumed negligible, $dP \approx 0$

t_{old}). Omitting the subscript $_{room}$ for better readability, the differential dh_{room} becomes the change in the average room enthalpy from t_{old} to t_{new} , which is $(h_{new} - h_{old})$. The air exits at room conditions, and its enthalpy is approximately the mean room enthalpy during the period Δt . This mean value is

$$h_{avg} = \frac{h_{old} + h_{new}}{2} \quad (5.29)$$

The energy balance equation then reads

$$\frac{m_a (h_{new} - h_{old})}{\Delta t} = \dot{Q}_{sens} + \dot{Q}_{lat} + \dot{m}_a (h_{sup} - h_{avg}) \quad (5.30)$$

An equation of similar structure is the room water mass balance:

$$\frac{m_a (w_{new} - w_{old})}{\Delta t} = \frac{\dot{Q}_{lat}}{\Delta h_{fg}} + \dot{m}_a (w_{sup} - w_{avg}) \quad (5.31)$$

where Δh_{fg} is the enthalpy of vaporization for water at T_{avg} . Again, the subscripts “new”, “old” and “avg” for the temperature and the humidity ratio refer to the room and are calculated as shown in equation 5.29. w_{avg} can be expressed with w_{new} and w_{old} , and further rearranging leads to an explicit solution for w_{new} :

$$\begin{aligned} m_a (w_{new} - w_{old}) &= \left(\frac{\dot{Q}_{lat}}{\Delta h_{fg}} + \dot{m}_a \left(w_{sup} - \frac{w_{old} + w_{new}}{2} \right) \right) \Delta t \\ \Leftrightarrow w_{new} &= \frac{\left(\dot{m}_a \left(w_{sup} - \frac{w_{old}}{2} \right) + \frac{\dot{Q}_{lat}}{\Delta h_{fg}} \right) \Delta t + m_a w_{old}}{m_a + \frac{\dot{m}_a}{2} \Delta t} \end{aligned} \quad (5.32)$$

Type 92 uses this equation to determine the new room humidity ratio.

Similar replacing and rearranging of equation 5.30 leads to an explicit solution for h_{new} :

$$m_a (h_{new} - h_{old}) = \left(\dot{Q}_{sens} + \dot{m}_a \left(h_{sup} - \frac{h_{old} + h_{new}}{2} \right) \right) \Delta t \quad (5.33)$$

$$\Leftrightarrow h_{new} = \frac{(\dot{m}_a (h_{sup} - \frac{h_{old}}{2}) + \dot{Q}_{sens}) \Delta t + m_a h_{old}}{m_a + \frac{\dot{m}_a}{2} \Delta t}$$

With w_{new} and h_{new} the temperature T_{new} can be determined.

The source code can be found in the appendix on page 140.

5.7 Controller - Type 99

The controller is the central system component and was designed exclusively for this simulation. Its task is to set all the valves (flow diverters, Type 11) and pumps (Type 3) in order to meet the load under the selected control strategy.

The cooling coil returns a desired mass flow rate of chilled water to the controller. The controller then determines how much of this load (in terms of mass flow rate) will be met by the tank (indirectly) and how much by the chiller (directly). This strongly depends on the selected storage medium and control strategy. For example, in full storage mode all of the on-peak load must be met with the tank alone, the chiller is not supposed to run (if the tank cannot meet the load, some component is not properly sized, and this is not necessarily the tank itself). In partial storage mode, the load is supposed to be met by tank and chiller simultaneously, and the proportion depends again on the selected strategy and the available tank and pump capacities.

The controller runs the chiller/harvester depending on the time of the day, the selected control strategy and the current tank charge status. It sets the tank charge pump such that the excess capacity of the chiller is used to charge the tank if needed.

In partial storage with tank priority, the controller basically follows the algorithm outline in section 4.6.3. This algorithm is comparatively simple and could be improved to increase the (economic) performance of the system in under this operation strategy, but this is beyond the scope of this project. The impact of a tank priority system with demand

leveling can be still be studied.

The controller buffers certain simulation parameters after each iteration. On the first call to the controller in each timestep ($\text{INFO}(7) = 0$), these buffered values are evaluated and warning and error messages are issued if necessary. The reason for this seemingly complicated setup is that in most cases only the final values, i.e. the values after convergence of all components has been achieved, are of interest. For example, the controller issues a warning message if the water supply temperature of the last timestep exceeds a user-specified limit. Without buffering, the same warning message would have been issued as often as the controller was called in this timestep (and the supply temperature was too high). With buffering, there will be only one warning message, which significantly reduces the amount of warnings. Also, the supply temperature might be too high in one iteration, but within range after the last iteration. In this case, checking the last timestep's value suppresses a confusing and misleading warning message that would occur otherwise.

Finally, the controller calculates the settings for the Type 11 flow diverters. In reality, the suction of the pumps located in the system would determine how much fluid leaves a tee-piece through exit one and how much through exit two. Not so in TRNSYS, because this violates the principle that the outputs are determined from the inputs and parameters using a set of equations. Plus, the mass flow rate given as input to a Type 3 pump is ignored by the pump, instead, the exiting mass flow rate is determined from a maximum flow rate and a control variable $\gamma = 0 \dots 1$. γ and the control variables for the flow diverters are both calculated in the controller.

The source code can be found in the appendix on page 142.

5.8 Other System Components

The pumps (Type 3) are assumed to convert 40% of the supplied electricity into thermal energy and to work against a pressure drop of 60 feet of water each. Since the system is fictitious, no real pressure drops are available, and therefore the basis head pressures from the equipment catalog were selected.

Each pump is then sized with the following equation:

$$P_{des} = \frac{\dot{m}_{max} \Delta P}{0.40 \rho_w} \quad (5.34)$$

An air fan was not included, since there would not be any difference between the systems. The flow diverters and tee-pieces (Type 11) are standard TRNSYS components which do not require sizing parameters. The same applies to the data reader (Type 9), the psychrometric calculator (Type 33), various integrators and printers. Friction and heat gains in pipes are neglected.

Chapter 6

Economics

This chapter provides a comparative economic analysis of the simulated systems. The differences are attributed to the operation strategy as well as the storage medium where possible. Principal advantages and disadvantages of one option over another are outlined. This is achieved by calculating the differences in installation and operation costs and comparing these values with the Life Cycle Savings method.

6.1 Investment costs

To determine differences in the initial investment, I compared the first costs of the major system components. This takes into account that there is for example a difference in the equipment sizes for similar systems (water vs. water-ethanol) as well as principle differences (liquid storage vs. ice storage). The main two sources of information on equipment cost for this chapter are the *Means* Catalog (Mossman, 1995), which is a compilation of mechanical cost data that is reviewed annually, and the Master's thesis of Kevin Cross (Cross, 1994), which has been used earlier. All curve-fits were done in EES. The investment cost analysis includes estimates for the chiller or harvester, a cooling tower,

the storage tank (either liquid or ice) and medium, all pumps shown in the simulation, and piping costs.

The unit of all cost variables is dollar, and all equipment sizes are given in British units. *Only within this chapter on economics*, the unit for all capacities is tons or HP (horse power), the tank size is measured in gallons, and lengths in feet. Helpful conversion factors are listed below:

$$1 \text{ kW} = \frac{1}{3.517} \text{ ton} \quad (6.1)$$

$$1 \text{ kW} = 1.341 * 550 \frac{\text{ft} - \text{lb}f}{s} = 1.341 \text{ HP ft} \quad (6.2)$$

$$1 \text{ m}^3 = \frac{1000}{3.74} \text{ gal} = 267.38 \text{ gal} \quad (6.3)$$

$$1 \text{ m} = 3.28 \text{ ft} \quad (6.4)$$

All dollar values are total values including overheads and profits.

6.1.1 Chiller for Liquid Systems

The chiller used for the water and water-ethanol system is a centrifugal chiller, as modeled in Type 68. *Means* provides cost data for centrifugal liquid chillers, water cooled, not including a cooling tower, with sizes ranging from 200 to 1500 tons for the packaged hermetic version (page 243). A curve fit of this data reads:

$$C_{chil} = 99791.78 + 92.37633 * \dot{Q}_{chil,des} + 0.1020788 * \dot{Q}_{chil,des}^2 \quad (6.5)$$

where $\dot{Q}_{chil,des}$ must be inserted in tons.

6.1.2 Cooling Tower

The cooling tower was modeled not as one unit in TRNSYS but by specifying the entering condenser water temperature as $T_{c,in} = \max(T_{amb,wb} + 4, 4)$ and the condenser mass flow rate \dot{m}_c such that the condenser temperature rise would be 6 K. It is a major cost factor in the investment cost analysis. Its capacity must be sufficient to allow the rejection of heat at the rate of the chiller's or harvester's capacity plus the rate of energy consumed by the chiller under design conditions. Therefore, the tower size varies with the chiller size, even if all systems meet the same load. If the chiller operates at its design COP, the cooling tower design capacity is approximately

$$\dot{Q}_{tower,des} = \dot{Q}_{chil,des} * \left(1 + \frac{1}{COP_{des}}\right) \quad (6.6)$$

This is a good enough estimate to calculate initial costs, and this formula was applied in the economic analysis. *Means* cost data for gear-driven cooling towers (page 255) was curve-fit to the equation below between the given data points to provide relative cost values (\$/ton), which were then multiplied with the design tower size in the calculation spread sheet.

$$C_{tower/ton} = 56.88463 - 0.002035702 * \dot{Q}_{tower,des} \quad (6.7)$$

6.1.3 Storage Tank

The simulation parameters are set for a sealed and insulated concrete tank for the liquid system (tank Type 60). *Means* offers cost data for a pre-stressed concrete ground storage tank (page 60), which is a similar type of tank. These cost values do not include pumps, piping and the diffuser system needed for stratification.

The ice storage tank uses Cross' tank model Type 71. In his thesis, Cross calculated

the cost for an ice storage tank with the pre-stressed concrete tank data from a previous edition of the *Means* catalog, and following this example C_{tank} is calculated with the same equation for both liquid and ice storage tanks.

The smallest size listed in the catalog is 250,000 gallons at a specific price of \$1/gal. This ratio was assumed to be valid for tanks smaller than 250,000 gallons, and the following (partially curve-fit) cost function was used:

$$\begin{aligned} C_{tank} &= 1 * V_{tank} \quad \text{if } V_{tank} < 250,000 \text{ gal} \\ C_{tank} &= 250,000 + 0.2226084 * V_{tank} \quad \text{if } V_{tank} \geq 250,000 \text{ gal} \end{aligned} \quad (6.8)$$

with V_{tank} in gallons. For the liquid systems, the tank cost were then increased by 3% to account for additional expenses for the diffuser.

6.1.4 Storage Medium

It is assumed that pure water is available at no cost and that the ethanol will have to be purchased. Calling different chemical companies in Wisconsin as well as nationally resulted in a price range of \$2.50 to \$5.00 per gallon of denaturated (non potable) ethanol of about 80% purity¹. Considering the required amount of ethanol, the price for a gallon of ethanol is assumed to be \$2.50, the lower end of the given spectrum. The total cost to fill each tank with 25 Vol.-% ethanol when the ethanol has a purity of 80% then calculated from

$$C_{ethanol} = V_{tank} * 0.25 / 0.80 * 2.50 \frac{\$}{gal} \quad (6.9)$$

¹Union Carbide Co., WI; DuPont Chemical Co.; Hydrite Co., WI

6.1.5 Pumps

The liquid system features three (physical) pumps. One circulates liquid through the chiller (pump 1 in TRNSYS), one charges and discharges the tank, and one circulates liquid through the cooling coil (pump 4). The tank pump is modeled with two pumps in TRNSYS (pump 2 and 3), because in TRNSYS a pump cannot be reverted (this would mean that the inlets become outlets and vice versa, which is not possible in TRNSYS). Another way to model the tank pump would have been to involve only one pump and a set of pipes connected to a reversing valve (which would have been a new but simple TRNSYS type). This is the way the problem would most likely be solved in practice. However, the modeling was easier with two pumps, and the results are equivalent (pump 2 and 3 are of identical size and do never operate at the same time).

The ice system requires only the coil pump (pump 4). Ice tank and ice harvester form a unit that does not require an additional pump (Cross, 1994).

Out of the different pump types in the *Means* catalog I chose data for circulating, cast iron pumps with flange connection, close coupled with end suction and bronze impeller (page 168). The data is based on a 60 ft head and 1800 RPM. This data was curve-fit to the following equation:

$$C_{pump_i} = 1089.384 + 2.704156 * GPM_i - 0.0005539739 * GPM_i^2 \quad (6.10)$$

where $i = 1 \dots 4$. The GPM_i rate (gallons per minute) was calculated from the $\dot{m}_{max,i}$ rate, taking into account the fluid density and several conversion factors.

For the simulation I had previously assumed the pumps to have a 60% engine efficiency, which means that 40% of the energy supplied to the pump are converted into heat, increasing the temperature of the liquid. The catalog data assumes a 60 ft head and 1800 RPM, and with this information, a design GPM rate and the assumed motor efficiency it

is possible to estimate a pump HP that would match the simulation parameters closely.

Or, it is possible to estimate the catalog pumps' engine efficiency.

The relevant equation in the TRNSYS decks uses the same head of 60 ft to calculate the design pump power. It is unlikely that the catalog HP values will match my assumption of $\eta_{pump} = 0.6$, but it is common engineering experience that 0.6 is a reasonable value, and the catalog data lies within this range, too ².

The incremental difference of the pump design power to the catalog value will not influence the overall results and has been neglected.

6.1.6 Variable Frequency Drive

Pump 1 is always operating at design capacity or not at all, but the flow rate of pump 2/3 and pump 4 is modulated by the controller, and therefore a variable frequency drive was accounted for in the first cost for these pumps. The *Means* catalog offers cost data as a function of pump size (in HP), and I used the pump size calculated in TRNSYS with the following curve fit of catalog data ranging from 5 to 25 HP:

$$C_{VFD} = 2869.188 + 27.02623 * HP + 3.916302 * HP^2 \quad (6.12)$$

6.1.7 Piping

The estimate of piping cost can only be a rough cut value, since there exists no actual system layout. It was accounted for linear feet of piping only, and an arbitrary length

²Inserting the catalog data into the equation that describes its energy consumption,

$$P_{pump} = \dot{V} * \Delta P / \eta_{pump} \quad (6.11)$$

results in a motor efficiency of $\eta_{pump} = 0.525$ for a 10 HP pump at 350 GPM and 60 ft head, which is close to the assumed value of 0.6. However, there seems to be an error in the data for another HP-GPM pair: η_{pump} comes out to be 0.999 for a 15 HP pump at 1000 GPM. This value would imply an ideal pump that converts all input energy into kinetic and potential energy.

of 2,000 ft for the liquid systems and 90% of this value were assumed for the ice system, which has fewer components and a simpler setup.

With a design fluid speed of 10 ft/s, the calculated minimum inside pipe diameter ranked from 4.84 to 5.61 inch. A PVC pipe SDR 26 (max. 160 psi) with couplings and hangers (*Means* catalog, page 83) was selected. None of the PVC pipes is available with a 5 inch diameter, so 6 inch pipes were chosen for all systems, though the water-ethanol system and the ice system require only 5 inch pipes, which would have lowered installation cost for these systems. But since the fraction of piping cost over total installation cost is below 2% this does not affect the overall results. The pipe cost are \$24.50 per linear foot.

As a rule of thumb, insulation can be accounted for by adding 10% to the total mechanical (piping) cost (page 197). The cost for insulated pipes then become \$26.95.

6.2 Annual Operating costs

The annual operating costs include electricity costs and demand charges. Not included are maintenance cost, repairs, etc. Two different rate structures are applied, which are described in table 6.1. The first is based on a large customer rate offered by Wisconsin Power & Light, while the second is fictitious. The on-peak period starts at 9 a.m. and ends at 7 p.m., Monday to Friday (WP&L's rate is from 10 a.m. to 10 p.m.). Official holidays on weekdays with no on-peak period were not taken into account³.

The second (fictitious) utility rate structure is an extrapolation of the first, intended to favor systems that avoid high on-peak demands and on-peak energy consumption. The on-peak (off-peak) rate is higher (lower) than in the first structure, and the demand charges are twice as high.

³The ECW data show an occupancy profile similar to weekends on these days, because they were no regular work days.

	unit	rate #1	rate #2
electricity charge			
on-peak	¢/kW	3.225	4.000
off-peak	¢/kW	2.040	2.000
demand charge			
on-peak	\$/kW	5.75	11.50
off-peak	\$/kW	0.00	0.00

Table 6.1: Utility charges that are applied to the simulation

6.3 Life Cycle Analysis

I applied the $P_1 P_2$ method described by Duffie and Beckman⁴ to calculate Life Cycle Savings (LCS), an economic figure representing the savings in today's dollars that the use of one system produces over the use of a second system. If the LCS are negative, it means that the second system is better than the first⁵. The LCS are the difference in Life Cycle Cost (LCC) or equivalently in Net Present Value (NPV) of two systems, but they are easier to calculate since only the *differences* between two systems need to be known. The LCC calculated in this analysis include only part of the total cost. The LCC value is therefore meaningless, but the difference between two LCC values is the LCS.

6.3.1 Assumptions and Economic Parameters

The utility rates and other economic parameters are assumed to be valid over the entire study period of 20 years (repeatability assumption), which is commonly regarded as the best guess in case no estimates on future values are available (Canada et al., 1996). Also,

⁴(Duffie and Beckman, 1991)

⁵This is not the original definition of LCS made by Duffie and Beckman. The original LCS evaluated fuel cost savings and investment cost expenses when using a solar system, so the LCS would tell how much money could be saved by using the solar system. The current interpretation of LCS is different, because two separate systems are compared. This interpretation can be found in (Mitchell and Braun, 1995). The $P_1 P_2$ method can still be successfully applied here.

the building load is assumed be representative for all the years of the study period. The study period is $N_e = 20$ years. Further economic parameters include an inflation rate of $i = 5\%$, an effective mortgage interest rate of $m = 4\%$ at which capital for the initial investment or part of it can be obtained, a market discount rate of $d = 10\%$, which is the rate of return one would get if one invested money elsewhere on the market, and an amount of \$300,000 available as initial down payment. This down payment might include utility subsidies. The mortgage period is the same as the study period. No property taxes were assumed, as well as no salvage value.

6.3.2 $P_1 P_2$ Method

With the $P_1 P_2$ method, Life Cycle Cost can be calculated with the following equation (adapted⁶ from (Duffie and Beckman, 1991)):

$$LCC = P_1 C_{annual} + P_2 C_{first} \quad (6.13)$$

Note that the LCC do not represent the *total* cost for any of the systems, because many actual cost which are identical in all systems are *not* included. Only the difference between two LCC values is meaningful. The difference between two LCC values are the Life Cycle Savings (in this equation: savings when alternative 2 is chosen rather than alternative 1):

$$\begin{aligned} LCS &= LCC_1 - LCC_2 \\ &= P_1 (C_{1,annual} - C_{2,annual}) + P_2 (C_{1,first} - C_{2,first}) \end{aligned} \quad (6.14)$$

The factors P_1 and P_2 incorporate interest, inflation, discount rate, taxes, mortgage pay-

⁶The adaption mainly consists of a change in sign, since the original equations considered C_{annual} as savings (incoming cash flow), whereas the first cost were an outgoing cash flow. For the purpose of this study, both values are outgoing cash flows and bear the same sign

ments, etc. as appropriate. A detailed definition can be found in (Duffie and Beckman, 1991), page 476. With the assumptions made for this project, the factors can be calculated from

$$P_1 = PWF(N_e, i, d) \quad (6.15)$$

$$P_2 = D + (1 - D) \frac{PWF(N_e, 0, d)}{PWF(N_e, i, m)} \quad (6.16)$$

where PWF is a Present Worth Factor, as defined in (Duffie and Beckman, 1991), page 463, and D is the fraction of the initial investment that can be paid down.

6.4 Spread Sheet Setup

The entire economic calculation is performed in Microsoft Excel, Version 5.0 spread sheets. Four sheets contain the calculations for

1. equipment sizing for design day (repeated until steady-periodic state)
2. investment cost of considered equipment
3. annual operating cost
4. economic parameters, life cycle cost of considered equipment and life cycle savings

On all sheets except the first, cells are referencing other cells on previous sheets such that information regarding e.g. the size of a chiller need to be entered only once on one sheet, and changes will affect all subsequent sheets automatically.

6.4.1 Equipment Sizing for Design Day

The first sheet contains the accumulated results of the sizing process for 6 repetitions of the design day. The key inputs are

- the heat capacity $c_{p,w}$ and the density ρ_w , with the index w indicating water as well as water-ethanol in contrast to a for air
- the tank volume
- the pump sizes, where the size of pump 1 determines the chiller size (compare equation 4.4)

The other fields are either calculated or provide simulation results:

- $T_{sup,des}$ and $T_{sup,max}$ are the design and maximum occurring water supply temperatures to the coil
- $T_{ret,min}$, $T_{ret,max}$ and $T_{ret,avg}$ are the minimum, maximum and average occurring water return temperatures from the coil.
- $\Delta T_{des} \approx T_{ret,avg} - T_{sup,des}$ is the design temperature difference for the tank and the chiller
- $Q_{tank,sup,des} = V_{tank} \rho_w c_{p,w} \Delta T_{des}$, the design tank storage capacity. For the ice tank, the enthalpy of sublimation is considered by adding $V_{tank} * 920 \frac{kg}{m^3} * 334.995 \frac{kJ}{kg}$ to $Q_{tank,sup,des}$
- the energy density is calculated as $Q_{tank,sup,des}/V_{tank}$
- the minimum and maximum tank charge level, as defined in section 4.3.2
- the maximum on-peak demand (fraction of design chiller mass flow rate)
- $\dot{Q}_{ch,des} = \dot{m}_{ch,des} c_{p,w} \Delta T_{des}$, the design chiller capacity
- $P_{ch,des} = Q_{ch,des}/COP_{ch,des}$, the chiller power demand at design conditions
- QLR_{max} and PLR_{max} , as defined in section 4.3.4

- the maximum total power (chiller plus pumps) consumption during on-peak and off-peak period
- the total energy consumed during on-peak and during off-peak (integrated total power values) and the sum of these two

6.4.2 Investment Cost of Considered Equipment

The second sheet lists the investment cost of the considered equipment (i.e. those parts of the equipment that are not identical for all systems). It follows the equations described in section 6.1.

6.4.3 Annual Operating Cost

The third sheet calculates annual operating expenses for different rate structures. The rate structures, which are listed in table 6.1, are part of another sheet and can be selected conveniently by specifying “1” or “2” on the fourth sheet.

The monthly maxima of the on-peak total power consumption are listed for the simulation period (May 1st to September 30th, 1990), and with the demand rate from line one the total demand charges can be calculated.

The total energy consumption during on- and off-peak periods for the entire 5 months are multiplied with the respective on- and off-peak rates and then added to the total demand charges, forming the annual operating cost.

6.4.4 Economic Parameters, LCC and LCS

The parameters on the last sheet include study period, initial down payment, interest rate, mortgage rate, and discount rate. The factor P_1 and the LCC for the considered equipment are calculated with these parameters as described previously.

The Life Cycle Savings are the differences in Life Cycle Cost between two systems. Using the last table, they can be obtained for any combination of two systems as follows:

1. select the row of system A and move right until the column of system B is reached
2. the value found must be interpreted as “*value* are the Life Cycle Savings if system A is chosen over system B”. “ < 0 ” indicates that system B is superior to system A, on a cost basis. The LCS of B over A can be found by exchanging row and column (negative values were omitted for better readability).

Chapter 7

Simulation Results

The following chapter presents the simulation results for all ten systems. The output is summarized using the four kinds of spread sheets introduced in section 6.4. Most of the presented data speak for themselves and need no special comment. Conclusions can be drawn from comparing systems operating with the same storage medium, and from comparing systems operating with the same strategy. It is important not to take the numbers at their face value but to see them in the context of all assumptions that were explained previously, which I will try in the following sections.

Figure 7.1 shows the load that is met by the coil on the design day, which mainly consists of the building and the ventilation load. Also shown are the on-peak and the off-peak period with their peak loads (in kW) and integrated loads (over the respective period, in kWh). Table 7.1 lists the relevant values in SI and British units.

7.1 Equipment Sizes

As previously mentioned, the equipment sizes are determined using 6 repetitions of design day cooling load and weather data to reach a steady-periodic state. The final sizes are

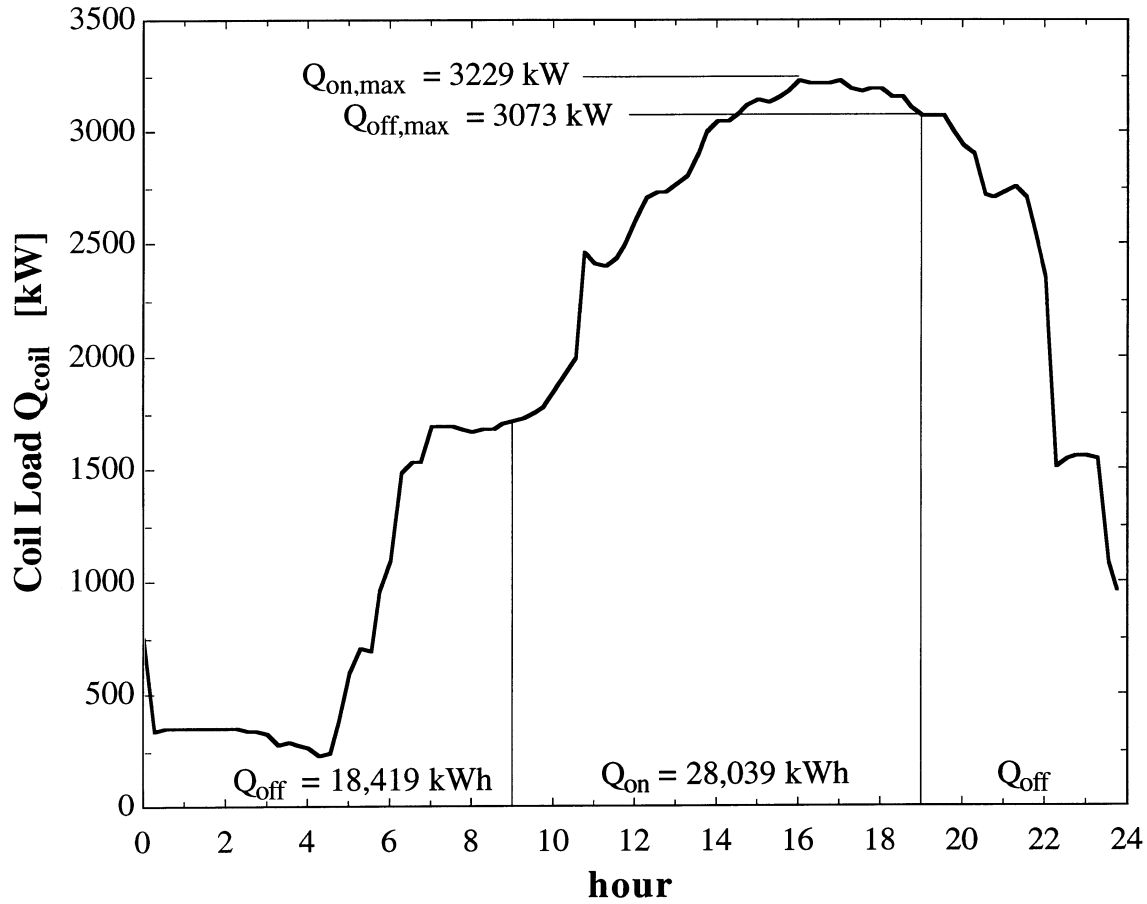


Figure 7.1: Coil Load on Design Day

summarized in table 7.2.

- Due to a lower ΔT_{des} , the pure water system has a bigger tank and pumps than the water-ethanol system, but the total energy consumption for the water-ethanol system is always higher.
- The full storage and the chiller priority systems were sized such that the tank would be discharged to a level of roughly 25%. More discharging would cause a rapid increase in tank discharge temperature, based on the storage tank representation in TRNSYS. With a higher resolution in modeling the thermocline these levels might

	Q_{coil}	$\dot{Q}_{coil,max}$
on-peak	28,039 kWh 7,972 ton-hr	3,229 kW 918 ton
off-peak	18,419 kWh 5,237 ton-hr	3,073 kW 873 ton

Table 7.1: Load and demand values for the design day on-peak and off-peak period

have turned out lower (thinner thermocline).

- The tank priority system could ideally be discharged down to a comparable tank level, but then the load prediction needs to be of much greater accuracy. The current prediction algorithm was designed to guarantee that the load could be met throughout the entire simulation period, and in fact the tank and chiller sizes resulting from the equipment sizing process had to be adjusted upwards in one case to make up for underestimated loads during the actual simulation¹
- As expected, the ice storage system has the highest energy storage density and requires the least storage volume. Due to the assumed tank performance curve the water supply temperature is always close to 0°C even at low tank levels, which in turn causes the required coil pump size to be slightly lower than for the water-ethanol system, which is supposed to supply at the same temperature, but where the tank models a non-ideal tank temperature profile.
- The water-ethanol system uses more energy than the pure water system, because the chiller must provide the medium at a lower temperature.

Figures 7.2 to 7.10 show the performance of the systems on the design day. They feature the coil load \dot{Q}_{coil} , which is identical for all systems, together with the change in internal

¹The design day data shows almost twice as high peaks than the previous day. The load estimation relies on the previous 24 hours' load values, and underestimation can occur easily. This can be avoided with more advanced load prediction.

Equipment Sizing for Design Day											
mode		water FS	water PS/CP	water PS/TP	water-eth. FS	water-eth. PS/CP	water-eth. PS/TP	ice FS	ice PS/CP	ice PS/TP	convent.
cp_w	kJ/kg K	4.19	4.19	4.19	4.00	4.00	4.00	4.19	4.19	4.19	4.19
rho_w	kg/m3	1,000	1,000	1,000	900	900	900	1,000	1,000	1,000	1,000
rho_ice	kg/m3	-	-	-	-	-	-	920	920	920	-
T_sup_des	C	5			0			0			5
T_sup_max	C	5.52	5.09	5.32	0.07	0.24	1.15	0.03	0.03	0.03	5.06
T_ret_min	C	21.37			21.96			21.98			21.40
T_ret_max	C	24.47			24.48			24.48			24.47
T_ret_avg	C	22.75			23.04			23.05			22.76
Delta_T_des	K	17.5	17.5	17.5	23	23	23	23	23	23	17.5
V_tank	m ³	1,800	600	1,300	1,500	550	1,100	450	180	300	-
Q_tank_sup_des	kWh	36,663	12,221	26,478	34,500	12,650	25,300	50,571	20,228	33,714	-
Energy Density	kWh/m3	20.4	20.4	20.4	23.0	23.0	23.0	112.4	112.4	112.4	-
Tank Level min/max	% / %	22 / 98	18 / 98	22 / 98	22 / 96	20 / 98	21 / 92	25 / 99	25 / 98	23 / 99	-
m_coil_des (4)	kg/hr	175,000	175,000	180,000	135,000	135,000	140,000	130,000	130,000	130,000	170,000
m_c/d_des (2=3)	kg/hr	180,000	100,000	180,000	140,000	80,000	140,000	-	-	-	-
m_ch_des (1)	kg/hr	180,000	100,000	155,000	140,000	80,000	110,000	-	-	-	170,000
max on-peak dmd	%	0	100	28	0	100	35	0	100	30	100
Q_ch_des	kW	3,666	2,037	3,157	3,578	2,044	2,811	3,400	2,000	2,700	3,463
P_ch_des	kW	922	512	794	900	514	707	894	526	710	871
QLR_max	-	1.06	1.09	1.07	1.18	1.20	1.20	1.13	1.13	1.13	0.75
PLR_max	-	1.04	1.06	1.05	1.02	1.04	1.04	1.01	1.01	1.01	0.75
P_max (onpeak)	kW	28	538	290	23	612	324	10	551	276	850
P_max (offpeak)	kW	1,011	576	879	1,088	632	865	923	547	735	804
W_onpeak (integr.)	kWh	233	5,124	2,661	189	5,823	3,003	87	5,169	2,537	7,091
W_offpeak	kWh	11,403	6,628	9,532	12,878	7,439	10,435	11,696	6,791	9,300	5,488
W_total	kWh	11,636	11,752	12,193	13,067	13,262	13,438	11,783	11,960	11,837	12,579
COP_ch_des	-	3.976									
COP_harv_des	-	3.805									

Table 7.2: Design Day Equipment Sizing. Values printed in a bold font are the important design parameters to size each system.

tank energy \dot{Q}_{tank} and the actual chiller capacity \dot{Q}_{chil} . The figures illustrate which device meets the load at what time.

7.2 Investment cost of considered equipment

Table 7.3 lists the investment cost of all items that are not common to all systems in order to assess the differences in investment cost. The main cost factors for all systems are the chiller or harvester, and the thermal storage tank. Figure 7.11 presents the same information graphically.

The water-ethanol storage system is more expensive than the pure water system, even

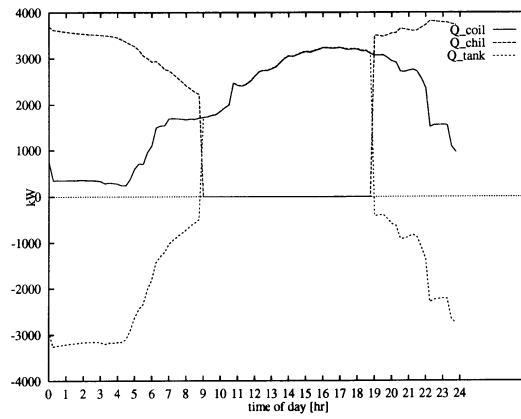


Figure 7.2: Mode 1 on design day: water, full storage

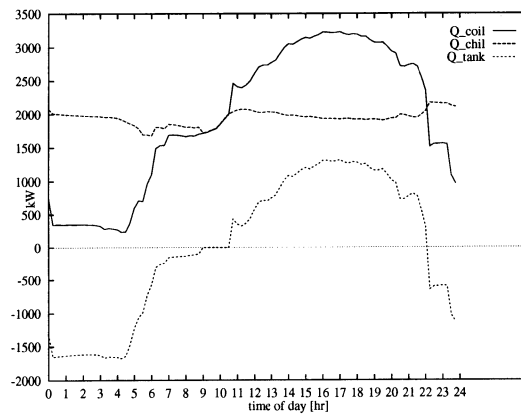


Figure 7.3: Mode 2 on design day: water, partial storage with chiller priority

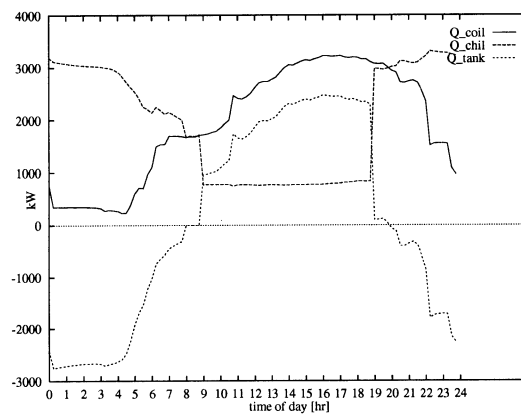


Figure 7.4: Mode 3 on design day: water, partial storage with tank priority

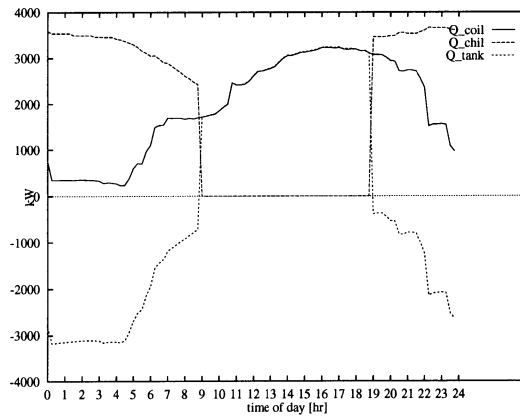


Figure 7.5: Mode 4 on design day: water-ethanol, full storage

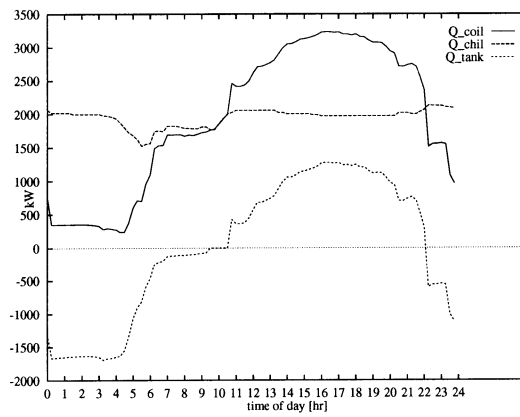


Figure 7.6: Mode 5 on design day: water-ethanol, partial storage with chiller priority

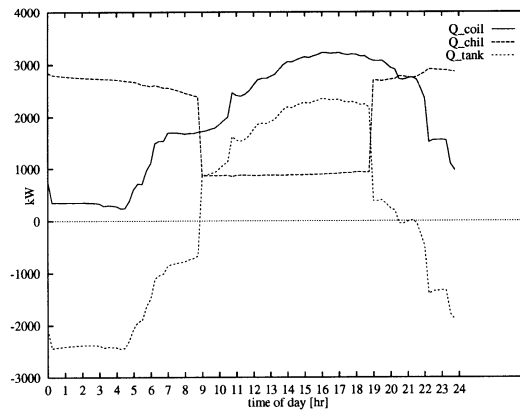


Figure 7.7: Mode 6 on design day: water-ethanol, partial storage with tank priority

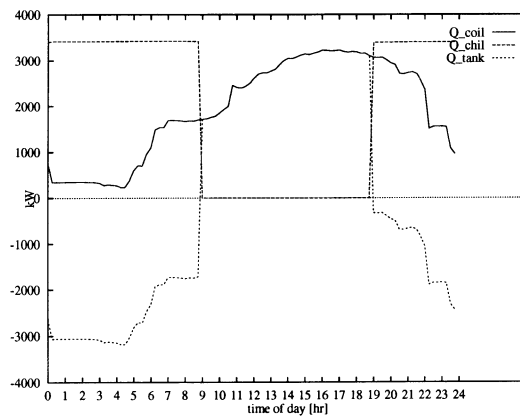


Figure 7.8: Mode 7 on design day: ice, full storage

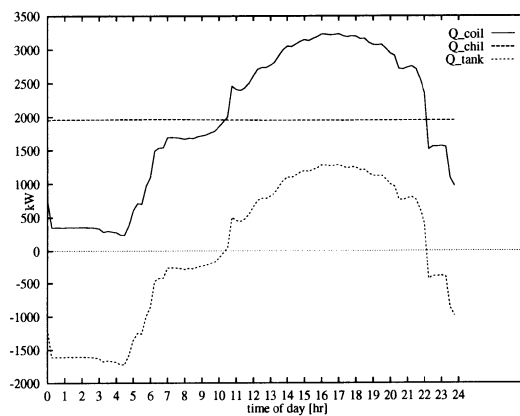


Figure 7.9: Mode 8 on design day: ice, partial storage with chiller priority

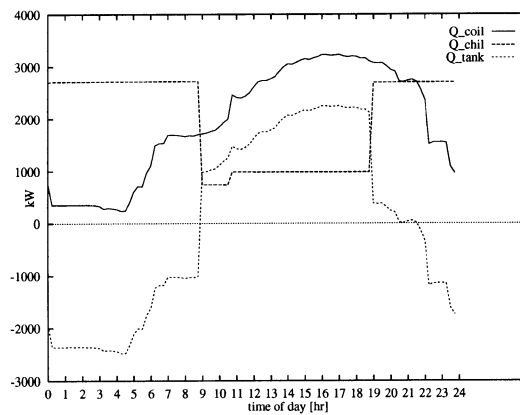


Figure 7.10: Mode 9 on design day: ice, partial storage with tank priority

Investment Cost of Considered Equipment											
mode		water FS	water PS/CP	water PS/TP	water-eth. FS	water-eth. PS/CP	water-eth. PS/TP	ice FS	ice PS/CP	ice PS/TP	convent.
Chiller design size	ton	1,042	579	898	1,017	581	799	967	569	768	985
Liquid Chiller	\$	307,015	187,526	264,967	299,402	187,984	238,842	0	0	0	289,682
Ice Harvester	\$	0	0	0	0	0	0	1,172,013	693,537	932,775	0
Cooling Tower size	ton	1,305	725	1,123	1,273	728	1,000	1,221	718	969	-
Cost per ton	\$/ton	54.23	55.41	54.60	54.29	55.40	54.85	54.40	55.42	54.91	-
Cooling Tower	\$	70,748	40,160	61,336	69,122	40,307	54,866	66,411	39,800	53,234	0
Tank Volume	gal	481,283	160,428	347,594	401,070	147,059	294,118	120,321	48,128	80,214	-
Thermal Storage Tank (3% added for liquid systems)	\$	367,852	165,241	337,199	349,460	151,471	324,937	120,321	48,128	80,214	0
Storage Medium	\$	0	0	0	313,336	114,890	229,779	0	0	0	0
Pump 1 design flow rate	GPM	802	446	691	693	396	545	-	-	-	758
Pump 1 (= chiller)	\$	2,902	2,184	2,693	2,698	2,074	2,398	0	0	0	2,820
Pump 2/3 design flow rate	GPM	802	446	802	693	396	693	-	-	-	-
Pump 2/3 design power	HP	20.3	11.3	20.3	17.5	10.0	17.5	-	-	-	-
Variable Freq. Drive for Pump 2/3	\$	5,024	3,669	5,024	4,542	3,531	4,542	0	0	0	0
Pump 4 design flow rate	GPM	780	780	802	668	668	693	579	579	579	758
Pump 4 (= coil)	\$	2,861	2,861	2,902	2,649	2,649	2,698	2,470	2,470	2,470	2,820
Pump 4 design power	HP	19.7	19.7	20.3	16.9	16.9	17.5	14.6	14.6	14.6	19.1
Variable Freq. Drive for Pump 4	\$	4,920	4,920	5,024	4,441	4,441	4,542	4,103	4,103	4,103	4,820
minimum req. pipe diameter	in	5.61	5.61	5.69	4.93	4.93	5.02	4.84	4.84	4.84	5.53
Piping Cost (incl. insulation)	\$/ft	26.95	26.95	26.95	26.95	26.95	26.95	26.95	26.95	26.95	26.95
assumed linear length	ft	2,000	2,000	2,000	2,000	2,000	2,000	1,800	1,800	1,800	1,500
Piping (linear footage only)	\$	53,900	53,900	53,900	53,900	53,900	53,900	48,510	48,510	48,510	40,425
Total First Cost	\$	815,222	460,462	733,044	1,099,550	561,247	916,505	1,413,828	836,548	1,121,306	340,567

Table 7.3: Investment cost of considered equipment

though the equipment is smaller. The reason is that for the water-ethanol system, a large amount of ethanol needs to be purchased.

The ice harvester shows a fairly major price difference to the liquid chillers, and it can be generalized that an ice harvester causes significantly higher investment cost. Other ice-making devices will be less expensive alternatives, which certainly will make the ice storage systems economically more favorable.

7.3 Operating cost of considered equipment

The annual operating cost for rate #1, which are given in table 7.4 as well as in figure 7.12, are a major influence factor on the economic desirability of any option, especially

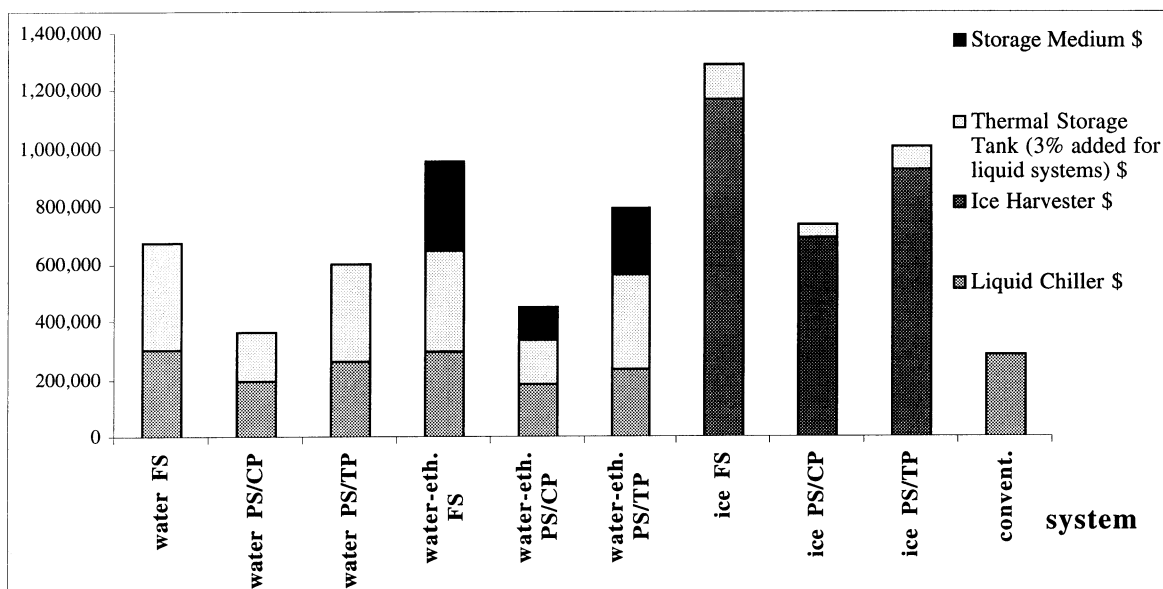


Figure 7.11: Investment cost of considered equipment

if the study period is long and the impact of the initial investment decreases. The same table for rate #2 can be found at the end of this chapter on page 126.

The rate structure is given in the first line (compare table 6.1). The following paragraphs refer to rate structure #1 (WP&L rate), unless stated.

- All full storage systems show very low annual cost. Since in full storage the chiller (harvester) does not operate during the on-peak period at all, only minor demand charges occur due to the pumps (lowest for the ice full storage system, because there is only one pump vs. three physical pumps in the other systems).
- High on-peak demand and high on-peak energy consumption are the reasons why all chiller priority systems are subject to higher annual cost than any other system with the same respective storage medium.
- Due to frequent part load operation, all tank priority systems consume more energy than the respective full storage and chiller priority systems. However, peak demand

Ann. Operating Costs @ rate #1		5.75 \$/kW			3.225 ¢/kWh on-pk.			2.040 ¢/kWh off-pk.			
mode		water FS	water PS/CP	water PS/TP	water-eth. FS	water-eth. PS/CP	water-eth. PS/TP	ice FS	ice PS/CP	ice PS/TP	convent.
P_max_on (May)	kW	11	323	188	9	366	185	4	463	461	362
P_max_on (Jun)	kW	19	557	216	15	622	210	7	533	135	601
P_max_on (Jul)	kW	26	569	233	21	630	223	10	566	210	787
P_max_on (Aug)	kW	23	615	230	18	674	222	9	560	142	703
P_max_on (Sep)	kW	28	578	739	23	640	761	10	550	736	849
total demand charges	\$	615	15,192	9,235	495	16,859	9,206	230	15,364	9,683	18,987
W_total on-peak	kWh	8,580	269,620	204,650	7,190	305,670	209,810	3,330	258,010	215,330	325,840
W_total off-peak	kWh	756,360	420,110	605,480	847,710	476,500	665,440	611,640	354,510	423,570	574,190
W_total	kWh	764,940	689,730	810,130	854,900	782,170	875,250	614,970	612,520	638,900	900,030
on-peak charges	\$	277	8,695	6,600	232	9,858	6,766	107	8,321	6,944	10,508
off-peak charges	\$	15,430	8,570	12,352	17,293	9,721	13,575	12,477	7,232	8,641	11,713
total annual cost	\$	16,322	32,457	28,186	18,020	36,437	29,547	12,815	30,917	25,268	41,208

Table 7.4: Annual operating cost of considered equipment (rate #1)

can clearly be reduced, compared to chiller priority, even with the simple tank priority control algorithm employed.

The monthly peak in the tank priority systems follows a pattern similar for all storage media: very moderate peaks in May though August are topped by (almost) design peak demand in September. This peak is a clear indicator that the load prediction algorithm failed to predict the correct load (it is actually the design day load on September 4). The demand level calculated during the sizing process, where identical data was used, is significantly lower than the peak demand during the actual simulation. Two possibilities exist that may have caused this peak:

1. The load was underestimated early in the day, and the controller had to set the high demand level to meet the afternoon load, or
2. The load curve contained a peak that was much higher than the corresponding value 24 hours earlier, and due to the employed algorithm the load was estimated to be very high and the demand level was set accordingly. If the high peak occurred late in the afternoon this effect will have been multiplied, since

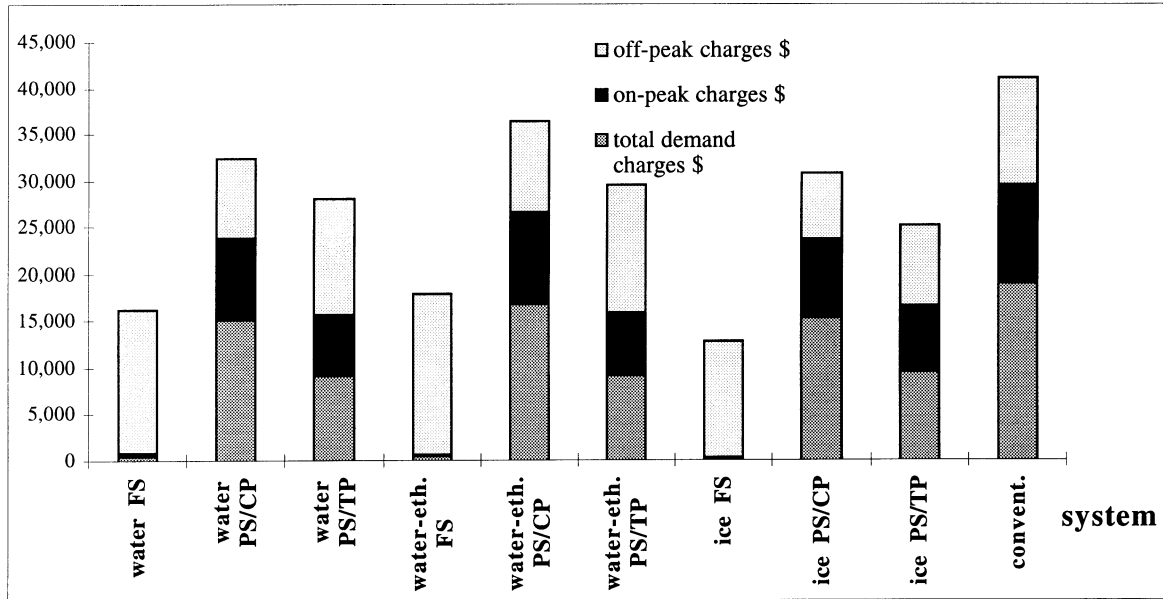


Figure 7.12: Annual operating cost of considered equipment (rate #1)

the algorithm divides the load estimate by remaining on-peak time and chiller design capacity to determine the necessary demand level.

The cost impact of this peak can be estimated for example for the water system by multiplying the demand charge with the increase in peak over the last month's level:

$$(739 - 230) \text{ kW} * 5.75 \text{ \$/kW} = \$2927 \quad (7.1)$$

even at the moderate conditions of rate #1 (\$5854 for rate #2).

- Another remarkable result is that *any* full storage appears to have lower operating costs than *any* other system, independent of the storage medium. This is again due to the absence of on-peak cost, indicating a principle advantage of full storage systems over partial storage systems (under the given conditions).

7.4 Life Cycle Savings

The results for rate #1 presented in figure 7.5 are subject to the choice of the economic parameters N_e , i , m , d and the initial down payment. It is however possible to identify general advantages of one system over another and to estimate the sensitivity of the outcome towards the economic parameters. The same table for rate #2 can be found at the end of this chapter on page 126.

Economic Parameters											
study period	yrs	20									
down payment	\$	300,000									
interest rate	%	5									
mortgage rate	%	4									
discount rate	%	10									
utility rate	#	2									
Life Cycle Cost (for considered equipment)											
mode		water FS	water PS/CP	water PS/TP	water-eth. FS	water-eth. PS/CP	water-eth. PS/TP	ice FS	ice PS/CP	ice PS/TP	convent.
P_1		12.11	12.11	12.11	12.11	12.11	12.11	12.11	12.11	12.11	12.11
annual cost	\$	16,322	32,457	28,186	18,020	36,437	29,547	12,815	30,917	25,268	41,208
D (fraction down)		0.37	0.63	0.41	0.27	0.53	0.33	0.21	0.36	0.27	0.88
initial cost	\$	815,222	477,132	733,044	1,099,550	561,247	916,505	1,413,828	836,548	1,121,306	340,567
P_2		0.62	0.78	0.65	0.57	0.72	0.60	0.53	0.62	0.56	0.93
LCC	\$	705,643	764,616	816,179	840,970	846,778	906,711	904,777	891,029	937,546	815,492
Life Cycle Savings of (row) over (col.)											
		water FS	water PS/CP	water PS/TP	water-eth. FS	water-eth. PS/CP	water-eth. PS/TP	ice FS	ice PS/CP	ice PS/TP	convent.
water FS	\$	0	58,972	110,536	135,326	141,135	201,067	199,134	185,385	231,903	109,849
water PS/CP	\$		0	51,563	76,354	82,162	142,095	140,161	126,413	172,930	50,876
water PS/TP	\$			0	24,791	30,599	90,531	88,598	74,849	121,367	
water-eth. FS	\$				0	5,808	65,741	63,807	50,059	96,576	
water-eth. PS/CP	\$					0	59,933	57,999	44,251	90,768	
water-eth. PS/TP	\$						0			30,835	
ice FS	\$						1,933	0		32,769	
ice PS/CP	\$						15,682	13,749	0	46,517	
ice PS/TP	\$									0	
conventional	\$			687	25,478	31,286	91,218	89,285	75,536	122,054	0

Table 7.5: Economic parameters, LCC and LCS (rate #1)

- The low operating cost for the full storage systems pay off in the LCS analysis, since all full storage systems save over both partial storage versions with the same

LCS of (row) over (col) in \$1,000	water FS	water PS/CP	water PS/TP	water-eth. FS	water-eth. PS/CP	water-eth. PS/TP	ice FS	ice PS/CP	ice PS/TP	conventional
water FS	0	59	111	135	141	201	199	185	232	110
water PS/CP		0	52	76	82	142	140	126	173	51
water PS/TP			0	25	31	91	89	75	121	
water-eth. FS				0	6	66	64	50	97	
water-eth. PS/CP					0	60	58	44	91	
water-eth. PS/TP						0			31	
ice FS						2	0		33	
ice PS/CP						16	14	0	47	
ice PS/TP									0	
conventional			1	25	31	91	89	76	122	0

Figure 7.13: Life Cycle Savings (rate #1)

medium. The water full storage system has LCS over any other system under both rate #1 and #2, and for the given economical parameters it is the optimum system.

- Under both rate structures the partial ice storage system has no LCS over any other system. This is due to the extremely expensive ice harvester and generally higher operating cost. However, if the initial cost could be reduced by 50%, e.g. by using another cheaper device for making ice, the ice storage system turns into an economical alternative: for rate #1, all ice storage variations produce LCS over any water-ethanol based system, and for rate #2, the ice full storage system saves over the water-ethanol full storage system, though not over the water full storage system².
- In direct comparison, each water-ethanol system is more expensive than the respective pure water based system. The savings in first cost due to slightly smaller equipment is more than offset by the cost for the ethanol itself, and the operat-

²this has been verified using the spread sheets

ing cost are higher due to the lower chiller supply temperature. This behavior is identical for both rate structures and a wide range of combinations of i , m , and d , and at this point additional safety measures are not yet considered. Except for minor savings in space due to a higher energy storage density of water-ethanol ($23.0 \frac{kWh}{m^3}$ versus only $20.3 \frac{kWh}{m^3}$ for pure water) there may be other advantages of using water-ethanol rather than pure water, i.e. cold air distribution.

- The conventional system, having the lowest first cost, is penalized by chiller operation below design capacity almost all the time, causing the highest operating cost of all systems. Still, for rate #1 it ranks second best after the water full storage system. This is because the differences in initial cost are so large (the difference to the second cheapest initial investment is roughly \$290,000, which is the 7-fold of the conventional system's annual operating expenses). The strong position of the conventional system is weakened if the mortgage rate is lowered (other systems will be penalized less for high investment cost) or if the study period is extended.
- Figure 7.1 shows that the on-peak period in effect is not optimal for the present load profile. The off-peak load is still significant, both in terms of integrated load and peak load. This decreases the (economical) performance of thermal energy storage (TES) systems since their ability to avoid using on-peak energy is not fully utilized. Large off-peak loads are often neglected in other analyses, when for example idealized load profiles are used.

Ann. Operating Costs @ rate #2		11.50 \$/kW			4.000 ¢/kWh on-pk.			2.000 ¢/kWh off-pk.			
mode		water FS	water PS/CP	water PS/TP	water-eth. FS	water-eth. PS/CP	water-eth. PS/TP	ice FS	ice PS/CP	ice PS/TP	convent.
P_max_on (May)	kW	11	323	188	9	366	185	4	463	461	362
P_max_on (Jun)	kW	19	557	216	15	622	210	7	533	135	601
P_max_on (Jul)	kW	26	569	233	21	630	223	10	566	210	787
P_max_on (Aug)	kW	23	615	230	18	674	222	9	560	142	703
P_max_on (Sep)	kW	28	578	739	23	640	761	10	550	736	849
total demand charges	\$	1,231	30,383	18,469	989	33,718	18,412	460	30,728	19,366	37,973
W_total on-peak	kWh	8,580	269,620	204,650	7,190	305,670	209,810	3,330	258,010	215,330	325,840
W_total off-peak	kWh	756,360	420,110	605,480	847,710	476,500	665,440	611,640	354,510	423,570	574,190
W_total	kWh	764,940	689,730	810,130	854,900	782,170	875,250	614,970	612,520	638,900	900,030
on-peak charges	\$	343	10,785	8,186	288	12,227	8,392	133	10,320	8,613	13,034
off-peak charges	\$	15,127	8,402	12,110	16,954	9,530	13,309	12,233	7,090	8,471	11,484
total annual cost	\$	16,701	49,570	38,765	18,231	55,475	40,113	12,826	48,139	36,451	62,490

Table 7.6: Annual operating cost of considered equipment (rate #2)

Economic Parameters		
study period	yrs	20
down payment	\$	300,000
interest rate	%	5
mortgage rate	%	4
discount rate	%	10
utility rate	#	2

Life Cycle Cost (for considered equipment)											
mode		water FS	water PS/CP	water PS/TP	water-eth. FS	water-eth. PS/CP	water-eth. PS/TP	ice FS	ice PS/CP	ice PS/TP	convent.
P_1		12.11	12.11	12.11	12.11	12.11	12.11	12.11	12.11	12.11	12.11
annual cost	\$	16,701	49,570	38,765	18,231	55,475	40,113	12,826	48,139	36,451	62,490
D (fraction down)		0.37	0.63	0.41	0.27	0.53	0.33	0.21	0.36	0.27	0.88
initial cost	\$	815,222	477,132	733,044	1,099,550	561,247	916,505	1,413,828	836,548	1,121,306	340,567
P_2		0.62	0.78	0.65	0.57	0.72	0.60	0.53	0.62	0.56	0.93
LCC	\$	710,236	971,890	944,305	843,527	1,077,360	1,034,682	904,912	1,099,620	1,072,988	1,073,262
Life Cycle Savings of (row) over (col.)											
		water FS	water PS/CP	water PS/TP	water-eth. FS	water-eth. PS/CP	water-eth. PS/TP	ice FS	ice PS/CP	ice PS/TP	convent.
water FS	\$	0	261,654	234,069	133,291	367,124	324,446	194,676	389,384	362,752	363,026
water PS/CP	\$		0			105,470	62,792		127,730	101,098	101,373
water PS/TP	\$		27,585	0		133,055	90,377		155,315	128,683	128,957
water-eth. FS	\$		128,363	100,778	0	233,833	191,155	61,385	256,093	229,461	229,735
water-eth. PS/CP	\$					0			22,260		
water-eth. PS/TP	\$					42,678	0		64,938	38,306	38,581
ice FS	\$		66,978	39,393		172,448	129,770	0	194,708	168,076	168,350
ice PS/CP	\$								0		
ice PS/TP	\$					4,372			26,632	0	275
conventional	\$					4,097			26,358		0

Table 7.7: Economic parameters, LCC and LCS (rate #2)

LCS of (row) over (col) in \$1,000	water FS	water PS/CP	water PS/TP	water-eth. FS	water-eth. PS/CP	water-eth. PS/TP	ice FS	ice PS/CP	ice PS/TP	conventional
water FS	0	262	234	133	367	324	195	389	363	363
water PS/CP		0			105	63		128	101	101
water PS/TP		28	0		133	90		155	129	129
water-eth. FS		128	101	0	234	191	61	256	229	230
water-eth. PS/CP					0			22		
water-eth. PS/TP					43	0		65	38	39
ice FS		67	39		172	130	0	195	168	168
ice PS/CP								0		
ice PS/TP					4			27	0	0
conventional					4			26		0

Figure 7.14: Life Cycle Savings (rate #2)

Chapter 8

Conclusions and Recommended Future Work

This chapter summarizes general conclusions that can be drawn from the results of this project and suggests areas where additional research or refinement is needed.

8.0.1 Conclusions

The use of water-ethanol provided no first cost or operating cost advantages over a pure water system under the given conditions. Refinement of the TRNSYS components towards better resemblance of a real system, such as more nodes to ensure proper modeling of the thermocline or inclusion of property functions into the components is unlikely to change this overall result (most of these changes would influence the simulation outcome of the water-based systems equally). Use of an ice harvester appears not useful economically, but the performance of ice storage systems based on different ice making devices will most likely lead to better results.

High demand charges penalize the use of on-peak energy and favor operation strategies that avoid using on-peak energy as much as possible, such as a full storage system. The

partial storage system, however, will be advantageous over the full storage system if the building's non-cooling load is taken into account and the chiller operates with reduced capacity during "shoulder hours" (compare figure 1.2, page 18).

To offset the incremental equipment cost over a conventional non-storage system the economical study period needs to be of sufficient length. There is always uncertainty about future energy prices and demand charges as well as other economic parameters.

8.0.2 Recommended Future Work

The thermophysical and transport properties of binary water-ethanol mixtures need to be measured over the investigated temperature and concentration range to ensure the validity of the approximating equations and figures in chapter 2.

The TRNSYS models do not make use of property functions in important devices such as chiller, coil, storage tank and pumps. For example, density and viscosity functions should be incorporated into the TRNSYS pump model, which would make the modeling of systems using highly viscous liquids such as ethylene-glycol more exact.

Using a different ice storage approach than harvester and open tank is clearly one of the next logical steps, since in practice ice storage systems installed today would rely on a cheaper alternative, too, which will influence the economics in favor of ice storage.

The effectiveness-NTU method should be extended to be capable of quickly predicting the air outlet state under any inlet conditions. The applied extensions and fixes can only be a start. The room model, combined with the externally provided building load data, should be replaced by a TRNSYS building model. This would eliminate the need for many assumptions made, allowing for thermal storage other than the air itself and for different zones inside the building.

The modeling of partial storage with tank priority requires more sophisticated ways to

predict future loads, and the tank model should be made suitable to support this kind of prediction, e.g. by providing a curve fit of the tank temperature profile. The chiller model should be equipped with a local controller (as in reality) that is capable of for example automatically limit the chiller power consumption to a certain level. As of today, most HVAC equipment is controlled locally by microprocessors, and the superposed (parent) control unit does not have to deal with equipment specific details.

A sensitivity analysis might prove useful for the many parameters that influence the dollar values in figures 7.5 and 7.7. The outcome also depends on the assumed building location which will influence the load profile.

Future energy cost and demand charges are uncertain, and assuming that they will remain at today's level is quick and convenient, however, there are several techniques available in engineering economics to predict future values on the basis of present and past data.

Appendix A

Nomenclature

All symbols used in chapter 2 can be found on page 52.

Roman Symbols

$c_{p,a}$	specific heat of air
$c_{p,s}$	saturation specific heat
$c_{p,w}$	specific heat of storage medium (water or water-ethanol)
C^*	capacitance ratio
C	capacitance, product of mass flow rate and specific heat
C	cost (in economic analysis)
COP	coefficient of performance, usually cooling capacity over power consumption
d	market discount rate
D	fraction of total initial cost that are paid down
GPM	value of pump flowrate in gallons per minute
h_{fg}	enthalpy of evaporation of water
h_{node}	height of each node in tank
h_{sub}	enthalpy of sublimation of water
h_{tank}	height of tank
h	enthalpy
HP	value of pump energy consumption in horse powers
i	interest rate
LCC	life cycle cost
LCS	life cycle savings
\dot{m}	mass flow rate
m^*	special capacitance ratio, see page 87
m	mortgage interest rate
m_a	mass of (dry) air in room (in Type 92)

N_e	length of study period for economic analysis
NTU	number of transfer units
P	power
PLR	chiller load ratio, based on power consumption
PWF	present worth factor
QLR	chiller load ratio, based on cooling capacity
Q_{day}	integrated coil load on the design day
Q_{est}	estimated future coil load
Q_{tank}	tank storage capacity
\dot{Q}_{bldg}	building load = energy flux into the building (read from data file)
\dot{Q}_{chg}	energy flux removed from tank during charging process
\dot{Q}_{chil}	chiller capacity
\dot{Q}_{coil}	coil load = energy flux transfered in the cooling coil
$\dot{Q}_{coil,lastday}$	same, but exactly 24 hours earlier
\dot{Q}_{dischg}	energy flux added to tank during discharging process
\dot{Q}_{lat}	latent part of building load \dot{Q}_{bldg}
$\dot{Q}_{pump,i}$	energy flux rejected by pump i to fluid
\dot{Q}_{sens}	sensible part of building load \dot{Q}_{bldg}
\dot{Q}_{vent}	ventilation load = energy flux due to exchange of room air with outdoor air
RH	relative humidity
RPM	rounds per minute
SHR	sensible heat ratio
t_{off}	hour of day when the off-peak period starts again and the on-peak period ends)
t_{on}	hour of day when the on-peak period starts (and the off-peak period ends)
T	temperature
$T_{a,set}$	temperature at which air is supposed to leave coil
$T_{c,in}$	condenser inlet temperature
$T_{e,out}$	evaporator outlet temperature
$T_{w,ret}$	temperature at which water returns from coil
$T_{w,set}$	temperature at which water is supposed to leave chiller
U_{tank}	overall tank heat transfer coefficient
(UA)	product of heat transfer coefficient U and area A effective for heat transfer
V	volume
w	humidity ratio

Greek Symbols

ΔP	pressure drop for pumps
ΔT	temperature difference
ΔU_{room}	change in internal energy of room for one timestep
ΔU_{tank}	change in internal energy of storage tank for one timestep
Δt_{off}	length of off-peak period
Δt_{on}	length of on-peak period
Δt	length of one timestep (0.25 hours in this simulation)
γ	control variable, usually $\gamma = 0 \dots 1$
ε	ice storage tank effectiveness
λ_{tank}	tank charge level, 1 = fully recharged, 0 = fully discharged
η_{motor}	pump motor efficiency
ρ	density

Common Indices

amb,wb	ambient wet bulb
amb	ambient
avg	average
a	air, air side
$chil$	chiller
des	design, means either “on the design day” or “design value”
e	ethanol
ice	ice
in	inlet, at inlet conditions
max	maximum
min	minimum
out	outlet, at outlet conditions
$peak$	peak, occurring at peak times or under peak conditions
ret	return
sat	saturation, at saturated conditions
set	setpoint value
sup	supply
VFD	variable frequency drive
$water$	water
w	water, water side

Appendix B

TRNSYS Modules Source Codes

This section contains only the source codes for Types 90 (cooling coil), 92 (conditioned room), and 99 (controller). The complete set of types can be found on the disk accompanying this thesis.

All types use a set of subroutines `checking.f` that provided the useful feature of printing user-specified lines of output from within a Type into a single file `*.chk`. This feature has only been tested under the Unix version of TRNSYS (Version 14.2). For use under DOS/Windows the appropriate lines should be removed (they can easily be identified).

Another important but not displayed file is `prop.f`, containing a function `PROP` that provides easy access to air property data. `PROP` is actually an interface to the TRNSYS procedure `PSYCH`.

B.1 Source Code Type 90

```
c      SUBROUTINE TYPE90(time,xin,out,t,dtdt,par,info,icntrl,*)
c      cooling coil, using effectiveness-NTU method
c      implicit none
c
c      trnsys variables
c      real time, t, dtdt, par(10), s
c      double precision xin(6), out(11)
c      integer info(15), icntrl, ptr, nstore, iav
c      common /store/ nstore,iav,s(5000)
```

```

c      variables for error handling
      integer dum, chk, err
      character filename*20
      logical check, check_it
      integer EI(3)
      real ET
      common /errorinfo/ EI, ET

c      local variables
      real T_A_ACC
      parameter (T_A_ACC = 0.01)
      integer MAX_IT
      parameter (MAX_IT = 30)

      integer mode, i, j
      double precision
      & T_a_in, w_a_in, m_a, T_w_in, m_w, T_a_set,
      & T_a_out, w_a_out, T_w_out,
      & epsilon, Q, SHR, Q_a, Q_w,
      & Q_sens, Q_lat,
      & cp_a, cp_w, h_a_in, h_a_out, h_a_out_des, h_a_x,
      & m_w_min, m_w_max, T_a_min, T_a_max,
      & dummy, press, store(MAX_IT,2)
      logical art_moist

c      statement / external functions
      double precision TSAT, WSAT, PWSAT, PROP
c      partial pressure of saturated air-water-vapor (Antoine Equation)
c      PSAT in atm, T in C; 1 atm = 1013.25 mbar
c      reference: Baehr, Thermodynamik, Springer Verlag
      PWSAT(TSAT) = 1 / 1013.25 * exp(19.0160-(4064.95/(TSAT+236.25)))
c      saturation value for humidity ratio
      WSAT(TSAT) = 0.622 * PWSAT(TSAT) / (1 - PWSAT(TSAT))

      if (info(7).LT.0) then
        info(9) = 0 ! time independent
        info(6) = 11 ! # outputs
        info(10) = 3 ! # storages
c      0: last timestep's T_a_out
c      1: last # iterations
c      2: last timestep's calculation result for m_w
        call TYPECK(-1,info,6,10,0)
        call init_checking(info, EI, filename)
      end if

c      ptr: index for location in s() array for storage over timesteps
      ptr = info(10)

      EI(1) = info(1)
      EI(2) = info(2)
      ET = time
      if (check) dum=chk('T:',dble(time))
      press = 1

c      inputs, parameters, initial values
      T_a_in = xin(1)
      w_a_in = xin(2)
      m_a = xin(3)
      T_w_in = xin(4)
      m_w = xin(5)
      T_a_set = xin(6)
      if (info(7).LT.0) s(ptr) = real(T_a_set)

      cp_a = par(4)
      cp_w = par(5)
      if (cp_w.LE. 0) dum=err('E:cp_w <= 0')
      mode = par(6)
      check = (par(7).GE.1)
      check_it = ((par(9).GE.1).AND.(check))
      m_w_max = par(10)

c      initial values for outputs
      T_a_out = T_a_set
      w_a_out = w_a_in
      T_w_out = T_w_in
      epsilon = 0
      SHR = 0
      Q = 0

c      general check if input data is reasonable
      if (w_a_in.GT. WSAT(T_a_in)) then
        dum=err('W:w_a_in > w_sat(T_a_in): assumed to be saturated')
        dum=chk('W:w_a_in',w_a_in)
        w_a_in = WSAT(T_a_in)

```

```

        dum=chk('W:w_sat',w_a_in)
    end if
    if (T_a_in .EQ. T_w_in) goto 30
    if (T_a_in .LT. T_w_in) then
        dum=err('W:T_a_in < T_w_in, coil bypassed')
        dum=chk('W:T_a_in',T_a_in)
        dum=chk('W:T_w_in',T_w_in)
        goto 30
    end if
    if (m_a .LT. 0) then
        dum=err('W:m_a < 0, reset to 0')
        dum=chk('W:m_a',m_a)
        m_a = 0
    end if
    if (m_w .LT. 0) then
        dum=err('W:m_w < 0, reset to 0')
        dum=chk('W:m_w',m_w)
        m_w = 0
    end if
c    first call each timestep
    if (info(7).eq.0) then
        if (s(ptr).gt.(T_a_set+0.1)) then
            dum=chk('W:T_a_out',dble(s(ptr)))
            dum=err('W:T_a_out was too high in last timestep')
        end if
        if (s(ptr+1).gt.MAX_IT) then
            dum=err('W:iteration for m_w did not converge last timestep')
        end if
        if (s(ptr+2).gt.m_w_max) then
            dum=err('W:coil calculated m_w > m_w_max last timestep')
            dum=chk('W:m_w_calc',dble(s(ptr+2)))
        end if
    end if
c    checking
    if (check) then
        dum = chk('T_a_in',T_a_in)
        dum = chk('w_a_in',w_a_in)
        dum = chk('m_a',m_a)
        dum = chk('T_w_in',T_w_in)
        dum = chk('m_w',m_w)
    end if

    select case (mode)
c    regular calculation: T_a_in, w_a_in, m_a, T_w_in, m_w are given
    case (1)
        call CALCULATION (time, info, par,
&         T_a_in, w_a_in, m_a, T_w_in, m_w,
&         T_a_min, w_a_out, T_w_out, epsilon, Q, SHR, art_moist)
c    find m_w with given T_a_out
    case (2)
        if ((m_a.EQ.0).OR.(T_a_in .LE. T_a_set)
&         .OR.(T_a_set .LE. T_w_in).OR.(m_w_max.LE.0)) then
            m_w = 0
            goto 25
        end if
c    Newton iteration starting from very low m_w
        m_w_min = 0
        T_a_max = T_a_in
        m_w = m_a * 0.1
        i = 1
c    Newton iteration block
20    continue
        call CALCULATION (time, info, par,
&         T_a_in, w_a_in, m_a, T_w_in, m_w,
&         T_a_out, w_a_out, T_w_out, epsilon, Q, SHR, art_moist)
        store(i,1) = m_w
        store(i,2) = T_a_out
        if (check_it) then
            dum=chk('i',dble(i))
            dum=chk('m_w',m_w)
            dum=chk('epsilon',epsilon)
            dum=chk('T_a_out',T_a_out)
            dummy = 0
            if (art_moist) dummy = 1
            dum=chk('art_moist',dummy)
            dum=chk('C:-----',dummy)
        end if
        if (abs(T_a_out-T_a_set).LT. T_A_ACC) goto 25
        if (i .gt. MAX_IT) then
            s(ptr+1) = real(i)
            if (.not. check_it) goto 25
            do 22, j=1, MAX_IT

```

```

                dum=chk('j',dble(j))
                dum=chk('W:m_w',store(j,1))
                dum=chk('W:T_a_out',store(j,2))
22          continue
            goto 25
        end if
c      new guess values for m_w
        m_w = (m_w-m_w_min) * (T_a_max-T_a_set) / (T_a_max - T_a_out)
        i = i + 1
c      goto 20
        end of iteration block

25      continue
c      store calculated value of m_w
        s(ptr+2) = m_w
c      if (m_w.GT. m_w_max) then
            reduce m_w and calculate T_w_out with m_w_max
            m_w = m_w_max
            call CALCULATION (time, info, par,
&              T_a_in, w_a_in, m_a, T_w_in, m_w,
&              T_a_out, w_a_out, T_w_out, epsilon, Q, SHR, art_moist)
            end if

            h_a_in      = PROP('TWH', T_a_in, w_a_in, press, time, info)
            h_a_out      = PROP('TWH', T_a_out, w_a_out, press, time, info)
            h_a_out_des = PROP('TWH', T_a_set, w_a_out, press, time, info)
            h_a_x        = PROP('TWH', T_a_in, w_a_out, press, time, info)
            Q_a = m_a * (h_a_in - h_a_out)
            Q_w = m_w * cp_w * (T_w_out - T_w_in)
            Q_sens = m_a * (h_a_x - h_a_out)
            Q_lat = Q_a - Q_sens

        case default
            dum=err('E:unknown mode of calculation requested')
            dum=chk('E:mode',dble(mode))
        end select

30      continue
c      outputs
        out(1) = T_a_out
        s(ptr) = T_a_out
        out(2) = w_a_out
        out(3) = m_a
        out(4) = T_w_out
        out(5) = m_w
        out(6) = T_a_set
        out(7) = epsilon
        out(8) = SHR
        out(9) = Q
        out(10) = Q_sens
        out(11) = Q_lat

c      checking if the user wishes to do so
        if (check) then
            dum = chk('T_a_out',T_a_out)
            dum = chk('w_a_out',w_a_out)
            dum = chk('T_w_out',T_w_out)
            dum = chk('m_w',m_w)
            dum = chk('epsilon',epsilon)
            dum = chk('Q',Q)
            dum = chk('# iterations',dble(i))
        end if

9999 return 1
end

c *****
SUBROUTINE CALCULATION (time, info, par,
&      T_a_in, w_a_in, m_a, T_w_in, m_w,
&      T_a_out, w_a_out, T_w_out, epsilon, Q, SHR, art_moist)

    implicit none

    real time, par(10), ET
    integer info(15), EI(3)
    double precision
    & NTU_a, NTU_w, UA_a, UA_w, UA_ratio, cp_a, cp_w,
    & T_a_in, w_a_in, m_a, T_w_in, m_w, T_a_out, T_w_out,
    & delta, cp_s, m_star, NTU, epsilon, h_a_ideal, w_a_ideal,
    & Q_wet_coil_max, Q_dry_coil_max, Q, h_surf_eff, T_surf_eff,
    & h_a_check, h_a_in, h_a_out, w_a_out, C_a, C_w, C_min, C_star,
    & Q_a_max, Q_w_max, SHR, press

    integer mode, error, dum, chk, err
    logical check_sub, art_moist

```

```

parameter (press = 1)

common /errorinfo/ EI, ET

c functions that are called
double precision PROP, H_FG
c statement functions
double precision TSAT, HSAT, PWSAT, WSAT, RHSAT, TT

c CORRELATION FOR SATURATION TEMPERATURE IN TERMS OF SATURATION
c ENTHALPY: CORRELATION IS IN SI UNITS AND IS GOOD FROM -50 KJ/KG
c TO 371 KJ/KG (-50 C TO 56 C).
c (CORRELATION IS FOR A TOTAL SYSTEM PRESSURE OF 1 ATMOSPHERE)
c valid for SI
TSAT(HSAT) = -5.922 + 0.674*HSAT - 4.662E-3*HSAT**2
& + 6.893E-6*HSAT**3 + 2.191E-7*HSAT**4 - 2.027E-9*HSAT**5
& + 7.896E-12*HSAT**6 - 1.473E-14*HSAT**7 + 1.075E-17*HSAT**8

c Enthalpy of airH2O at P=1 atm and RH=1.0 for T=-50 .. +56 C in kJ/kg
HSAT(TT) = 9.244 + 1.714*TT + 2.344E-2*TT**2 + 4.433E-4*TT**3
& + 5.075E-6*TT**4 + 4.194E-8*TT**5 + 7.897E-10*TT**6
& + 1.014E-11*TT**7

c partial pressure of saturated air-water-vapor (Antoine Equation)
c PSAT in atm, T in C; 1 atm = 1013.25 mbar
c reference: Baehr, Thermodynamik, Springer Verlag
PWSAT(TT) = 1 / 1013.25 * exp(19.0160-(4064.95/(TT+236.25)))

c saturation value for humidity ratio
WSAT(TT) = 0.622 * PWSAT(TT) / (1 - PWSAT(TT))

c RHSAT is not 1.0 to avoid trouble with PSYCH
RHSAT = 0.999

c m in kg/hr, UA in W/K, cp in kJ/kg-K
mode = par(1)
UA_a = par(2)
if (UA_a.LE. 0) dum=err('E:UA_a <= 0')
c U_ratio is the ratio of U_w over U_a
c UA_a and UA_w are both based on the area for air
UA_ratio = par(3)
if (UA_ratio.LE. 0) dum=err('E:UA_ratio <= 0')
UA_w = UA_a * UA_ratio
cp_a = par(4)
if (cp_a.LE. 0) dum=err('E:cp_a <= 0')
cp_w = par(5)
if (cp_w.LE. 0) dum=err('E:cp_w <= 0')
check_sub = .FALSE.
if ((par(8).GE.1).AND.(par(7).GE.1)) check_sub = .TRUE.

c factor 1000 for cp and 3600 for m to be in basic SI units
NTU_a = UA_a/(m_a/3600)/(cp_a*1000)
NTU_w = UA_w/(m_w/3600)/(cp_w*1000)

delta = 1
cp_s = (HSAT(T_w_in+delta)-HSAT(T_w_in-delta))/(2*delta)
C_a = m_a * cp_s
C_w = m_w * cp_w
C_min = min(C_a, C_w)
m_star = C_a/C_w
C_star = min(m_star, 1/m_star)

c NTU_wet will be written as NTU (shorter)
NTU = NTU_a/(1+m_star*NTU_a/NTU_w) * C_a / C_min

c effectiveness for different modes of operation
select case (mode)
case (1)
counterflow
if (C_star.EQ.1) then
epsilon = NTU/(1+NTU)
else
epsilon = (1-exp(-NTU*(1-C_star)))/
& (1-C_star*exp(-NTU*(1-C_star)))
end if
case (2)
parallel flow
epsilon = (1-exp(-NTU*(1-C_star)))/(1+C_star)
case (3)
crossflow, both fluids mixed
epsilon = 1-exp(NTU**0.22/C_star*(exp(-C_star*NTU**0.78)-1))
case (4)
crossflow, C_max (water) mixed and C_min (air) unmixed
epsilon = 1/C_star*(1-exp(-C_star*(1-exp(-NTU))))

```

```

case (5)
c   crossflow, C_max (water) unmixed and C_min (air) mixed
   epsilon = 1-exp(-1/C_star*(1-exp(-C_star*NTU)))
case default
   dum=err('E:unknown coiltype (1..5) requested')
   dum=chk('W:mode',dble(mode))
end select

if (epsilon.LT.0) then
   dum=err('W:data such that epsilon < 0 (has been set to 0)')
   dum=chk('W:epsilon',epsilon)
   epsilon = 0
end if
if (epsilon.GT.1) then
   dum=err('W:data such that epsilon > 1 (has been set to 1)')
   dum=chk('W:epsilon',epsilon)
   epsilon = 1
end if

h_a_in = PROP('TWH', T_a_in, w_a_in, press, time, info)
c   values of h and w at apparatus dew point ADP
h_a_ideal = HSAT(T_w_in)
w_a_ideal = WSAT(T_w_in)

c   calculate Q's assuming coil is either completely dry or wet
Q_wet_coil_max = m_a*(h_a_in-h_a_ideal)
Q_dry_coil_max = 0 * m_a*cp_a*(T_a_in-T_w_in)
c   maximum value is good approximation within engr. accuracy
c   *** always assume wet coil ***
c   determine which side can transfer the smaller amount of heat
Q_a_max = max((Q_dry_coil_max*0), Q_wet_coil_max)
Q_w_max = m_w * cp_w * (T_a_in - T_w_in)
Q = epsilon * min(Q_a_max, Q_w_max)

c   h_a_out from energy balance
h_a_out = h_a_in - Q / m_a
c   effective surface temperature and enthalpy
h_surf_eff = h_a_in + (h_a_out-h_a_in)/(1-exp(-NTU_a))
c   take curve fit value as initial guess
T_surf_eff = TSAT(h_surf_eff)
c   then match h_a_check with h_a_eff
c   (this procedure is necessary as PSYCH produces warnings if
c   it is called with over-saturated inputs)
20 h_a_check = PROP('TRH', T_surf_eff, RHSAT, press, time, info)
   if (check_sub) then
      dum=chk('T_surf_eff',T_surf_eff)
      dum=chk('h_a_check',h_a_check)
      dum=chk('h_surf_eff',h_surf_eff)
   end if
   if (abs(h_a_check-h_surf_eff) .GT. 0.01) then
      T_surf_eff = T_surf_eff - (h_a_check-h_surf_eff)/cp_s
      goto 20
   end if

T_a_out = T_surf_eff + (T_a_in-T_surf_eff)*exp(-NTU_a)

if (h_a_out .LT. HSAT(T_a_out)) then
   w_a_out = PROP('TWH', T_a_out, h_a_out, press, time, info)
else
   w_a_out = WSAT(T_a_out)
end if

c   check for artificial humidification and eliminate it
art_moist = .false.
if (w_a_out.gt.w_a_in) then
   w_a_out = w_a_in
   T_a_out = T_a_in - Q/m_a/cp_a
   art_moist = .true.
end if

if (Q.GT.0) then
   SHR = 1 - (w_a_in - w_a_out) * m_a
&   * H_FG((T_a_out+T_a_in)/2) / Q
   else
   dum=err('W:Q is lower than zero')
   dum=chk('W:Q',Q)
   Q = 0
   SHR = 0
end if

T_w_out = T_w_in
if (m_w.GT.0) T_w_out = T_w_in + Q/m_w/cp_w

c   checking
if (check_sub) then

```

```

      dum = chk(' cp_s',cp_s)
      dum = chk(' NTU_a',NTU_a)
      dum = chk(' NTU_w',NTU_w)
      dum = chk(' NTU_wet',NTU )
      dum = chk(' T_surf_eff',T_surf_eff)
      dum = chk(' m_star',m_star)
    end if

    return
  end

```

B.2 Source Code Type 92

```

      SUBROUTINE TYPE92(TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
c      air conditioned room
      implicit none

c      VARIABLES
c      trnsys variables
      real time, t, dttd, par(7), s
      double precision xin(7),out(12)
      integer info(15), icntrl, ptr, nstore, iav
      common /store/ nstore,iav,s(5000)
      real TIMEO,TFINAL,DELT
      integer IWARN
      COMMON /SIM/ TIMEO,TFINAL,DELT,IWARN

c      variables for error handling
      integer dum, chk, err
      character filename*20
      logical check
      integer EI(3)
      real ET
      common /errorinfo/ EI, ET

c      local variables
      double precision T_new, T_old, w_new, w_old, T_avg, w_avg,
&      T_max, w_max, T_sup, w_sup, T_set, w_set,
&      h_new, h_old, h_avg, h_sup, h_max,
&      h_x,
&      Q_sens, Q_lat, Q_tot, SHR, Q_sens_act, Q_lat_act,
&      Q_tot_act, m_a,
&      delta_w, delta_E1, delta_E2,
&      V_room, rho_a, M_a_room, h_fg2, press
      integer mode, i

      integer MAX_RUN
      parameter (MAX_RUN = 30)
      parameter (press = 1)

c      functions that are called
      double precision PROP, H_FG

c      statement functions

      if (info(7).LT.0) then
        info(9) = 1 ! time dependent
        info(6) = 12 ! # outputs
        info (10) = 4 ! # storage variables
        call TYPECK(-1,info,7,7,0)
        call init_checking(info, EI, filename)
      end if

      ptr = info(10)
      EI(1) = info(1)
      EI(2) = info(2)
      ET = time
      if (check) dum=chk('T:',dble(time))

c      inputs, parameters, initial values
      T_sup = xin(1)
      w_sup = xin(2)
      m_a = xin(3) !currently ignored

```

```

c      Q's come in kWh and are converted to kJ/hr
      Q_tot = xin(4) / DELT * 3600 ! kWh/hr*3600 = kJ/hr
      SHR = xin(5)
      Q_sens = xin(6) / DELT * 3600 ! kWh/hr*3600 = kJ/hr
      Q_lat = xin(7) / DELT * 3600 ! kWh/hr*3600 = kJ/hr

      mode = int(par(1))
      T_set = par(2)
      w_set = par(3)
c      on first call, store initial room T and w to s(ptr+2/3)
c      2/3 is storage for last timestep's values
      if (info(7).LT.0) then
        s(ptr+2) = real(T_set)
        s(ptr+3) = real(w_set)
      end if
c      on first call in each timestep, save 0/1 values into 2/3
c      0/1 is storage for last iteration's values
      if (info(7).EQ.0) then
        s(ptr+2) = s(ptr+0)
        s(ptr+3) = s(ptr+1)
      end if
      V_room = par(4)
      rho_a = par(5)
      h_fg2 = par(6)
      check = .FALSE.
      if (par(7) .GE. 1) check = .TRUE.

      M_a_room = V_room * rho_a
c      in every call, read last timestep's values from 2/3
      T_old = dble(s(ptr+2))
      w_old = dble(s(ptr+3))
      h_sup = PROP('TWH', T_sup, w_sup, press, time, info)
      h_old = PROP('TWH', T_old, w_old, press, time, info)
      h_x = PROP('TWH', T_old, w_sup, press, time, info)

      select case (mode)
c      case (1)
        Load specified as Q_tot and SHR
        Q_sens = Q_tot * SHR
        Q_lat = Q_tot * (1-SHR)
c      case (2)
        Load specified as Q_sens and Q_lat
        Q_tot = Q_sens + Q_lat
        if (Q_tot.NE.0) then
          SHR = Q_sens / Q_tot
        else
          SHR = 0
        end if
      case default
        dum=err('E:unknown mode of calculation requested')
      end select

      if (check) then
        dum = chk('T_sup',T_sup)
        dum = chk('w_sup',w_sup)
        dum = chk('m_a',m_a)
        dum = chk('T_old',T_old)
        dum = chk('w_old',w_old)
      end if

c      maximum values if m_a = 0
      w_max = Q_lat / H_FG(T_old) * DELT / M_a_room + w_old
      h_max = Q_tot * DELT / M_a_room + h_old
      T_max = PROP('WHT', w_max, h_max, press, time, info)

c      initial guess for m_a
      if (h_x .LE. h_sup) then
        m_a = 0
        T_new = T_old
        w_new = w_old
        h_new = h_old
        dum=chk('W:room does not need sensible cooling',dble(0))
        goto 20
      else
        m_a = Q_sens / (h_x - h_sup)
      end if

      i = 1
c      10 continue
      leave iteration if i > MAX_RUN
c      if (i.gt.MAX_RUN) then
        dum=err('W:iteration did not converge')
        dum=chk('W:m_a',m_a)
        dum=chk('W:T_new',T_new)

```



```

        goto 20
    end if

    w_new = ((m_a*(w_sup-w_old/2) + Q_lat/H_FG(T_old)) * DELT
&         + w_old*M_a_room) / (M_a_room + m_a/2 * DELT)

    h_new = ((m_a*(h_sup-h_old/2) + Q_tot) * DELT + M_a_room * h_old)
&         / (M_a_room + m_a/2 * DELT)

    T_new = PROP('WHT', w_new, h_new, press, time, info)

c    leave iteration if T_new close to T_set
    if (abs(T_new-T_set).LT.0.01) goto 20

c    new value for m_a
    m_a = m_a * (T_max-T_set) / (T_max-T_new)
    i = i + 1
    goto 10

20   continue
c   check: change in mass of water (kg) and in energy (kJ) for room
    T_avg = (T_new+T_old)/2
    w_avg = (w_new+w_old)/2
    delta_w = (Q_lat/H_FG(T_avg) - m_a*((w_old+w_new)/2-w_sup))*DELT
    h_avg = PROP('TWH', T_avg, w_avg, press, time, info)

    h_x = PROP('TWH', T_avg, w_sup, press, time, info)
    Q_sens_act = m_a * (h_x - h_sup)
    Q_lat_act = m_a * (h_avg - h_x)
    Q_tot_act = Q_sens_act + Q_lat_act

    delta_E1 = (Q_tot - Q_tot_act) * DELT
    delta_E2 = M_a_room * (h_new - h_old)

40   continue
c   outputs
    out(1) = T_avg
    out(2) = w_avg
    out(3) = m_a
    s(ptr+0) = T_new
    s(ptr+1) = w_new
    out(4) = T_new
    out(5) = w_new
    out(6) = Q_sens_act
    out(7) = Q_lat_act
    out(8) = Q_tot_act
    out(9) = delta_w
    out(10) = delta_E2
    out(11) = h_new
    out(12) = h_avg
    if (check) then
        dum = chk('# it',dble(i))
        dum = chk('T_avg',T_avg)
        dum = chk('w_avg',w_avg)
        dum = chk('T_new',T_new)
        dum = chk('w_new',w_new)
        dum = chk('delta_w (kg)',delta_w)
        dum = chk('Q_sens',Q_sens)
        dum = chk('Q_sens_act',Q_sens_act)
        dum = chk('Q_lat',Q_lat)
        dum = chk('Q_lat_act',Q_lat_act)
        dum = chk('Q_tot',Q_tot)
        dum = chk('Q_tot_act',Q_tot_act)
        dum = chk('delta_E1',delta_E1)
        dum = chk('delta_E2',delta_E2)
    end if

    return 1
end

```

B.3 Source Code Type 99

SUBROUTINE TYPE99(TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)

```

c      controller
      implicit none

c      *****
c      VARIABLES
c      *****
c      trnsys variables
      real time, t, dt, par(22), s
      double precision xin(5), out(14)
      integer info(15), icntrl, ptr, nstore, iav
      common /store/ nstore, iav, s(5000)
      real TIMEO, TFINAL, DELT
      integer IWARN
      COMMON /SIM/ TIMEO, TFINAL, DELT, IWARN

c      variables for error handling
      integer dum, chk, err
      character filename*20
      logical check
      integer EI(3)
      real ET
      common /errorinfo/ EI, ET

c      local variables
      double precision
      &      T_a_set, T_w_in, Tcap, Tcap_des, T_w_set, T_w_max,
      &      T_w_out,
      &      Q_coil, Q_ch_des,
      &      m_w, cp_w, hour, onpeak_hr, offpeak_hr, dt_on,
      &      pump(5), m_max(5), y(60:64),
      &      Tank_lvl, Lvl_high, Lvl_low,
      &      dummy, rho_w, V_tank, Est_Cap, Avl_Cap, factor,
      &      QLR_reset, QLR_min, QLR, QLR_max, Needed_Cap,
      integer i, mode, medium, strategy, slot, pumpscheme
      integer startday, day, Saturday, Sunday, MONTH(12)
      logical onpeak, weekend, weekendrate, charge_tank,
      &      dmd_leveling

      double precision press, h_sub_w
      parameter (press = 1)
      parameter (h_sub_w = 334.994) ! kJ/kg

      DATA MONTH/744,1416,2160,2880,3624,4344,5088,5832,6552,
      .       7296,8016,8760/

c      statement functions
      double precision m
      m(i) = pump(i) * m_max(i)

c      *****
c      INITIAL CALL
c      *****
      if (info(7).LT.0) then
        &      if ((1/DELT).ne.INT(1/DELT))
        &      dum=err('E:This timestep cannot be used with Type 99')
        info(9) = 1 ! time dependent
        info(6) = 14 ! # outputs
        info(10) = 11+(24/DELT) ! # storage variables
        0 = last timestep's value for Tank_cap
        1 = last time when tank was fully charged
        2-5 = storage for pump warnings (0 or 1)
        6 = last timestep's value for T_w_in
        7,8,9 = currently unused
        10 = QLR_max (maximum demand level)
        11..11+(24/DELT)-1 = last 24 hr's Load
        call TYPECK(-1,info,5,22,0)
        call init_checking(info, EI, filename)
      end if

c      *****
c      INPUTS, PARAMS, INITIAL VALUES, CONSTANTS
c      *****
      ptr: index for location in s() array for storage over timesteps
      ptr = info(10)
      if (TIME.eq.TIMEO) then
        s(ptr+1) = TIME
        s(ptr+10) = QLR_min
      end if

c      error/checking variables
      EI(1) = info(1)
      EI(2) = info(2)
      ET = time
      if (check) dum=chk('T:',dble(time))

```

```

c      inputs
c      Q_coil means "energy used in this timestep", so it is kW * hr
c      Q_coil = xin(1) * DELT          ! in kWh
c      T_w_in = xin(2)
c      m_w = xin(3)
c      if (m_w.LT. 0) then
c          dum=err('W:m_w < 0, reset to 0')
c          dum=chk('W:m_w',m_w)
c          m_w = 0
c      end if
c      Tcap is available tank energy (liquid system) in kWh
c      or available Tank ice mass (ice system) in kg
c      Tcap = abs(xin(4))
c      T_w_out = xin(5)

c      parameters
c      T_a_set = par(1)
c      T_w_set = par(2)
c      T_w_max = par(3)
c      Lvl_high = par(4)
c      if (Lvl_high.gt. 1) dum=err('E:Lvl_high must be <= 1')
c      Lvl_low = par(5)
c      if (Lvl_low.gt. Lvl_high)
c          & dum=err('E:Lvl_high must be < Lvl_low')
c      if (Lvl_low.lt. 0) dum=err('E:Lvl_low must be >= 0')
c      cp_w = par(6)
c      m_max(1) = par(7)
c      if (m_max(1).LT. 0) dum=err('E:m_max(1) is < 0')
c      m_max(2) = par(8)
c      if (m_max(2).LT. 0) dum=err('E:m_max(2) is < 0')
c      m_max(3) = m_max(2)
c      m_max(4) = par(9)
c      m_max(5) = par(10)
c      if (m_max(5).LT. 0) dum=err('E:m_max(5) is < 0')
c      check = (par(11).ge. 1)
c      mode = par(12)
c      Tcap_des = par(13)      ! kWh or kg
c      rho_w = par(14)        ! kg/m^3
c      V_tank = par(15)        ! m^3
c      onpeak_hr = par(16)
c      offpeak_hr = par(17)
c      dt_on = offpeak_hr - onpeak_hr
c      startday = 0 indicates that first day of simulation is a
c      Sunday, 1=Monday, etc. Simulation is always assumed to
c      start at midnight of this day.
c      startday = mod(int(par(18)), 7)
c      weekendrate = (par(19).ge.1)
c      QLR_min = par(20)
c      Q_ch_des = par(21) / 3600    ! kJ/hr into kW
c      QLR_reset = par(22)

c      determine hour, day, onpeak
c      hour = 0 indicates hour from 0 to 1 a.m., etc.
c      hour = dmod(dble(time), dble(24.0))
c      day = int(time/24)
c      Sunday = mod(7-startday, 7)
c      Saturday = mod(6+Sunday, 7)
c      weekend = .false.
c      dum = mod(day,7)
c      if ((dum.EQ. Saturday).OR.(dum.EQ. Sunday)) weekend = .true.
c      onpeak = .true.
c      if ((hour.ge.offpeak_hr).or.(hour.lt.onpeak_hr)
c          & .or.((weekend).and.(weekendrate))) onpeak = .false.

c      mode determines the controller strategy
c      0,1,2,3 = water; 4,5,6 = water-ethanol; 7,8,9 = ice
c      1,4,7 = full storage; 2,5,8 = PS/CP; 3,6,9 = PS/TP
c      select case (mode)
c      case(0,1,2,3)
c          medium = 1
c      case(4,5,6)
c          medium = 2
c      case(7,8,9)
c          medium = 3
c      case default
c          dum=err('E:wrong calculation mode')
c      end select
c      strategy = mode + 3 - medium*3

c      pump(1) = 0

```

```

pump(2) = 0
pump(3) = 0
pump(4) = 0

c *****
c FIRST CALL OF EACH TIMESTEP
c *****
if (info(7).GT.0) goto 100

c 1- check for Tank_lvl at beginning of each timestep if mode > 0
if (mode.gt.0) then
c   s(ptr) = 1: tank can be charged this timestep
   charge_tank = (Tcap .lt. (Tcap_des*Lvl_high))
   s(ptr) = 0
   if (charge_tank) s(ptr) = 1
   Tank_lvl = Tcap / Tcap_des
c   store current time if tank is fully charged right now
   if (Tank_lvl .GE. Lvl_high) then
     s(ptr+1)=time
   end if

c   issue warning if onpeak starts and tank is not fully recharged
   if ((hour.eq.onpeak_hr).and.(Tank_lvl.lt.(Lvl_high*0.97))
&     .and.(s(ptr+1).lt.(hour-DELT))) then
     dum=err('W:Tank is not fully recharged at start of onpeak')
     dum=chk('W:hour',hour)
     dum=chk('W:last_full',dble(abs(s(ptr+1))))
     dum=chk('W:Tank_lvl',Tank_lvl)
   end if
end if

c 2- check if there has been a pump warning in last timestep
if (s(ptr+1 + 1) .gt. 1) then
  dum=err('W:Pump 1 was too small in PREVIOUS timestep.')
  dum=chk('W:pump(1)',dble(s(ptr+1 + 1)))
  s(ptr+1 + 1) = 0
end if
if (s(ptr+1 + 2) .gt. 1) then
  dum=err('W:Pump 2 was too small in PREVIOUS timestep.')
  dum=chk('W:pump(2)',dble(s(ptr+1 + 2)))
  s(ptr+1 + 2) = 0
end if
if (s(ptr+1 + 3) .gt. 1) then
  dum=err('W:Pump 3 was too small in PREVIOUS timestep.')
  dum=chk('W:pump(3)',dble(s(ptr+1 + 3)))
  s(ptr+1 + 3) = 0
end if
if (s(ptr+1 + 4) .gt. 1) then
  dum=err('W:Pump 4 was too small in PREVIOUS timestep.')
  dum=chk('W:pump(4)',dble(s(ptr+1 + 4)))
  dum=err('E:Calculation halted (Pump 4 too small).')
  s(ptr+1 + 4) = 0
end if

c 3- special things for p.s.w/tank priority
if (strategy.eq.3) then
c   store load value in s() array
   slot = 11 + hour/DELT      ! slot = position for this hour
   factor = (Q_coil / max(s(ptr+slot),1.))*0.5
c   factor = max(factor,-8.)
c   factor = min(factor, 8.)
   if (check) dum=chk('factor',factor)
   s(ptr+slot) = Q_coil

c   reset demand level
   if (QLR_reset.eq.-1) then
c     units of months assumed
     do 3, i=1,12
       if (TIME.ge.MONTH(i) .and. (TIME-DELT).lt.MONTH(i)) then
         s(ptr+10) = QLR_min
         QLR_max = dble(s(ptr+10))
         if (check) dum=chk('QLR_max now',QLR_max)
         goto 3
       end if
3     continue
   else
c     units of hours assumed
     QLR_reset MUST be multiple of timestep
     if (mod(TIME,QLR_reset).eq.0) then
       s(ptr+10) = QLR_min

```

```

                QLR_max = dble(s(ptr+10))
                if (check) dum=chk('QLR_max now',QLR_max)
            end if
        end if
    end if

    if ((strategy.eq.3).and.((TIME-TIME0).gt.24).and.onpeak) then
        Est_Cap = 0 ! est'd capacity demand
c      estimated integrated coil load (kWh)
        do 2, i = slot, 11+(offpeak_hr/DELT)
            Est_Cap = Est_Cap + s(ptr+i)
2      continue
        Est_Cap = Est_Cap * factor
        if (medium .eq. 3) then
            Avl_Cap = Tcap_des * (Tank_lvl - Lvl_low) * h_sub_w / 3600
        else
            Avl_Cap = Tcap_des * (Tank_lvl - Lvl_low)
        end if
        Needed_Cap = Est_Cap-Avl_Cap
        QLR = min (Needed_Cap / Q_ch_des / (offpeak_hr-hour), 1.)
        if ((QLR.gt.QLR_max).and.(hour.le.(offpeak_hr-1))) QLR_max=QLR
        s(ptr+10) = real(QLR_max)
    end if

100 continue

c      *****
c      START OF CALCULATION
c      *****

        charge_tank = (s(ptr) .eq. 1)
        Tank_lvl = Tcap / Tcap_des
        if (T_w_in .ge. T_w_max) s(ptr+7) = 1
        QLR_max = dble(s(ptr+10))
        if ((time.LT.(TIME0+24)).and.(strategy.eq.3)) then
c          first 24 hours: no load data yet, use PS/CP instead of PS/TP
            mode = mode -1
            strategy = 2
        end if

c      *****
c      set pumps

        pump(4) = m_w / m_max(4)
        if (pump(4) .ge. 1) s(ptr+1 + 4) = pump(4)

        include 'modes.f'
        include 'pumps.f'

c      *****
c      set valve positions. Zero is default value
        y(61) = 0
        y(62) = 0
        y(64) = 0
        if (medium.LE.2) then
c          liquid storage modes only have valves
            dummy = m(2) + m(4) - m(3)
            if (pump(1) .GT. 0) y(61) = dummy / m(1)
            if (dummy .GT. 0) y(62) = m(2) / dummy
            if (m(4) .GT. 0) y(64) = m(3) / m(4)
        end if

c      *****
c      outputs
        out(1) = T_w_in
        s(ptr+6) = T_w_in
        out(2) = m_w
        out(3) = T_a_set
        out(4) = y(61)
        out(5) = y(62)
        out(6) = y(64)
        out(7) = pump(1)
        out(8) = pump(2)
        out(9) = pump(3)
        out(10) = pump(4)
        out(11) = Tank_lvl
        out(12) = 0
        if (onpeak) out(12) = 1
        out(13) = factor
        if (strategy .eq. 3) then
            out(14) = QLR_max

```

```

else
  out(14) = 0
end if

c *****
c checking
if (check) then
  if (info(7).LT.0) then
    dum = chk('m_max(1)',m_max(1))
    dum = chk('m_max(2)',m_max(2))
    dum = chk('m_max(3)',m_max(3))
    dum = chk('m_max(4)',m_max(4))
  end if
  if (info(7).EQ.0) then
    dum = chk('hour',hour)
    dum = chk('last full charge',dble(s(ptr+1)))
    dum = chk('Q_coil kWh',Q_coil)
    dum = chk('strategy',dble(strategy))
    dum = chk('medium',dble(medium))
    dum = chk('Q_ch_des',Q_ch_des)
    dum = chk('Tcap_des',Tcap_des)
    dum = chk('Est_Cap',Est_Cap)
    dum = chk('Avl_Cap',Avl_Cap)
    dum = chk('Weeded_Cap',Weeded_Cap)
    dum = chk('factor',factor)
    dum = chk('T_w_in',T_w_in)
    dum = chk('QLR_max',QLR_max)
  end if
  dum = chk('call #',dble(info(7)))
  if (onpeak) dum = chk('C:onpeak',dummy)
  if (charge_tank) dum = chk('C:chg_tank',dummy)
  dum = chk('m_w',m_w)
  dum = chk('pumpsc.',dble(pumpscheme))
  dum = chk('pump(1)',pump(1))
  dum = chk('pump(2)',pump(2))
  dum = chk('pump(3)',pump(3))
  dum = chk('pump(4)',pump(4))
  dum = chk('y(61)',y(61))
  dum = chk('y(62)',y(62))
  dum = chk('y(64)',y(64))
  dum = chk('T_w_out',T_w_out)
  dum = chk('Tank_lvl',Tank_lvl)
end if

c *****
c security check of pumps and valves
do 8000, i=1,4
  if (pump(i).LT.0) then
    dum=err('number',dble(i))
    dum=err('pump',pump(i))
    dum=err('E:error in pump calculation')
  end if
8000 continue
  if ((y(61).LT. 0).or.(y(61). GT. 1.01)) then
    dum=chk('E y(61)',y(61))
    dum=err('E:error calculating y(61)')
  end if
  if ((y(62).LT. 0).or.(y(62). GT. 1.01)) then
    dum=chk('E y(62)',y(62))
    dum=err('E:error calculating y(62)')
  end if
  if ((y(64).LT. 0).or.(y(64). GT. 1.01)) then
    dum=chk('E y(64)',y(64))
    dum=err('E:error calculating y(64)')
  end if
  return 1
end

```

Include File modes.f modes.f contains the information on what each control strategy built into Type 99 actually does.

```

c select case (mode)
c *****
c case (0)
c conventional system w/o storage

```

```

pumpscheme = 0
c *****
c case (1,4)
c * liquid full storage system *
c if (onpeak) then
c   onpeak: tank meets load alone, chiller off, no recharge
c   pumpscheme = 4
c else
c   offpeak: chiller charges tank and meets load
c   no tank discharge offpeak
c   pumpscheme = 5
c end if
c *****
c case (2,5)
c * partial storage liquid system with chiller priority *
c chiller meets load, tank adds only when chiller too small,
c tank charged no matter what time of day
c
pumpscheme = 1
c *****
c case (3,6)
c * partial storage liquid system with tank priority *
c
c if (onpeak) then
c   tank, then chiller, no recharge
c   pumpscheme = 3
c else
c   chiller, then tank, recharge allowed
c   pumpscheme = 1
c end if
c *****
c case (7)
c * ice, full storage *
c there are no pumps 2 and 3 in any ice system
c harvester variable is pump(1)
c pumpscheme = -1
c if (onpeak) then
c   harvester off, no charging of tank,
c   pump(1) = 0
c   if tank_lvl low, make ice but warn
c   if (Tank_lvl .LE. Lvl_low) then
c     pump(1) = 1
c     dum=err('W:Tank level low -> harvester runs onpeak')
c     dum=chk('W:Tank_lvl',Tank_lvl)
c   end if
c else
c   harvester on
c   pump(1) = 0
c   if (charge_tank) pump(1) = 1
c end if
c
c case (8)
c * ice, PS/chiller priority
c pumpscheme = -1
c pump(1) = 0
c if (charge_tank) pump(1) = 1
c if ((m_w .le. 0) .and. (Tank_lvl .gt. Lvl_low)) pump(1) = 0
c
c case (9)
c * ice, PS/tank priority
c pumpscheme = -1
c if (onpeak) then
c   pump(1) = QLR_max
c else
c   harvester on if tank needs to be charged
c   pump(1) = 0
c   if (charge_tank) pump(1) = 1
c end if
c *****
c case default
c dum=err('E:unsupported control mode')
c dum=chk('E:mode',dble(mode))
c
end select

```

Include File pumps.f pumps.f sets the control variable for the pumps in Type 99.

```

select case (pumpscheme)
c *****
c case (-1)
c ice system: do nothing here

c *****
c case (0)
c simple system setup without storage
c pump(1) = 1
c pump(2) = 0
c pump(3) = 0
c pump(4) = m_w/m_max(4)
c if (pump(4).GT.1) s(ptr+1 + 4) = pump(4)
c turn off pump(1) if there is no load
c if (m(4).EQ.0) pump(1) = 0

c *****
c case (1,2)
c 1) chiller, then tank, recharge possible
c   used for: all PS/CP, PS/TP offpeak
c 2) chiller, then tank, no recharge allowed
c   used for: PS/TP when tank runs low

c pump(1) = 1
c if (m(4).GT.m(1)) then
c   chiller cannot meet load alone,
c   try to meet load with additional water from tank
c   pump(2) = 0
c   pump(3) = (m(4)-m(1)) / m_max(3)
c   if (pump(3).GT.1) then
c     s(ptr+1 + 3) = pump(3)
c     pump(3) = 1
c     pump(4) = (m(3)+m(1)) / m_max(4)
c   end if
c else
c   chiller has cap. left, tank can be charged if needed
c   pump(3) = 0
c   if (charge_tank .and. (pumpscheme.eq.1)) then
c     set pump_2 such that it uses not more than pump_1's
c     remaining mass flow
c     dummy = (m(1) - m(4)) / m_max(2)
c     pump(2) = min(dummy, dble(1))
c   else
c     no charging needed
c     pump(2) = 0
c   end if
c end if

c turn off pump(1) if there is neither load nor need for charge
c if ((m(4).EQ.0).AND.(m(2).EQ.0)) pump(1) = 0

c *****
c case (3)
c tank, then chiller, no recharge allowed
c used for: PS/TP when tank has sufficient capacity

c pump(1) = 1
c pump(2) = 0
c dummy = m_max(1) * QLR_max
c pump(3) = max ((m(4) - dummy) / m_max(3), dble(0))

c *****
c case (4)
c tank alone, chiller off, no recharge
c used for: FS onpeak

c pump(1) = 0
c pump(2) = 0
c if (m_w .le. m_max(3)) then
c   pump(3) big enough
c   pump(3) = m_w / m_max(3)
c else
c   pump(3) too small: reduce pump(4)
c   s(ptr+1 + 3) = pump(3)
c   pump(3) = 1
c   pump(4) = m(3) / m_max(4)
c end if

c *****
c case (5)
c chiller alone, recharge possible

```

```

c      used for: FS offpeak
      pump(1) = 1
      pump(3) = 0
      if (m(4).GT.m(1)) then
c        chiller too small
          s(ptr+1 + 1) = m(4) / m(1)
          pump(4) = m(1) / m_max(4)
      else
c        chiller has capacity left to charge tank
          if (charge_tank) then
            dummy = m(1)-m(4)
            pump(2) = min(dummy, m_max(2)) / m_max(2)
          else
            pump(2) = 0
          end if
      end if

c      turn off pump(1) if there is neither load nor need for charge
      if ((m(4).EQ.0).AND.(m(2).EQ.0)) pump(1) = 0

      case default
      dum=err('E:the value for pumpscheme should be 1 to 5')
      dum=chk('E:pumpscheme',dble(pumpscheme))

      end select

```

Appendix C

TRNSYS deck for water, full storage

The TRNSYS deck for a full storage pure water system is printed below as an example¹. It is slightly modified to increase readability. I used variable names for all important quantities, which increases the file length but reduces the source of errors greatly. This deck was used with TRNSYS under a Unix operating system and might not be immediately compatible under other TRNSYS implementations. One would have to remove all Type 93 printers (or replace them by Type 25 printers, of which the latest version of TRNSYS allows 13 per deck now).

I automatically regenerated the decks for each system before every single TRNSYS run by merging several mini-decks containing units that were relevant for the respective mode, which is an easy task with a Unix script. This way I had to change global information such as output settings or simulation period only once while affecting all decks.

The order of units and equations should not be changed, since variables are often referencing previously defined variables, and TRNSYS is not capable of resolving EQUATIONS backwards.

¹The numbers (mode = 0 . . . 9) correspond to those in the four spread sheets.

units in order of appearance
 (#) indicates a new TYPE, written for this project

```

UNIT 41      TYPE 9 data reader
UNIT 40,42   TYPE 33 psychrometrics
UNIT 67      TYPE 11 T Piece (ambient + room air)
UNIT 10      TYPE 99 Controller (#)
UNIT 31      TYPE 68 Kevin Cross' modified Chiller 53
UNIT 61,62,64 TYPE 11 Diverters
UNIT 63,65,66 TYPE 11 T pieces
UNIT 20      TYPE 90 Cooling Coil
UNIT 71      TYPE 3 Primary Pump
UNIT 72      TYPE 3 Tank Pump (charge)
UNIT 73      TYPE 3 Tank Pump (discharge)
UNIT 74      TYPE 3 Coil Pump
UNIT 45      TYPE 92 Conditioned Room (#)
UNIT 52      TYPE 60 stratified storage tank
UNIT 7       TYPE 24 Integrator
UNIT 9       TYPE 24 Integrator
UNIT 3       TYPE 95 COP calculator (#)
UNIT 4       TYPE 55 Periodic Integrator
UNIT 5       TYPE 55 Periodic Integrator
UNIT 6       TYPE 55 Periodic Integrator
UNIT 19,21   TYPE 25 Printer for output after SUMMARY period
UNIT 8       TYPE 55 Periodic Integrator
UNIT 11-16   TYPE 93 Printer (#) for output after PRINTOUT period
UNIT 17,18   TYPE 25 Printer for output after SUMMARY period

```

* TRNSYS DECK STARTS HERE (delete previous lines before use)

* liquid water, full storage

* Design Day Repetition

ASSIGN list 6

EQUATIONS 9

WE_RATE = 0

STARTDAY = 1

DAYS = 6

TIMESTEP = 0.25

SUMMARY = 24

PRINTOUT = 1 * TIMESTEP

CHK_CTRL = 1 * LT(DAYS, 10)

CHK_COIL = 1 * LT(DAYS, 10)

CHK = 0 * LT(DAYS, 10)

ASSIGN 11.out 11

ASSIGN 12.out 12

ASSIGN 13.out 13

ASSIGN 14.out 14

ASSIGN 15.out 15

ASSIGN 16.out 16

ASSIGN 17.out 17

ASSIGN 18.out 18

ASSIGN 19.out 19

ASSIGN 21.out 21

* ASSIGN ../data/data90_corr 20

ASSIGN ../data/data_designday 20

* SOLVER 1

* LIMITS 40 20

* default is 25 10

EQUATIONS 6

START = MAX((STARTDAY-1)*24, 0.25)

LENGTH = DAYS * 24

END = MIN (START + LENGTH, 8759.75)

HOURL = MOD(TIME, 24)

PSTART = START

PEND = END

SIMULATION START END TIMESTEP

WIDTH 100

EQUATIONS 26

T_AMB = [41,8]

W_AMB = [41,10]

H_AMB = [40,3]

RH_AMB = [41,9]

T_AMB_WB = [40,2]

T_R_SET = 24

W_R_SET = 0.0093

T_ROOM = [45,4]

W_ROOM = [45,5]

H_ROOM = [45,11]

RH_ROOM = [42,6]

T_EXIT = [45,1]

W_EXIT = [45,2]

```

H_EXIT = [45,12]
CP_A = 1.0042
U_TANK = 0.0248 * 3.6
LOAD = [41,6]
TANK_LVL = [10,11]
SHR = 0.8
M_A = [45,3]
M_W = [20,5]
T_A_SET = 12
W_INIT = 0
M_INIT = 0
DAY = INT(TIME/24)+1
DMD_LVL = [10,14]

* common variables for piping
EQUATIONS 9
Y_61 = [10,4]
Y_62 = [10,5]
Y_64 = [10,6]
PUMP_1 = [10,7]
PUMP_2 = [10,8]
PUMP_3 = [10,9]
PUMP_4 = [10,10]
PRESS_DROP = 60 * 12 * 249.082
LOSSFRAC = 0.40

* other common variables
EQUATIONS 1
ONPEAK = [10,12]

*****
UNIT 41 TYPE 9 DATA READER (FREE = MODE 2)
PARAMS 35
* mode, # values, timestep
-2, 10, TIMESTEP
* day, month, hour, E_tot (kWh), E_cool (kWh), Cap_cool (kWh), COP
-1,1,0 -2,1,0 -3,1,0 -4,1,0 -5,1,0 -6,1,0 -7,1,0
* T_amb (C), RH_amb (%), w_amb
-8,1,0 -9,1,0 -10,1,0
* LUNIT, formatted reading
20, 0

*****
UNIT 40 TYPE 33 psychrometrics
PARAMS 4
2, 1, 1, 2
INPUTS 2
* T_amb RH_amb
T_AMB RH_AMB
20 0
* out(2) = T_amb_wb
* out(3) = h_amb

*****
UNIT 42 TYPE 33 psychrometrics
PARAMS 4
4, 1, 0, 2
INPUTS 2
* T_room w_room
T_ROOM W_ROOM
20 0
* out(3) = h_room
* out(6) = RH_room

*****
EQUATIONS 2
OUTDOOR = 0.15
RECYC = 1 - OUTDOOR

UNIT 67 TYPE 11 Tee-Piece (ambient + room air)
PARAMS 1
6
INPUTS 6
T_AMB W_AMB 0,0 T_EXIT W_EXIT 0,0
0 0 OUTDOOR 0 0 RECYC
* mass flow is not of interest

TOLERANCES 0.0005 0.0005

*****
EQUATIONS 14
MODE = 1
M_MAX_4 = 175000
M_MAX_2 = 180000
M_MAX_3 = M_MAX_2
M_MAX_1 = 180000
CP_W = 4.19

```

```

RHO_W = 1000
V_TANK = 1800
DT_TANK_DES = 17.5
T_W_SET = 5
T_W_MAX = 6
Q_CH_DES = M_MAX_1 * CP_W * DT_TANK_DES
UA_A_COIL = 350000
FACTOR = 5

* common variables for water/water-ethanol storage
EQUATIONS 9
H_TANK = (V_TANK*2.86)**(1/3)
T_INIT = T_W_SET
CAP_T_DES = V_TANK*RHO_W*CP_W*DT_TANK_DES/3600
CAP_T_ACT = CAP_T_DES - [52,16] / 3600
M_1 = PUMP_1 * M_MAX_1
M_2 = PUMP_2 * M_MAX_2
M_3 = PUMP_3 * M_MAX_3
M_4 = PUMP_4 * M_MAX_4
M_DIS = M_3 - M_2

* converting all power outputs into kilowatts (pump comes in kJ/hr)
EQUATIONS 3
P_PUMP = ([71,3] + [72,3] + [73,3] + [74,3]) / 3600
P_CHIL = [31,6] * GT(PUMP_1, 0)
P_TOTAL = P_PUMP + P_CHIL

* energy variables in kW
EQUATIONS 10
Q_LOAD = LOAD / TIMESTEP
Q_CHIL = [31,5] / 3600
Q_COIL = [20,9] / 3600
Q_SENS = [20,13] / 3600
Q_LAT = [20,14] / 3600
Q_VENT = OUTDOOR * M_A * (H_EXIT-H_AMB) / 3600
Q_ROOM = [45,10] / TIMESTEP / 3600
Q_TANK = (M_3*([64,3]-[52,5])-M_2*([52,6]-[72,1])) * CP_W / 3600
Q_T_AMB = [52,7] / 3600
COP_ACT = Q_CHIL / MAX (P_CHIL,1) * GT(TIME,START)

*****
UNIT 10 TYPE 99 Controller
PARAMS 22
* T_a_set T_w_set T_w_max Lvl_high Lvl_low cp_w
  T_A_SET T_W_SET T_W_MAX 0.98 0.20 CP_W
* m_max_1 m_max_2 m_max_4 m_max_5
  M_MAX_1 M_MAX_2 M_MAX_4 0
* check, mode, Cap_T_des
  CHK_CTRL MODE CAP_T_DES
* rho_w V_tank on_hr off_hr startday weekend dmd_min Q_ch_des
  RHO_W V_TANK 9 19 2 WE_RATE 0.15 Q_CH_DES
* May 1st, 1990 = Tuesday = startday 2
* reset_dmd
SUMMARY
INPUTS 5
* CoillLoad T_w_in m_w Tank_Cap T_w_ret
  Q_COIL 74,1 20,5 CAP_T_ACT 20,4
  0 T_INIT M_INIT CAP_T_DES T_INIT

*****
EQUATIONS 6
COP_CH_DES = 3.976
P_CH_DES = Q_CH_DES / 3600 / COP_CH_DES
Q_CH_MAX = Q_CH_DES * 1.35
Q_CH_MIN = Q_CH_DES * 0.00
M_COND = Q_CH_DES / 4.19 / 6
T_C_IN = MAX(T_AMB_WB+4, 16)

UNIT 31 TYPE 68 Kevin Cross' modified Chiller 53
PARAMS 11
* Q_max Q_min Q_des P_des a b c d e
  Q_CH_MAX Q_CH_MIN Q_CH_DES P_CH_DES 0.140 0.544 0.316 0.012 0.015
* Cp_evap Cp_cond
  CP_W 4.19
INPUTS 6
* T_w_set T_evap_in m_evap T_cond_in m_cond on/off (0/1)
  T_W_SET 66,1 66,2 T_C_IN 0,0
  T_INIT T_INIT M_INIT T_INIT M_COND 1

*****
UNIT 61 TYPE 11 Diverter
PARAMS 1
2
INPUTS 3
* T_in, m_in, gamma (0 = out to 1+2, 1 = out to 3+4)
  71,1 M_1 7_61
  T_INIT M_INIT 0
*****

```

```

UNIT 62 TYPE 11 Diverter
PARAMS 1
2
INPUTS 3
* T_in, m_in, gamma (0 = out to 1+2, 1 = out to 3+4)
  61,3 61,4 7,62
T_INIT M_INIT 0
*****
UNIT 63 TYPE 11 T_piece
PARAMS 1
1
INPUTS 4
* T_in_1, m_in_1, T_in_2, m_in_2
  62,1 62,2 73,1 M_3
T_INIT M_INIT T_INIT M_INIT
*****
UNIT 64 TYPE 11 Diverter
PARAMS 1
2
INPUTS 3
* T_in, m_in, gamma (0 = out to 1+2, 1 = out to 3+4)
  20,4 20,5 7,64
T_INIT M_INIT 0
*****
UNIT 65 TYPE 11 T_piece
PARAMS 1
1
INPUTS 4
* T_in_1, m_in_1, T_in_2, m_in_2
  64,1 64,2 52,6 52,4
T_INIT M_INIT T_INIT M_INIT
*****
UNIT 66 TYPE 11 T_piece
PARAMS 1
1
INPUTS 4
* T_in_1, m_in_1, T_in_2, m_in_2
  65,1 65,2 61,1 61,2
T_INIT M_INIT T_INIT M_INIT
*****
UNIT 20 TYPE 90 Cooling Coil
PARAMS 10
* calcmode = 2: iterate m_w with given T_A_SET
* coilmode, UA_a, factor, cp_a, cp_w, calcmode
  1, UA_A_COIL, FACTOR CP_A, CP_W, 2
* check, check_sub, check_it, m_w_max
  CHK_COIL 0, 0 M_MAX_4
INPUTS 6
* T_a_in, w_a_in, m_a, T_w_in, m_w, T_a_set
  67,1 67,2 M_A 74,1 0,0 T_A_SET
T_INIT W_INIT M_INIT T_W_SET M_INIT T_A_SET

* pressure drop = 100 ft of water, converted into Pascal
EQUATIONS 4
P_DES_1 = M_MAX_1 / RHO_W * PRESS_DROP / (1-LOSSFRAC) / 1000
P_DES_2 = M_MAX_2 / RHO_W * PRESS_DROP / (1-LOSSFRAC) / 1000
P_DES_3 = M_MAX_3 / RHO_W * PRESS_DROP / (1-LOSSFRAC) / 1000
P_DES_4 = M_MAX_4 / RHO_W * PRESS_DROP / (1-LOSSFRAC) / 1000
* press_drop in Pa, P_des in kJ/hr (Type 3 needs P_des in kJ/hr !!)
*****
UNIT 71 TYPE 3 Primary Pump
PARAMS 4
M_MAX_1, CP_W, P_DES_1, LOSSFRAC
INPUTS 3
31,1 31,2 PUMP_1
T_INIT M_INIT 1 0
*****
UNIT 72 TYPE 3 Tank Pump (charge)
PARAMS 4
M_MAX_2, CP_W, P_DES_2, LOSSFRAC
INPUTS 3
62,3 62,4 PUMP_2
T_INIT M_INIT 0
*****
UNIT 73 TYPE 3 Tank Pump (discharge)
PARAMS 4
M_MAX_3, CP_W, P_DES_3, LOSSFRAC
INPUTS 3
52,5 52,2 PUMP_3
T_INIT M_INIT 0

```

```

*****
UNIT 74 TYPE 3 Coil Pump
PARAMS 4
M_MAX_4, CP_W, P_DES_4, LOSSFRAC
INPUTS 3
63,1 63,2 PUMP_4
T_INIT M_INIT 0

*****
EQUATIONS 1
V_ROOM = 390000 * 12 * 0.028
* cooled area * height * conversion factor

UNIT 45 TYPE 92 Conditioned Room
PARAMS 7
* mode T_init w_init V_room rho_a h_fg chk
1 T_R_SET W_R_SET V_ROOM 1 2256.8 CHK
INPUTS 7
* T_sup w_sup m_a Load SHR L_sens L_lat
20,1 20,2 M_A LOAD 0,0 0,0 0,0
T_INIT W_INIT M_INIT 0 SHR 0 0

*****
UNIT 52 TYPE 60 new strat. tank
EQUATIONS 2
H_1 = 0.1
H_2 = H_TANK-0.1

* k_fluid = 2.0538
PARAMS 8
* inlets, vol, height, perimeter, h_in_1, h_out_1 h_in_2 h_out_2
1 V_TANK H_TANK -1 H_1 H_1 H_2 H_2
PARAMS 17
* cp_w rho_w U_tank k_fluid de_k_fluid T_boil AuxMode
CP_W RHO_W U_TANK 2.0538 0 80 2
0,0,0,1,0,0,0,0,1,0
PARAMS 7
* UA_flue T_flue CritFrac GasAuxMode hxMode HMode UMode
0 6 0 0 0 0
DERIVATIVES 50
*100, T_W_SET
T_W_SET, T_W_SET, T_W_SET, T_W_SET, T_W_SET,
T_W_SET, T_W_SET, T_W_SET, T_W_SET, T_W_SET,
T_W_SET, T_W_SET, T_W_SET, T_W_SET, T_W_SET,
T_W_SET, T_W_SET, T_W_SET, T_W_SET, T_W_SET,
T_W_SET, T_W_SET, T_W_SET, T_W_SET, T_W_SET,
T_W_SET, T_W_SET, T_W_SET, T_W_SET, T_W_SET,
T_W_SET, T_W_SET, T_W_SET, T_W_SET, T_W_SET,
T_W_SET, T_W_SET, T_W_SET, T_W_SET, T_W_SET,
T_W_SET, T_W_SET, T_W_SET, T_W_SET, T_W_SET,
T_W_SET, T_W_SET, T_W_SET, T_W_SET, T_W_SET
INPUTS 4
* m_in_1 m_out_1 m_in_2 m_out_2
72,2 64,4 64,4 0,0
0 0 0 -2
INPUTS 5
* T_in_1 T_in_2 T_amb gamma_1 gamma_2
72,1 64,3 T_AMB 0,0 0,0
T_INIT T_INIT T_INIT 0 0

EQUATIONS 4
P_ON = P_TOTAL * GT(ONPEAK,0)
P_OFF = P_TOTAL - P_ON
Q_T_DIS = MAX(0, (Q_COIL - Q_CHIL))
Q_T_CHG = - MIN(0, (Q_COIL - Q_CHIL))

*****
UNIT 7 TYPE 24 Integrator
INPUTS 2
P_ON P_OFF
0 0

*****
UNIT 9 TYPE 24 Integrator
INPUTS 10
Q_CHIL Q_COIL Q_SENS Q_LAT Q_VENT Q_ROOM Q_TANK P_PUMP P_CHIL P_TOTAL
0 0 0 0 0 0 0 0 0 0

*****
UNIT 3 TYPE 95 COP calculator
INPUTS 2
P_CHIL Q_CHIL
0 0

*****
* energy variables in kW for checking balances

```

```

EQUATIONS 2
Q_A_CHK = (Q_LOAD - Q_COIL - Q_VENT - Q_ROOM) * GT(TIME,START)
Q_W_CHK = (Q_CHIL + Q_TANK - Q_COIL - P_PUMP * LOSSFRAC) * GT(TIME,START)

```

```

UNIT 4 TYPE 55 Periodic Integrator
PARAMS 35
1, START, SUMMARY, SUMMARY, SUMMARY, START, END
2, START, SUMMARY, SUMMARY, SUMMARY, START, END
3, START, SUMMARY, SUMMARY, SUMMARY, START, END
4, START, SUMMARY, SUMMARY, SUMMARY, START, END
5, START, SUMMARY, SUMMARY, SUMMARY, START, END

```

```

INPUTS 5
Q_A_CHK Q_W_CHK Q_LOAD COP_ACT DMD_LVL
0 0 0 0 0

```

```

EQUATIONS 4
A_CHK = [4,1]
W_CHK = [4,11]
DMD_MAX = [4,49]
DMD_MIN = [4,47]

```

```

*****
* energy variables in MWh, cost in $

```

```

EQUATIONS 12
E_ON_TOT = [7,1] / 1000
E_OFF_TOT = [7,2] / 1000
E_CHIL = [9,1] / 1000
E_COIL = [9,2] / 1000
E_SENS = [9,3] / 1000
E_LAT = [9,4] / 1000
E_VENT = [9,5] / 1000
E_ROOM = [9,6] / 1000
E_TANK = [9,7] / 1000
E_PUMP = [9,8] / 1000
E_CHIL2 = [9,9] / 1000
E_TOTAL = [9,10] / 1000
* E_CHIL = air side, E_CHIL2 = water side

```

```

*****

```

```

UNIT 5 TYPE 55 Periodic Integrator
PARAMS 70
1, START, SUMMARY, SUMMARY, SUMMARY, START, END
2, START, SUMMARY, SUMMARY, SUMMARY, START, END
3, START, SUMMARY, SUMMARY, SUMMARY, START, END
4, START, SUMMARY, SUMMARY, SUMMARY, START, END
5, START, SUMMARY, SUMMARY, SUMMARY, START, END
6, START, SUMMARY, SUMMARY, SUMMARY, START, END
7, START, SUMMARY, SUMMARY, SUMMARY, START, END
8, START, SUMMARY, SUMMARY, SUMMARY, START, END
9, START, SUMMARY, SUMMARY, SUMMARY, START, END
10, START, SUMMARY, SUMMARY, SUMMARY, START, END

```

```

INPUTS 10
* T_w_in T_w_out
74,1 20,4 0,0 0,0 0,0 0,0 0,0 0,0 0,0
0 0 0 0 0 0 0 0 0
* 1 2 3 4 5 6 7 8 9

```

```

UNIT 6 TYPE 55 Periodic Integrator
PARAMS 70
1, START, SUMMARY, SUMMARY, SUMMARY, START, END
2, START, SUMMARY, SUMMARY, SUMMARY, START, END
3, START, SUMMARY, SUMMARY, SUMMARY, START, END
4, START, SUMMARY, SUMMARY, SUMMARY, START, END
5, START, SUMMARY, SUMMARY, SUMMARY, START, END
6, START, SUMMARY, SUMMARY, SUMMARY, START, END
7, START, SUMMARY, SUMMARY, SUMMARY, START, END
8, START, SUMMARY, SUMMARY, SUMMARY, START, END
9, START, SUMMARY, SUMMARY, SUMMARY, START, END
10, START, SUMMARY, SUMMARY, SUMMARY, START, END

```

```

INPUTS 10
TANK_LVL P_CHIL Q_T_CHG Q_T_DIS P_ON P_OFF 0 0 M_W M_A
0 0 0 0 0 0 0 0 0
* 1 2 3 4 5 6 7 8 9

```

```

EQUATIONS 7
E_T_CHG = [6,21]
E_T_DIS = [6,31]
W_ON = [6,41]
W_OFF = [6,51]
W_FRAC = W_ON / MAX(1,W_ON+W_OFF)
PEAK_ON = [6,49]
PEAK_OFF = [6,59]

```

```

*****

```

```

UNIT 19 TYPE 25 Total Summary Printer
PARAMS 4

```



```

1, 15, 1, PSTART, PEND
PARAMS 18
6,2 7,0 7,0 7,0 7,0 7,0 8,1 7,1
INPUTS 9
HOUR Q_LOAD Q_COIL Q_CHIL Q_TANK Q_VENT Q_ROOM Q_A_CHK Q_W_CHK
1 2 3 4 5 6 7 8 9 10 11 12 13 14

*****
UNIT 16 TYPE 93
PARAMS 5
1, 16, 1, PSTART, PEND
PARAMS 22
7,2 6,2 8,0 6,2 6,2 7,2 6,0 7,2 7,2 7,2 7,2
INPUTS 11
DAY HOUR 61,4 Y_61 DMD_LVL PLR P_CHIL QLR 66,1 31,1 10,13
1 2 3 4 5 6 7 8 9 10 11 12 13 14

*****
UNIT 17 TYPE 25 Summary Printer
PARAMS 4
SUMMARY START END 17
INPUTS 10
5,17 5,19 5,13 6,7 6,9 5,9 8,11 8,1 8,49 8,59
ret_mn ret_mx ret_av mn_lvl mx_lvl mx_Tsup E_c_sup E_t_sup PLR_mx QLR_mx
FORMAT
(F5.0, F7.2, F7.2, F7.2, F6.2, F5.2, F7.2, F8.0, F7.0, F7.2, F6.2)

*****
UNIT 18 TYPE 25 Summary Printer
PARAMS 4
SUMMARY START END 18
INPUTS 9
6,19 6,89 6,99 W_ON W_OFF E_T_CHG E_T_DIS 8,29 8,39
P_ch_mx mx_coil mx_m_a W_on W_off E_chg E_dis mx_chg mx_dis
FORMAT
(F5.0, F6.0, F8.0, F8.0, F8.0, F8.0, F8.0, F8.0, F8.0)
END

```

Appendix D

Detailed System Illustrations

Figures D.1 and D.2 show more details on the assignment of TRNSYS units, the linkage between TRNSYS inputs and outputs and the information flow than the respective figures 3.3 and 3.4.

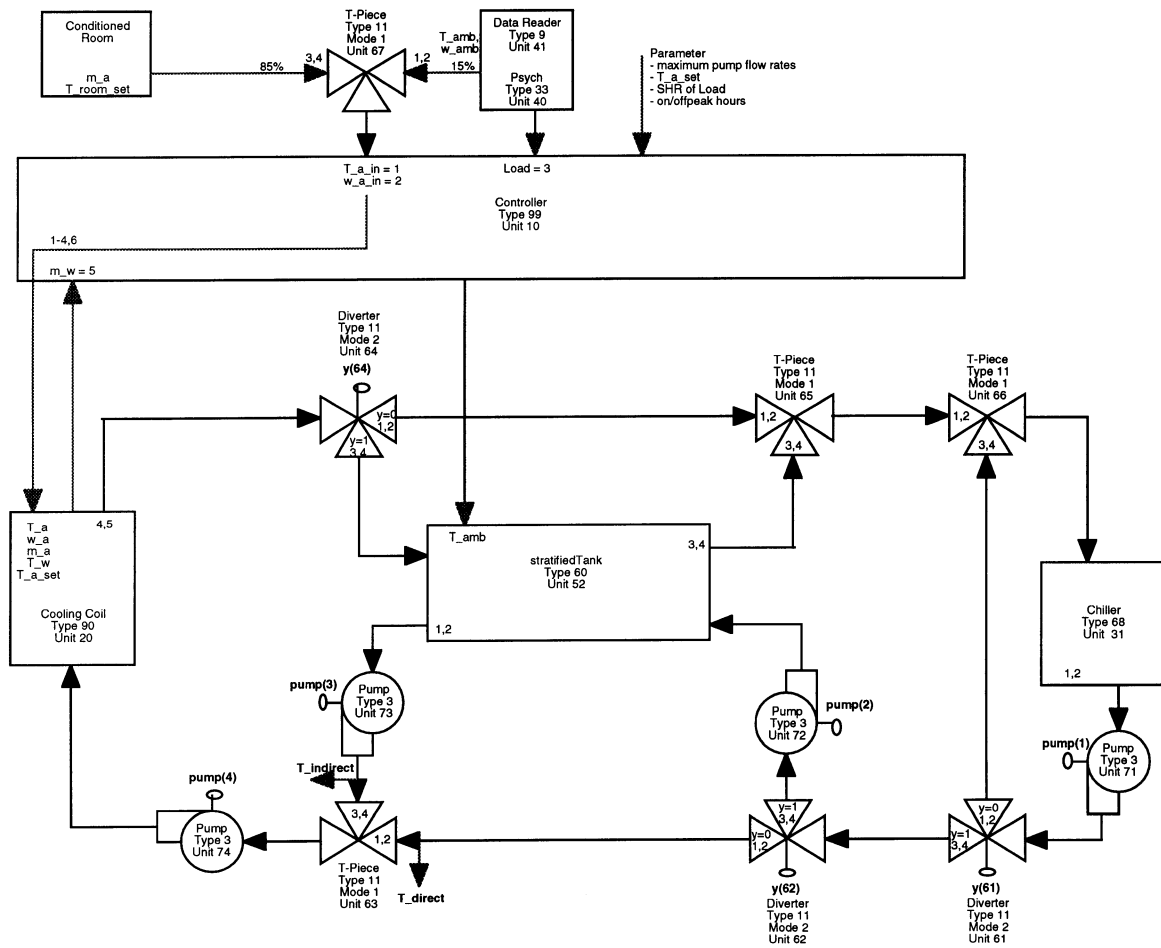


Figure D.1: Detailed overview of the liquid storage system

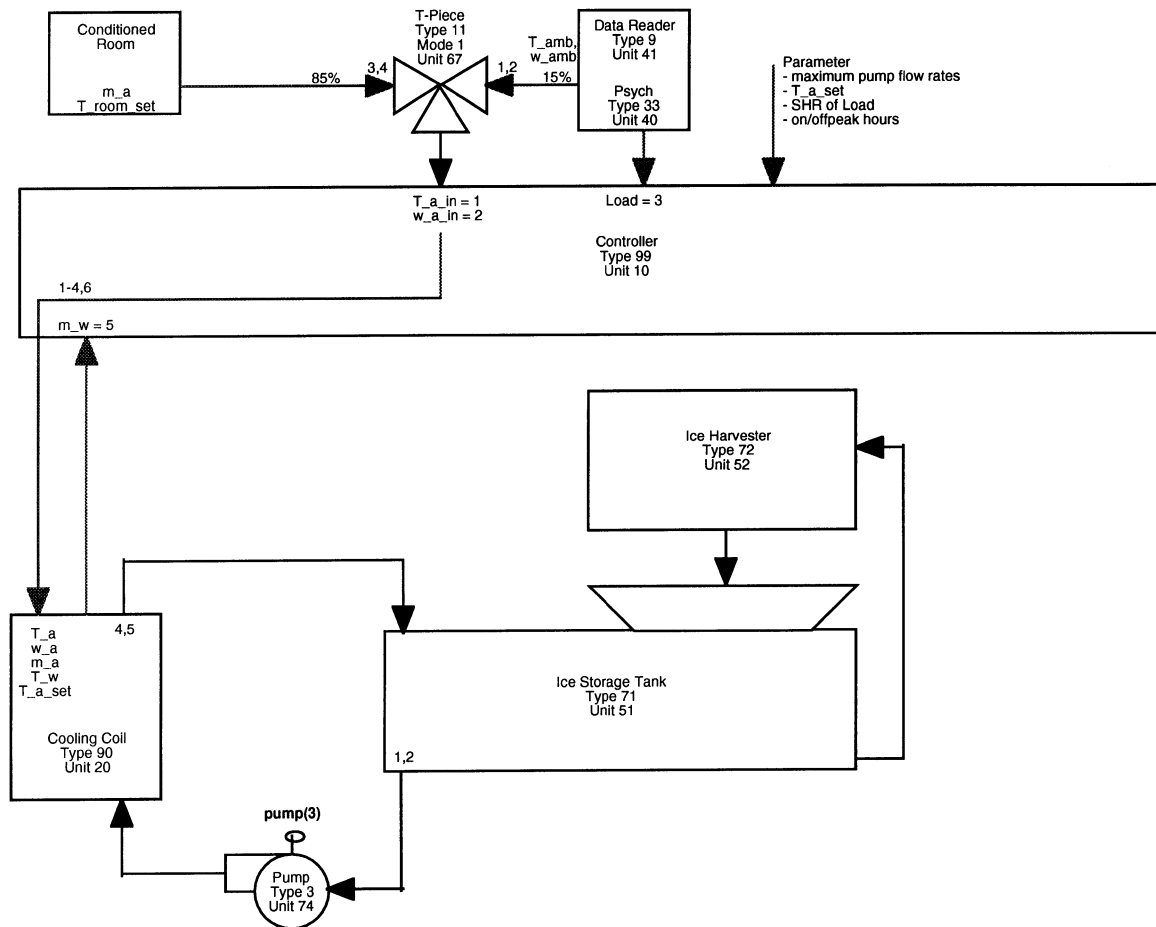


Figure D.2: Detailed overview of the ice storage system

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