

# Use of Air Conditioning Heat Rejection for Swimming Pool Heating

by

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## Abstract

Residential swimming pools are common in Wisconsin. However, pool heaters are needed in this climate to allow the pool to be used during the summer and to extend the period of use from late spring to early fall. Swimming pool heaters commonly use natural gas or propane as fuel. Although pool covers are often used to reduce the evaporation loss, the heating needs of an outdoor pool can result in significant operating expense and unnecessary use of natural resources. Even though the available solar energy is at a maximum at the time that pool heating is needed, solar heating systems are not commonly employed. Central air conditioning systems are common in Wisconsin. Central systems are routinely installed in most new homes, especially in those that have residential swimming pools. Air conditioners are electrically driven, and the energy removed from the cooled space plus the electrical energy are rejected to the ambient through air-cooled condensers. Even though the air conditioning season is relatively short in Wisconsin, air conditioning is estimated to contribute 10 to 15 % to the electric demand in the state. The objective of this study is to explore and evaluate different methods of combining air conditioning and pool heating to reduce the energy requirements and electrical demand.



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# Nomenclature

## Roman Symbols

$C$	Compressor clearance factor
$C_i$	Cost for $i$
$C_p$	Fan pressure coefficient
$c_p$	Heat capacity
$C_v$	Fan capacity coefficient
$C_w$	Fan power coefficient
COP	Coefficient of performance
$h$	Pool-air convection heat transfer coefficient
$h_{pc}$	Pool-Air convection heat transfer coefficient with cover
$\Delta h_{\text{evap}}$	Enthalpy of evaporation
LCC	Life Cycle Cost
LCS	Life Cycle Savings
$\dot{m}$	Mass flow rate
$N$	Isentropic exponent
NTU	Number of transfer units
$P$	Pressure
$P_{\text{amb}}$	Saturation vapor pressure at ambient temperature
$P_{\text{atm}}$	Atmospheric pressure
$P_{\text{dis}}$	Discharge pressure
$P_{\text{pool}}$	Saturation vapor pressure at pool temperature
$P_s$	Fan static pressure rise
$P_{\text{suc}}$	Suction pressure
$P_1$	Ratio of the life cycle fuel cost savings to the first-year fuel cost savings

$P_2$	Ratio of the life cycle expenditures incurred to initial investment
$\Delta P_V$	Pressure drop
$\dot{Q}_{act}$	Actual heat transfer rate
$\dot{Q}_{cond}$	Condenser heat transfer rate
$\dot{Q}_{evap}$	Evaporative energy rate
$\dot{Q}_{max}$	Maximum possible heat transfer rate
$q''_{evap}$	Evaporation heat flux
$q''_{con}$	Convection heat flux
$R_{bowen}$	Bowen ratio
$R_c$	R-Value of pool cover
$T_{amb}$	Ambient temperature
$T_{c,i}$	Cold fluid inlet temperature
$T_{cover}$	Pool cover temperature
$T_{dp}$	Dew point temperature
$T_{h,i}$	Hot fluid inlet temperature
$T_{pool}$	Swimming pool temperature
$T_{sky}$	Sky temperature
TTD	Terminal temperature difference
U	Fluid velocity
UA	Overall heat loss coefficient
$\dot{V}$	Fan capacity
$\dot{V}_w$	Water loss per unit time
v	Specific volume
$v_{dis}$	Specific suction volume
$v_G$	Mean wind velocity measured at ground level
$v_{suc}$	Specific discharge volume

$v_{wind}$	Mean wind velocity measured at weather station
$\dot{v}$	Volumetric flow rate
$\dot{v}_{dis}$	Compressor displacement rate
$\dot{W}$	Power
$\dot{W}_{comp}$	Compressor power
$\dot{W}_{pump/fan}$	Pump/fan power

### Greek Symbols

$\varepsilon$	Heat exchanger effectiveness
$\varepsilon_s$	Cloudy sky emissivity
$\varepsilon_{sc}$	Clear sky emissivity
$\delta$	Thickness of cover
$\eta_{vol}$	Volumetric efficiency
$\rho$	Density
$\sigma$	Stefan Boltzmann constant
$\xi$	Hydraulic loss figure

### Additional Subscripts

e	Electricity
eq	Equipment
g	Natural gas
SPAC	Swimming pool air conditioner



# Chapter 1

## Introduction

### 1.1 Objective

In the Wisconsin climate residential swimming pools need to be heated throughout the season. Gas pool heaters are commonly used to supply the energy required to maintain the comfort temperature of the swimming pool. However, most residences that host a swimming pool nearby have air conditioning systems to reduce the temperature inside the building. Consequently, there is one device that rejects heat and another one that needs heat. The objective of this research is to explore and evaluate different methods of combining air conditioning and pool heating to reduce the energy requirements and electrical demand.

Both air conditioners and gas pool heaters require purchased energy to operate. If the heating demand of the pool can be satisfied using the rejected heat from the building, the gas energy for pool heating can be reduced or possibly eliminated. Additionally, a water-cooled air conditioner performs better than a conventional air-cooled air conditioner because of the water properties. Consequently, the homeowner saves purchased energy by implementing a swimming pool heating system that uses the swimming pool as the condenser for the air conditioner.

More than six million American families own a swimming pool. Consequently, reducing the energy demand for pool heating and air conditioning helps saving natural resources.

To investigate the performance of the improved swimming pool air conditioner and to discuss the benefits of such a system, a computer simulation has been implemented. A transient simulation program called TRNSYS was employed to simulate the required components, where each component (building, air conditioner, swimming pool) is based on equations that describe its physical behavior. This simulation can be used for different places by changing the weather data, which is an input to the program. It will be shown that the swimming pool air conditioner lowers the operation cost and reduces the energy consumption for almost every location in the United States.

## **1.2 An Introduction to TRNSYS**

TRNSYS is a transient system simulation program with a modular structure. The program is well suited to simulate the performance of systems, the behavior of which is a function of the passage of time. This is the case if outside conditions that influence the system behavior change, such as weather conditions, or if the system components themselves go through conditions that vary with time.

Modular simulation of a system requires the identification of components whose collective performance describes the performance of the system. Each component is formulated by mathematical equations that describe its physical behavior. The mathematical models for each component are formulated in FORTRAN code, so that they

can be used within the TRNSYS program. Formulation of the components has to be in accordance with the required TRNSYS format. A basic principle in this format is the specification of PARAMETERS, INPUTS and OUTPUTS for each component. Parameters are constant values that are used to model a component; these can be for example, the geometric parameters of the swimming pool such as length, depth and width. Inputs are time-dependent variables that can come from a user supplied data source such as weather data or from outputs of other components.

There can be several components of the same type specified in one simulation. The way this identification is accomplished is that each component is assigned an identifying type number that is component specific. A second number, the unit number, is unique and can only be used once in a simulation. Different unit numbers can be associated with the same type number, although there are limitations on how many types of one kind can be used in one simulation.

A system is set up in TRNSYS by means of an input file, called a TRNSYS deck. This deck contains all the information that specifies the components and how the components interact. The system is set up by connecting all inputs and outputs in an appropriate way to simulate the real system. For example the cooling demand for the building unit is the evaporator energy of the air conditioner unit. Once a system is set up in a TRNSYS deck, the program can be run over a user defined time interval. The time interval is divided into equal number of time steps. At each time step the program calls each component and solves all the mathematical equations that specify the component performance. The program iteratively calls the system component until a stationary state

is reached. The stationary state is reached when all the calculated inputs to the components remain constant between two iterations. Naturally, in a numerical solution such as calculated by TRNSYS, there will always be a difference in results between two iterations. Therefore the user has to specify tolerances that define a stationary state.

Aside from the components that simulate actual physical parts of the system, there are predefined utility components that can be used in the simulation. One of them is the data reader. The data reader is able to read data from a user supplied data file that has to be assigned in the TRNSYS deck. Every time step of the simulation the data file then reads the desired values from the file and makes them accessible to the components.

Another kind of utility component is a printer that stores output data in a file. Several printers can be defined in one deck. These output files can be imported into a spreadsheet program and the results further examined. The online plotter can be used to make the progress of the simulation visible on the screen, so that the user can immediately decide whether a run was useful or not. Additionally, a quantity integrator is available to integrate values over time.

A special feature of the TRNSYS program package is the possibility to create a user-friendly input file called a TRNSED file. When the TRNSED program is started, the user only has to supply the important parameters and can change these easily for different simulations. In this way the program is accessible to users who are not experienced in using TRNSYS but are only interested in examining a particular system.

### 1.3 Software Selection

Based on the features mentioned in section 1.2, TRNSYS was selected as the primary tool to perform the swimming pool air conditioner analysis.

Although the main product of the present work is a swimming pool air conditioner simulation in TRNSYS appearing in user-friendly TRNSED format, parts of the studies were done using EES (Engineering Equation Solver). The basic function provided by EES is the solution of a set of algebraic equations. EES can also solve differential equations, equations with complex variables, do optimization, provide linear and non-linear regression and generate plots. The program was especially useful for the examination of the refrigeration cycle because of its built-in thermophysical property functions. A Diagram window provides a place to display important input and output values and a schematic diagram can help to interpret their meaning.



# Chapter 2

## The Swimming Pool Simulation

### 2.1 Brief Literature Survey

The following section gives an overview of some of the studies found in the literature that discuss the energy transfer across an air water interface for large bodies of water, such as a swimming pool.

Carrier (1918) did a series of measurements on pans and small tanks in wind tunnels where the evaporation rate was formulated in terms of the partial pressure difference of water and the air above it, and the velocity of the air.

Ryan and Harleman (1973) introduced a study of transient cooling pond behavior and developed an algorithm to simulate the thermal and hydraulic behavior of a cooling pond or lake. They gave relations to estimate the surface energy flux. The convective energy transfer terms were divided into a forced and free convection part. The forced convection was estimated by an empirical function. The free convection terms were derived from a basic heat and mass transfer analogy for a flat plate. The flat plate relations were refined by accounting for the effect of the water vapor in the air above the surface. A so called wind function was introduced, that combined free and forced convection effects. Relations to estimate long wave radiation from the water to the sky were also given in the study of Ryan and Harleman.

The American Society of Heating, Refrigeration and Air Conditioning Engineers Handbook ASHRAE (1991) uses the Carrier equations but notes that the equation, when applied to swimming pools, may predict high values for the heat loss.

Wei, Sigworth et al. (1979) performed a swimming pool analysis. This analysis gives a relation for heat and mass transfer across a pool surface that is similar to the one proposed by Ryan and Harleman, but neglects the effect of water vapor in the air with respect to free convection. Estimations of radiation heat transfer are also made.

Smith, Loeff et al. (1994) made measurements on the evaporation losses from an outdoor swimming pool by measuring the reduction of pool water volume over time due to evaporation as well as measuring evaporation losses from pans floated in the pool. They also measured the temperature change of the pool and correlated the heat loss with the evaporation and measured the radiation exchange between the pool surface and the sky. The data were analyzed and compared to the commonly used evaporation rate equations found in the ASHRAE Applications Handbook. The result found was lower than the predicted result by ASHRAE and a modified version of the ASHRAE equation was developed.

Hahne and Kuebler (1994) made measurements on two heated outdoor swimming pools located in Stuttgart, Germany. They applied formulas for evaporation, radiation, convection, conduction and fresh water supply developed by Richter (1969), Richter (1979) to predict the heat balance of the pools. Using the most suitable correlation for the evaporative losses of the pool the temperature was found to have less than 0.5 K standard

deviation between measured and simulated temperature. The result was implemented in a TRNSYS subroutine (TYPE 144).

## **2.2 A Comparison of Four Swimming Pool Simulations**

### **2.2.1. Introduction**

The following comparison of four different computer programs for simulating swimming pools has been made in an attempt to find the most reliable model for swimming pool heat losses. The emphasis of the comparison is on evaporation models since evaporation accounts for such a large percentage of the heat loss. All of these programs are based on measurements made on pools, lakes or ponds. In each case the measured results were used to find model parameters so that the model and measurements agree. In spite of this experimental verification of the programs, the programs predict different evaporative losses. Part of this difference may be due to the very different nature of the experiments as discussed below.

Table 2.1 provides an overview of the algorithms used in the various programs for convection, evaporation and radiation. Information is also provided on how each program obtains certain thermal parameters. For example, the cover transmittance for solar radiation is a program input (i.e., set by the user) for POOLS and TRANSSOLAR but is a fixed, but different, value for F-Chart and ESP. The following sections give an overview of the four programs and then discuss the various assumptions in more detail.

		Pools [LBL]	F-Chart [F-Chart Software]	Energy Smart Pools [DOE]	Transsolar Type 144 [Transsolar]
<b>Evaporation Losses</b>		Wei, Lederer and Rosenfeld	Wei, Lederer and Rosenfeld	Löf	D.Richter
<b>Convection Losses</b>		Bowen	Bowen	$H = H(T_{Pool}, v_{wind})$	J.T. Czarnecki
<b>Solar Radiation</b>		Hourly	Monthly/Nrel	Monthly	Hourly
<b>Film Cover</b>	<b>Transmittance</b>	Input	0.837	0.75	Input
	<b>R-Value</b> Hr-ft <sup>2</sup> -F/BTU	Input	~ 0.0	0.1	See convection section
<b>Bubble Cover</b>	<b>Transmittance</b>	Input	0.783	0.85	Input
	<b>R-Value</b> Hr-ft <sup>2</sup> -F/BTU	Input	1.87	1.5	See convection section
<b>Wind Velocity</b>		$v = \kappa \cdot v_{airport}$ $\kappa = 0.5$ very windy, unsheltered $\kappa = 0.1..0.2$ normal wind fractions $\kappa = 0.05$ well sheltered pool	Input (monthly average) The manual recommends: $v = v_{airport}/10$ for well sheltered and $v = v_{airport}/5$ for moderate shelter	$v = \xi \cdot v_{airport}$ $\xi = 0.3$ no windbreak $\xi = 0.2$ moderate windbreak $\xi = 0.1$ good windbreak $\xi = 0.15$ for estimation	$v = v_{airport} \cdot \frac{h_{rel}}{h_0} \frac{1}{S_{fac}}$ S <sub>fac</sub> =2 strong shelter S <sub>fac</sub> =2-4 normal shelter S <sub>fac</sub> =3-6 wooded area S <sub>fac</sub> =6-8 unsheltered S <sub>fac</sub> =8-10 open water h <sub>rel</sub> =height of wind measurement h <sub>0</sub> =3 m
<b>Sky Temperature</b>		Berdahl	M. Martin and P. Berdahl	$T_{sky} = T_{amb} - 20$ [F]	M. Martin and P. Berdahl

Table 2.1 Comparison of Different Swimming Pool Models

### 2.2.2. The different heat transfer mechanisms

To predict the pool energy use, an energy balance is set up that specifies the heat fluxes through the water surface. Table 2.1 shows schematically the heat gains and losses from a pool. Both the ESP and F-Chart programs use monthly average energy balances that ignore the thermal capacitance of the pool. The POOLS and TRYSYS programs use hourly energy balances that include the thermal capacitance of the pool. In all the programs the energy gains that increase water temperature come from the direct sun, a solar collector, or a gas (typically) pool heater. The energy losses that decrease the water temperature come from convection, evaporation and radiation to the ambient. Conduction to the ground is small and is neglected in all of the programs.

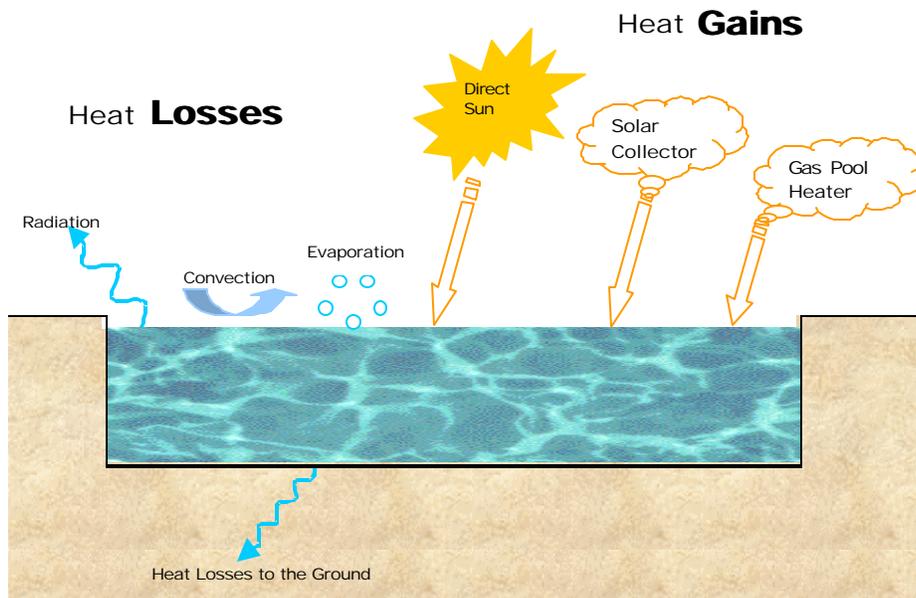


Figure 2.1 Heat transfer mechanisms associated with a swimming pool.

### **2.2.3. Heat Losses**

Of the three heat losses from an uncovered pool the most important is evaporation, often accounting for 45-60% of the total loss. The convection loss is a function of wind speed above the water surface, which is difficult to estimate and the difference in water vapor pressure between the ambient air and the pool surface. The radiation heat loss from the pool to the sky is also difficult to estimate due to problems with estimating the effective sky temperature. All three of these heat losses can be reduced by means of pool covers whenever the pool is not used.

### **2.2.4. Heat Gains**

The largest heat gain is due to the incoming solar radiation, approximately 80% of which is absorbed by the water. Using a solar heater with a collector area that is 75% of the pool area can nearly double the amount of solar energy available to the pool, as noted by Wei, Sigworth et al. (1979).

### **2.2.5. Comparison of Four Computer Models**

#### **2.2.5.1 POOLS (LBL)**

The computer program called POOLS has been developed by Wei, Sigworth et al. (1979) at the Lawrence Berkeley Laboratory (LBL) to compare a wide range of pool energy conservation measures including pool covers and solar collectors. The computer simulations are reported to agree with measurements within a range of 10%. To determine

the accuracy of the computer model, heating gas requirements of existing pools were simulated, using the model, and compared with actual gas consumption in the San Francisco bay area. The solar system gains to the pool are based on hourly calculations of the collector performance. The program source code is no longer available but the algorithms used in the program are available in the LBL report by Wei, Sigworth et al. (1979).

#### 2.2.5.2 F-Chart (F-Chart Software)

The F-Chart Software (Beckman, Klein et al. (1977)) is a program based on methods developed by S. A. Klein and W. A. Beckman. The F-Chart solar design method (Beckman, Klein et al. (1977)) was developed to select the size and type of solar collector that, in conjunction with an auxiliary furnace, will supply the entire heating load at the least possible cost. The solar collector performance is based upon utilizability methods developed by Klein and Beckman and reported by Duffie and Beckman (Beckman, Klein et al. (1977)). Results of the POOLS program were used in the development of the F-Chart program.

#### 2.2.5.3 Energy Smart Pools Software (DOE)

The Energy Smart Pools Software (Gunn, Jones et al. ), developed by the U.S. Department of Energy (DOE), is designed to estimate the annual cost of heating both indoor and outdoor swimming pools and spas. The software is designed to analyze energy and water savings from pool covers and solar pool heating systems. It can also be used to

determine differences between conventional heating and high efficiency heating systems and conventional and high efficiency electric motors. The calculation of the solar gain from the solar collector system is based upon a monthly average day. If the collector has a high loss coefficient, as might be expected of an uncovered collector, then this method of estimating collector performance can lead to significant errors.

#### 2.2.5.4 TRNSYS TYPE 144 (Transsolar)

TRNSYS (Klein (1996)) is a transient system simulation program with a modular structure. The modular structure of TRNSYS gives the program tremendous flexibility, and facilitates the addition to the program of models not included in the standard library. The TYPE 144 (Auer (1996)) is a non-standard TRNSYS model which was developed by the TRANSSOLAR Company in Stuttgart, Germany, to simulate an outdoor or indoor swimming pool. The solar system model is similar to that used by the POOLS program in that the collector output is calculated on an hourly basis.

### 2.2.6. Evaporation Calculations

#### 2.2.6.1 POOLS 9 (LBL)

The evaporative heat flux due to combined forced and free convection was obtained from Wei, Sigworth et al. (1979). This algorithm is a result of a study by Klotz (1977) at Stanford and Ryan and Harleman (1973), at MIT. Wei, Lederer and Rosenfeld verified these algorithms with measurements on pools in the San Francisco bay area.

$$q''_{evap,LBL} = 0.0417 \cdot \left[ 254 \cdot v_{wind} + 569 \cdot \left( \frac{T_{pool} + 460}{1 - 0.378 \cdot \frac{P_{pool}}{P_{atm}}} - \frac{T_{amb} + 460}{1 - 0.378 \cdot \frac{P_{amb}}{P_{atm}}} \right) \right]^{1/3} \cdot (P_{pool} - P_{amb}) \cdot \left[ 2.035859 \cdot \frac{inHg}{psi} \right] \quad (2.1)$$

Where

$q''_{evap}$  = Evaporation heat flux [BTU/hr-ft<sup>2</sup>]

$P_{pool}$  = saturation vapor pressure at the pool temperature [psia]

$P_{amb}$  = saturation vapor pressure at the ambient temperature [psia]

$P_{atm}$  = atmospheric pressure, typically 14.7 [psia]

### 2.2.6.2 F-Chart (F-Chart Software)

The F-Chart Software uses the same evaporation equation used by Wei, Sigworth et al. (1979) implemented in the LBL computer program. However, as F-Chart is a monthly average program, it uses a monthly average ambient temperature and relative humidity.

### 2.2.6.3 Energy Smart Pools (DOE)

Energy Smart Pools (Gunn, Jones et al. ) uses the evaporation algorithm reported in Smith, Jones et al. (1993): Evaporation rates are based on an experimental study of an indoor pool. Based on a single indoor pool experiment, the energy loss rates were set to be 74% of that predicted by the Carrier equation (Carrier (1918)) used in the ASHRAE

Applications Handbook (ASHRAE (1982)). The Carrier algorithm was developed from measurements on a small shallow pond in an outdoor environment. It is difficult to make any judgment as to how the wind velocity in this indoor environment relates to the wind in an outdoor environment.

$$q''_{evap,ESP} = (C_1 + C_2 \cdot v_{wind}) \cdot (P_{pool} - P_{amb}) \cdot \left[ 2.035859 \cdot \frac{inHg}{psia} \right] \quad (2.2)$$

Where:

$$q''_{evap,esp} = \text{Evaporation Heat Flux [BTU/hr-ft}^2\text{]}$$

$$v_{wind} = \text{Wind Speed [mph]}$$

$$P = \text{Pressure [psia]}$$

$$C_1 = 69.4 \text{ [Btu/hr-ft}^2\text{-inHg]}$$

$$C_2 = 30.8 \text{ [Btu/hr-ft}^2\text{-inHg-mph]}$$

#### 2.2.6.4 TRNSYS TYPE 144 (Transsolar)

TRNSYS TYPE 144 (Auer (1996)) uses an empirical correlation based on a report by Richter (1969), who investigated evaporation losses on a cooling pond in Germany.

$$q''_{evap,Transsolar} = \left[ 3.6 \cdot \left( 42.39 + 56.52 \cdot \left[ v_{wind} \cdot \left( 0.44704 \cdot \frac{m/s}{mph} \right) \right]^{0.5} \right) \cdot (P_{dw,pool} - relhum \cdot P_{dw,amb}) \right] \cdot \left[ 0.08805508 \cdot \frac{Btu / ft^2 \cdot hr}{KJ / hr \cdot m^2} \right] \quad (2.3)$$

Where:

$$T'_{pool} = (T_{pool} - 32) \cdot 5/9 \quad (2.4)$$

$$T'_{amb} = (T_{amb} - 32) \cdot 5/9 \quad (2.5)$$

$$P_{dw,pool} = \left[ 0.004802 + 0.0007109T'_{pool} - 0.00000352 \cdot T'_{pool}{}^2 + 7.2200 \times 10^{-7} \cdot T'_{pool}{}^3 \right] \cdot \left[ 101.325 \cdot \frac{kPa}{atm} \right] \quad (2.6)$$

$$P_{dw,amb} = \left[ 0.004802 + 0.0007109T'_{amb} - 0.00000352 \cdot T'_{amb}{}^2 + 7.2200 \times 10^{-7} \cdot T'_{amb}{}^3 \right] \cdot \left[ 101.325 \cdot \frac{kPa}{atm} \right] \quad (2.7)$$

### 2.2.7. Comparison of the evaporation calculation methods

In Figures 2-5, the evaporative heat flux is shown versus the ambient temperature. The pool water temperature is chosen to be a constant at 27°C. The plots are arranged with increasing relative humidity (40 to 100%) and in each chart the wind speed was chosen to be either 0 or 3.2 km/h.

#### 2.2.7.1 Effect of relative humidity

For a relative humidity (RH) of 100% and an ambient temperature of 27°C the evaporation for all methods is zero, because in this point the ambient temperature and the pool water temperature are the same and the difference in relative humidity is zero.

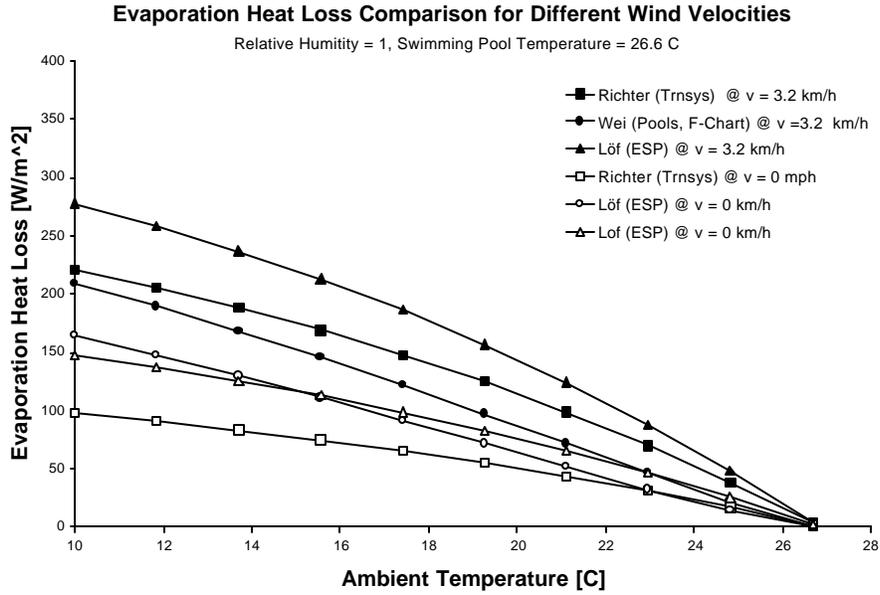


Figure 2.2 Evaporation Comparison at different wind speeds and relative humidity of 1

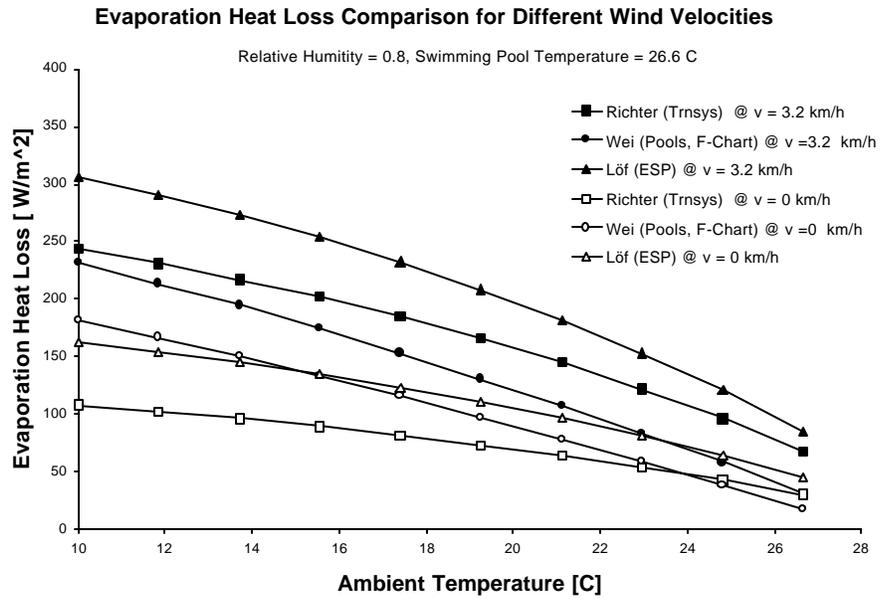


Figure 2.3 Evaporation Comparison at different wind speeds and relative humidity of 0.8

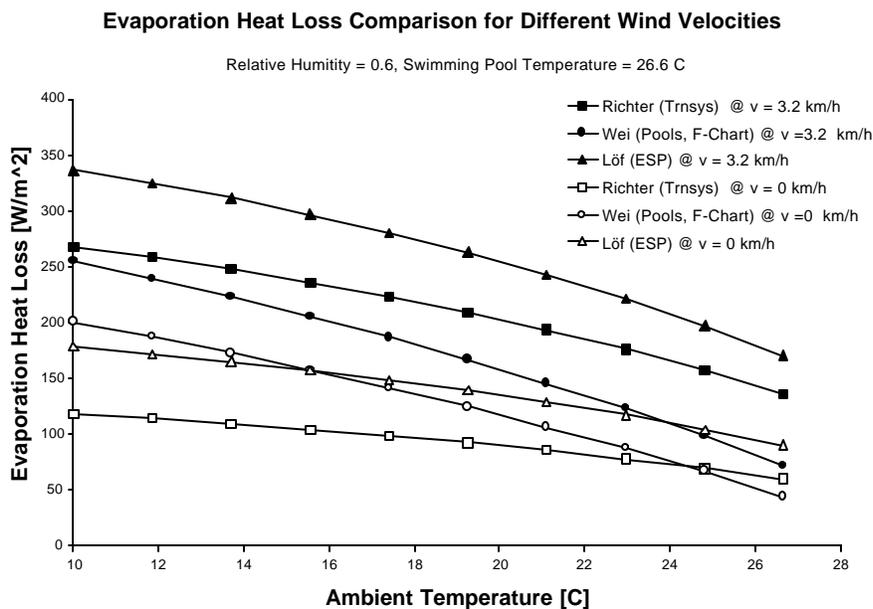


Figure 2.4 Evaporation Comparison at different wind speeds and relative humidity of 0.6

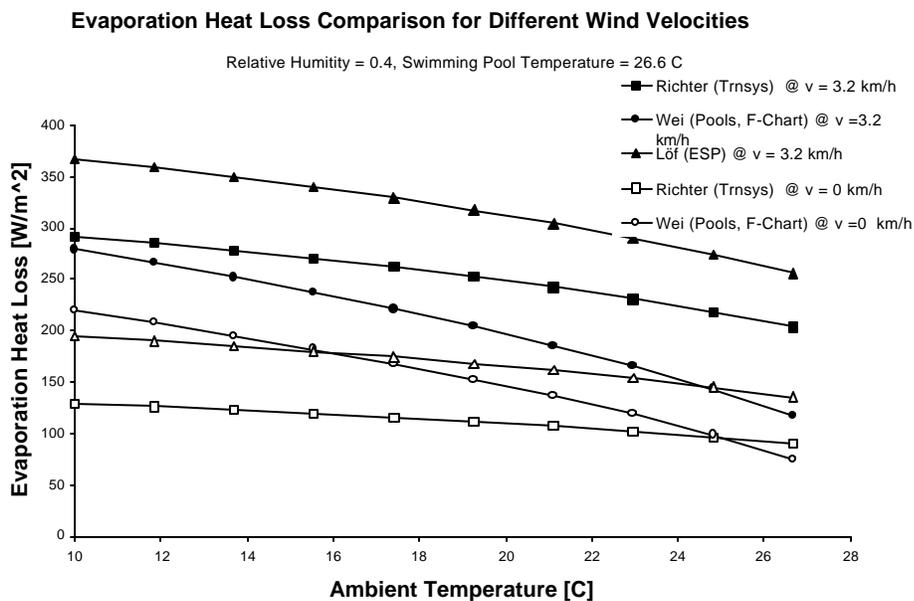


Figure 2.5 Evaporation Comparison at different wind speeds and relative humidity of 0.4

From Figure 2.2 to Figure 2.5, at a constant ambient temperature and decreasing RH, all three methods predict an increase in the evaporation energy loss. For a pool with zero wind speed, the Richter (TRANSSOLAR) and the Löff (ESP) models have essentially the same slope but the Löff algorithm predicts about 30% higher heat loss than the Richter model. The Wei (LBL) curve crosses both the Richter and Löff curves for all of the different relative humidities.

#### 2.2.7.2 Effect of wind speed

With the relative humidity held at 60% the heat transfer predicted by the Richter (TRANSSOLAR) and Löff (ESP) yields by a factor of 2 between a wind speed of 0 and 3.2 km/h. On the other hand the Wei (LBL) correlation varies by a factor of only 1/3. For a wind speed of 3.2 km/h the Löff (ESP) model gives the highest evaporation heat loss as the Wei model (LBL) is constantly about 70 W/m<sup>2</sup> lower than the Löff.

### 2.2.8. Convection/Conduction Calculations

#### 2.2.8.1 POOLS (LBL)

Bowen (1926) related the heat loss by conduction directly to the evaporation heat loss, by considering the process of molecular diffusion from a water surface in the presence of forced convection. This leads to

$$q''_{con,LBL} = R_{bowen} \cdot q''_{evap,LBL} \quad (2.8)$$

and  $R_{\text{bowen}}$  is known as the Bowen Ratio

$$R_{\text{bowen}} = 0.01 \frac{T_{\text{pool}} - T_{\text{amb}}}{P_{\text{pool}} - P_{\text{amb}}} \cdot \frac{P_{\text{atm}}}{29.92} \quad (2.9)$$

Where:

$q''_{\text{con, LBL}}$  = Convection Heat Flux [Btu/hr-ft<sup>2</sup>]

$P_{\text{pool}}$  = Saturated Water Vapor Pressure at  $T_{\text{pool}}$  [inHg]

$P_{\text{amb}}$  = Water Vapor Pressure at  $T_{\text{amb}}$  [inHg]

$T_{\text{pool}}$  = Pool Temperature [F]

$T_{\text{amb}}$  = Ambient Temperature [F]

### 2.2.8.2 F-Chart (F-Chart Software)

The F-Chart program uses the same equations used in the LBL model\*.

### 2.2.8.3 Energy Smart Pools (DOE)

Convection losses are calculated as a function of a convection coefficient and pool water temperature – air temperature difference. The convection coefficient is a function of pool surface wind speed as given by:

$$q''_{\text{con,ESP}} = \text{cover\%} \cdot h_{pc} + (1 - \text{cover\%}) \cdot h \cdot (T_{\text{pool}} - T_{\text{drybulb}}) \quad (2.10)$$

---

\* Eqn (9.7.2) in Solar Engineering of Thermal Processes, by J. A. Duffie and W. A. Beckman is in error. The coefficient 0.0006 should be 0.00022.

and

$$h_{pc} = \frac{1}{0.15 + R_c + \frac{1}{h}} \quad (2.11)$$

$$h = 1 + 0.3 \cdot v_G \quad (2.12)$$

$$v_G = 0.15 \cdot v_{wind} \quad (2.13)$$

Where:

$q''_{con,ESP}$  = Hourly Convection Energy Load [Btu/hr]

Cover% = Pool Area Covered [%]

$T_{pool}$  = Pool Temperature [F]

$T_{drybulb}$  = Dry Bulb Temperature of Air [F]

$h_{pc}$  = Pool-Air Convection Coefficient with Cover [F]

$h$  = Pool –Air Convection Heat Transfer Coeff. [Btu/hr-ft<sup>2</sup>-F]

$v_G$  = Mean Wind Velocity Measured at Ground Level [mph]

$v_{wind}$  = Mean Wind Velocity Measured at Weather Station [mph]

$R_c$  = R-Value of Pool Cover [hr-ft<sup>2</sup>-F/Btu]

#### 2.2.8.4 TRNSYS TYPE 144 (Transsolar)

The TRNSYS model uses a switch to change between a covered and uncovered pool. In the case of an uncovered pool (cover% = 0) the heat loss is a convection loss, and

for a covered surface (cover% = 1) it becomes a conduction heat loss with  $k=\alpha/\delta$  as follows. This heat transfer relationship was developed by Czarnecki (1978):

$$q''_{conv,Type144} = \left[ \alpha \cdot (T_{pool} - T_{amb}) \cdot (1 - cover\%) + cover\% \cdot \frac{\alpha}{\delta} \cdot (T_{pool} - T_{cover}) \right] \cdot \left[ 0.08805508 \cdot \frac{Btu / ft^2 \cdot hr}{KJ / hr \cdot m^2} \right] \cdot 3.6 \quad (2.14)$$

$$\alpha = 3.1 + 4.1 \cdot v_{wind} \cdot \left( 0.44704 \cdot \frac{m/s}{mph} \right) \quad (2.15)$$

Where:

$q''_{conv,Type144}$  = Hourly Convection Energy Load [kJ/hr-m<sup>2</sup>]

$\delta$  = Thickness of Cover [m]

$T_{cover}$  = Temperature of Cover [C]

$T_{pool}$  = Temperature of Pool [C]

### 2.2.9. Comparison of the convection calculation methods

The four models were compared for temperature differences between 10 and 27° C at a relative humidity of 60%. All calculations have been made for an uncovered pool.

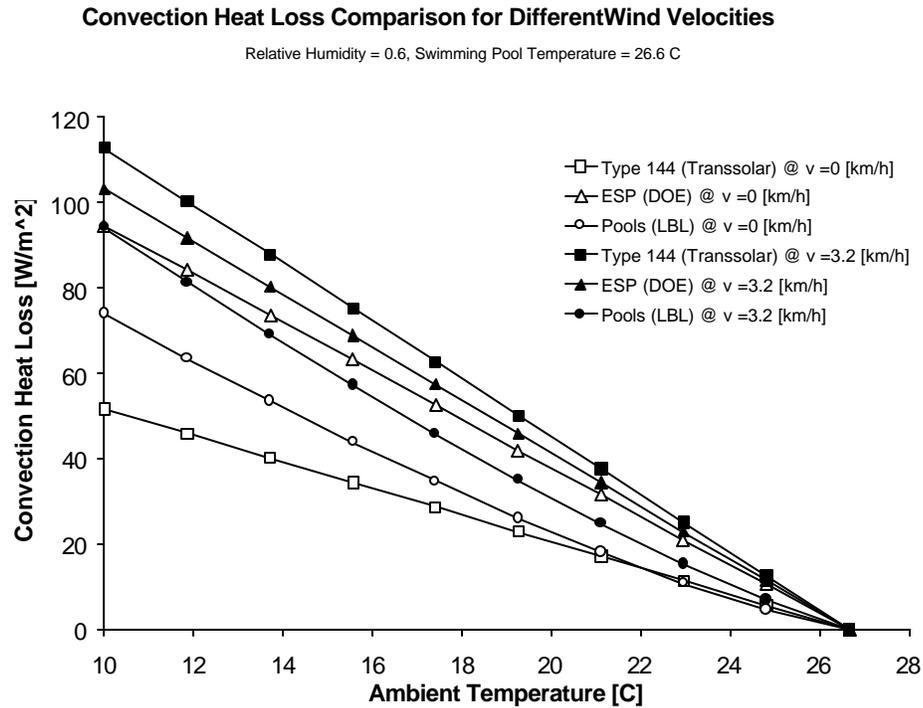


Figure 2.6 Convection losses with no pool cover for a relative humidity=0.6 and two wind speeds

The F-Chart program also uses the Bowen ratio that is implemented in the LBL model.

The Bowen method takes a temperature difference and vapor pressure difference into account to calculate the heat flux due to the convection whereas the ESP and TRANSSOLAR programs use only the temperature difference. All four models consider the wind speed as a variable. The TRANSSOLAR predictions vary by 100% when the changes from 0 to 3.2 km/h. The ESP model is relatively insensitive to different wind velocities. The change predicted by the LBL model is between that predicted by the other two methods. At 27 °F the ambient temperature and the pool temperature are equal and all convective losses are zero.

### **2.2.10. Wind Velocity**

In all of the programs the wind speed to use in the evaporation and convection loss terms is related to the wind speed at the airport. Table 2.1 gives the relationships between the airport wind speed and the local wind speed. The TRANSSOLAR program permits the user to input wind speed measurements at any height. In the other programs a standard “airport measurement height” is assumed. In the TRANSSOLAR program an exponential form is used to convert the airport measurement to local value with 5 levels of shelter. In the other programs a linear “airport reduction factor” is used that typically varies from 0.1 to 0.2 although the POOLS program suggests 0.5 for unsheltered areas and the ESP program suggests 0.3 for no windbreak. At this time it is unclear how these adjustments were determined. Furthermore, by selecting different wind speeds (or “airport reduction factors”), for the different programs, the evaporation and convection correlations can be made to agree for most conditions. There is insufficient experimental information available to pick one correlation over the other.

### **2.2.11. Sky Temperature**

The radiation exchanged between the pool surfaces and the ambient is a function of the pool and sky temperatures and the pool emittance. A gas (and consequently the atmosphere) has the ability to emit and absorb radiation. The so-called “sky temperature” is an equivalent radiation temperature to be used in a simple two-body radiation problem. This sky temperature is not equal to the ambient temperature. The sky is considered to be

a blackbody at an equivalent sky temperature  $T_{sky}$  such that the radiation heat flux can be calculated by:

$$q''_{rad} = \epsilon \sigma (T^4 - T_{sky}^4) \quad (2.16)$$

where  $\sigma$  is the Stefan Boltzmann constant and  $\epsilon$  is the pool emittance. All of the programs use an emittance of about 0.9. The difference in the programs is how they estimate the sky temperature.

### 2.2.11.1 POOLS (LBL)

The LBL model computes the sky temperature in degrees Rankine and is given by a equation suggested by Behrdahl, Grether et al. (1978):

$$T_{sky} = (\epsilon_s T_{amb}^4)^{1/4} \quad (2.17)$$

For clear skies, the empirical relation for sky emissivity,  $\epsilon_s$  is proposed by Brunt (1938):

$$\epsilon_{sc} = 0.605 + 0.048\sqrt{P} \quad (2.18)$$

where P is the water vapor pressure near the ground in millibars.

For cloudy skies, the approximation is made that the emissivity varies linearly from its nominal clear value.

$$\epsilon_s = \epsilon_{sc} + (1 - \epsilon_{sc}) \cdot C \quad (2.19)$$

where C is the cloudiness index between 0 for clear day and 1 for cloud covered days.

### 2.2.11.2 F-Chart Software and TRANSSOLAR

Both methods calculate the sky temperature by using the equation developed by Martin and Behrdahl (1984)

$$T_{sky} = T_{amb} [0.711 + 0.0056 \cdot T_{dp} + 0.000073 \cdot T_{dp}^2 + 0.013 \cdot \cos(15 \cdot t)]^{1/4} \quad (2.20)$$

Where  $T_{sky}$  and  $T_{amb}$  are in degrees Kelvin,  $T_{dp}$  is the dew point temperature in degrees Celsius and  $t$  is the hour from midnight. This equation is based on experimental data that covered a temperature range from  $-20$  to  $+30$  degrees Celsius.

### 2.2.11.3 Energy Smart Pools (DOE)

The Energy Smart Pools Program estimates the sky temperature to be the air temperature minus 20 degrees F.

## 2.2.12. Comparison of the radiation calculation methods

The radiative heat fluxes shown in Figure 7 vary between 90 and 190  $\text{W}/\text{m}^2$  for a pool temperature of  $10^\circ\text{C}$  and between 5 and 110  $\text{W}/\text{m}^2$  for a pool temperature of  $27^\circ\text{C}$ . Only TRANSSOLAR and F-Chart use humidity sensitive equations and they vary as shown. For most conditions the ESP and LBL programs predict lower values than the F-Chart and TRANSSOLAR programs. The LBL predictions approach zero at  $T_{amb}=T_{pool}$ , which is clearly incorrect. F-Chart and TRANSSOLAR use the more recent LBL sky temperature correlations, which generally predicts a higher radiation heat loss than ESP.

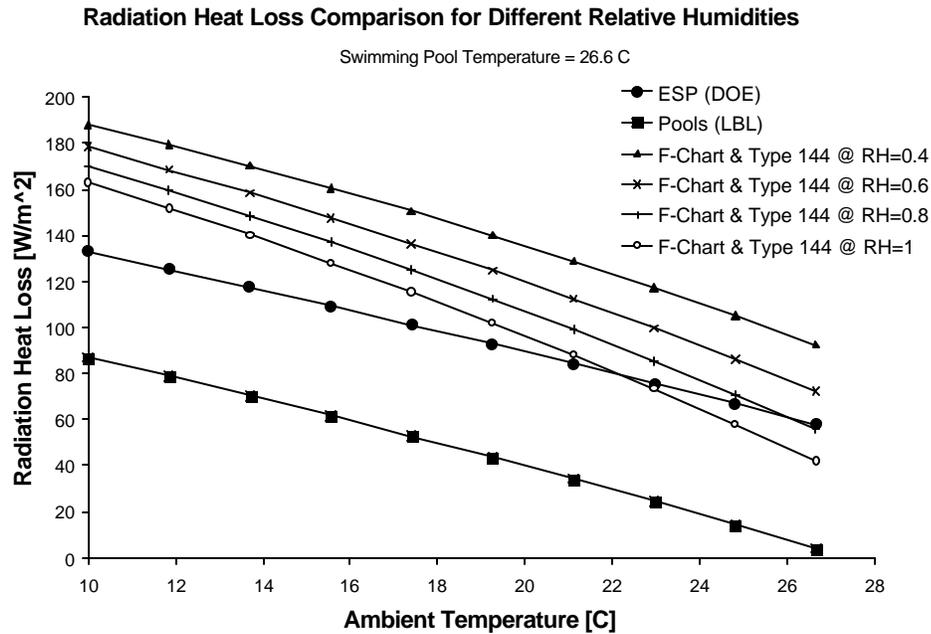


Figure 2.7 Radiation Losses from Uncovered Pool Surface

## 2.3 Summary

To compare the four computer simulations, all losses from an uncovered outdoor swimming pool have been added up and the fractions of evaporation, radiation and convection were evaluated as a function of ambient temperature. The relative humidity was set to 60 %. Figure 2.8 shows the predicted heat losses from all four computer models at two different wind speeds. As expected the energy lost increases at increasing wind speed and decreasing ambient temperature. The trends are similar for all of the programs but the magnitudes are different.

The highest loss without wind is calculated by the F-Chart program and is about 440 W/m<sup>2</sup>. For a wind of 3.2 km/h the TRANSSOLAR and ESP programs predict the

highest heat loss of about  $570 \text{ W/m}^2$  at 10 degrees Celsius and  $220 \text{ W/m}^2$  at 27 degree Celsius.

For no wind the evaporation heat flux behaves very similarly in the correlations used in the ESP, LBL, and the F-Chart model. The TRANSSOLAR program estimates the evaporative losses to be somewhat lower than the other programs. The ESP model for a wind velocity of 3.2 km/h calculates the highest evaporation loss. The evaporative losses for ESP are higher than f-chart while the TRANSSOLAR evaporative losses are smaller than fchart at zero wind speed and about the same as fchart at a 3.2 km/h wind speed. With the large uncertainty in estimating the evaporative losses, and with no definitive experiments available to test the predictions, it is unclear which algorithm should be used. All models show the same characteristics and about the same magnitude for the convective heat loss. At zero wind speed TRANSSOLAR predicts the smallest convection loss and at 3.2 km/h TRANSSOLAR predicts the largest convection heat loss. Since the convection heat loss in all of the programs is small relative to the evaporation and radiation heat losses, the algorithm choice does not make much difference in the overall heat loss. Perhaps the most surprising result is the variation in the radiation heat loss predicted by the four programs. The different assumptions made for the sky temperature have a very significant impact on the radiative losses.

## Pool Losses for an Uncovered Pool

Relative Humidity=0.6, Pool Temperature = 27 C

Evaporation Convection Radiation

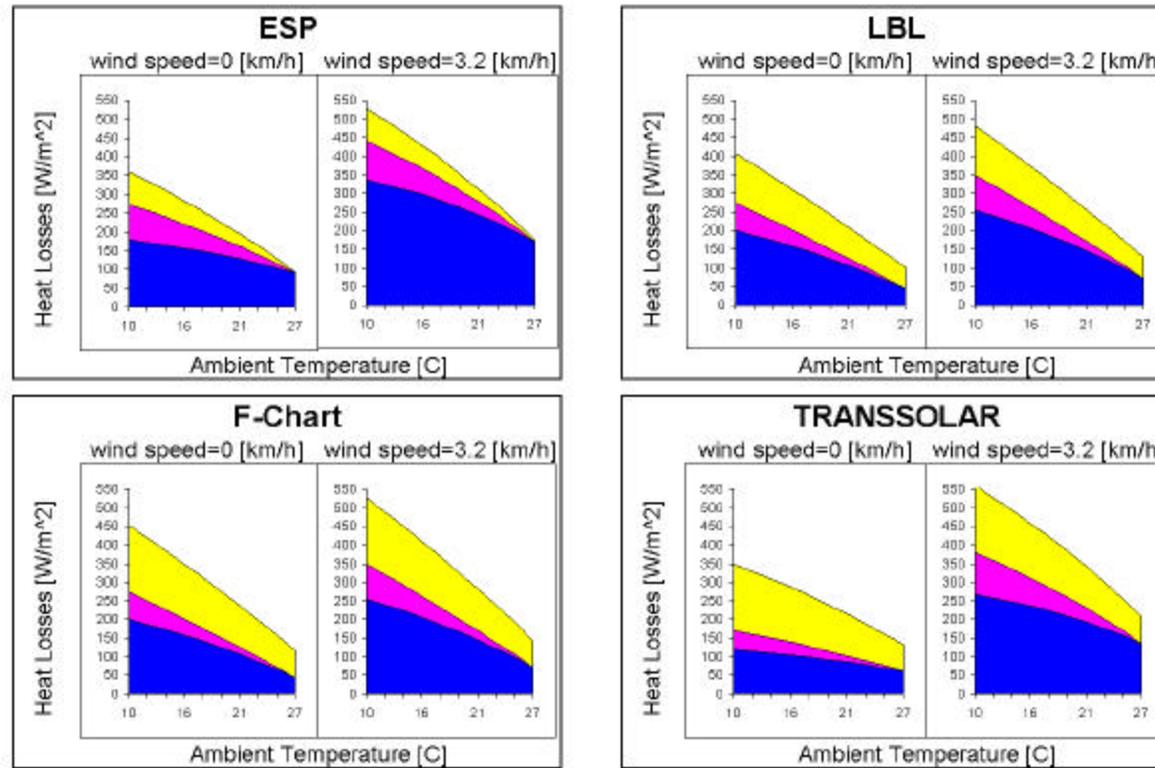


Figure 2.8 Total Energy Losses

The LBL program used a version of the sky temperature developed by Martin and Berdhal in 1978 but this algorithm was modified in 1984, after the POOLS program was developed. The 1984 algorithm is used in both the F-chart and TRANSSOLAR programs and predicts the highest radiative heat flux from the pool surface to the surroundings. The assumption of using  $T_{\text{amb}}$  minus 20 °F for the sky temperature, as used in ESP program, appears to underestimate radiative losses.

Since the overall losses are relatively equal and the correlations behave similar there is no advantage of choosing one approach over the other. For further studies the TYPE 144 developed by the TRANSSOLAR Company has been used to model the swimming pool because it was already available as a subroutine for TRNSYS.



# Chapter 3

## The Air Conditioner Model

### 3.1 The Refrigeration Cycle

For the purpose of this research two different air conditioner models were developed. First, an approach using a constant coefficient of performance (COP) was applied, because house and building temperatures are nearly constant (since the pool is heated). Also, it will be shown for energy observations concerning the swimming pool a constant COP model is sufficient. Second, when examining the energy consumption of an air conditioner that rejects heat either to the ambient air or to the swimming pool water, it is important to have a more precise model in order to be able to obtain a temperature sensitive result. This leads to an air conditioner model that has a COP as a function of the condenser inlet temperature and therefore allows more precise economic observations. The following section describes the components of both the constant and the variable COP model.

### 3.2 The Constant COP Model

As a first approximation for the air conditioner model, a simple thermodynamic approach was used where the coefficient of performance was assumed to be constant.

Figure 3.1 shows an idealized refrigeration system described by energy fluxes into the system boundary at certain temperatures of  $T_{pool}$  and  $T_{house}$ .

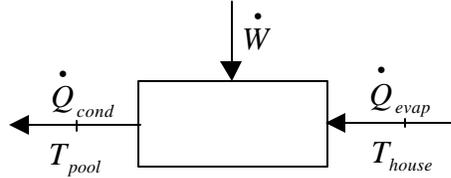


Figure 3.1 A simple thermodynamic approach for an air conditioner

For the present work  $\dot{Q}_{evap}$  is the building cooling demand and  $\dot{Q}_{cond}$  the amount of energy added to the pool.

Performance of a refrigeration cycle is usually described by a coefficient of performance. The COP is defined as ratio of the amount of removed heat to the required energy input to operate the cycle

$$COP = \frac{\text{Useful refrigerant effect}}{\text{Net energy supplied}} = \frac{\dot{Q}_{evap}}{\dot{W}} \quad (3.1)$$

Applying the first Law of Thermodynamics for the system in Figure 3.1 yields:

$$\dot{Q}_{cond} = \dot{W} - \dot{Q}_{evap} \quad (3.2)$$

Combining ( 3.1 ) and ( 3.2 ) leads to an equation that is only a function of the building cooling demand and the COP. By fixing the COP and calculating the cooling demand of the building the energy rejected to the swimming pool can be obtained by the following equation:

$$\dot{Q}_{cond} = -\dot{Q}_{evap} \left( 1 - \frac{1}{COP} \right) \quad (3.3)$$

The constant COP model that was built into the Swimming Pool Air Conditioner Simulation is based on the single equation model given by equation ( 3.3 ). It was found that for energy balance purposes of the swimming pool the constant COP model sufficiently predicts the air conditioner behavior. For further information see paragraph 3.5.

### **3.3 The Variable COP Model**

#### **3.3.1. Introduction**

In order to obtain detailed information on the system performance as a function of environmental impacts a more precise model of a vapor compression air conditioner was modeled using EES (Engineering Equation Solver). The standard thermodynamic approach was used where the pressure drop in the evaporator and the condenser were neglected and the isentropic compressor efficiency was assumed to be constant. To transfer thermal energy from an enclosure such as a refrigerator or a building it is necessary to have a fluid that has the ability to absorb the energy from one area and reject it to another area, usually through condensation and evaporation. The first and second laws of thermodynamics can be applied to individual components to determine mass and energy balances and the irreversibility of the components.

### 3.3.2. The Vapor-Compression Cycle

Vapor-compression cycles are most common in air conditioning systems today. A general illustration of a vapor-compression cycle is shown in Figure 3.2. The pressure-enthalpy diagram visualizes the phase changes throughout the process Figure 3.3

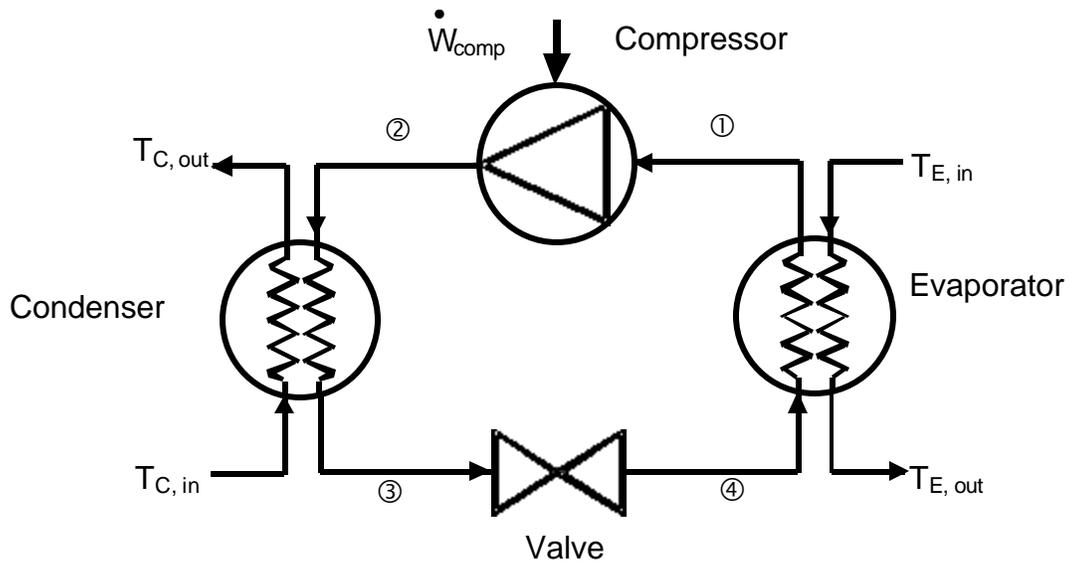


Figure 3.2 Schematic Diagram of a vapor – compression cycle

The evaporator, where the desired refrigeration effect is achieved, was chosen to start the analysis. As the refrigerant passes through the evaporator, heat transfer from the refrigerated space results in the vaporization of the refrigerant.

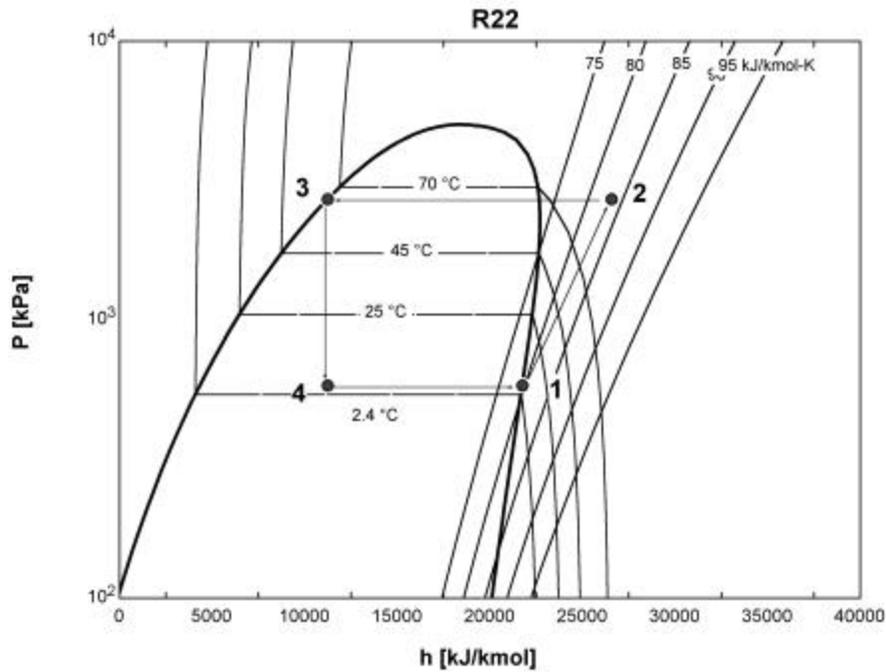


Figure 3.3 The pressure – enthalpy diagram for the vapor - compression cycle

For a control volume enclosing the refrigerant side of the evaporator, the mass and energy rate balances reduce to give the rate of heat transfer per unit mass of refrigerant flow

$$\dot{Q}_{41} = \dot{m} \cdot (h_1 - h_4) \quad (3.4)$$

where  $\dot{m}$  is the mass flow rate of the refrigerant. The heat transfer rate  $\dot{Q}_{41}$  is referred to as the refrigerant capacity. The refrigerant leaving the evaporator is compressed to a relatively high pressure and temperature by the compressor. Assuming no heat transfer to or from the compressor, the mass and energy balances for a control volume enclosing the compressor give

$$\dot{W}_{12} = \dot{m} \cdot (h_2 - h_1) \quad (3.5)$$

where  $\dot{W}_{12}$  is the power input to the refrigeration cycle. Because the enthalpy at state 2 remains unknown, an isentropic compression can be assumed to calculate an ideal compressor work and is later corrected by the compressor efficiency to derive the actual compressor work.

Next, the refrigerant passes through the condenser, where the refrigerant condenses and heat is transferred from the refrigerant to the cooling fluid, which is usually air or water. For a control volume enclosing the refrigerant side of the condenser, the rate of heat transfer of the refrigerant is

$$\dot{Q}_{23} = \dot{m} \cdot (h_3 - h_2) \quad (3.6)$$

Finally the refrigerant enters the expansion valve and expands to the evaporator pressure. This process is usually modeled as a throttling process, for which

$$h_3 = h_4 \quad (3.7)$$

The refrigerant pressure increases in the irreversible adiabatic expansion, and there is an accompanying increase in specific entropy. The refrigerant exits the valve at state 4 as a two-phase liquid-vapor mixture.

### 3.3.3. The Refrigerant

The refrigerant is the working fluid for an air conditioning cycle. Although the Montreal Protocol, which controls the production of ozone-depleting substances, prescribes a phase-out of HCFC (Hydrochloroflouorocarbons) in the next 30 years, the

HCFC R22 is still used in air conditioning systems in the U.S. To be able to calibrate a computer model and to compare results HCFC R22 has been used for the computer simulation.

### 3.3.4. Performance of a Vapor-Compression Cycle

As shown in equation ( 3.1 ) the coefficient of performance is defined as the ratio of net energy that is supplied to the system to the work that is needed to run the cycle. For a mechanical vapor compression system, the net energy supplied is usually in form of work, mechanical or electrical, and may include work to the compressor and fans or pumps.

$$COP = \frac{\dot{Q}_{evap}}{\dot{W}_{comp} + \dot{W}_{pump/fan}} \quad (3.8)$$

### 3.3.5. The Volumetric Compressor Efficiency Model

The compressor efficiency is not constant for all conditions. The effect will be shown in the following section.

The compressor refrigerant flow rate is a decreasing function of the pressure ratio due to the re-expansion of the vapor in the clearance volume. With the refrigerant vapor modeled as an ideal gas, the volumetric flow rate is given in the ASHRAE Fundamentals Handbook ASHRAE (1997) by the following:

$$\dot{v} = \dot{v}_{dis} \left( 1 + C - C \left( \frac{P_{dis}}{P_{suc}} \right)^{1/n} \right) \quad (3.9)$$

where  $v$  is the volume flow rate,  $v_{dis}$  is the displacement of the compressor,  $C$  is the clearance factor ( $v_{clearance}/v_{cylinder}$ ),  $P_{dis}/P_{suc}$  is the cylinder pressure ratio and  $n$  the polytropic exponent.

The volumetric compressor efficiency is defined as ratio of the volume flow rate to the compressor displacement rate and is therefore affected by equation (3.9). Applying the definition for polytropic expansion and compression

$$pv^n = const \quad (3.10)$$

the volumetric efficiency can be calculated by

$$\eta_{vol} = 1 - C \cdot \left( \frac{v_{suc}}{v_{dis}} - 1 \right) \quad (3.11)$$

### 3.3.6. The Effectiveness-NTU Heat Exchanger Model

The evaporator and the condenser heat exchangers are modeled using the effectiveness-NTU method. The effectiveness of the heat exchanger is defined as the ratio of the actual heat transfer for a heat exchanger to the maximum possible heat transfer rate

$$\varepsilon = \frac{\dot{Q}_{act}}{\dot{Q}_{max}} \quad (3.12)$$

where the maximum heat transfer is

$$\dot{Q}_{\max} = \left( c_p \cdot \dot{m} \right)_{\min} \cdot (T_{h,i} - T_{c,i}) \quad (3.13)$$

where  $\left( c_p \cdot \dot{m} \right)_{\min}$  is the minimum of the heat capacity and mass flow rate product

for either the hot or the cold fluid.

The number of transfer units (NTU) is a dimensionless parameter that is widely used for heat exchanger analysis and is defined as

$$NTU = \frac{UA}{C_{\min}} \quad (3.14)$$

where  $C_{\min} = \left( c_p \cdot \dot{m} \right)_{\min}$  and  $UA$  is the overall heat loss coefficient.

From these equations the effectiveness-NTU relation can be determined for different heat exchanger designs and were found in Incropera and DeWitt (1985). For a counterflow arrangement the effectiveness is

$$\varepsilon = \frac{1 - \exp(-NTU(1 - C_r))}{1 - C_r} \quad (3.15)$$

$$\text{where } C_r = \frac{C_{\min}}{C_{\max}} \quad (3.16)$$

For heat exchangers in which condensation occurs, the temperature remains constant and  $C_h \rightarrow \infty$ . Conversely, in an evaporator it is the cold fluid that experiences a change in phase and remains at a nearly uniform temperature ( $C_c \rightarrow \infty$ ). In that case the value of  $C_r$  is defined to be zero and for all heat exchangers the effectiveness is calculated by

$$\varepsilon = 1 - \exp(-NTU). \quad (3.17)$$

### 3.3.7. Fan and Pump

For completeness in the evaluation of the coefficient of performance the fan and pump work have to be included into the calculations.

#### 3.3.7.1 Fan Laws

Residential split air conditioners usually have a fan built into their outdoor unit that blows air through the condenser to achieve the desired condensing effect. The fan characteristics can be described by three relationships that predict the effect on fan performance of changing such quantities as the density of the air ( $\rho$ ), operating speed (N), and size of the fan (d). These equations are known as the fan laws and can be found in Roberson and Crowe (1985):

$$\dot{V} = C_v \cdot N \cdot d^3 \quad (3.18)$$

$$P_s = C_p \cdot N^2 \cdot d^2 \cdot \rho \quad (3.19)$$

$$\dot{W} = C_w \cdot N^3 \cdot d^5 \cdot \rho \quad (3.20)$$

$\dot{V}$  is the capacity,  $P_s$  is the static pressure rise and  $\dot{W}$  is the power. The various coefficients ( $C_v$  - capacity coefficient,  $C_p$  - pressure coefficient,  $C_w$  - power coefficient) can be determined for a fixed diameter, speed and inlet air density taken from manufacturer data.

### 3.3.7.2 Pump Modeling

The swimming pool water is moved through the condenser by a pump. For flow through pipes the Bernoulli Equation can be applied:

$$\dot{W} = \Delta P_v \cdot \dot{V} \quad (3.21)$$

$$\Delta P_v = \zeta \cdot \frac{\rho}{2} \cdot u^2 \quad (3.22)$$

$\dot{W}$  is the pump power,  $\Delta P_v$  is the pressure drop because of friction,  $\dot{V}$  is the volumetric flow,  $\zeta$  is the hydraulic loss figure,  $\rho$  is the density of water and  $u$  is the velocity. Again, the model can be validated by applying manufacturer data for a pump. The hydraulic loss figure is assumed to be unity for corroded pipes.

## 3.4 The EES Air Conditioner Simulation

### 3.4.1. Introduction

A useful engineering tool, the Engineering Equation Solver (EES), was employed to solve the coupled non-linear equations governing an air conditioning system. Besides a fast algorithm EES provides property data for refrigerants and other species, that make the calculations much easier. Additionally a diagram window, as shown in Figure 3.4, can be used to present the results in a window that may include a graphic of the simulated process. In this work the main diagram window provides an overview about input and output variables. Two child windows carry information of the pump and fan settings.

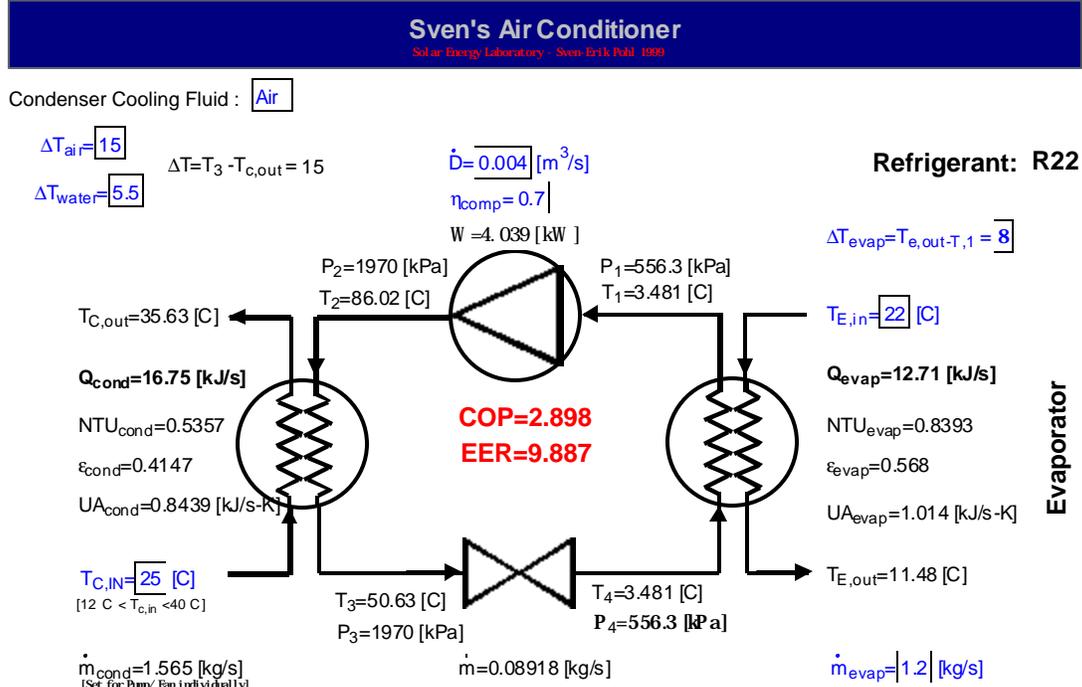


Figure 3.4 The EES Diagram window provides a user-friendly input and output screen

The objective of the EES program is to study the changes in the coefficient of performance for the different cooling fluids of the condenser. A conventional air conditioner rejects the energy to the ambient air that is moved through the heat exchanger by a fan. The air conditioner that transfers the condenser heat to the swimming pool uses water as cooling fluid. The idea is to generate curve fits of the coefficient of performance versus condenser inlet temperature, i.e. the pool temperature. To simulate the air conditioner curve fits were also developed for the change in the capacity as a function of the condenser inlet temperature.

### 3.4.2. Calibration of the Simulation

To achieve results that reflect the real behavior of a conventional residential air conditioner the EES simulation was calibrated using manufacturer performance data. The

available information covered performance and capacity data as a function of the condenser coolant inlet temperature. Also, the fan parameters were taken from catalog data and the building conditions were fixed. Two simulation parameters, the compressor displacement rate and the compressor efficiency, were varied to minimize the error in COP between catalog data and simulation results.

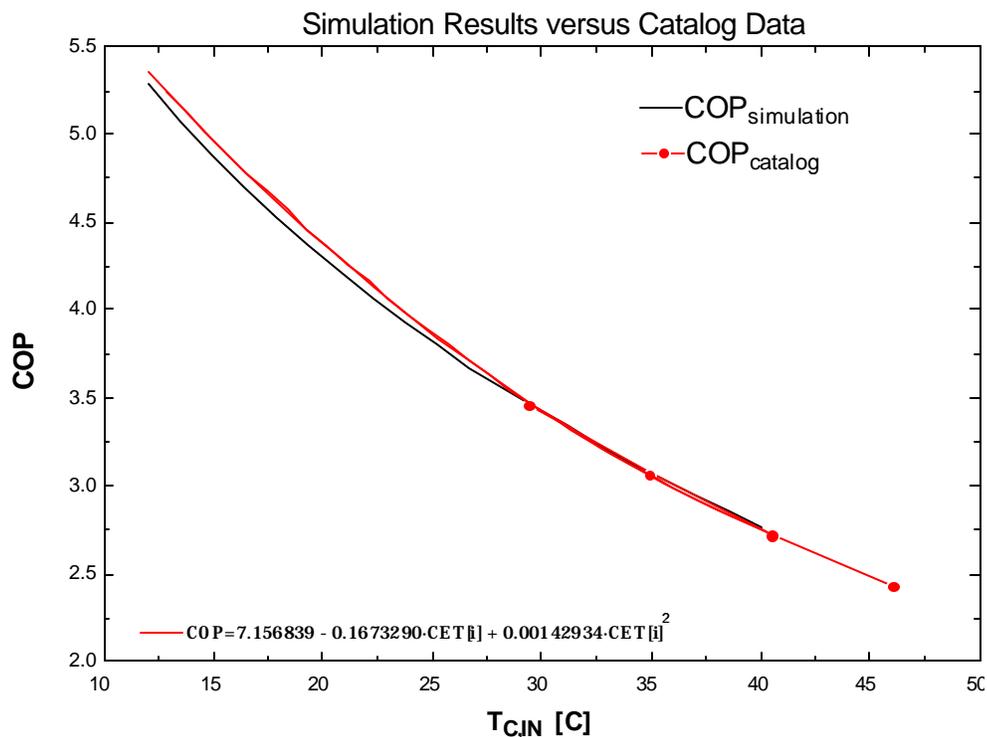


Figure 3.5 Manufacturer data and simulation results agree within a small difference

Figure 3.5 shows the four data points given by manufactures catalog data and the simulation results after the minimization. A curve fit represents the catalog data over the temperature range of interest. The behavior as given in the catalog can be predicted within a small error.

### 3.4.3. A Fair Comparison

A major problem was how to make a fair comparison of a conventional air-cooled air conditioner to a water-cooled air conditioner. Manufacturers data are available for residential air conditioners, but for a swimming pool air conditioner such information is not available. Properties such as the heat exchanger effectiveness or the overall heat loss coefficient were examined but led always to the problem that there is no information for a water-cooled system. Finally the terminal temperature difference (TTD) was taken to compare the different systems, because values were available from air conditioner designer experience. The approximate values for the terminal temperature difference are assumed to be about 10 C for air and 5.5 C for water.

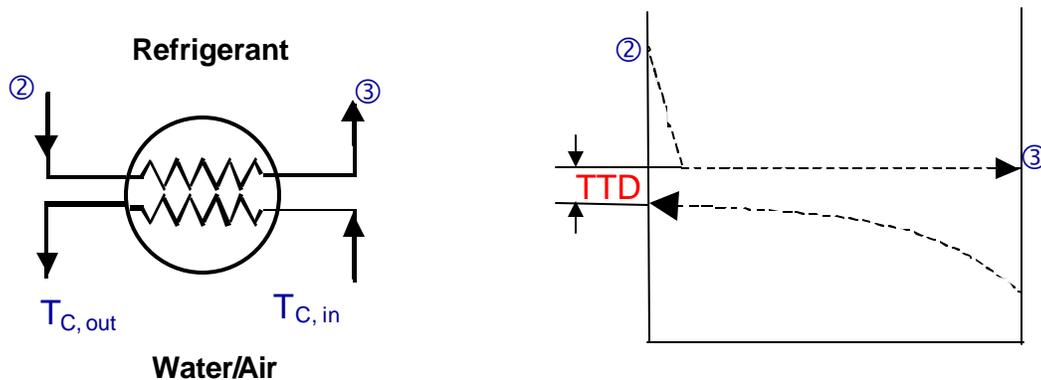


Figure 3.6 The definition of the terminal temperature difference (TTD) for a condenser.

The TTD is the temperature difference between the condenser cooling fluid outlet temperature and the condensation temperature of the refrigerant. Figure 3.6 shows the definition of the TTD for the vapor-compression cycle described above. The variation in the terminal temperature differences for water and air is based on their properties and different mass flow rates through the heat exchanger. Naturally, water performs better

than air in a heat exchanger. Thus, by changing the working fluid of the condenser the terminal temperature difference changes. It is also necessary to include the designer knowledge of the mass flow rate. The TTD for water of 5.5 C goes along with a water flow rate of 36 cm<sup>3</sup>/s per kW of refrigeration effect. Accordingly, a 10 kW air conditioner would have a water flow rate of 360 cm<sup>3</sup>/s and a TTD of 5.5 C.

The temperature approach was used to run calculations for a range of condenser cooling fluid inlet temperatures for both, water and air. The performance of the two systems (water-cooled and air-cooled) can be seen in Figure 3.7. For a better understanding, a change in the TTD can be explained as a change in the physical size of the heat exchanger. In other words, a smaller heat exchanger results in a larger TTD. Therefore, for an increasing TTD for water, which means a decreasing heat exchanger size, the performance is decreasing.

By comparing a water-cooled heat exchanger with a large temperature difference to an air-cooled heat exchanger with a small temperature difference a condition can be found where both air-cooled and water-cooled perform the same.

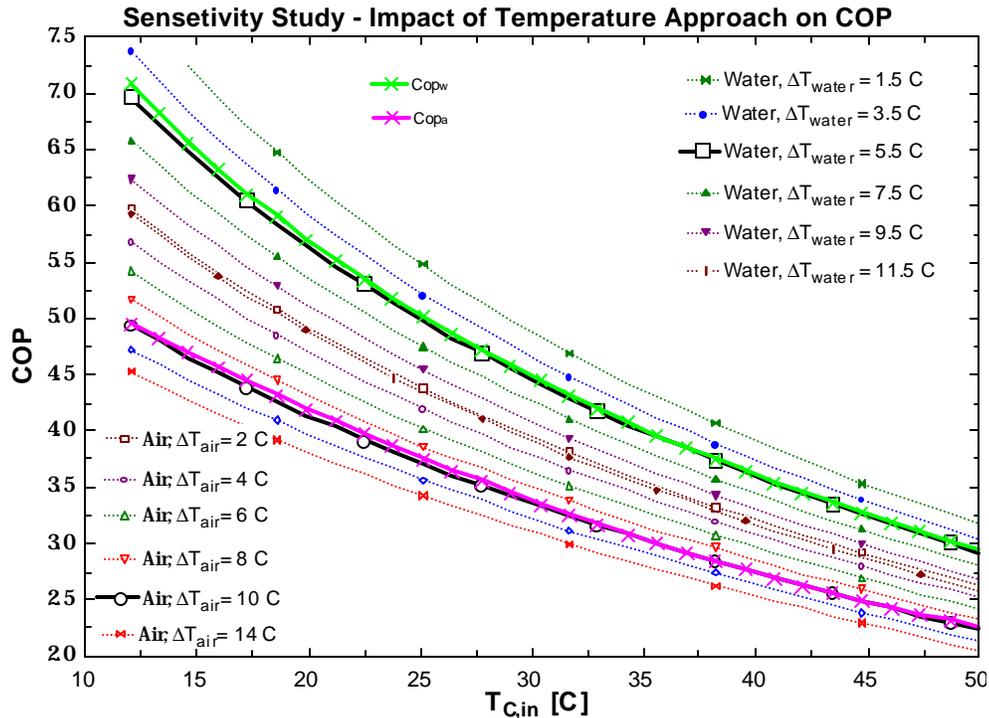


Figure 3.7 Performance of air conditioner simulation for water and air for different temperature approaches. If the TTD increases, the performance decreases.

As the condenser inlet temperature increases the performance decreases. That means the hotter the ambient conditions the more energy consumed by the air conditioner. Ironically, an air conditioner performs best when its not needed.

#### 3.4.4. Implementation of the AC Model into TRNSED Simulation

The air conditioner model has been analyzed in EES but the transient system simulation is to be performed in TRNSYS. Consequently, it is necessary to implement the results from the EES model into TRNSYS. The best realization would be to take the equations from the EES program and use them in a TRNSYS Type. Unfortunately a number of iterations are needed to solve for the system equation resulting in a significant

solution time. This fact makes it difficult to transfer the EES simulation into TRNSYS. For this purpose another feature of EES was used – the results were transferred using curve fits. These curve fits were combined in a way that for each TTD the coefficient of performance and the capacity can be determined. In Figure 3.7 the result is shown for the desired TTD's of 5.5C and 10C. Curve fits of the coefficient of performance and the capacity as the function of the condenser inlet temperature were implemented in a Fortran subroutine. This subroutine is then called by the TRNSYS, which returns information about the air conditioner performance.

#### **3.4.5. Sensitivity Study for Temperature Approach**

In order to investigate the sensitivity of a change in the TTD the TRNSYS model was run for a range of terminal temperature differences. Figure 3.8 presents the result by showing the seasonal savings as a function of the TTD for water and air. The savings are based on the difference between the cost of an air-cooled system and a water-cooled system over the time period of May 1<sup>st</sup> and October 1<sup>st</sup>. It can be seen that the savings for the proposed approach of 5.5 C for water and 10 C for air would be about \$36 of air conditioner operation cost. Assuming an uncertainty of  $\pm 10\%$  in the TTD the amount of saved money differs about  $\pm \$5$ , which is about 14 % of the original savings.

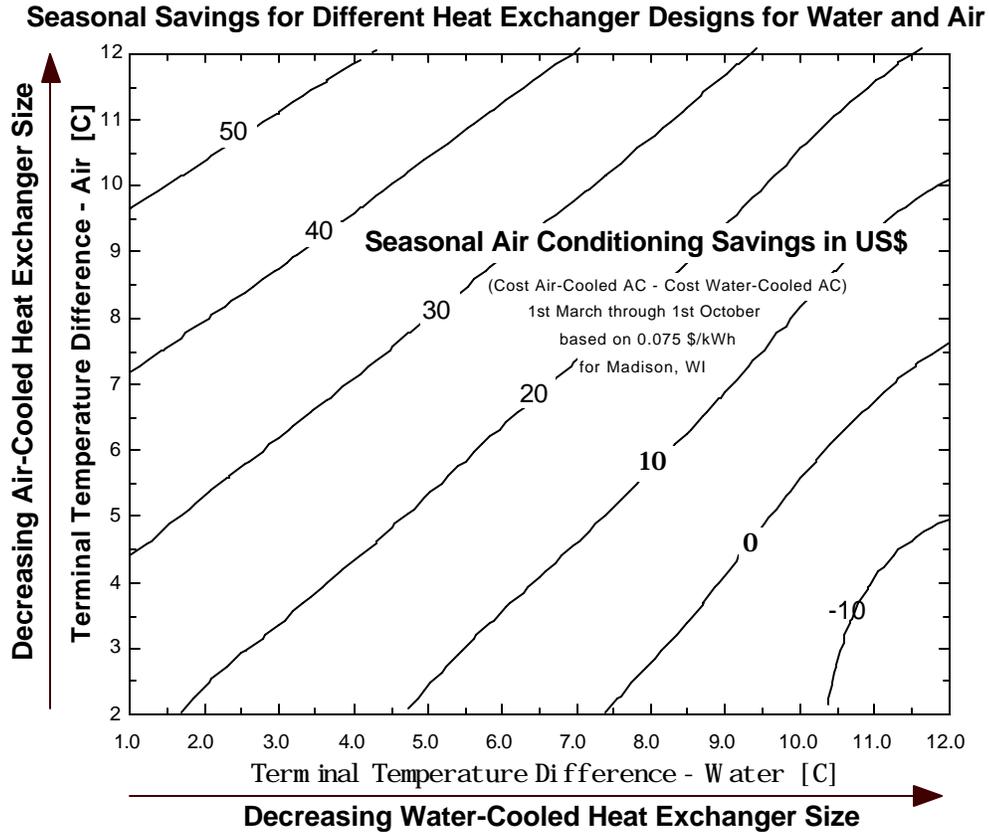


Figure 3.8 Seasonal Air Conditioning Savings for different temperature approaches

### 3.4.6. Generalization of the Air Conditioner Correlations

To make the correlations for coefficient of performance and capacity suitable for other air conditioners than the one employed in this work the equations have been normalized by dividing the correlations by their corresponding value at a rating temperature of 35 C. The rating temperature is a standard rating point for air conditioners tested using the Air-Conditioning and Refrigerant Institute's (ARI) rating system. The Air-Conditioning and Refrigeration Institute (ARI) provides a certification program for unitary air conditioners in their ARI 210/240-89 Norm. The idea is that a user of the

TRNSYS program enters the performance values for his specific air conditioner that has been tested under ARI conditions and the program itself doesn't have to be changed.

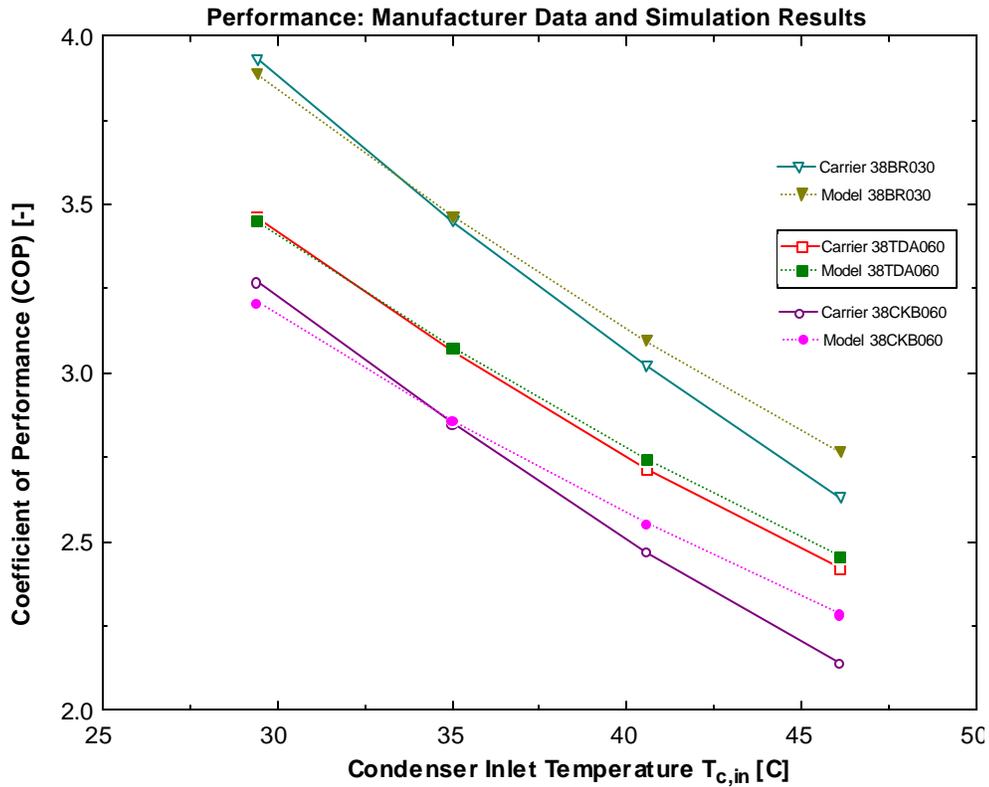


Figure 3.9 Application of the air conditioner correlations on different unit sizes

Figure 3.9 shows a comparison of different air conditioner manufacturer data and the fitted correlations. The plot indicates that the generalization of the correlations is limited. It works fine for the original set of data but for other unit sizes it only can give an approximate performance map. An objective for future research could be an investigation of a better way to generalize the gained air conditioner correlations.

### 3.5 Summary

In this section two air conditioner models were presented. The first model is based on a simple thermodynamic approach that assumes a constant coefficient of performance. It was found that this one-equation simulation is sufficient to predict the swimming pool temperature.

To obtain information about the energy consumption of a water-cooled air conditioner a second more detailed model was implemented that calculates the performance and the capacity of two cooling fluids, water and air based on the condenser inlet temperature and the terminal temperature difference.

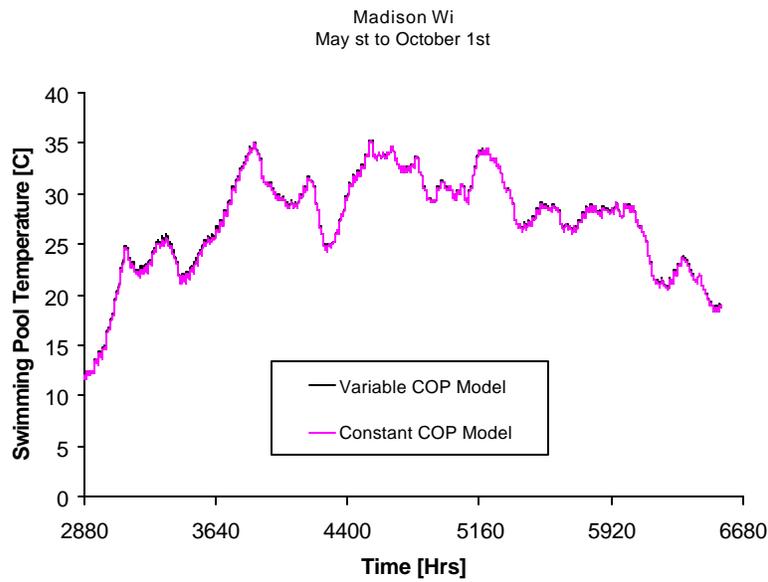


Figure 3.10 Effect of different air conditioner models on swimming pool temperature

Figure 3.10 shows a comparison of the constant COP model and the variable COP model. It can be seen that there is no effect on the swimming pool temperature although

the two models have a different performance and do not deliver the same amount of heat to the pool.

For predicting the swimming pool temperature the changes due to a varying COP can be neglected. But for an economic analysis, a more detailed model needs to be used, because of the sensitivity of the power demand of the air conditioner to changes in the COP.



# Chapter 4

## Weather Data

### 4.1 Introduction

Since this work is based on theoretical simulations of components that are influenced by the environmental conditions, good estimates of these environmental conditions are needed in order to gain reliable results. Two approaches are compared in this section.

### 4.2 The Typical Meteorological Year (TMY and TMY2)

The National Renewable Energy Laboratory NREL (1995) derived weather data from the National Solar Radiation Data Base (NSRDB), which contains measured or modeled solar radiation and meteorological data for 239 US stations for the 30-year period from 1961-1990. A typical meteorological year (TMY) is a data set of hourly values of solar radiation and meteorological elements for a 1-year period. It consists of months selected from individual years, concatenated to form a complete year. TMY weather data represents conditions judged to be typical over a long period of time, such as 30 years. To distinguish between recent TMY data files and earlier releases, the new TMY data sets are referred to as TMY2.

### **4.3 Generated Weather**

For some locations only monthly average weather information is available. To allow hourly simulations for places with limited weather data, a TRNSYS TYPE “Weather Generator” has been developed by Knight, Klein et al. (1991). This component generates hourly weather data given the monthly average values of solar radiation, dry bulb temperature, humidity ratio and wind speed. The data are generated in a manner such that their associated statistics are approximately equal to the long-term statistics at the specified location. The purpose of this method is to generate a single year of typical data, similar to a Typical Meteorological Year.

### **4.4 Comparison of TMY Data and Generated Weather**

For the purpose of this study, the TMY2 data and the generated weather set have been compared to check the agreement of both methods. For locations in the US, the TMY2 weather data set provides almost as much information as the generated weather. However, the file size for the generated data is much smaller than TMY2 data files. For example, the whole data file for the weather generator that includes 329 locations has only 91 kByte compared to 1.2 Mbytes for one location of TMY2 data.

For a fair comparison, of the generated weather is based on monthly average ambient temperatures and monthly average solar radiation values obtained from TMY2 weather data for Madison, WI. These monthly average data are inputs to the weather generator.

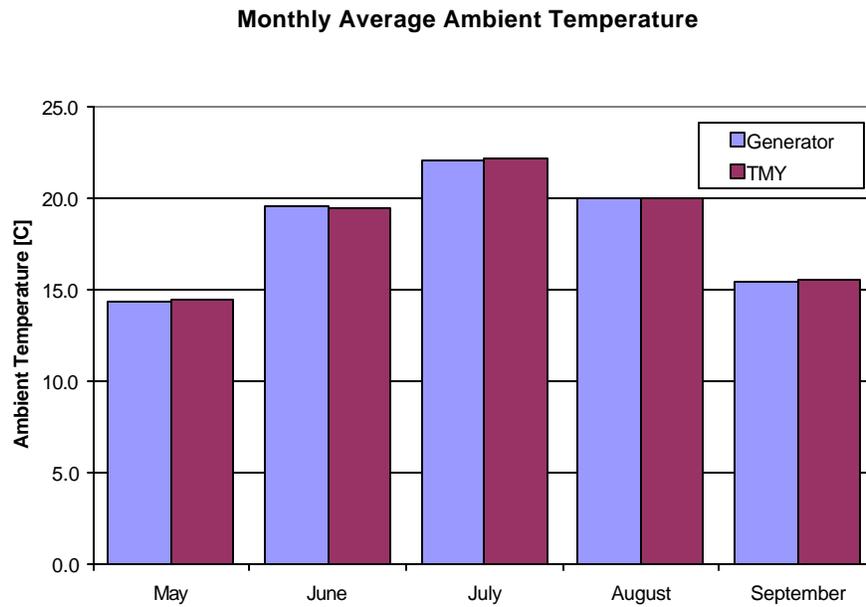


Figure 4.1 Monthly Average Ambient Temperatures for Generated Weather and TMY Weather Data

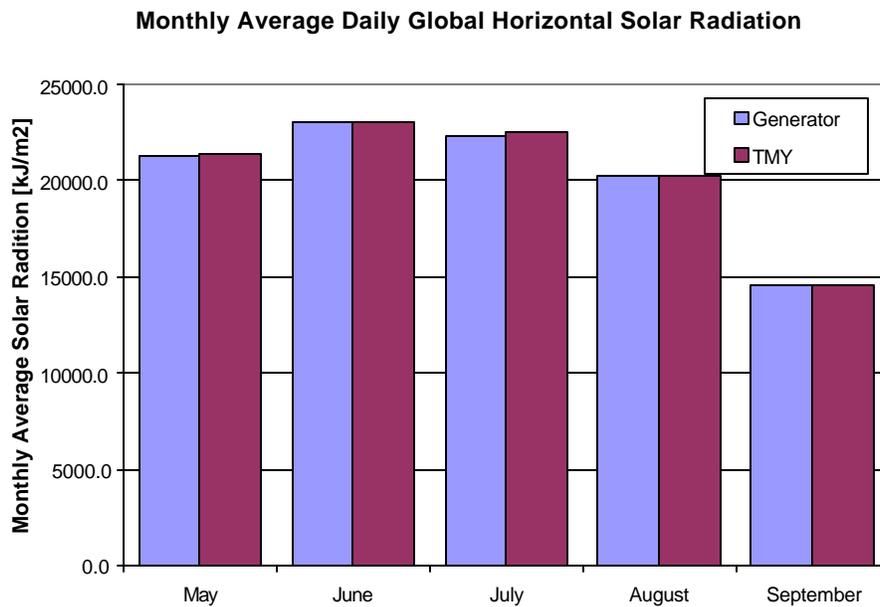


Figure 4.2 Monthly Average Daily Global Horizontal Solar Radiation for Generated Weather and TMY Weather Data.

The output of the weather generator is hourly values of ambient temperature and solar radiation. Figure 4.1 and Figure 4.2 show the monthly averages of the ambient temperatures and solar radiation for the simulation results. It can be seen that the monthly averages are the same for the both data sources. This verifies that the weather generator produces hourly data that maintains the same monthly averages.

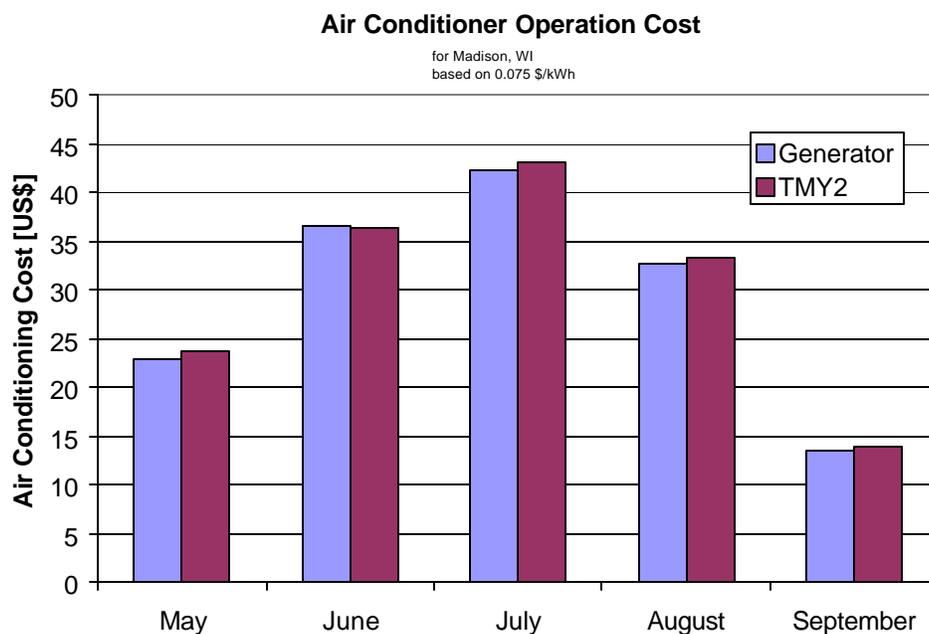


Figure 4.3 Monthly Air Conditioning Cost Impacted by Generated Weather and TMY Weather Data

Examining the impact of both weather sources on the cooling load of a building, Figure 4.3 shows the monthly cost for air conditioning for a season between May and October. Since ambient parameters, temperature and solar radiation influence the indoor climate, the cooling load reflects the impact of different weather conditions. The simulation was set to remove heat whenever the building temperature exceeds 25°C. Figure 4.3 shows a good agreement for both generated and TMY2 weather data for

Madison, WI. Thus, for analytical purpose, generated weather can be used along with TMY2 weather data.

## **4.5 Conclusions**

Two sources of weather data have been presented in this section. Weather data that has been generated from monthly-average weather information was found to lead to almost the same results than TMY data. Because the TMY data is based on actual weather conditions, while the weather generator uses a statistical approach to estimate the weather information from monthly values, the TMY data is assumed to be more accurate. For the remainder of this thesis the typical meteorological year data will be used for the simulations. But for locations where hourly weather data is not available the generated weather provides a good estimate.



# Chapter 5

## The Swimming Pool Air Conditioner (SPAC)

### 5.1 Introduction

The following section describes the software that has been written to simulate a swimming pool air conditioning system (SPAC). The objective is to show the major components of the SPAC program and to explain how it is used.

The Swimming Pool Air Conditioner Simulation (SPAC) is written for TRNSYS 14.2 and contains information for a TRNSYS desktop application, called TRNSED. The SPAC Simulation was developed to investigate different swimming pool heating concepts influenced by the ambient conditions.

### 5.2 SPAC Features

The SPAC simulation is composed of two main sections: the general information (e.g. simulation start and end and location) and the simulated components, such as the swimming pool, the gas pool heater, the air conditioner and the building. The program can be used to simulate for different heating modes that can be specified by the user.

Possible modes are:

- Gas pool heater on-off
- Reject air conditioner heat to the environment or to the pool water

- Pool cover opening and closing time
- Automatic pool cover control

These modes can be used to investigate different alternatives. An economic analysis can then discover the best component combination to achieve minimal operating cost.

## **5.3 The SPAC Simulation**

### **5.3.1. General Information**

After opening the SPAC program in TRNSHELL the user is confronted with a desktop input mask. This screen contains all information that is necessary to run the simulation. Figure 5.1 shows the input window for some general information, such as:

- Simulation start
- Simulation end
- Ground reflectance
- Weather data mode
- Location of Simulation

The ground reflectance is needed to accomplish the radiation that is reflected to the building.

**Simulation Parameters**

Month of the simulation start	May
Day of Month for Simulation Start	1
Month of the simulation Stop	October
Day of Month for Simulation Stop	1

Ground Reflectance	0.15
--------------------	------

**Weather Data Mode**

TMY2 Weather Data  
 Weather Generator

**Location: TMY2 Weather Data**

City for Simulation	Madison WI
---------------------	------------

Figure 5.1 General information window for SPAC

### 5.3.2. The Weather Data Mode

According to Chapter 4 two sources for weather data, TMY2 data and generated weather can be used within this program. Radio buttons allow switching between the weather data formats. The weather generator provides data for 200 locations in the US. Generated weather data is calculated from monthly average weather information such as radiation, ambient temperature and relative humidity. It has been shown in Chapter 4 that generated weather leads to similar results than the TMY2 data.

The program provides 10 cities of TMY2 data that can be selected from the location pull down menu. For further use locations based on TMY2 data can be added as mentioned in section 5.3.3.

### 5.3.3. Adding TMY2 locations to the SPAC simulation

The number of cities that are based on TMY2 weather data is limited to ten locations spread over the United States. The main reason for this limitation is the file size of each weather file. If necessary more locations can be added by copying the desired \*.tm2 file into the C:/spac/weather/ folder. In the “C:/spac/weather.dat”-file the first row contains total the number of cities listed in that file and has to be corrected according to the number of cities added. Each information row consists of: Name of the Location, Latitude, Shift in Solar Time Hour Angle, File-location on Hard drive. For example:

Madison WI,43.13,0.670,C:\spac\weather\madison\_wi.tm2

The shift in solar time hour angle can be calculated by knowing the longitude of the city (Duffie and Beckman (1991)) and is calculated by the following:

$$\text{Shift in Solar Hour Angel} = \text{Standard Meridian} - \text{Longitude of Location} \quad (5.1)$$

Where the Standard Meridian is defined as 75°W (Eastern), 90°W(Central), 105°W (Mountain), 120°W (Pacific)

Note that only NREL TMY2 Data files can be used in this simulation!

### 5.3.4. Swimming Pool Water Loss Calculations

The swimming pool is losing water over the season due to evaporation. This loss is compensated by precipitation. Since the TMY2 weather files do not include provide precipitation information, additional input is required. There are two ways in SPAC (Figure 5.2) to activate a water loss calculation.

**Precipitation Mode**

Provide Precipitation Data in a File  
 Enter Data in a Table  
 Don't Calculate Water Loss

Figure 5.2 Radio buttons switch between the precipitation modes

First, the monthly average precipitation can be provided in a data file(C:/spac/weather/\*.pre) that contains a single row of twelve precipitation values. The SPAC program provides information for Madison, Wisconsin that was obtained from NCDC (1998) in a file called mad.pre. The second possibility is to enter the twelve values directly in the SPAC program. Notice that the entered information can't be saved separately from the SPAC program. A third option disables the water loss calculation.

### 5.3.5. Economic Analysis

If the economics check box is enabled the SPAC program provides an economic analysis for the operation cost for each component that is involved in the current simulation. Inputs are the cost per kilowatt-hour for electricity and natural gas. The result can be seen in output file 5, described in section 5.8.

Enable Economic Analysis

**Economics**

Electricity Cost	<input type="text" value="0.075"/>	\$/kwh
Natural Gas Cost	<input type="text" value="0.020"/>	\$/kwh

Figure 5.3 Economics Analysis input mask in SPAC

## 5.4 The Building Simulation

Included to the TRNSYS package is a subroutine that simulates a multizone building. A software program called PREBID Transsolar (1997) has been developed by the TRANSSOLAR Company to create a special input file that is required by the building simulation. This tool has been used to set up a simple one-zone building with attic. This one story building represents a common ranch style house that has a stratified zone temperature. The model was considered to produce results that are accurate enough for the task of this work.

### 5.4.1. Simple One-Zone Building with Attic

Using the PREBID program a one-zone building with attic has been created. The building modeled in this example has an area of 250 m<sup>2</sup>. Table 5.1, Table 5.2 and Table 5.3 show the parameters that characterizes the building in PREBID.

---

#### Living Zone

---

Volume [m <sup>3</sup> ]	900
Capacitance [kJ/K]	1080
Initial Zone Temperature [C]	20
Initial Zone Humidity [%]	50

---

Table 5.1 Living Zone Parameters for multizone building

---

#### Attic Zone

---

---

Volume [m <sup>3</sup> ]	450
Capacitance [kJ/K]	540
Initial Zone Temperature [C]	20
Initial Zone Humidity [%]	50

---

Table 5.2 Attic zone parameters for multizone building

---

### Walls

---

Construction board	0.006 m
Insulation material – mineral wool	0.102 m
Wooden material – spruce pine	0.051 m
Wooden material – ply wood	0.006 m
Covering material – poly-vinyl chloride	0.013 m

---

Table 5.3 Wall Materials for multizone building

The building has only one main zone. Additionally, a tilted roof is attached by a ceiling. Solar gain is possible through the walls, the roof and windows that are integrated in the walls. Internal gains due to people inside the building were not included.

#### 5.4.2. SPAC Building Input

The appearance of the building simulation in the SPAC program is shown in Figure 5.4. If there are multiple buildings available the user can choose between them in the pull down menu. The desired building temperature can be specified below. Whenever

the building temperature exceeds this temperature, the air conditioner turns on and removes the heat from the building.

### Building Parameters

Building Type	1 Zone + Attic A=300 m2 ▾
Comfort Room Temperature (25 C ASHRAE)	25.00 C

Figure 5.4 Input mask of the building simulation in SPAC

#### 5.4.3. Adding a Building to the SPAC Program

With a little effort other buildings can be added to the SPAC program. First, the user has to create a building using the PREBID tool. For more information refer to the TRNSYS Manual Klein (1996) or contact the Solar Energy Laboratory ([trnsys@sel.me.wisc.edu](mailto:trnsys@sel.me.wisc.edu)). Next, the building information (the files \*.trn and \*.bld.) need to be added in the file C:/spac/buildings.dat that contains the building name (e.g. "1 Zone + Attic = 300m2") followed by the file locations for the \*.bld and the \*.trn files. Additionally, the first row of "buildings.dat" must contain the total number of buildings described in this file.

The user-defined building can contain one additional input that sets the desired building temperature and can be modified in the PREBID program.

## 5.5 The Swimming Pool

The swimming pool is represented by the physical size, information about the water surface activity, pool cover specifications and the cover control settings as shown in

Figure 5.5. A check box enables the pool set temperature control using automatic pool cover control. This option will be explained in Chapter 6.

### 5.5.1. Base Case Swimming Pool Settings

For the present work a base case swimming pool of 55 m<sup>2</sup> and 1.5 m depth was used. In general the pool cover was removed from the pool between 11 am and 2 pm. The pool was remained closed for the rest of the day to reduce energy loss due to evaporation. The water surface activity was set to slight surface motion as an average over the day, because the pool is unused and covered for the majority of the day.

**Swimming Pool Parameters**

<b>General Information</b>	
Pool Area	<input type="text" value="55.00"/> m <sup>2</sup>
Pool Depth	<input type="text" value="1.50"/> m
Pool Start Temperature	<input type="text" value="12.00"/> C
Shelter mode	<input type="text" value="Normal Shelter"/>
Water Surface Activity	<input type="text" value="slight surface motion(private pool)"/>
<b>Swimming Pool Cover Information</b>	
Thickness	<input type="text" value="0.01"/> m
Conductivity	<input type="text" value="0.18"/> kJ/h-m-K
Emittance/Absorption	<input type="text" value="0.60"/>
Pool Cover Opening Time	<input type="text" value="11.00"/> 24h
Pool Cover Closing Time	<input type="text" value="14.00"/> 24h

Enable Automatic Pool Cover Controller

Pool Set Temperature

Figure 5.5 Swimming Pool input mask in SPAC

The shelter factor specifies how well the pool is sheltered against wind. For this work a normal shelter has been assumed. The swimming pool cover is defined by the

thickness, the conductivity, the emittance and absorptance. Assuming Kirchoffs law to be valid, emittance and absorptance are equal.

## **5.6 The Air Conditioner**

The SPAC program provides two different air conditioner simulations as mentioned in Chapter 3 and has a switch to change between the cooling fluids (ambient air or swimming pool water).

### **5.6.1. The Constant COP Model**

The only input that is required for the constant COP model is the coefficient of performance itself. Based on this information the air conditioner is simulated as explained in Chapter 3.

### **5.6.2. The Variable COP Model**

The input screen for the variable COP model is shown in Figure 5.6. Required information is of the capacity and the COP of the modeled air conditioner at the ARI-Condition of 35 C.

**Air Conditioner**

Heat is rejected to

Constant COP model  
 Variable COP model

**Variable COP Air Conditioner Model**  
**Air cooled System Information from Manufacturer Data**

Capacity at Tcin=35 C (ARI-Condition)	<input type="text" value="18.60"/>	kW
COP at Tcin=35 C (ARI-Condition)	<input type="text" value="3.04"/>	kW

Figure 5.6 Information required by the SPAC program for the Air Conditioner

Both data points can be taken from manufacturer information for the specific air conditioner. Notice the limited accuracy mentioned in Chapter 3 by applying air conditioner characteristics other than used in the present work.

## 5.7 The Gas Pool Heater

If the Gas Pool Heater is enabled (checkbox: “Use Gas Pool Heater” is checked) the SPAC program maintains the desired swimming pool temperature using a gas pool heater system that consists of a gas furnace a pump and a controller.

**Gas Pool Heater System**

Use a Gas Pool Heater ?

**Gas Furnace**

Maximum Heating Rate	<input type="text" value="158258.00"/>	<i>kJ/hr</i>
Set Pool Temperature	<input type="text" value="25.00"/>	<i>C</i>
Efficiency	<input type="text" value="0.70"/>	-

**Pump**

maximum flowrate	<input type="text" value="14732.00"/>	<i>kg/hr</i>
maximum Power Consumption	<input type="text" value="1342.00"/>	<i>kJ/hr</i>

Figure 5.7 Input Mask for the Gas Pool Heater in SPAC

The appearance in the SPAC program is shown in Figure 5.7. The gas furnace requires input for the maximum heating rate in kJ/hr, the overall heat loss coefficient and the efficiency. If given by manufacturer data, the overall heat loss coefficient and the efficiency should be used for accuracy; otherwise the system is kept at zero heat loss and an efficiency of 1. The gas pool heater system turns on whenever the actual swimming pool temperature is lower than the pool set temperature. For further information on control strategies see Chapter 6.

The pump is used to cycle the pool water through the gas furnace. Since this pump is also needed to maintain a certain level of disinfecting chemicals in the water this device needs to run independent of the swimming pool heating mode. Two characteristic input values are required: the maximum mass flow rate and the maximum power consumption, both obtained from manufacturer data.

## **5.8 Output**

The SPAC program provides two modes of presenting output information. First, an online plotter shows up-to-date temperature results for swimming pool, building and ambient conditions. The other source for output information is a variety of output files that contains detailed data for specific system components.

### **The Online Plotter**

The online plotter entertains the user during the calculation with system information of the swimming pool. The plotter can be disabled by unchecking the

corresponding checkbox in the SPAC program. For further information consult the TRNSYS Users Manual (Klein (1996)).

### **5.8.1. Output Files**

Five output files that include information on the swimming pool, the weather data, the air conditioner, the economics and general system data. The output files are stored in the C:/spac folder and are labeled \*.ou1 to \*.ou5. Each file contains the hour of the year in the first column for orientation. Appendix C provides detailed information on the output files.



# Chapter 6

## Simulation Results

### 6.1 Introduction

The Swimming Pool Air Conditioner Simulation Program described in Chapter 5 has been employed to investigate different modes of swimming pool heating and their effect on swimming pool temperature and economic measures. The following section describes the simulation results and the findings of the present work. It will be shown that the proposed swimming pool air conditioner performs better than conventional air conditioners.

### 6.2 Swimming Pool Cover Control Strategies

A swimming pool cover is the best mechanism to prevent heat losses from an outdoor swimming pool. Evaporation has been shown to be the major heat loss from a water surface in Chapter 2. A cover that is placed on the water surface minimizes the evaporation heat loss. In some climates the swimming pool cover alone can provide a comfortable swimming pool temperature. In warmer regions the swimming pool might exceed the comfort temperature. In order to maintain a comfortable swimming pool temperature an automatic swimming pool cover is used that is controlled by the pool temperature. The cover then automatically opens whenever the swimming pool gets too

hot and allows heat dissipation from the pool to the environment. The pool owner can achieve the same effect by manually uncovering the pool if the water is above a personal comfort temperature.

### **6.2.1. Swimming Pool Covers**

Swimming pool covers eliminate most evaporative heat losses while they are on, and can also reduce convection, and thermal radiation heat loss. In general, there are four different kinds of swimming pool covers available.

1. Transparent plastic bubble cover, 0.6 cm thick, composed of a layer of plastic “floating” on a number of small bubbles formed by a second layer of plastic. The cover allows direct absorption of sunlight by pool water. The air spaces between the top layer of plastic and the water provide insulation value. Assumed properties of the plastic allow for a relatively high radiation heat loss.
2. An opaque foam cover, 0.25 cm thick, composed of insulating foam with a plastic film backing material. Since the cover is not transparent, solar heat must be transferred through the foam to the water. The insulating qualities reduce the amount of solar energy transferred to the pool, but also serve to retain heat in the pool.
3. Clear single layer plastic film cover that floats on the water surface. The clear plastic permits direct absorption of sunlight by the water, but does not provide much insulating value beyond suppression of evaporation.

4. Black, single-layer plastic film cover that floats on the surface. Since it is not transparent, transfer of solar heat to the water is somewhat less efficient than for clear plastic. Like the clear plastic, it does not provide much insulating value beyond the suppression of evaporation.

For the purpose of this study a bubble pool cover was chosen. The physical specifications are as follows:

Thickness = 0.01 m

Conductivity = 0.18 kJ/hr-m-K

Emittance / absorptance = 0.6

### **6.2.2. Comfortable Swimming Pool Temperature**

A comfortable swimming pool temperature has been assumed to be at 27°C (80°F). Correspondence with swimming pool manufacturers verified a temperature between 25 °C and 29 °C as desirable. The ASHRAE Applications Handbook also recommends a swimming pool temperature of 27°C (ASHRAE (1999))

### **6.2.3. Effect of Swimming Pool Cover Control Strategies**

The swimming pool behavior for different control strategies has been investigated for four cities in the United States displayed in Figure 6.1. Seattle, WA, Madison WI, New York City and Austin have been chosen to represent various climates throughout the US. However, impacts of three different pool cover treatments have been investigated:

1. The pool remains uncovered the entire season
2. The pool remains uncovered daily between 11 am and 2 pm.
3. The pool remains uncovered whenever the swimming pool temperature is above a critical temperature ( $26^{\circ}\text{C}$ ) and follows a daily schedule if the temperature is below the set temperature.

Additionally, in two cases of heat rejecting to the pool from a residential air conditioner have been studied.



Figure 6.1 US Cities that were examined for different pool cover strategies.

Figure 6.2 compares swimming pool temperatures for an unheated and uncovered pool in four different locations. The water surface is completely exposed to the ambient conditions and therefore is very sensitive to changes during the day. It can be seen that without any protection against environmental influences, the comfortable temperature cannot be reached in New York, Madison and Seattle. Only in Austin and for a short

season between June and September is the swimming pool water above the desired temperature.

Adding a swimming pool cover to the pool changes the situation completely. Figure 6.3 shows a scenario where a swimming pool cover remains on the pool for the entire season except during a daily swimming time between 11 am and 2 pm. The changes in the pool temperature over a day are a lot smaller than without a pool cover since the heat losses are reduced. The temperature of the covered pool is about 4°C higher than the uncovered pool. In Austin, Texas the swimming pool temperature exceeds the desired temperature for much of the summer while Madison and New York are heated to an acceptable temperature. The installation of a pool cover in Seattle raises the temperature, but the pool temperature remains below 25°C.

In the next scenario, the pool cover strategy remained the same, but the residential air conditioner rejected heat into the pool. Figure 6.4 shows that in Austin, where the cooling demand is high, the swimming pool temperature almost reaches 40°C which is a temperature that is definitely too high for recreational purpose. New York and Madison provide an agreeable temperature for part of season but approach 33°C more than once. Due to the small air conditioning demand in Seattle the pool heating effect is small and keeps the pool below 27°C for almost the entire season.

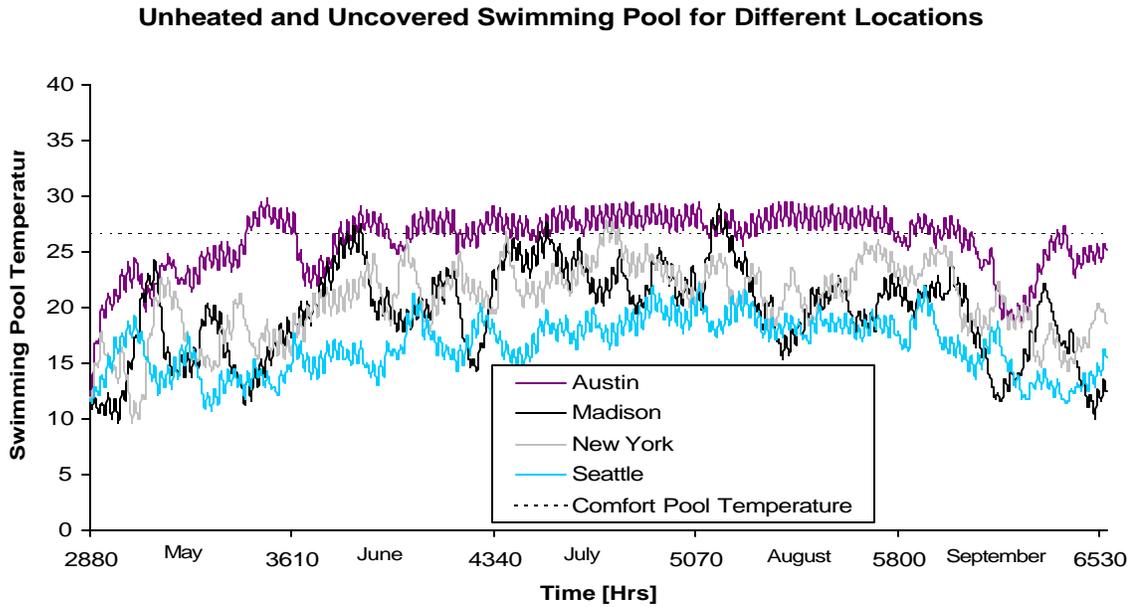


Figure 6.2 Swimming pool temperature for an uncovered and unheated pool

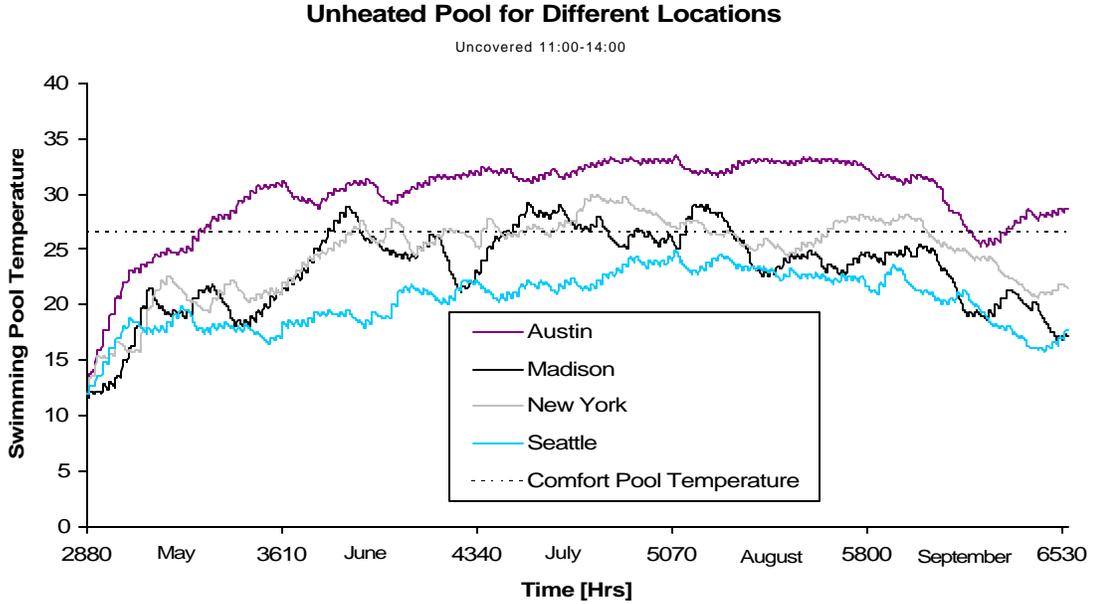


Figure 6.3 Pool temperature for an unheated and uncovered pool between 11am and 2pm

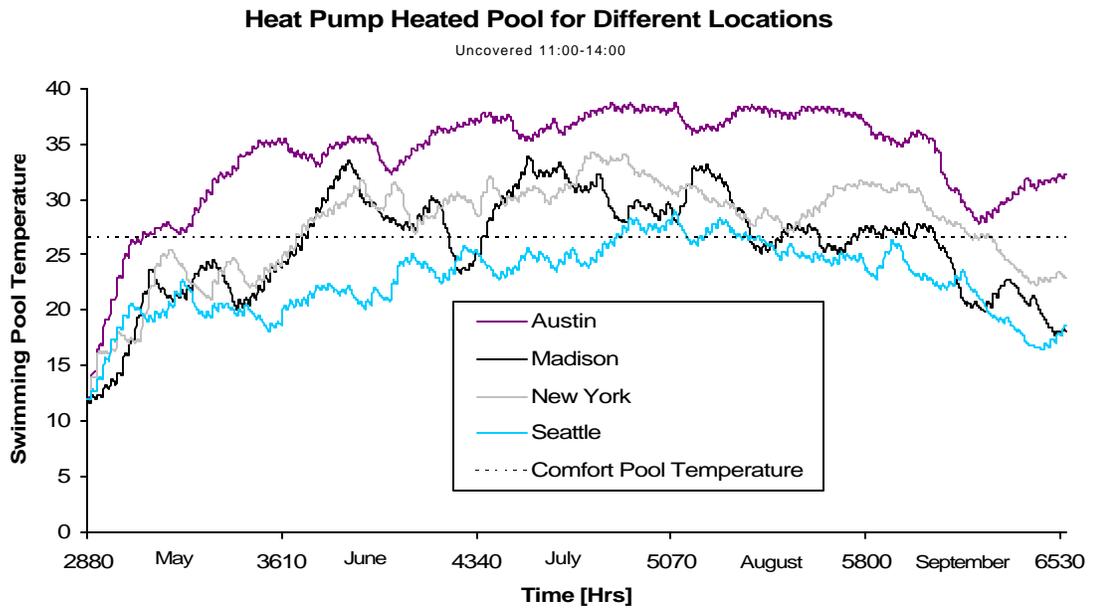


Figure 6.4 Pool temperature for a heated and part time covered pool

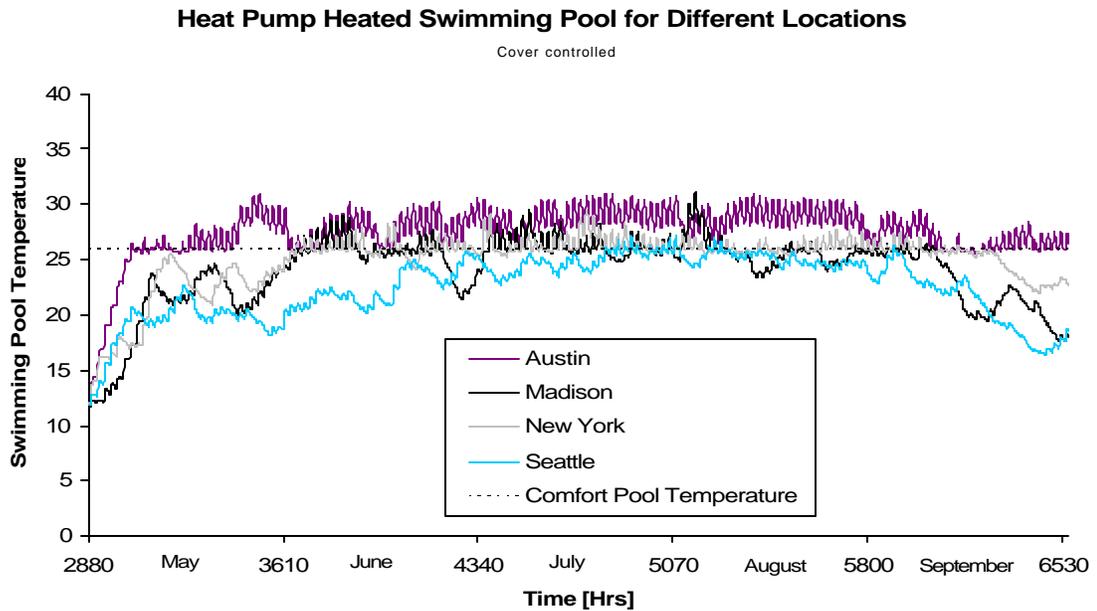


Figure 6.5 Automatic pool cover controlled swimming pool temperature

Applying the advanced pool cover control strategy where the swimming pool remains uncovered if the temperature exceeds 26°C Figure 6.5 shows that a substantial cooling effect can be achieved. Again, the heat removal from the building is added to the pool. The extreme pool temperatures for Austin of the last example can be reduced to a maximum of 30°C. Madison and New York can be maintained oscillating around the desired temperature. Since the control strategy only effects high pool temperatures the results for Seattle are not affected. In this case a swimming pool gas heater would be needed.

#### 6.2.4. Summary

It has been shown that a swimming pool cover influences the swimming pool temperature significantly. An overview of average swimming pool temperatures is provided in Table 6.1. Adding a cover to the swimming pool raises the pool temperature about 4°C. Combining a pool cover with the swimming pool air conditioner increases the temperature another 4°C. To avoid overheating, the automatic pool cover controller adjusts the pool temperature again to a lower temperature level.

Cities	Average Swimming Pool Temperature			
	Uncovered Unheated	Uncovered 11:00 – 14:00 Unheated	Uncovered 11:00-14:00 Heat Pump heated	Automatic Pool Cover Heat Pump heated
Austin	26.0	29.8	34.0	27.5
Madison	19.6	23.2	26.0	24.0
New York	20.6	24.7	27.7	24.9
Seattle	16.5	20.4	22.7	22.5

Table 6.1 Comparison of average swimming pool temperatures for different locations.

Note that Table 6.1 shows seasonal averages that can be lower than the comfortable temperatures. However, the cooling effect results in a higher pool water loss that has to be equalized from time to time. The water loss has been investigated in Section 6.4.

### **6.3 Benefits for the Customer**

In this section the operation costs for both conventional air conditioning and gas heating and the combined house cooling and pool heating are presented. The separate air conditioning and gas pool heating is referred to as conventional system, while the combined swimming pool heater and air conditioner will be called the swimming pool air conditioner (SPAC).

#### **6.3.1. System Control Strategies**

To compare the two systems on a fair basis the following control strategy has been developed. First, for swimming pool temperatures less than 25°C a gas pool heater adds heat to the pool. The gas heater maintains this temperature for both system configurations. Since the swimming pool air conditioner rejects heat whenever a cooling demand exists, the additional heat from the gas pool heater will be less.

If the swimming pool temperature exceeds 26°C the automatic swimming pool cover controller opens the pool and regulates the temperature by evaporation. The building temperature is controlled during the entire season so that heat is removed by the air conditioner whenever the temperature exceeds 25°C. One possible arrangement of the

system components modeled in the SPAC simulation program are shown in Figure 6.6 where the air conditioner is in water-cooling mode and the gas furnace joins to maintain the desired swimming pool temperature.

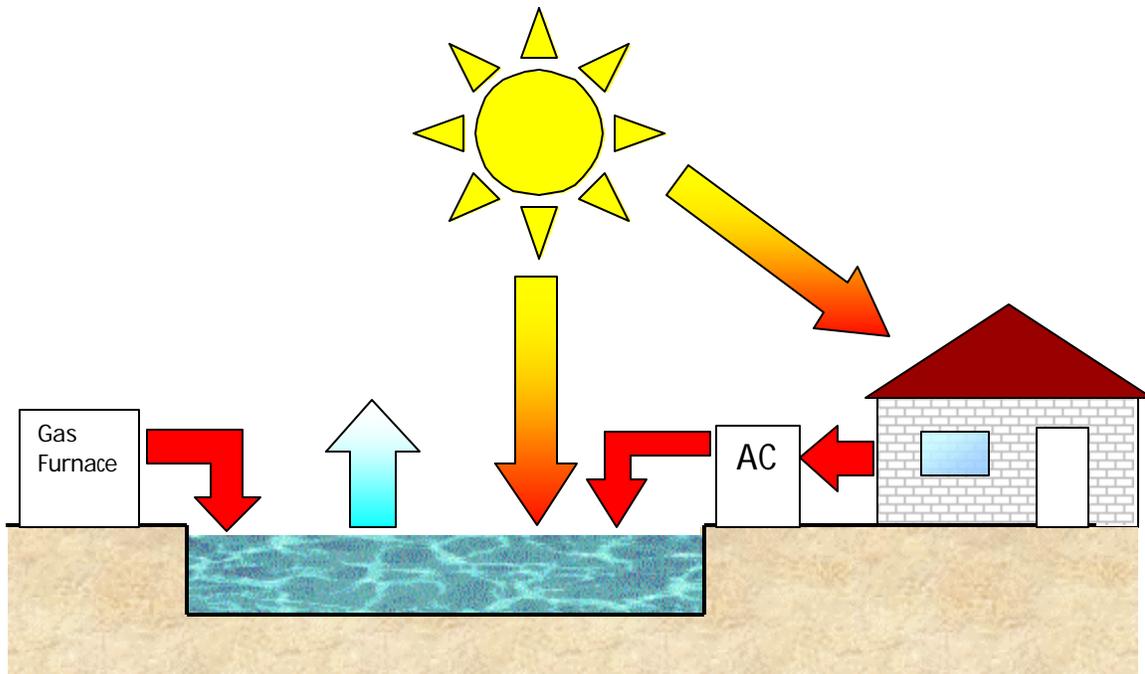


Figure 6.6 The Swimming Pool Air Conditioner Configuration

Based on these control mechanisms the SPAC simulation has been modified to run for different locations in the United States. Ten Cities have been chosen for an economic analysis and are shown in Figure 6.7. Different climates were examined for pool behavior and heating requirements to observe the impact on the economic analysis and narrow down regions where the swimming pool air conditioner performs best.



Figure 6.7 Cities for economic analysis

### 6.3.2. Seasonal Operation Cost

For each location the monthly energy requirement has been calculated for a conventional air conditioner with a gas pool heater and for a swimming pool air conditioner with a gas pool heater. The simulation was started in the beginning of May and continued until the beginning of October. For this period the swimming pool temperature was maintained to be at least 25°C using the gas pool heater when necessary. For some locations the pool might exceed the desired temperature resulting from solar gains and heat removed from the building that cannot be controlled by the cover controller. Table 6.2 shows that the maximum pool temperature stays below 32°C even for warm climates like Miami or Austin.

City	Conventional System					SPAC System				Seasonal Savings [\$] (May 1st - October 1st)
	Average Pool Temp [C]	Max Pool Temp [C]	Cost [\$]		Average Pool Temp [C]	Max Pool Temp [C]	Cost [\$]			
			AC	GPH			SPAC	GPH		
Atlanta	GA	26.2	29.9	202	3	26.6	31.3	154	1	50
Austin	TX	27.2	32.0	279	0	27.8	33.6	203	0	76
Baltimore	MD	25.8	29.2	171	46	26.2	30.2	133	24	60
Los Angeles	CA	25.7	27.6	182	31	26.0	28.4	147	11	56
Madison	WI	25.4	28.1	136	150	25.7	29.0	105	105	77
Miami	FL	27.3	31.3	251	0	28.0	32.7	186	0	65
New York	NY	25.6	28.6	154	75	25.9	29.6	120	44	65
Phoenix	AZ	26.7	30.5	394	0	27.4	32.2	268	0	126
Seattle	WA	25.1	26.8	113	224	25.4	27.4	95	154	88
St.Louis	MO	26.1	29.8	209	16	26.5	31.2	158	8	59

Control Strategies	Economic Analysis
<i>Gas Pool Heater(GPH):</i> Cover opens if Tpool > 26 C , Else the pool is open between 11am - 2 pm Heater activates if Tpool < 25 C <i>Swimming Pool Air Conditioner(SPAC):</i> Cover opens if Tpool > 26 C . Else the pool is open between 11am - 2 pm	Gas: 0.02 \$/kWh Electricity: 0.075 \$/kWh

Table 6.2 Temperatures and Operation Cost for the examined systems. GPH is a Gas Pool Heater, AC is a Conventional Air Conditioner and SPAC is a Swimming Pool Air Conditioner.

The table also shows the component operation cost for cooling and heating for both, the conventional and the SPAC system. Figure 6.8 visualizes this result by showing columns for each system and location. The left column represents the conventional system, where the air conditioning operation cost is added to the seasonal pool heating cost. The right column shows the cost for the SPAC system and if additional pool heating is necessary the cost for natural gas. The cities are ordered by decreasing cooling demand.

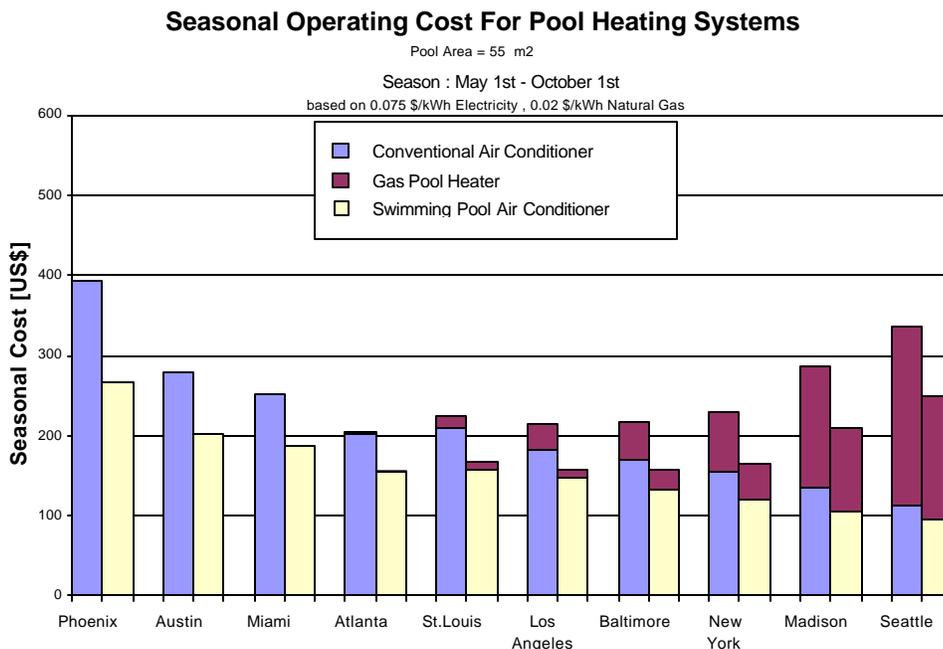


Figure 6.8 Economic analysis for locations in different climates in the United States. For each City the left column shows the conventional house cooling and heating, while the right includes the swimming pool air conditioner plus additional pool heating cost.

In Phoenix, for example, no gas heating is necessary for either the conventional or SPAC system to maintain a comfortable swimming pool temperature. But, due to the better performance of a water-cooled air conditioner the operation cost is lower. By comparing the swimming pool temperatures for both systems, Table 6.2 clarifies that there is no major change in swimming comfort due to the heat rejection to the pool. The maximum temperature increases only about 1.5°C, while the average only changes about 1°C.

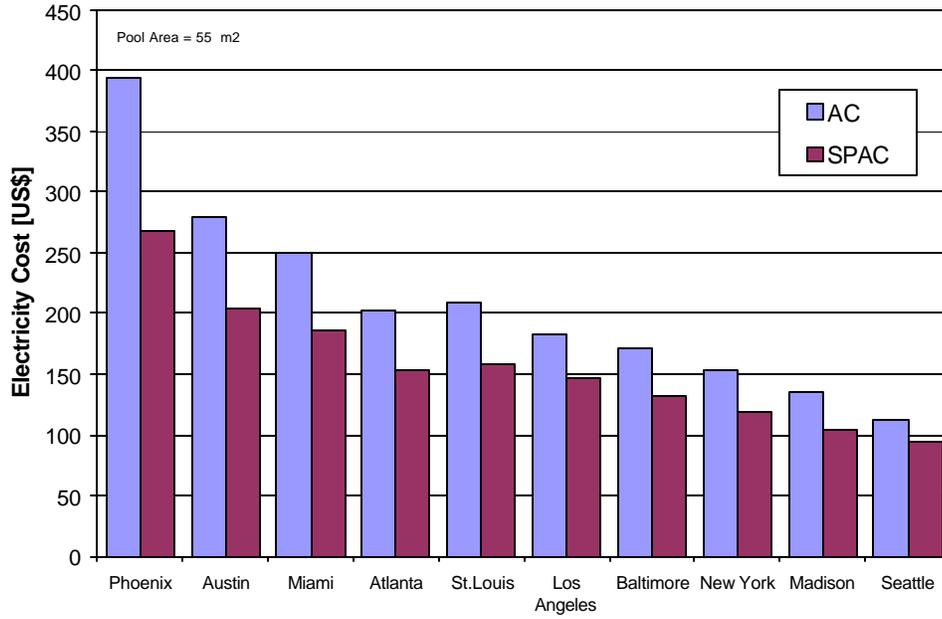


Figure 6.9 Electricity Cost for house cooling for a season between May 1<sup>st</sup> to October 1<sup>st</sup> based on 0.075 \$/kWh

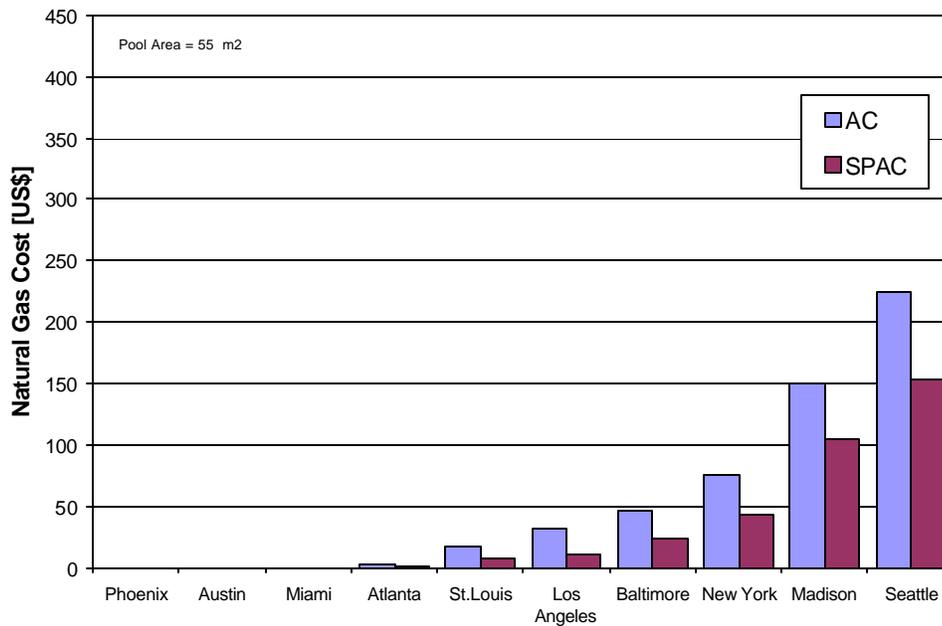


Figure 6.10 Natural Gas Cost for swimming pool heating between May 1<sup>st</sup> to October 1<sup>st</sup> based on 0.02 \$/kWh

With decreasing ambient temperatures, the cooling demand for a building decreases. Accordingly, the rejected heat to the swimming pool is less and pool heating becomes more important. Also, due to the lower ambient temperatures in cooler climates the general need for gas pool heating increases. Thus, the electricity cost decreases while the cost for natural gas increases.

For a better comparison, Figure 6.9 and Figure 6.10 show the expenses for electricity and natural gas separated in two charts. The left column for each city represents the conventional system configuration and the right column the swimming pool air conditioner.

The cooling demand is the major impact on the electricity cost. Thus, the warmer the ambient conditions at the specific location the higher the required energy to maintain the building temperature. However, there is still an advantage of the swimming pool air conditioner because of its better general performance due to the lower cooling fluid of the condenser.

Based on the natural gas consumption two city groups can be identified. The first group includes cities where gas pool heating is not necessary at all. Phoenix, Austin, Miami and Atlanta can be counted to this group. The second group consists of cities that need gas pool heating for both system configurations to maintain the desired swimming pool temperature as seen in New York, Seattle, Madison, St. Louis and Baltimore. A few locations cannot be assigned to one of the groups easily. Los Angeles for example has almost no heating demand for the swimming pool air conditioner and could be counted to the second group.

### 6.3.3. Seasonal Savings for Different Locations

Compared to the conventional system, the customer can save on seasonal expenses using the swimming pool air conditioner. Because of the better performance, the SPAC saves electricity. Figure 6.11 shows the electricity savings as a function of the cooling demand at different locations. The higher the expenses for the conventional air conditioner, the higher the seasonal savings.

Because the SPAC rejects the heat to the pool that is usually released to the ambient, the cost for heating is reduced. Figure 6.12 shows the seasonal gas savings.

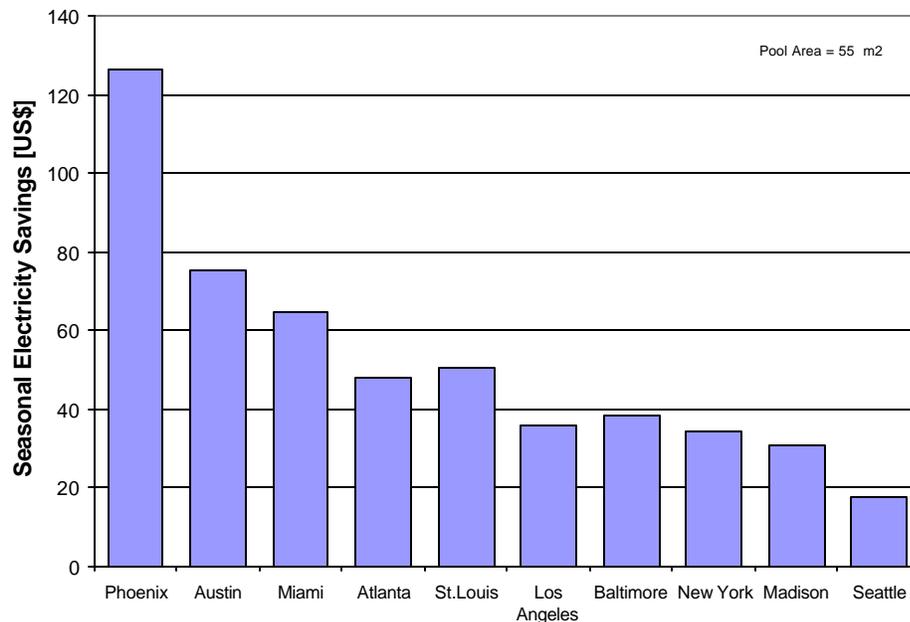


Figure 6.11 Electricity savings for SPAC compared to a conventional system for a season from May 1<sup>st</sup> to October 1<sup>st</sup> based on 0.075 \$/kWh

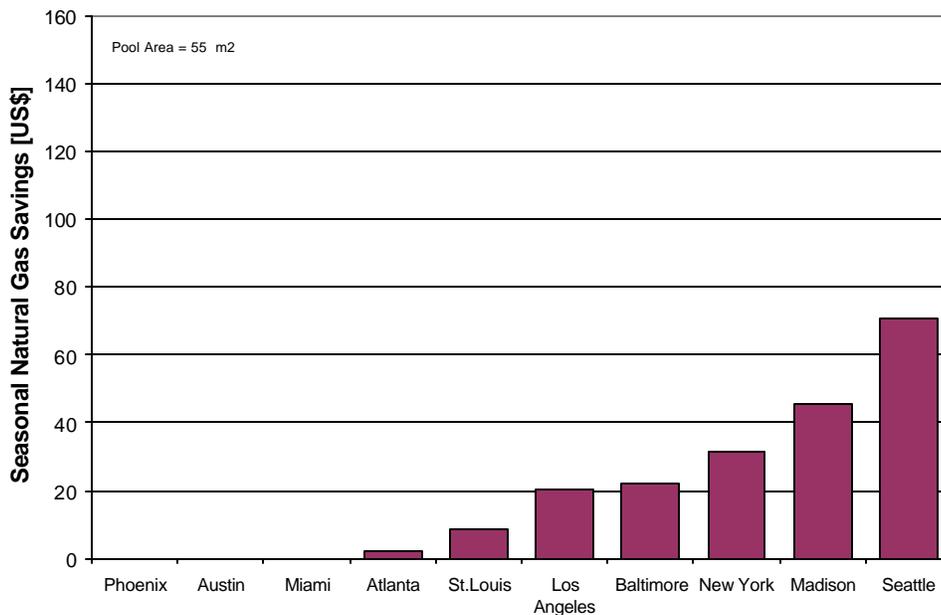


Figure 6.12 Natural Gas Savings for SPAC compared to a conventional system for a season from May 1<sup>st</sup> to October 1<sup>st</sup> based on 0.02 \$/kWh

#### 6.3.4. Impact of Different Swimming Pool Sizes

In addition to the swimming pool investigated above, the impact of varying pool sizes is examined. Starting with the base case pool size of 55 m<sup>2</sup> a smaller swimming pool (27.5 m<sup>2</sup>) and a larger pool (110 m<sup>2</sup>) are investigated. Figure 6.13 and Figure 6.14 show the effect on the economic analysis. Since the building and the air conditioning system remained the same, the amount of rejected heat is constant for the three cases. Independent of the pool sizes the pool conditions can be adjusted without supplemental heat from a gas pool heater in Phoenix, Austin, Miami and Atlanta. For locations that require pool heating to maintain the swimming pool at the comfortable swimming pool temperature of 27 °C the natural gas consumption increases for increasing pool area.

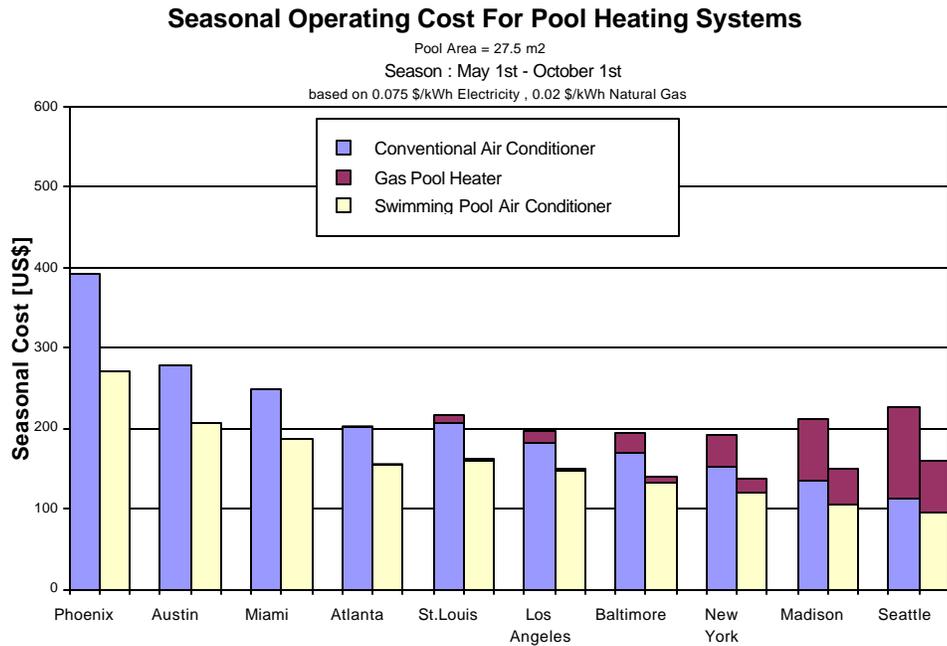


Figure 6.13 Economic analysis for locations in different climates in the United States for a 27.5 m<sup>2</sup> pool.

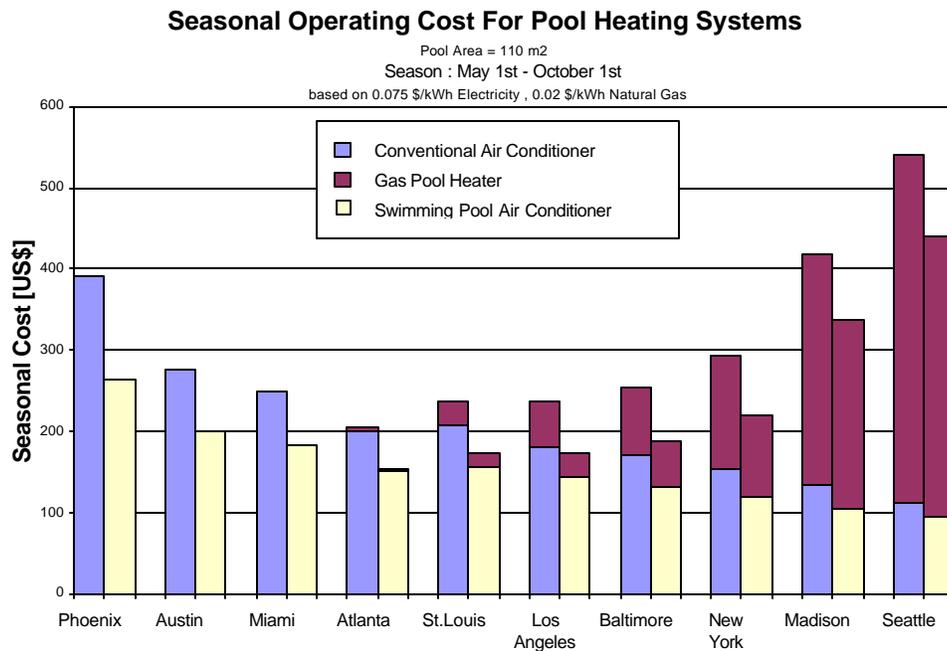


Figure 6.14 Economic analysis for locations in different climates in the United States for a 110 m<sup>2</sup> pool.

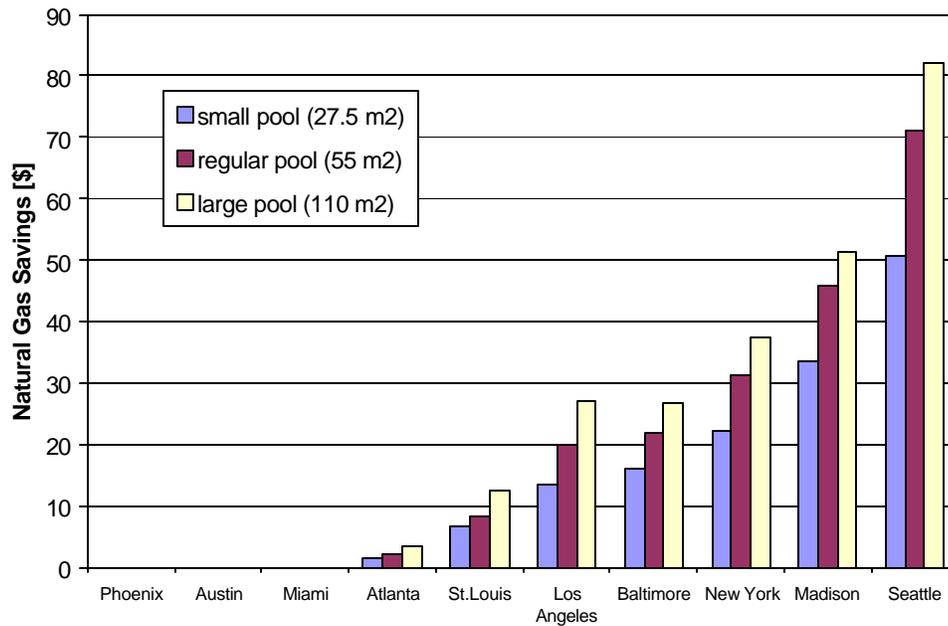


Figure 6.15 Natural Gas Savings for different swimming pool sizes for a season between May and November.

The seasonal savings in natural gas also increase with increasing swimming pool size as shown in Figure 6.15. The largest impact of changing the swimming pool size can be observed for Seattle, because the energy gains are by far not sufficient to meet the energy demand of the swimming pool. Note that the maximum savings go along with a much higher cost. Thus, it cannot be concluded that a larger pool is more efficient.

### 6.3.5. Sensitivity of Deviation in Evaporation Calculation Methods

It is shown in Chapter 2 that evaporation is the major heat loss from a swimming pool. To account for the uncertainties in the examined evaporation correlations presented by various researchers, the impact of changes in the evaporation heat loss on the

simulation results were investigated. As can be seen in Figure 6.16 an uncertainty of  $\pm 10\%$  results in a change in the natural gas cost of  $\pm 10\%$ .

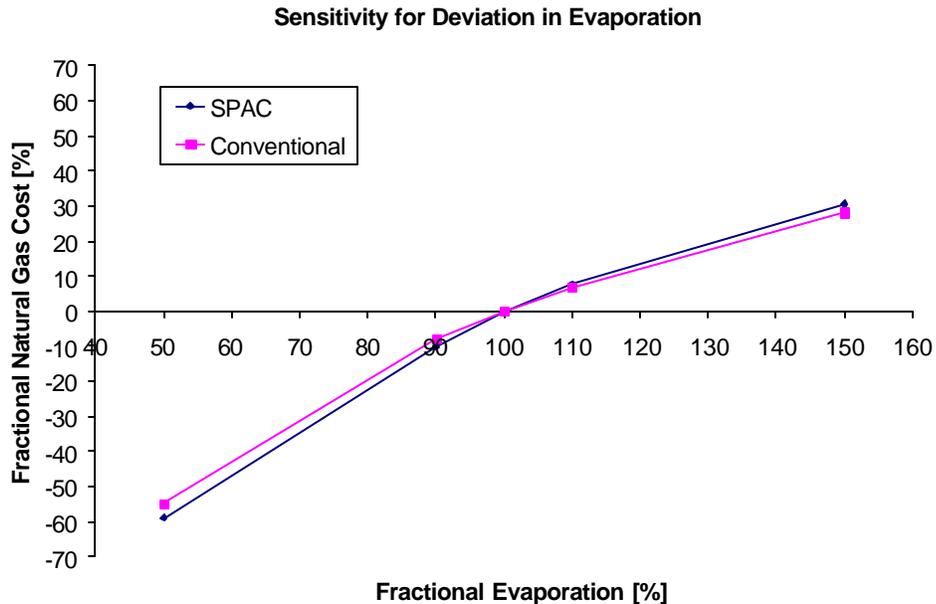


Figure 6.16 Impact of uncertainties in evaporation calculation methods on the natural gas cost for the SPAC and conventional systems.

### 6.3.6. SPAC Equipment Cost

Further investigations have been made to examine the extra budget that is available to manufacture a swimming pool air conditioner. The fact that SPAC is more efficient than a conventional air conditioner provides the advantage for the manufacturer to make it more expensive. The maximum possible additional product cost is reached when the life-cycle cost of the SPAC is the same as the conventional system. Duffie and Beckman (1991) provide a method to calculate the life cycle cost for a system. The life cycle cost (LCC) is the sum of all the costs associated with an energy delivery system

over its lifetime or over a selected period of analysis, in today's dollars, and takes into account the time value of money. Life cycle savings is defined as the difference between the life cycle costs of a conventional air conditioner and pool heating system and the life cycle cost for the swimming pool air conditioner system. Duffie and Beckman (1991) have shown how all economic parameter can be cast in only two parameters,  $P_1$  and  $P_2$ . Thus, the life cycle cost for the conventional system and the SPAC can be written as

$$LCC_{Conv} = P_1(C_{e,conv} + C_{g,conv}) + P_2 \cdot C_{eq,conv} \quad (6.1)$$

$$LCC_{SPAC} = P_1(C_{e,SPAC} + C_{g,SPAC}) + P_2 \cdot C_{eq,SPAC} \quad (6.2)$$

Where  $C_e$  is the electricity cost for the first year of analysis,  $C_g$  the natural gas cost for the first year of the analysis and  $C_{eq}$  the equipment cost of the system.

The difference of equation ( 6.1 ) and equation ( 6.2 ) results in the an expression for the life cycle savings of the SPAC system. For a break-even calculation the life cycle savings are zero.

$$LCS = P_1(\Delta C_e + \Delta C_g) - P_2 \cdot \Delta C_{eq} = 0 \quad (6.3)$$

Since the savings  $\Delta C_e$  and  $\Delta C_g$  are known, equation ( 6.3 ) can be solved for the difference in equipment cost  $\Delta C_{eq}$ .

$$\Delta C_{eq} = \frac{P_1}{P_2} (\Delta C_e + \Delta C_g) \quad (6.4)$$

The difference of the equipment cost is the maximum money that can be charged by the manufacturer for a swimming pool air conditioner that has the same life cycle cost as a conventional air conditioner plus pool heating system.

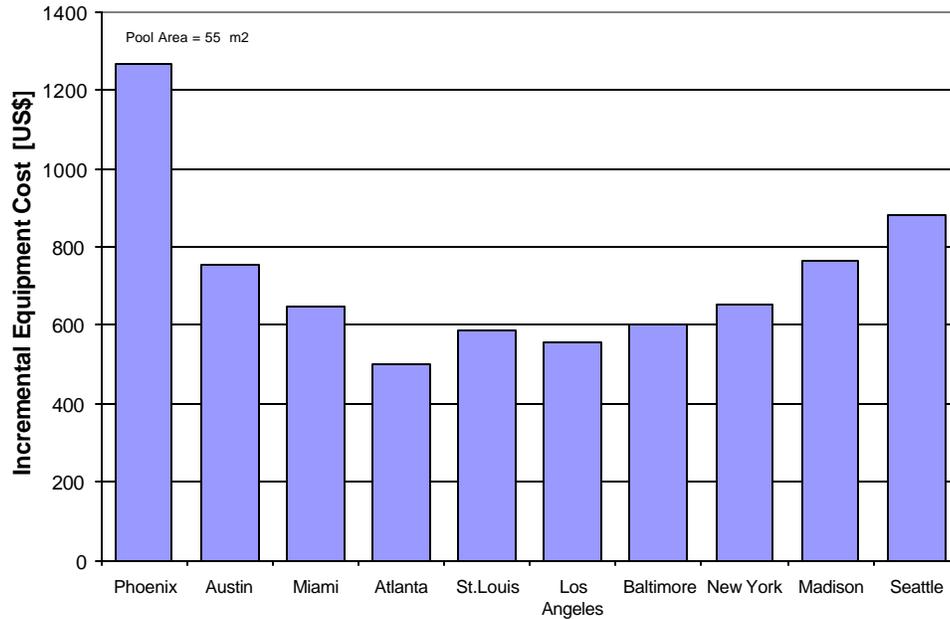


Figure 6.17 Incremental equipment cost for a SPAC system compared to a conventional system because of better performance. Time period: May 1<sup>st</sup> to October 1<sup>st</sup>.

Because the detailed system component cost and various economic parameters especially for the swimming pool air conditioner, are not available an approximate value for the ratio of  $P_1/P_2$  can be obtained by the following assumptions: If the inflation rate of fuel (electricity and gas) is of the order of the general inflation rate, then  $P_1$  is of the order of the period of the economic analysis.  $P_2$  is unity if the system is paid for in cash. Therefore, the ratio of  $P_1/P_2$  equals the period of the economic analysis. For the present work, a period of ten years has been chosen. Figure 6.17 shows the result of this approach. The allowable costs are between about \$600 and \$1000 for most locations, with higher values in the very hot climates of Austin and Phoenix. A minimum is found for Atlanta, which can be taken as additional budget that is available to finance the redesign of the air conditioner.

### 6.3.7. Air Conditioner Power Demand

An interesting aspect for power plant companies is the power demand for air conditioners. Since air conditioning consumes power mostly during the daytime where the energy demand is high, a reduction would be benefited to the power company. Figure 6.18 shows a comparison of the maximum power demand for a conventional air conditioner that uses air as cooling fluid to a swimming pool air conditioner. The air-cooled power demand is almost uniformly at about 6 kW. The SPAC system energy demand is about 5 kW, which results in a demand saving of 1 kW compared to conventional air conditioning systems. Applying the swimming pool air conditioner this demand can be reduce by about 20%.

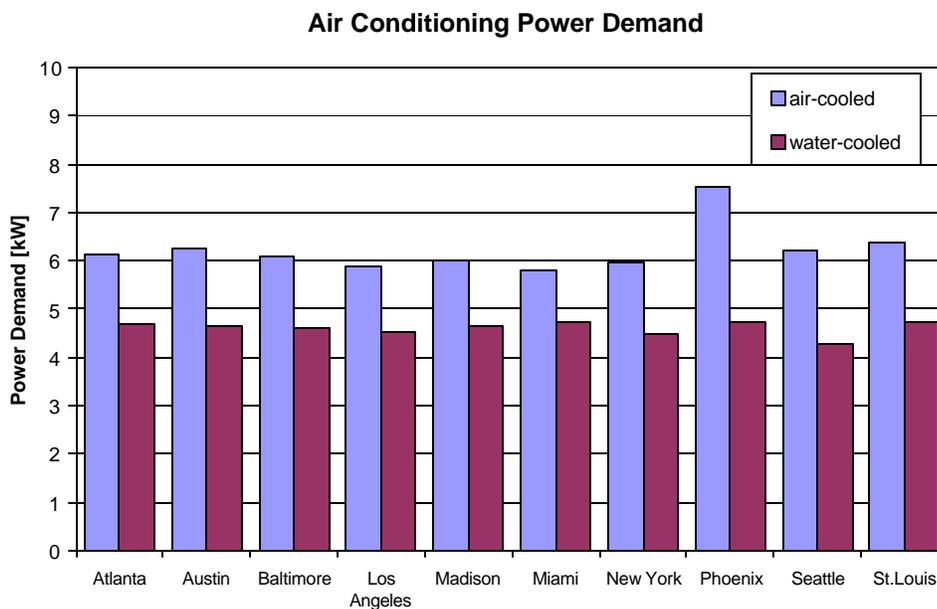


Figure 6.18 Power demand comparison for air conditioner systems.

### 6.3.8. Summary

In this section it has been shown that water-cooled air conditioners that reject heat to a swimming pool work more efficiently than conventional air-cooled air conditioners. In some climates additional swimming pool heating is not necessary with the proposed swimming pool air conditioner because the heating demand is accomplished by the rejected heat. In warmer regions where swimming pool heating is not necessary at all, the improved performance of a water-cooled air conditioner reduces the operation cost. For most locations in the United States additional pool heating is necessary. Despite this fact, the SPAC system still performs better than conventional methods according to the heat rejection to the pool.

In conclusion, the proposed swimming pool air conditioner is shown to perform better than conventional solutions. The amount of savings is dependent of the location of the customer but saves at least \$60 per season.

## 6.4 Swimming Pool Water Level Calculations

Every water body has water losses due to evaporation. Thermodynamically, the evaporation heat loss is proportional to the amount of water that is lost by the pool:

$$\dot{Q}_{evap} = \rho \cdot \dot{V}_w \cdot \Delta h_{evap} \quad (6.5)$$

Where  $\dot{Q}_{evap}$  is the evaporation energy,  $\rho$  is the density of water,  $\Delta h_{evap}$  is the enthalpy of evaporation and  $\dot{V}_w$  is the volume of evaporated water, e.g. the water loss per

unit time [ $\text{m}^3/\text{hr}$ ]. The enthalpy of evaporation is a function of the swimming pool temperature and can be linearly approximated by

$$\Delta h_{evap} = 1602.652 - 6.659 \cdot T_{pool}. \quad (6.6)$$

Where the  $T_{pool}$  is the pool temperature is in degree Celsius.

The water loss can be calculated from equation ( 6.6 ) by knowing the amount of energy that is evaporated from the swimming pool surface.

Information for the monthly average precipitation has been obtained from the annual weather summary for Madison, WI (NCDC (1998)). The amount of rain per square meter has been multiplied by the total pool area to calculate the total amount of water added to the swimming pool due to precipitation.

Applying the water loss calculation to the SPAC simulation for Madison over a season from May to October using the automatic cover control and heat rejection to the pool results in a water loss of  $27.4 \text{ m}^3$ . This evaporation about one third of the total pool volume ( $82.5 \text{ m}^3$ ). Precipitation adds  $24.6 \text{ m}^3$  water to the pool. Therefore, in Madison almost no water has to be replaced due to evaporation. This can only be an approximation since other effects influence the water loss of a swimming pool.



# Chapter 7

## Conclusions and Recommendations

The goal of this project was to investigate the performance of an air conditioner that rejects energy to a swimming pool instead of to the ambient air. Swimming pools and air conditioners have been examined separately in Chapter 2 and 3. Both components are available as TRNSYS types that were implemented in the swimming pool air conditioner simulation (SPAC).

Chapter 6 concludes that it is generally possible to heat a pool and to keep the building at a comfortable temperature using a swimming pool air conditioner system. In some climates a swimming pool heater is not necessary because the heat rejected by the swimming pool air conditioner alone maintains a comfortable pool temperature. Due to the fact that water is used as the cooling fluid, a swimming pool air conditioner performs better than conventional air conditioners. Thus, the purchased energy for pool heating and house cooling can be reduced. The manufacturer can most likely include higher manufacturing costs into the first cost while the customer enjoys the benefit of lower life cycle cost.

The swimming pool air conditioner simulation program provides a research tool for swimming pool manufactures and customers. The program is suitable for directly estimating the economic impact of different scenarios. The SPAC program can be used to investigate various configurations of a swimming pool, a gas pool heater, an air

conditioner and a building for different time periods and locations. SPAC provides information on the component performance as well as hourly information on system parameters. The cost advantage of one alternative over the other can be determined. The present system can be analyzed and the impact of an additional device studied before an investment is made.

Compared to a conventional system, the customer can save on seasonal expenses using the swimming pool air conditioner. Because of the better performance, the SPAC saves electricity. The seasonal electricity savings vary for different climates but are between \$40 and \$80 for most locations. Because the SPAC rejects the heat to the pool that is usually released to the ambient, the cost for swimming pool heating is reduced. The customer can save about \$40 on natural gas by using the SPAC system.

The allowable incremental equipment costs for a swimming pool heater system compared to a conventional system configuration, are between about \$600 and \$1000 for most locations, with higher values in the very hot climates of Austin and Phoenix. The SPAC system energy demand is about 5 kW, which results in a demand saving of 1 kW compared to conventional air conditioning systems.

The cooling requirements of a building and the pool heating demand are dependent on the climate. In warmer climates a swimming pool air conditioner rejects more heat to the swimming pool than in moderate climates. Thus, overheating is possible. In this work the swimming pool temperature was reduced by removing a pool cover from the surface to allow evaporation. An object of further investigation could be an air conditioning device that hosts both air and water-cooling. After determining if the

swimming pool requires heating a control mechanism would then reject the heat either to the pool or the environment.

The mathematical descriptions of the modeled components are considered to produce results that are accurate enough for the task of this work. However, as with all theoretical studies, simulations can only approximate reality. Thus, for further investigation a swimming pool air conditioner has to be designed and manufactured to obtain measurements that can verify the results of this research.



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# Appendix A

## Description of TRNSYS Type 144

### A.1 General Description

TRNSYS TYPE 144, developed at TRANSSOLAR in Germany, simulates both indoor and outdoor swimming pools. The inputs and outputs as well as the parameters () are described in this section.

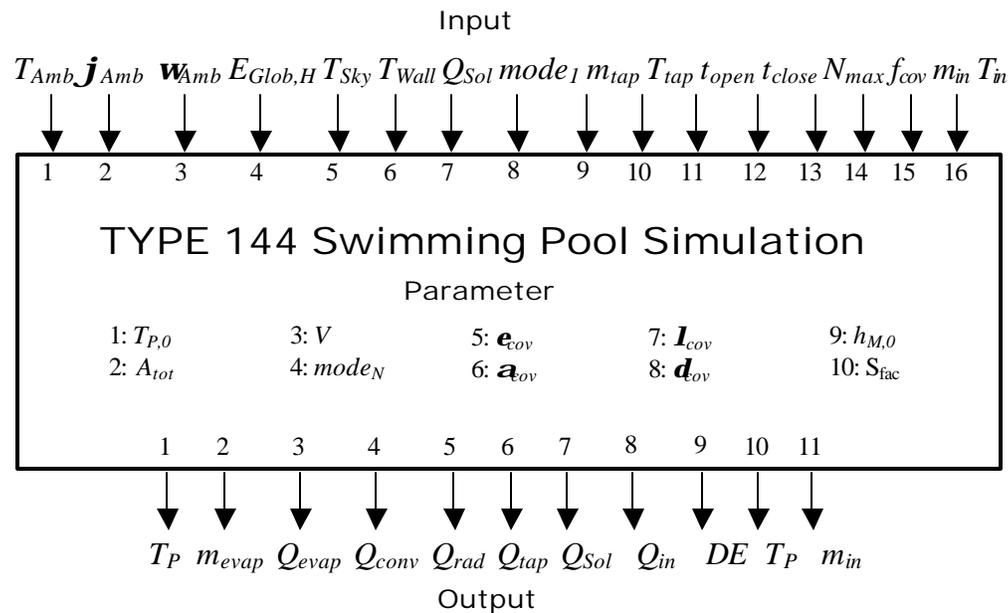


Figure A.1 Input and output for TYPE 144

### A.2 Description of the Inputs

To simulate an outdoor or indoor swimming pool the following inputs are used:

Input no.	Symbol	Description	Unit
1	$T_{Amb}$	ambient air temperature	[ °C ]

2	$\mathbf{j}_{Amb}$	relative humidity of ambient air	[%]
3	$\mathbf{w}_{Amb}$	velocity of ambient air	[m/s]
4	$E_{Glob,H}$	global radiation on horizontal surface	[kJ/h-m <sup>2</sup> ]
5	$T_{Sky}$	sky temperature	[°C]
6	$T_{Wall}$	temperature of pool walls	[°C]
7	$Q_{Sol}$	window radiative energy gains	[kJ/h]
8	$Mode_1$	water surface activity	[-]
9	$m_{tap}$	mass flow rate of tap water	[kg/h]
10	$T_{tap}$	temperature of tap water	[°C]
11	$T_{open}$	pool opening time	[-]
12	$T_{close}$	pool closing time	[-]
13	$N_{max}$	daily maximum number of people in the pool	[-]
14	$f_{cov}$	fractional coverage of water surface	[-]
15	$m_{in}$	mass rate of incoming warm water	[kg/h]
16	$T_{in}$	temperature of incoming warm water	[°C]

#### 1. Ambient Air Temperature ( $T_{Amb}$ )

For an outdoor pool this temperature should be the outside air temperature. For an indoor pool room temperature should be used.

#### 2. Relative Humidity of Ambient Air ( $\mathbf{j}_{Amb}$ )

Depending upon the ambient air temperature chosen above ,the relative humidity is taken to be the relative humidity of either the outdoor or the indoor air.

#### 3. Velocity of Ambient Air ( $\mathbf{w}_{Amb}$ )

This input is used only for an outdoor swimming pool calculation. Because the wind speed is a function of the height above ground and the microclimate around the pool, two additional parameters have been added to determine this value. (See Parameters)

#### 4. Global Radiation on Horizontal Surface ( $E_{Glob,H}$ )

To calculate the radiation heat gains to an outdoor pool the radiation on a horizontal surface is needed.

#### 5. Sky Temperature ( $T_{Sky}$ )

To calculate the exchange of long wave radiation a sky temperature is necessary. This can be found using TYPE 69.

#### 6. Temperature of Pool Walls ( $T_{Wall}$ )

To calculate the exchange of long wave radiation for an indoor pool an average wall temperature of the pool is needed.

#### 7. Window Radiation Gains ( $Q_{Sol}$ )

Only of use for the calculation of the energy gains of an indoor pool. The window radiation gains are energy inputs to the pool due to sunlight shining through windows. This input can be calculated by TYPE 56, output 21.

#### 8. Water Surface Activity ( $mode_1$ )

The motion of the water surface has a strong influence on evaporation and convection. Therefore a switch is used to set one of the following modes:

$mode_1 = 0$	<i>quiet water surface</i>
$mode_1 = 1$	<i>slight surface motion (private pool)</i>
$mode_1 = 2$	<i>slight surface motion (public pool)</i>
$mode_1 = 3$	<i>moderate surface motion (recreational pool)</i>
$mode_1 = 4$	<i>intense surface motion (wave pool)</i>
$mode_1 = -1$	<i>activity function</i>

The activity function ( $mode_1 = -1$ ) computes a parabola between opening and closing time of the pool. The maximum is located in the middle. Therefore it is necessary to use inputs 11, 12 and 13.

#### 9. Mass Flow Rate of Tap Water ( $m_{tap}$ )

Input 9 describes only the water rejection for hygienic purposes. The loss of water due to evaporation is compensated automatically.

#### 10. Temperature of Tap Water ( $T_{tap}$ )

Temperature of incoming tap water (input 9). Tap water flows for both hygienic and evaporative compensation are assumed to have the same temperature.

#### 11. Pool Opening Time ( $t_{open}$ )

The time of day (0-24) when pool usage begins. (needed for the calculation of the activity function.)

#### 12. Pool Closing Time ( $t_{close}$ )

(see input 11 and 8)

#### 13. Daily Maximum Number of People in the Pool ( $N_{max}$ )

Input 13 is also used to calculate the activity function (input 8). The input is the daily maximum number of people in the pool. In a pool with an area of 100m<sup>2</sup> and 100 people a day for example, the activity function has a maximum of 4 compared to an unused pool.

#### 14. Fractional Coverage of Water Surface ( $f_{cov}$ )

The percentage of time pool surface is covered ( $f_{cov} = 0..1$ )

#### 15. Mass Flow Rate of Incoming Warm Water ( $m_{in}$ )

Mass flowrate entering pool from the heating system

#### 16. Temperature of Incoming Warm Water ( $T_{in}$ )

Temperature of water entering the pool from the heating system.

### A.3 Description of Parameters

To simulate an outdoor or indoor swimming pool the following parameters are used:

Input no.	Symbol	Description	Unit
1	$T_{P,0}$	initial temperature of pool water	[ °C ]

2	$A_{tot}$	total surface area of pool	$[m^2]$
3	$V$	volume of pool water	$[m^3]$
4	$Mode_N$	switch between outdoor and indoor pool	$[-]$
5	$e_{cov}$	emissivity of pool cover	$[-]$
6	$a_{cov}$	Absorption of cover	$[-]$
7	$I_{cov}$	heat transfer coefficient of cover	$[kJ/h-m-K]$
8	$d_{cov}$	thickness of cover	$[m]$
9	$h_{M,0}$	height of wind measurement	$[m]$
10	$S_{fac}$	Shelter factor	$[-]$

1. Initial Temperature of Pool Water ( $TP,0$ )

Temperature of the pool at the time when simulation starts.

2. Total Surface Area of the Pool ( $A_{tot}$ )

Surface Area of the swimming pool including the spillway.

3. Pool Water Volume ( $V$ )

4. Switch between outdoor and indoor pool ( $mode_N$ )

The parameter  $mode_N$  switches between the calculation of an indoor and an outdoor swimming pool:

$$\begin{array}{ll} mode_N = 0 & \text{Indoor Pool} \\ mode_N = 1 & \text{Outdoor Pool} \end{array}$$

5. Emissivity of Pool Cover ( $e_{cov}$ )

6. Absorption of Pool Cover ( $a_{cov}$ )

7. Heat Transfer Coefficient of Cover ( $I_{cov}$ )

8. Thickness of Cover ( $d_{cov}$ )

### 9. Height of Wind Measurement ( $h_{M,0}$ )

(See parameter 10)

### 10. Shelter Factor ( $S_{fac}$ )

The heat loss of an outdoor swimming pool depends strongly on the wind speed. In this program a wind speed measurement height of 3m above ground is assumed. Because this is not necessarily the height at which the wind speed was actually measured, a correction term is included. The correction term depends on the shelter of the pool due to the surroundings. Figure A.2 shows an example where the wind velocity measured is 5m/s at a height of 10m.

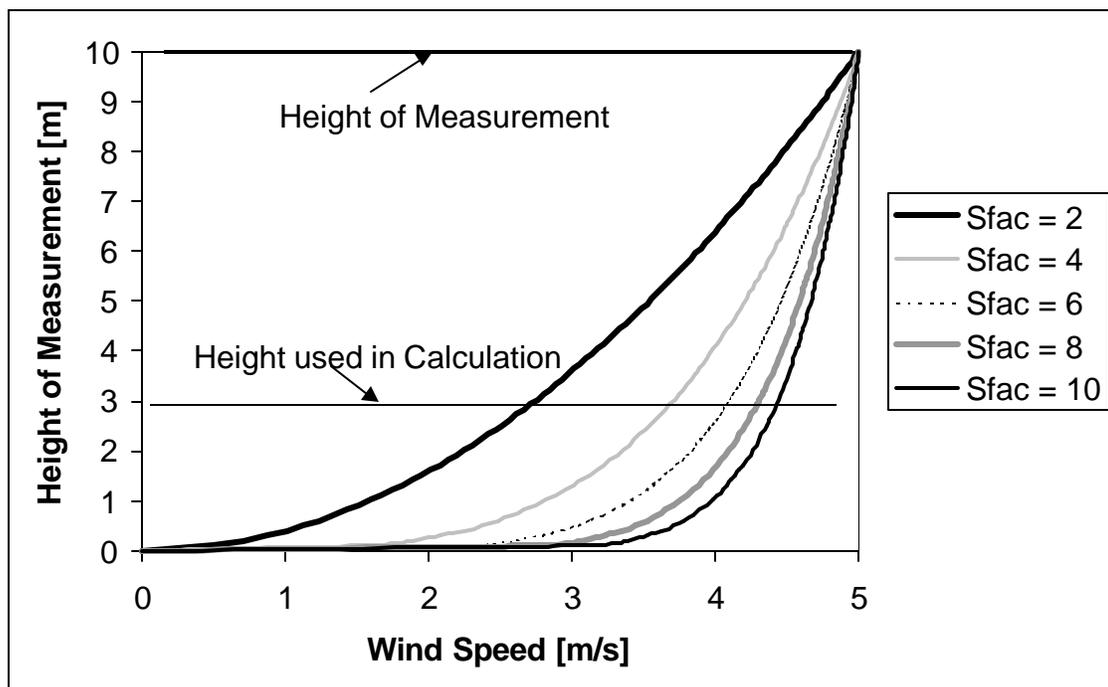


Figure A.2 The height as a function of wind speed. Measured wind velocity at the airport: 5 m/s at a height of 10m

The following relation is used to compute the modified wind velocities:

$$v = v_{airport} \cdot \frac{h_{rel}}{h_0} \frac{1}{S_{fac}}$$

$S_{fac} = 2$  strong shelter

$S_{fac} = 2-4$  normal shelter

$S_{fac} = 3-6$  wooded area

$S_{fac} = 6-8$  unsheltered

$S_{fac} = 8-10$  open water

$h_{rel} =$  height of wind measurement

$h_0 = 3$  m

#### A.4 Description of the Outputs

Input no.	Symbol	Description	Unit
1	$T_P$	pool water temperature	[ °C]
2	$m_{evap}$	mass flow rate due to evaporation	[kg/h]
3	$Q_{evap}$	evaporation heat flux	[kJ/h]
4	$Q_{conv}$	convection heat flux	[kJ/h]
5	$Q_{rad}$	radiation heat flux	[kJ/h]
6	$Q_{tap}$	heat loss due to tap water	[kJ/h]
7	$Q_{Sol}$	solar heat gain	[kJ/h]
8	$Q_{in}$	added heat flux	[kJ/h]
9	DE	energy stored in the pool	[kJ]
10	$T_P$	pool water temperature	[ °C]
11	$m_{in}$	mass rate of incoming warm water	[kg/h]

The component outputs are basically self-explanatory.



# Appendix B

## SPAC Output Files

This section includes the description of the output of the swimming pool air conditioner simulation (SPAC). The output files can be found in C:\SPAC\.

### B.1 General System Information

The file spac.ou1 provides general system information on an hourly basis.

Output Parameter	Description
Time	Hour of Year of Simulation
Tamb	Ambient Dry Bulp Temperature [C]
Tpool	Swimming Pool Temperature [C]
Tbuild	Building Temperature [C]
Qhouse	Building Cooling Demand [kJ/hr]
Qcond	Air Conditioner Energy Output [kJ/hr]
Power	Air Conditioner Power Consumption [kJ/hr]

### B.2 Swimming Pool Information

Output file spac.ou2 provides hourly values for the swimming pool

Output Parameter	Description
Time	Hour of Year of Simulation
Qevap	Pool Evaporation Heat Loss [kJ/hr]
Qconv	Pool Convection Heat Loss [kJ/hr]
Qrad	Pool Radiation Heat Loss [kJ/hr]
Qsol	Pool Solar Gains [kJ/hr]
Qheater	Heat added to the pool by pool heater [kJ/hr]
Qcooler	Energy removed from the pool by the cooler [kJ/hr]

### B.3 Weather Information

Weather information for the desired location is available in output file spac.ou3.

Output Parameter	Description
Time	Hour of Year of Simulation
Ibeam	Direct Normal Solar Radiation [kJ/hr]
Iglob	Global Solar Radiation on a horizontal surface [kJ/hr]
Tamb	Ambient Temperature [C]
Humrat	Humidity Ratio
WindVel	Wind Velocity [m/s]

### B.4 Air Conditioner Information

Output file spac.ou4 provides hourly information on air conditioner performance.

Output Parameter	Description
Time	Hour of Year of Simulation
$T_{c,in,w}$	Condenser Inlet Temperature (Pool Water) [C]
$COP_{water}$	Coefficient of Performance for Pool Water [-]
$W_{water}$	Air Conditioner Power for Water-Cooling[kW]
$Q_{cond,w}$	Condenser Energy for Water-Cooling [kJ/hr]
$T_{c,in,a}$	Condenser Inlet Temperature (Ambient Air) [C]
$COP_{air}$	Coefficient of Performance for Ambient Air [-]
$W_{air}$	Air Conditioner Power for Air-Cooling[kW]
$Q_{cond,a}$	Condenser Energy for Air-Cooling [kJ/hr]

### B.5 Economics Information

Monthly information of the economic analysis is available in output file \*.ou5.

Output Parameter	Description
Time	Hour of Year of Month
$Cost_{water}$	Cost for Water-Cooled Air Conditioning [\$/month]
$Cost_{air}$	Cost for Air-Cooled Air Conditioning [\$/month]
$Cost_{gas}$	Cost for Gas Pool Heating [\$/month]

$Cost_{pumpheat}$	Cost for Water Pump for Pool Heating [\$/month]
$Cost_{cool}$	Cost for Water Cooling [\$/month]
$Cost_{pumpcool}$	Cost for Water Pump for Pool Cooling [\$/month]

## B.6 Power Consumption Information

Monthly information of the power consumption of different devices is available in output file spac.ou6.

Output Parameter	Description
Time	Hour of Year of Month
$P_{ac,air}$	Air-Cooled AC Power Consumption [kJ/month]
$P_{ac,water}$	Water-Cooled AC Power Consumption [kJ/month]
$P_{heat}$	Power Consumption of Gas Pool Heater [kJ/month]
$P_{cool}$	Power Consumption of Water Cooler [kJ/month]
$P_{pump, heater}$	Heater Pump Power Consumption [kJ/month]
$P_{pump, cooler}$	Cooler Pump Power Consumption [kJ/month]

## B.7 Water Loss Information

Hourly information of the water loss and water gain is available in output file spac.ou7.

Output Parameter	Description
Time	Hour of Year of Simulation
Waterloss	Hourly Water Loss due to Evaporation [mm/hr]
SumWaterLoss	Integrated Hourly Water Loss [mm]
SumWaterGain	Integrated Hourly Water Gain from Precipitation [mm]



# Appendix C

## SPAC Source Code

This section contains the TRNSYS source code for the Swimming Pool Air Conditioner Simulation Program (SPAC).

```
*TRNSED
*-----
*           Swimming Pool Air Conditioner Simulation
*                   - SPAC -
*                   Sven-Erik Pohl 1999
*                   Master of Science Project
*                   Solar Energy Laboratory
*                   University of Wisconsin, Madison
*-----
ASSIGN C:\spac\spac.lst      6

*/ * |<BACKGROUND> SILVER
* |<BACKGROUND> WHITE
* |<ALIGN1> CENTER
* |<COLOR1> NAVY
* |<SIZE1> 16
* |<STYLE1> BOLD

* |<COLOR2> BLACK
* |<SIZE2> 10
* |<STYLE2> NONE

* |<COLOR3> BLACK
* |<SIZE3> 10
* |<STYLE3> ITALIC

* * Swimming Pool Air Conditioner Simulation
* * - S P A C -
* |<SIZE1> 8
* * Sven-Erik Pohl
* * Solar Energy Laboratory 1999
* |<PICTURE> \spac\acpoollinked.bmp
* * Online Help: Click on the input box and press F1
* |<APPLINK1> TRNINFO.bmp Brochure.pdf LEFT
* |<SIZE1> 14
* *
* * Simulation Parameters
```

```

*|[SIMULATION|
EQUATIONS 10
STARTMONTH= 2880
*|<Month of the simulation start          |\spac\Month1.dat|1|2|1
DAY1= 1.000000000000000E+00
*|Day of Month for Simulation Start       |||0|1|1|31|2
STARTDAY=(STARTMONTH)/24+DAY1
*|*
STOPMONTH= 6552
*|<Month of the simulation Stop          |\spac\Month1.dat|1|2|3
DAY2= 1.000000000000000E+00
*|Day of Month for Simulation Stop       |||0|1|1|31|4
STOPDAY=(STOPMONTH)/24+DAY2
START=24*(STARTDAY-1)+1
STOP=24*(STOPDAY-1)+1
STADAY = (START+23)/24
TSTEP = 1

```

```

* Start time End time Time step
SIMULATION START STOP TSTEP
* Integration Convergence
TOLERANCES 0.001 0.001
* Max iterations; Max warnings; Trace limit;
LIMITS 40 40 40
* TRNSYS output file width, number of characters
WIDTH 80
* TRNSYS numerical integration solver method
DFQ 1

```

```

CONSTANTS 4
BILDER = 1
GRIDNR = 12
ori=0
slope=0
*|]

```

```

*|[GRNDREF|
EQUATIONS 1
GROUNDREF= 1.500000000000000E-01
*|Ground Reflectance                      |||0.00|1.00|1|1.00|5
*|]

```

```

*|(LOCATION| Weather Data Mode
*| TMY2 Weather Data          |TMY|_GENERATOR
*| Weather Generator          |_TMY|GENERATOR
*|)

```

```

* ----- D A T A R E A D E R -----
---
*|#*|[TMY| Location: TMY2 Weather Data
*|#*|ASSIGN C:\spac\weather\madison_wi.tm2 13
*|#*|<City for Simulation
|c:\spac\weather.dat|1|4|6

```

```

* #EQUATIONS 2
* #LAT= 43.13
* #*|<Latitude of City
|c:\spac\weather.dat|0|2|7
* #DevSolar= 0.670
* #*|<Shift in solar time hour angle (degrees)
|c:\spac\weather.dat|0|3|8
* #
* #UNIT 1 TYPE 9 DATA READER
* #PARAMETERS 2
* # -3 13
* #
* #EQUATIONS 10
* #month= [1,1]
* #I=[1,4]
* #Ib=[1,3]
* #Id=[2,5]
* #Tamb=[1,5]
* #humRat=[1,6]
* #Timelast= [1,19]
* #Timenext= [1,20]
* #windVel=[1,7]
* #*/from psychometrics:
* #relhum=[3,6]
* #*|]

* ----- W E A T H E R   G E N E R A T O R -----
----

*|[GENERATOR| Location: Weather Generator
ASSIGN C:\spac\WEATHER\WDATA.DAT 10
EQUATIONS 3
CITY= 127
*|<City for Simulation
|C:\spac\weather\Cities2.dat|2|1|6
LAT= 43.13
*|<Latitude of City
|C:\spac\weather\Cities2.dat|0|3|10
DevSolar= 0.67
*|<Shift in solar time hour angle (degrees)
|c:\spac\weather\Cities2.dat|0|4|11

UNIT 54 TYPE 54 WEATHER GENERATOR
PARAMETERS 6
* UNITS LU CITY# TEMP-MODEL RAD-CORR RAND
    1 10 City 1 1 1

EQUATIONS 10
month=[54,1]
I=[54,7]
Ib=[54,8]
Id=[54,9]
Tamb=[54,4]
relhum=[54,6]

```

```

Timelast=[54,19]
Timenext=[54,20]
windVel=[54,10]
humrat=0.005
*|]

```

```

UNIT 3 TYPE 33 PSYCHROMETRICS PRESIM TYPE 233
PARAMS 4
* 1 drybulb->HR Press[atm] WBMODE EMODE
  4 1 0 1
INPUTS 2
* 1 Tamb 2Humrat
Tamb humrat
* INPUT INITIAL VALUES
* 1 2
  20 0.0028
*/OUTPUT 3,6 : rel. humidity

```

```

* ----- P R E C I P I T A T I O N -----

```

```

* |(precipitationmode|Precipitation Mode
* | Provide Precipitation Data in a File
|prefile|_prettable|rainmaker|waterlossprint
* | Enter Data in a Table
|_prefile|prettable|_rainmaker|waterlossprint
* | Don't Calculate Water Loss
|norain|_prefile|_prettable|_rainmaker|_waterlossprint
* |)

```

```

* |[PREFILE| Monthly Average Precipitation
ASSIGN C:\spac\WEATHER\mad.pre 11
* |? Precipitation Data File Location |12
* |<ALIGN1> LEFT
* |<COLOR1> BLACK
* |<SIZE1> 10
* |<STYLE1> PLAIN
* |*
* |* File has to contain one row and 12 columns with monthly average
precipitation separated by a space
* |*
* |<ALIGN1> CENTER
* |<COLOR1> NAVY
* |<SIZE1> 14
* |<STYLE1> BOLD
* |]

```

```

* |#*|[PRETABLE| Monthly Average Precipitation
* |#Equations 13
* |#Qevap=[6,3]
* |#Jan= 2.7180000000000E+01
* |#*|January
|mm/month||0.00|1.00|1|1000.00|13
* |#Feb= 2.7430000000000E+01

```

```

* | #* | February
| mm/month | | 0.00 | 1.00 | 1 | 1000.00 | 14
* | #mar= 5.5120000000000E+01
* | #* | March
| mm/month | | 0.00 | 1.00 | 1 | 1000.00 | 15
* | #apr= 7.2640000000000E+01
* | #* | April
| mm/month | | 0.00 | 1.00 | 1 | 1000.00 | 16
* | #may= 7.9760000000000E+01
* | #* | May
| mm/month | | 0.00 | 1.00 | 1 | 1000.00 | 17
* | #jun= 9.2960000000000E+01
* | #* | June
| mm/month | | 0.00 | 1.00 | 1 | 1000.00 | 18
* | #jul= 8.6110000000000E+01
* | #* | July
| mm/month | | 0.00 | 1.00 | 1 | 1000.00 | 19
* | #aug= 1.0260000000000E+02
* | #* | August
| mm/month | | 0.00 | 1.00 | 1 | 1000.00 | 20
* | #sep= 8.5600000000000E+01
* | #* | September
| mm/month | | 0.00 | 1.00 | 1 | 1000.00 | 21
* | #oct= 5.5120000000000E+01
* | #* | October
| mm/month | | 0.00 | 1.00 | 1 | 1000.00 | 22
* | #nov= 5.3090000000000E+01
* | #* | November
| mm/month | | 0.00 | 1.00 | 1 | 1000.00 | 23
* | #dec= 4.6740000000000E+01
* | #* | December
| mm/month | | 0.00 | 1.00 | 1 | 1000.00 | 24
* | #* | ]

```

```

* | #* | [NORAIN|
* | #Equations 13
* | #Jan=0
* | #Feb=0
* | #mar=0
* | #apr=0
* | #may=0
* | #jun=0
* | #jul=0
* | #aug=0
* | #sep=0
* | #oct=0
* | #nov=0
* | #dec=0
* | #Qevap=0
* | #* | ]

```

```

* ----- S W I M M I N G   P O O L -----

```

```

* | *Swimming Pool Parameters

```

```

*|[POOL|
EQUATIONS 12
*|* General Information
poolarea= 5.5000000000000E+01
*| Pool Area |m2||0|1|0|1000.00|25
pooldepth= 1.5000000000000E+00
*| Pool Depth |m||0|1|0|20.00|26
Tpoolstart= 1.2000000000000E+01
*| Pool Start Temperature |C||0|1|0|40.00|27
sfac= 3
*|< Shelter mode
|\spac\sheltermode.dat|1|2|28
mode= 1
*|< Water Surface Activity |\spac\poolmode.dat|1|2|29

*|* Swimming Pool Cover Information
covthick= 1.0000000000000E-02
*| Thickness |m||0|1|0|1.00|30
condcov= 1.8000000000000E-01
*| Conductivity |kJ/h-m-K||0|1|0|1.00|31
poolemis= 6.0000000000000E-01
*| Emittance/Absorption |||0|1|0|1.00|32
topen = 1.1000000000000E+01
*| Pool Cover Opening Time |24h||0|1|0|24.00|33
tclose = 1.4000000000000E+01
*| Pool Cover Closing Time |24h||0|1|0|24.00|34
poolvol=poolarea*pooldepth
Fcover = [143,1]
*|]

*|{SETTEMP|
*| Enable Automatic Pool Cover Controller |SET1|_SET2
*|}

*|[SET1|
EQUATIONS 1
Tset= 2.6000000000000E+01
*| Pool Set Temperature |||0|1|0|40.00|35
*|]

*|#*|[SET2|
*|#EQUATIONS 1
*|#Tset=0
*|#*|]

*/*|{SETTEMP|
*/*| Connect a Building to the Swimming Pool ?|AIR-COND|HOUSE|_AC-
OFF|56BUILDING|AC-ON1|AC-ON2
*/*|}

*----- G A S P O O L H E A T E R S Y S T E M -----
--
*|* Gas Pool Heater System

```

```

*|{heater|
*| Connect Gas Pool Heater to the Pool ?|Poolheater|_noheater
*|}

*|[Poolheater|
CONSTANTS 7

*|* Gas Furnace
QmaxkW= 4.4000000000000E+01
*| Maximum Heating Rate |kW||0|1|0|10000000.00|36
Tsetgas= 2.5000000000000E+01
*| Set Pool Temperature |C||0|1|0|40.00|37
cp=4.176
UA= 0.0000000000000E+00
*/| Overall Loss Coefficient |kJ/hr-C||0|1|0|40.00|38
eta= 7.0000000000000E-01
*| Efficiency |-||0|1|0|40.00|39

*|* Pump
m_max= 1.4732000000000E+04
*| maximum flowrate |kg/hr||0|1|0|100000.00|40
PmaxkW= 3.8000000000000E-01
*| maximum Power Consumption |kW||0|1|0|10000.00|41

EQUATIONS 2
P_max=PmaxkW*3600
Qmax=QmaxkW*3600

UNIT 82 TYPE 2 Controller
PARAMETER 4
*NSTK DThigh DTlow Tmax
5 Tsetgas TSetgas 40
INPUTS 4
6,1 0,0 0,0 82,1
*INITIAL VALUES
12 0 0 0

EQUATIONS 1
onoff=1-[82,1]

UNIT 81 TYPE 3 PUMP
PARAMETERS 4
*m_max cp Pmax fpar
m_max cp P_max 0
INPUTS 3
*T_in m_o onoff
6,1 81,2 onoff
*INITIAL VALUES
0 0 0

EQUATIONS 2
m_in=[81,2]

```

```

Q_pump=[81,3]

UNIT 80 TYPE 6 GAS POOL HEATER
PARAMETERS 5
Qmax Tsetgas cp UA eta
INPUTS 4
*T_in m_in onoff Tamb
  6,1 m_in onoff Tamb
*INITIAL VALUES
12 0 0 0 0

EQUATIONS 2
Qfluidheat=[80,5]
Qaux_gas=[80,3]

*|]

*|#*|[noheater|
*|#Equations 3
*|#Qfluidheat=0
*|#Qaux_gas=0
*|#Q_pump=0
*|#*|]

* ----- P O O L C O O L I N G S Y S T E M -----
*|* Pool Cooling System

*|{cooler|
*| Connect Pool Cooling System to the Swimming Pool
?|Poolcooler|_nocooler
*|}

*|[Poolcooler|
CONSTANTS 7

*|* Cooling System
Qmaxcoolkw= 4.4000000000000E+01
*| Maximum Cooling Rate |kW||0|1|0|10000000.00|42
Tsetcool= 2.5000000000000E+01
*| Set Pool Temperature |C||0|1|0|40.00|43
cp_cool=4.176
UA_cool= 0.0000000000000E+00
copcool= 2.0000000000000E+00
*| Coefficient of Performance |-||0|1|0|10.00|45

*|* Pump
m_max_cool= 1.4732000000000E+04
*| maximum flowrate |kg/hr||0|1|0|100000.00|46
Pmaxcoolkw= 3.8000000000000E-01
*| maximum Power Consumption |kW||0|1|0|10000.00|47

equation 3

```

```

etacool=1/copcool
Qmaxcool=Qmaxcoolkw*3600
P_max_cool=Pmaxcoolkw*3600

```

```

UNIT 72 TYPE 2 Controller
PARAMETER 4
*NSTK DThigh DTlow Tmax
7 Tsetcool TSetcool 40
INPUTS 4
6,1 0,0 0,0 72,1
*INITIAL VALUES
12 0 0 0

```

```

EQUATIONS 1
onoffcool=[72,1]

```

```

UNIT 71 TYPE 3 PUMP
PARAMETERS 4
*m_max cp Pmax fpar
m_max_cool cp_cool P_max_cool 0
INPUTS 3
*T_in m_o onoff
6,1 71,2 onoffcool
*INITIAL VALUES
0 0 0

```

```

EQUATIONS 2
m_in_cool=[71,2]
Q_pump_cool=[71,3]

```

```

UNIT 70 TYPE 92 POOL COOLER
PARAMETERS 4
Qmaxcool cp_cool UA_cool etacool
INPUTS 5
*T_in m_in onoffcool Tsetcool Tamb
6,1 m_in_cool onoffcool Tsetcool Tamb
*INITIAL VALUES
12 0 0 0 0 0

```

```

EQUATIONS 2
Qfluidcool=[70,5]
Qaux_cool=[70,3]
*|]

```

```

*|#*|[nocooler|
*|#Equations 3
*|#Qfluidcool=0
*|#Qaux_cool=0
*|#Q_pump_cool=0
*|#*|]

```

```

*----- A I R   C O N D I T I O N E R -----
*|* Air Conditioner

```

```

* |(AIR-COND|
* | No Air Conditioner installed |_AC-ON1|_AC-ON2|AC-OFF|_VARCOP-
PRINT|_fluid|_HOUSE
* | Constant COP model |_AC-ON1|_AC-ON2|_AC-OFF|_VARCOP-PRINT|fluid|HOUSE
* | Variable COP model |_AC-ON1|_AC-ON2|_AC-OFF|VARCOP-PRINT|fluid|HOUSE
* |)

* |[fluid|
equations 1
fluidmode= 1
* |<Heat is rejected to | \spac\fluid.dat|1|2|48
* |]

* |#*|[AC-ON1|Constant COP model
* |#
* |#*|<PICTURE> \spac\acsimple.bmp
* |#CONSTANTS 1
* |#COP= 3.000000000000000E+00
* |#*| Coefficient of Performance ||0|1|0|10.00|49
* |#EQUATIONS 11
* |#Qcond=[56,3]*(1+1/COP)*fluidmode
* |#Qdemand= [56,3]
* |#Tbuild = [56,1]
* |#Qback = 0
* |#Power=Qdemand/COP
* |#W_water=Power
* |#W_air=Power
* |#DT_w=0
* |#DT_a=0
* |#T_c_in_w=[6,1]
* |#T_c_in_a=Tamb
* |#Qac=Qcond
* |#*|]

* |[AC-on2| Variable COP Air Conditioner Model
EQUATIONS 3
T_c_in_w=[6,1]
T_c_in_a=Tamb
Qdemand= [56,3]

CONSTANTS 4

* |<SIZE1> 10
* |* Air cooled System Information from Manufacturer Data
* |*
Cap_air= 1.860000000000000E+01
* |Capacity at Tcin=35 C (ARI-Condition) |kW||0.00|1.00|0|100.00|50
COP_air= 3.040000000000000E+00
* |COP at Tcin=35 C (ARI-Condition) |kW||0.00|1.00|0|100.00|51
* |*
* |*|* Temperature Approach
dT_a= 1.000000000000000E+01
* |*|dT_air |C||0.00|1.00|0|100.00|52

```

```

dT_w= 5.500000000000000E+00
*/*|dT_water |C||0.00|1.00|0|100.00|53
*|<SIZE1> 14

UNIT 44 TYPE 140 AC
PARAMS 4
COP_air CAP_air dt_w dt_a
INPUTS 3
T_c_in_a T_c_in_w Qdemand
*INTial VAlues
12 12 0

Equations 8
*ac outputs
W_water=[44,4]
W_air=[44,5]
QcondW=[44,2]
*misc
Qback=0
Power=fluidmode*W_water+(1-fluidmode)*W_air
*Qcond=0 for air, no heat is rejected to the pool...
Qcond=fluidmode*QcondW
Tbuild = [56,1]
Qac=fluidmode*QcondW+(1-fluidmode)*[44,3]
*|]

*|#*|[AC-OFF|
*|#CONSTANTS 10
*|#Qac=0
*|#Qcond=0
*|#Qdemand= 0
*|#Tbuild = 0
*|#Power=0
*|#Qback=0
*|#T_c_out_water=45
*|#T_c_in=25
*|#W_water=0
*|#W_air=0
*|#*|]

*----- B U I L D I N G -----
*|*

*|[HOUSE|Building Parameters
ASSIGN c:\spac\bid\lib\w4-libe.dat 43
ASSIGN c:\spac\myhouse\house.bld 41
*|<Building Type
|c:\spac\buildings.dat|1|2|54
ASSIGN c:\spac\myhouse\house.trn 42
*|<Building Type
|c:\spac\buildings.dat|0|3|55
Constant 1
ROOMTEMP= 2.500000000000000E+01

```

```
*|Comfort Room Temperature (25 C ASHRAE) |C||0.00|1.00|0|100.00|56
```

```
UNIT 56 TYPE 56 MULTIZONE BUILDING
```

```
PARAMS 5
```

```
* 1 BuildDescr. 2 WallTrns 3 WinLib 4 T_mode 5 WeightingFac
      41          42          43          0          1
```

```
INPUTS 25
```

```
* Tamb rh Tsky
Tamb relhum 4,1
```

```
* NRad SRad ERad WRad HRad NslopRad SSlopRad
91,14 91,6 91,17 91,11 2,6 92,11 92,6
```

```
* NBeamRad SBeamRad EBeamRad WBeamRad HBeamRad NslopBeamRad SSlopBeamRad
91,15 91,7 91,18 91,12 2,7 92,12 92,7
```

```
* NIncAng SIncAng EIncAng WIncAng HIncAng NslopIncAng SSlopIncAng
91,16 91,9 91,19 91,13 2,8 92,13 92,9
```

```
* Set Room Temperature
ROOMTEMP
```

```
10 40 10 0 0 0 0 0 0 0 0 0
0 0 0 0 0 0 0 0 0 0 0 0
```

```
* INPUT INITIAL VALUES
```

```
* 1 2 3 4 5 6 7 8 9 10 12
```

```
* 13 14 15 16 17 18 19 20 21 22 23 24
```

```
*|]
```

```
*|#*|[NOHOUSE|
*|#*|]
```

```
*----- E C O N O M I C S -----
```

```
UNIT 39 TYPE 24 INTEGRATOR
```

```
PARAMETERS 1
```

```
* monthly output
```

```
-1
```

```
INPUT 6
```

```
W_water W_air Qaux_gas Q_pump Qaux_cool Q_pump_cool
```

```
*INITIAL VALUE
```

```
0 0 0 0 0 0
```

```
*|[Econ|Economics
```

```
EQUATIONS 8
```

```
ecost= 7.500000000000000E-02
```

```
*| Electricity Cost
```

```
|$/kwh||0|1|0|1.000|57
```

```
gcost= 2.000000000000000E-02
```

```

* | Natural Gas Cost
|$/kwh||0|1|0|1.000|58
costwater=[39,1]/3600*ecost
costair=[39,2]/3600*ecost
costgas=[39,3]/3600*gcost
costpump=[39,4]/3600*ecost
costcool=[39,5]/3600*ecost
costpumpcool=[39,6]/3600*ecost
Equations 1
deltacost=costair-costwater
* | ]

* ----- O U T P U T   I N F O R M A T I O N -----
-
* | * Output Information

* | {PRINTERMODE|Printer Mode
* | Show Online Printer ? |ONLINEPRN
* | }

* | {Outputmode|Output Mode
* | General Information (Output 1 : SPAC.OU1) |BuildingPrint
* | Energy Information (Output 2 : SPAC.OU2) |PoolenergyPrint
* | Weather Information (Output 3 : SPAC.OU3) |Weatherprint1
* | Air-Conditioner Information (Output 4 : SPAC.OU4) |Varcop-Print
* | Economic Information (Output 5 : SPAC.OU5) |Econ|Printecon
* | System Power Consumption Information (Output 6 : SPAC.OU6)|PowerPrint
* | }

* | *
* | * Press F8 to RUN simulation !

*====T R N S Y S - O N L Y =====
*----- R A D I A T I O N -----

UNIT 2  TYPE 16 RADIATION PROCESSOR          FOR SWIMMING POOL
PARAMS 9
*1 Mode 2 Tracking 3 Rad.mode 4 start_day
  7          1          1          STADAY

*5 Latitude 6 Solar_const
  Lat          4871.1

*7 Shift in solar time hour angle (degrees)
  DevSolar

*8 Rad_mode 9 sim-time
  2          1

INPUTS 9
* 1 globrad 2Beam_rad 3 Time_last_reading 4 Time_next_reading
  I          Ib          TimeLast          TimeNext

```

```

* 5 Gr_reflect  6 Slope/axis  7 AziAngle  8 I_next  9 Ib_next
groundref      0,0          0,0          0,0          0,0

* INPUT INITIAL VALUES
* 1  2  3  4  5  6  7  8  9
  0  0  0  1  0.2 SLOPE ORI  0  0

UNIT 91 TYPE 16 RADIATION PROCESSOR          FOR HOUSE
PARAMS 9
*1 Mode 2 Tracking 3 Rad.mode 4 start_day
  7          1          1          STADAY

*5 Latitude 6 Solar_const
  Lat          4871.1

*7 Shift in solar time hour angle (degrees)
  DevSolar

*8 Rad_smooth 9 sim-time
  1          1

INPUTS 15
* 1 Glob_rad 2 Beam_rad 3 Time_last_reading 4 Time_next_reading  5
Gr_reflect
  I          Ib          Timelast          Timenext
groundref

* 6 Sl-S  7 Azi-S  8 SL-W  9 Azi-W  10 Sl-N  11 Azi-N  12 Sl-E  13 Azi-E
  0,0  0,0  0,0  0,0  0,0  0,0  0,0  0,0

* 14 I_next  15 Ib_next
  0,0          0,0

* INPUT INITIAL VALUES
* 1  2  3  4  5  6  7  8  9  10  11  12  13  14  15
  0  0  0  1  0.2  0  0  0 -90  0  180  0  90  0  0

UNIT 92 TYPE 16 RADIATION PROCESSOR          FOR ROOF
PARAMS 9
*1 Mode 2 Tracking 3 Rad.mode 4 start_day
  7          1          1          STADAY

*5 Latitude 6 Solar_const
  Lat          4871.1

*7 Shift in solar time hour angle (degrees)
  DevSolar

*8 Rad_smooth 9 sim-time
  1          1

INPUTS 11

```

```

* 1 glob_rad  2 beam_rad  3 Time_last_reading  4 Time_next_reading  5
Gr_reflect
      I          Ib          TimeLast          TimeNext
groundref
* 6 Sl-S  7 Azi-S  8 SL-N  9 Azi-N
      0, 0    0, 0    0, 0    0, 0

* 10 I_next  11 Ib_next
      0,0      0,0

* INPUT INITIAL VALUES
* 1  2  3  4  5  6  7  8  9  10  11
      0  0  0  1  0.2  14  0  14  180  0  0

UNIT 4 TYPE 69 Sky Temperature
PARAMS 2
* mode height
      0 450
INPUTS 5
* Ta Tdp Ib Id
Tamb 3,8 Ib Id 0,0
* INPUT INITIAL VALUES
      0 0 0 0 0

*-----S W I M M I N G   P O O L -----

EQUATIONS 1
Qpoolin=Qfluidheat+Qcond-Qfluidcool

UNIT 143 TYPE 143 PREPOOL - FCOVER
PARAMETER 3
* topen tclose settemp
topen tclose Tset

INPUTS 2
* Tpool fcover
6,1 143,1

*INITIAL VALUES
Tpoolstart 0

UNIT 6 TYPE 144 SWIMMING POOL SIMULATION
PARAMS 10

* 1 pooltemp_start  2 poolarea  3 poolvol  4 ModeN  5 epsilon_cover
Tpoolstart      poolarea      poolvol      1      poolemis

* 6 absorbt  7 Cond_cover  8 Thick_cover  9 Heigth_windspeed
poolemis      condcov      covthick      10

* 10 Shelter_factor
      sfac
INPUTS 15

```

```
* 1Tamb 2relhum 3windspeed 4RadHori 5Tsky 6Tpoolwall 7HeatTrans
Tamb relhum windVel 2,4 4,1 0,0 0,0
```

```
* 8mode 9m_dotTab 10Ttab 11lopen 12close 13maxpeople 14cover%
Qpoolin
mode 7,1 0,0 0,0 0,0 0,0 Fcover Qpoolin
```

```
*mPool TpoolIn
```

```
* INPUT INITIAL VALUES
```

```
* 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16
20 60 0 0 0 0 0 0 0 12 0 0 0 0 0 12
```

```
* ----- TAB WATER -----
```

```
UNIT 7 TYPE 14 FORCING FUNCTION PRESIM TYPE 1014
```

```
PARAMS 12
```

```
0 0
48 0
*/48 1000
48 0
58 0
*/58 1000
58 0
168 0
```

```
* ----- MAKE UP WATER -----
```

```
*|[RAINMAKER|
```

```
UNIT 29 TYPE 9 DATA READER - THE RAINMAKER !
```

```
PARAMETER 41
```

```
* mode #ofvalues timestep
```

```
2 12 1
```

```
*value mult add
```

```
-1 1 0
```

```
-2 1 0
```

```
-3 1 0
```

```
-4 1 0
```

```
-5 1 0
```

```
-6 1 0
```

```
-7 1 0
```

```
-8 1 0
```

```
-9 1 0
```

```
-10 1 0
```

```
-11 1 0
```

```
-12 1 0
```

```
*unit# format
```

```
11 0
```

```
EQUATIONS 13
```

```
jan=[29,1]
```

```
feb=[29,2]
```

```
mar=[29,3]
```

```

apr=[29,4]
may=[29,5]
jun=[29,6]
jul=[29,7]
aug=[29,8]
sep=[29,9]
oct=[29,10]
nov=[29,11]
dec=[29,12]
Qevap=[6,3]
*|]
UNIT 31 TYPE 130 Calculates hourly precipitation
INPUTS 12
jan feb mar apr may jun jul aug sep oct nov dec
*INITIAL VALUES
jan feb mar apr may jun jul aug sep oct nov dec

EQUATIONS 3
prewater=[31,1]
* waterloss=Qevap/(rho*dh)*3.6 [m3/hr]
waterloss = 0.27777*3.6*Qevap/(997.9*(1602.652-6.659528*[6,1]))
watergain = prewater/1000*poolarea

UNIT 30 TYPE 24 INTEGRATOR
INPUT 2
waterloss watergain
*INITIAL VALUE
0 0

* ----- O U T P U T -----
*|[BuildingPrint|
ASSIGN C:\spac\spac.oul 31
UNIT 19 TYPE 25 PRINTER OUTPUT 1 General Information
PARAMS 5
*STEP START STOP LOGICAL-UNIT UNITS
1 Start Stop 31 1
INPUTS 6
Tamb 6,1 Tbuild Qdemand Qcond Power
Tamb Tpool Tbuild Qhouse Qcond ACPower
C C C kJ/hr kJ/hr kJ/hr
*|]

*|[PoolenergyPrint|
ASSIGN C:\spac\spac.ou2 32
UNIT 99 TYPE 25 PRINTER PRESIM TYPE 1125 OUTPUT 2 Energy
PARAMS 5
*STEP START STOP LOGICAL-UNIT UNITS
1 start stop 32 1
INPUTS 7
6,3 6,4 6,5 6,7 Qac Qfluidheat Qfluidcool

```

```

Qevap Qconv Qrad Qsol Qac Qheater Qcooler
kJ/hr kJ/hr kJ/hr kJ/hr kJ/hr kJ/hr kJ/hr
*|]

```

```

*|[WEATHERPRINT1|
ASSIGN C:\spac\spac.ou3 33
UNIT 100 TYPE 25 PRINTER OUTPUT 3 Weather
PARAMS 5
*STEP START STOP LOGICAL-UNIT UNITS
1 START STOP 33 1
INPUTS 5
Ib I Tamb humrat windvel
Ib Iglob Tamb humrat windVel
kJ/hr-m2 kJ/hr-m2 C % m/s
*|]

```

```

*|[VARCOP-PRINT|
ASSIGN C:\spac\spac.ou4 34
UNIT 101 TYPE 25 OUTPUT 4 AC
PARAMS 5
*STEP START STOP LOGICAL-UNIT UNITS
1 0 10000 34 1
*T_c_in COP_water W_water QcondW COP_air W_air Qconda
INPUTS 8
6,1 44,6 44,4 44,2 Tamb 44,7 44,5 44,3
T_c_in_w COP_water W_water QcondW T_c_in_a COP_air W_air Qconda
C - kW kJ/hr C - kW kJ/hr
*|]

```

```

*|[PrintEcon|
ASSIGN C:\spac\spac.ou5 35

UNIT 102 TYPE 25 Output 5 Economics
PARAMETERS 5
*STEP START STOP LOGICAL-UNIT UNITS
-1 Start STOP 35 1
INPUTS 6
costwater costair costgas costpump costcool costpumpcool
ac_water ac_air cost_heating pumph heating poolcooling pumphcooling
$ $ $ $ $ $
*|]

```

```

*|[PowerPrint|
ASSIGN C:\spac\spac.ou6 37

UNIT 104 TYPE 25 Output 6 Integrated Power Consumption
PARAMETERS 5
*STEP START STOP LOGICAL-UNIT UNITS
-1 Start STOP 37 1
INPUTS 6
39,1 39,2 39,3 39,5 39,4 39,6
P_AC_water P_AC_air P_heating P_cooling P_pump_heat P_pump_cool

```

```
kJ kJ kJ kJ kJ kJ
*|]
```

```
*|[waterlossprint|
ASSIGN C:\spac\spac.ou7 36
UNIT 103 TYPE 25 PRINTER Output 7 Waterloss
PARAMS 5
*STEP START STOP LOGICAL-UNIT UNITS
1 START STOP 36 1
INPUTS 3
waterloss 30,1 30,2
waterloss sumwloss sumwgain
m3/hr m3 m3
*|]
```

```
*\*----- O N L I N E P R I N T E R -----
*|[ONLINEPRN|
UNIT 20 TYPE 65 ONLINE PLOTTER
PARAMS 14
* 1#left 2#right 3minylef 4maxylef 5minyrig 6maxyrig 7updateplot
8updatenum
3 0 10 40 0 0 1 1
* 9units 10Npic 11grid 12stop 13symbols 14on/off
3 BILDER GRIDNR 2 2 0
INPUTS 3
* 1Ta 2Tpool 3Tbuild
Tamb
6,1
Tbuild
*INPUT INITIAL VALUES
*1 2 3
Tamb Tpool
Thouse

LABELS 4
øC
øC
Temperatures [øC]
-
*|]
```

```
END
```



# Appendix D

## EES Air Conditioner Model

The source code for the EES program is shown below. Various inputs can be modified in the diagram window.

```

"!----- Air Conditioner Model -----"

FUNCTION fluidmode (fluid$)
{determines condenser cooling fluid set in diagram window}
IF fluid$ = 'water' THEN
    fluidmode=1
ENDIF
IF fluid$='air' THEN
    fluidmode=0
ENDIF
END

R$='R22'

"!Basis: Fixed compressor displacement"
{D_dot      =0.0016 "m3/s      compressor displacement rate:  "}
V_dot      =D_dot*Eta_volumetric
Eta_volumetric=1-C*(v[1]/v[2]-1)
C          =0.03 "ratio of clearance volume to displacement"

"!Compressor "
{eta_comp      =0.5}
"!First determine isentropic conditions at state 2 designated with the `
symbol"
h2`          =enthalpy(R$,P=P[2],s=s[1])
W_id        =(h2`-h[1])*n_dot "ideal compressor work"
W           =W_id/Eta_comp "actual compressor work"
W           =(h[2]-h[1])*n_dot "energy balance to determine enthalpy
at compressor outlet"
T[2]        =Temperature(R$,H=h[2],P=P[2])
v[2]        =volume(R$,H=h[2],P=P[2])
x[2]        =quality(R$,H=h[2],P=P[2])

"!Condenser"
P[2]        =P[3]
h[3]        =enthalpy(R$,P=P[3],x=0)
T[3]        =temperature(R$,P=P[3],x=0)

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Q_cond      =(h[2]-h[3])*(n_dot) "heat transfer from condenser from an
energy balance"
Q_cond      =epsilon_cond*C_min_cond*(T[3]-T_C_in) "heat transfer rate
equation"
epsilon_cond =1-exp(-NTU_cond) "effectiveness with Cr=0 (constant
temperature condensation)"
NTU_cond    =UA_cond/C_min_cond "condenser NTU"
C_min_cond  =C_waterair
Q_cond=C_waterair*(T_C_out-T_C_in)"Poolwaterpump-massflowrate=const"
c_p        =SPECHEAT(Water,T=T_c_in,P=101.3)
n_dot_cond=m_dot_cond/MOLARMASS(fluid$)
c_air       = SPECHEAT(Air,T=T_C_in) "[kJ/kmole-K]"
c_water     = SPECHEAT(Water,T=T_c_in,P=101.3) "[kJ/kmole-K]"
C_waterair  = (1-
fluidmode(fluid$))*n_dot_cond*c_air+fluidmode(fluid$)*n_dot_cond*c_water
epsilon_c[1] =epsilon_cond
NTU_c[1]    =NTU_cond

"!Throttle - isenthalpic flashing"
x[4]        =quality(R$,P=P[4],h=h[4])
h[4]        =h[3]
T[4]        =temperature(R$,P=P[4],h=h[4])

"!Evaporator"
P[1]        =P[4]
h[1]        =enthalpy(R$,P=P[1], x=1)
v[1]        =volume(R$,P=P[1],x=1)
s[1]        =entropy(R$,P=P[1],x=1)
T[1]        =Temperature(R$,P=P[1],x=1)
V_dot       =v[1]*n_dot "this statement determines n_dot, the
refrigerant molar flowrate"
m_dot       =n_dot*MOLARMASS(R$)
C_min_evap  =Q_evap/(T_E_in-T_E_out) "minimum capacitance rate is the
air"
Q_evap      =epsilon_evap*C_min_evap*(T_E_in-T[4]) "heat transfer rate
equation"
Q_evap      =(h[1]-h[4])*n_dot "evaporator heat transfer rate from an
energy balance"
NTU_evap    =UA_evap/C_min_evap "NTU of evaporator"
epsilon_evap =1-exp(-NTU_evap)
epsilon_e[1] =epsilon_evap
NTU_e[1]    =NTU_evap

Q_evap      =n_dot_evap*c_air_e*(T_e_in-T_e_out)
m_dot_evap  =n_dot_evap*MOLARMASS(Air)
c_air_e     =SPECHEAT(Air,T=T_e_in)

m_dot_cond=(1-fluidmode(fluid$))*m_dot_air+fluidmode(fluid$)*m_dot_water

"TEMPERATURE APPROACH"

T[3]        =T_c_in+DELTAT_cond
DELTAT_cond =DELTAT_water*fluidmode(fluid$)+DELTAT_air*(1-
fluidmode(fluid$))

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T_e_out      =DELTA_T_evap+T[1]

"Volumetric Flows"
V_dot_cond=m_dot_cond/DENSITY(fluid$,T=T_c_in,P=101.3)/MOLARMASS(fluid$)
V_dot_cond_cfm=V_dot_cond*convert(m^3/s, cfm)

V_dot_evap   =m_dot_evap/DENSITY(air,T=T_e_in,P=101.3)/MOLARMASS(air)
V_dot_evap_cfm=V_dot_evap*convert(m^3/s, cfm)

"!end of ac-model"

"!System"
COP          =Q_evap/(W+W_dot_fan+W_dot_pump+W_dot_fan_evap)
W_total      =W+W_dot_fan+W_dot_pump+W_dot_fan_evap
EER          =q_evap*convert(kW,Btu/hr)/
((W+W_dot_fan+W_dot_pump)*convert(kW,W))

"! FAN-tastic"
{N_ref      =2800                                "[rpm]" }
{d          =28*convert(inch,m)                  "[m]" }
{W_dot_ref  =0.345                                "[kW]" }
{V_dot_ref  =2800*convert(cfm, m^3/s)            "[m^3/s]" }

V_dot_ref    =V_dot_ref_cfm*convert(cfm, m^3/s)
W_dot_ref    =C_w*(N_ref*convert(rpm, rps))^3*d^5*rho*convert(W, kW)
V_dot_ref    =C_v*N_ref*convert(rpm, rps)*d^3
rho          =DENSITY(air,T=20,P=101.3)*MOLARMASS(air) "[kg/m^3]"

"Fanlaws"
W_dot_fan    =C_w*(N*convert(rpm, rps))^3*d^5*rho*convert(W, kW)*(1-
fluidmode(fluid$))
V_dot_fan    =C_v*N*convert(rpm, rps)*d^3
V_dot_fan    =V_dot_cond

W_dot_fan_evap=C_w*(N2*convert(rpm, rps))^3*d^5*rho*convert(W, kW)
V_dot_fan_evap=C_v*N2*convert(rpm, rps)*d^3
V_dot_fan_evap=V_dot_evap

"!PUMP-astic"

rho_pump     =DENSITY(water,T=T_C_in,P=101.3)*MOLARMASS(water)
              "[kg/m^3]"
W_dot_pump   =DELTA_p_v*V_dot_pump*convert(W,kW)/eta_pump

DELTA_p_v    =ceta*rho_pump/2*v_pump^2           "[kg/m-s^2]"
ceta         =1
v_pump       =V_dot_cond/A_pipe                  "[m/s] velocity of
water"
A_pipe       =d_pipe^2*pi/4                      "[m^2]"
{d_pipe      =0.015                              "[m]" }

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V_dot_pump =V_dot_cond*fluidmode(fluid$)
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