AN EVALUATION OF ICE AND CHILLED WATER AS THERMAL STORAGE MEDIA FOR COMBUSTION TURBINE INLET AIR COOLING SYSTEMS

by

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ABSTRACT

Utilities are relying increasingly on combustion turbines to generate both on- and off-peak electricity. The capacity and conversion efficiency of combustion turbines decline as the ambient dry bulb temperature increases. Turbine performance can be improved substantially by cooling the air before it enters the compressor stage. This study evaluates the use of chilled water and ice as thermal storage media for inlet air cooling systems designed for a simple cycle combustion turbine power plant located in the upper mid-Western United States.

Combustion turbine, ice harvester, ice storage tank, and evaporative cooler models were developed and implemented as TRNSYS computer simulation components. These components were used together with standard subroutines to build a model of a combustion turbine inlet air cooling system based on both chilled water and ice storage. The overall system model can also be used to simulate the performance of cooling systems based on chilled water or ice storage alone. EES programs modeling both the chilled water and ice storage sections of the overall cooling system were also written for use in the system design process.

Cooling systems based on each of the two storage media alone and on an optimized combination of chilled water and ice storage were designed for four different power plant load profiles. Two general cases were considered: base mode combustion turbine operation without evaporative cooling and power augmentation mode combustion turbine operation with evaporative cooling upstream of the thermal storage based cooling system. Annual simulations were performed for each cooling system design based on the average expected number of hours of power plant operation during the cooling season.

Chilled water storage based inlet air cooling systems lead to a power plant capacity increase of 16.0% and a maximum overall conversion efficiency increase of 1.9% at design conditions in the first general case, and result in a capacity increase of 9.5% and a maximum overall conversion efficiency increase of 0.8% in the second general case. Inlet air cooling systems based on ice storage alone result in capacity increases of 17.9% and 11.3% in the first and second general cases, respectively. The capacity increases associated with the optimized combination of storage media are 0.1% lower than those for systems based on ice storage alone. The maximum overall conversion efficiency is approximately the same for all storage capacity splits.

Cooling systems were also compared based on the associated power plant capacity enhancement cost, the peak capacity enhancement cost, and the cost of the incremental power generated with inlet air cooling assuming a 20 year system payback period. The peak capacity enhancement cost constitutes a measure of the value of the incremental capacity provided by ice storage in comparison to water storage. The incremental power generation cost takes into account operating costs in addition to first costs.

Cooling systems based on chilled water storage alone yield the lowest capacity enhancement and incremental power generation costs for all power plant load profiles and operating conditions considered, while cooling systems based on ice storage alone yield the highest capacity enhancement costs and incremental power generation costs. The peak capacity enhancement costs for systems based on ice storage alone are 1.3 to 4.3 times as great as those for systems based on a combination of storage media. In general, the most appropriate storage capacity split will have to be determined on a case by case basis. The models developed in this study are useful tools for making such a determination.

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NOMENCLATURE

A	cooling coil face area
AMFR	air mass flow rate
AWB	ambient wet bulb temperature
BEP	combustion turbine capacity for inlet air at 59° F and 14.2 psia; no water injection
Cap	capacity
CChrs	effective number of weekly hours of operation for second cooling coil
CCMF	cooling coil water mass flow rate
CCWV	cooling coil water velocity
CEC	inlet air cooling system capacity enhancement cost
C_F	fuel cost
C_{OPE}	wholesale cost of off-peak electricity
C_{PE}	wholesale cost of on-peak electricity/cost of incremental electricity produced with inlet air cooling
C_{pw}	water heat capacity at constant pressure
Cstar	ratio of minimum to maximum cooling coil capacitance rates
CV1	first cooling coil pump control variable
CV1min	minimum value of first cooling coil pump control variable
CV2	second cooling coil pump control variable
CV2min	minimum value of second cooling coil pump control variable
D	diameter
d	discount rate
dPin	combustion turbine inlet system pressure loss

dPout	combustion turbine exhaust system pressure loss
DECWT	chiller design entering condenser water temperature
Dhrs	effective number of daily hours of operation for first cooling coil
divf	diverted fraction of water flow from chiller evaporator outlet
DLEWT	chiller design leaving evaporator water temperature
DWB	design wet bulb temperature
EE_{OP}	annual cooling system electrical energy consumption
ECWT	chiller entering condenser water temperature
EDB	entering dry bulb temperature
eff	effectiveness
EP	combustion turbine gross electric power output
EPNC	combustion turbine net electric power output without inlet air cooling
eps_{pass}	effectiveness for single cooling coil tube pass
EWB	entering wet bulb temperature
EWT	entering water temperature
f_{air}	air friction factor
f_{wat}	water friction factor
FMFR	fuel mass flow rate
G	air mass flux
g1min	ratio between minimum and maximum water mass flow rates for first cooling coil
g2min	ratio between minimum and maximum water mass flow rates for second cooling coil
g_c	force - mass conversion in English units
H_a	initial position of thermocline in chilled water storage tank

HHV	higher heating value
HRCF	evaporative condenser heat rejection correction factor
i	fuel inflation rate
IPLCM	inlet pressure loss capacity multiplier
IPLEM	inlet pressure loss efficiency multiplier
K_c	cooling coil air side entrance pressure loss coefficient
K_e	cooling coil air side exit pressure loss coefficient
L	length
LDB	leaving dry bulb temperature
LEWT	chiller leaving evaporator water temperature
LWT	leaving water temperature
MEP	combustion turbine capacity
MLWT	maximum chiller evaporator inlet temperature
n_{base}	combustion turbine efficiency for inlet air at 59°F and 14.2 psia; no water injection
NCap	ice harvester net capacity
NEP	combustion turbine net electric power output
n_{HHV}	combustion turbine efficiency
NomCap	ice harvester compressor nominal capacity
N_P	discounted payback period
NPower	ice harvester net power requirement
n_{rel}	combustion turbine relative efficiency
Nrows	number of cooling coil rows
Ntu	number of transfer units
Ntup	number of transfer units per cooling coil tube pass

num	ratio of cooling coil duct width to duct height
OPEC	average annual cost of off-peak electricity required by cooling system
OPLCEM	outlet pressure loss capacity and efficiency multiplier
PCEC	inlet air cooling system peak capacity enhancement cost
P_{des}	chiller design electric power requirement
PLF	combustion turbine part load factor
P_{req}	chiller electric power requirement
PWF	present worth factor
Q_{des}	chiller design refrigeration load
Q_{load}	chiller refrigeration load
Re_D	Reynold's number based on inner tube diameter
RefCap	ice harvester reference capacity
RefPow	ice harvester reference brake horsepower
rho	water density
RPO	combustion turbine relative power output
s	ratio of free flow area to cooling coil face area
SCT	saturated condensing temperature
SDT	saturated discharge temperature
SPT	chiller set point temperature
SST	saturated suction temperature
T_I	initial temperature of lower portion of chilled water storage tank
T_{set}	initial temperature of upper portion of chilled water storage tank
v	specific volume
VCOL	chilled water storage tank flow rate to volume ratio
WFR	ratio of water and fuel mass flow rates

WFRCM	water-fuel ratio capacity multiplier
WFREM	water-fuel ratio efficiency multiplier
WMFR	water mass flow rate
WVFR	water volumetric flow rate
ΔEE_{NC}	annual incremental electrical energy produced with inlet air cooling
ΔEE_{LDB}	annual difference between electrical energy that could be produced with 40° F inlet air and the electrical energy actually produced
ΔEP	difference between desired and actual net power plant output
ΔFuel	annual incremental fuel consumption due to inlet air cooling
ΔP_{air}	cooling coil air side pressure drop
ΔP_{wat}	cooling coil water side pressure drop
ΣCV	sum of the two cooling coil pump control variables

CHAPTER 1: INTRODUCTION

Most power plants built in the United States since the mid-1980's have been combustion turbines, which use either natural gas or fuel oil as the energy source (Brown et al. 1994, 66). Simple cycle combustion turbines are typically used to produce power during periods of peak electricity demand, since they are more expensive to operate than steam or hydro-driven turbines. These periods of peak electricity demand often occur during hot summer months due to high air conditioning loads. Unfortunately, both the generating capacity and conversion efficiency of combustion turbines decrease with increasing ambient dry bulb temperature.

An approach to solving the problems associated with operating combustion turbines in hot weather that has received attention in recent years involves using ice as a thermal storage medium for cooling the inlet air stream. Ice is generated and stored during off-peak hours, and then melted by circulating water which is exposed directly or indirectly to the combustion turbine inlet air. The effectiveness of this new technology was first demonstrated in 1991 at the Rokeby Power Station operated by the Lincoln Electric System in Lincoln, Nebraska. A second facility was retrofitted with an ice-based inlet air cooling system in 1993 for the city of Fayetteville, North Carolina (Ebeling et al. 1994).

Refrigeration equipment used to produce ice is expensive compared to that required to produce chilled water. A promising alternative to ice storage would hence appear to be stratified chilled water storage. The main drawback to using chilled water as the storage medium for combustion turbine inlet air cooling is that it cannot be used to cool the air stream to as low a temperature as can ice. For design ambient dry bulb temperatures of 90° to 100° F, systems based on ice storage can lower the inlet air temperature to roughly 40° F, while systems based on chilled water storage can only reach air temperatures of about 46° F. Combustion turbine performance improves as the inlet dry bulb temperature decreases to some minimum value, which varies between 40° F and 60° F for different combustion turbine types. Temperatures lower than the minimum value can lead to condensation and icing at the inlet, which can damage the combustion turbine (Andrepont 1994).

The results of the research described in this thesis include a detailed comparison of combustion turbine inlet air cooling systems based on ice, water, and a hybrid combination of the two storage media. This introductory chapter reviews the fundamentals of combustion turbine operation, describes other work related to combustion turbine inlet air cooling, and outlines the scope of the present study.

1.1 Fundamentals of Combustion Turbine Operation

A simple combustion turbine power plant consists of the four components shown in Figure 1.1.1: a compressor, combustion chamber, expansion section, and electric generator.

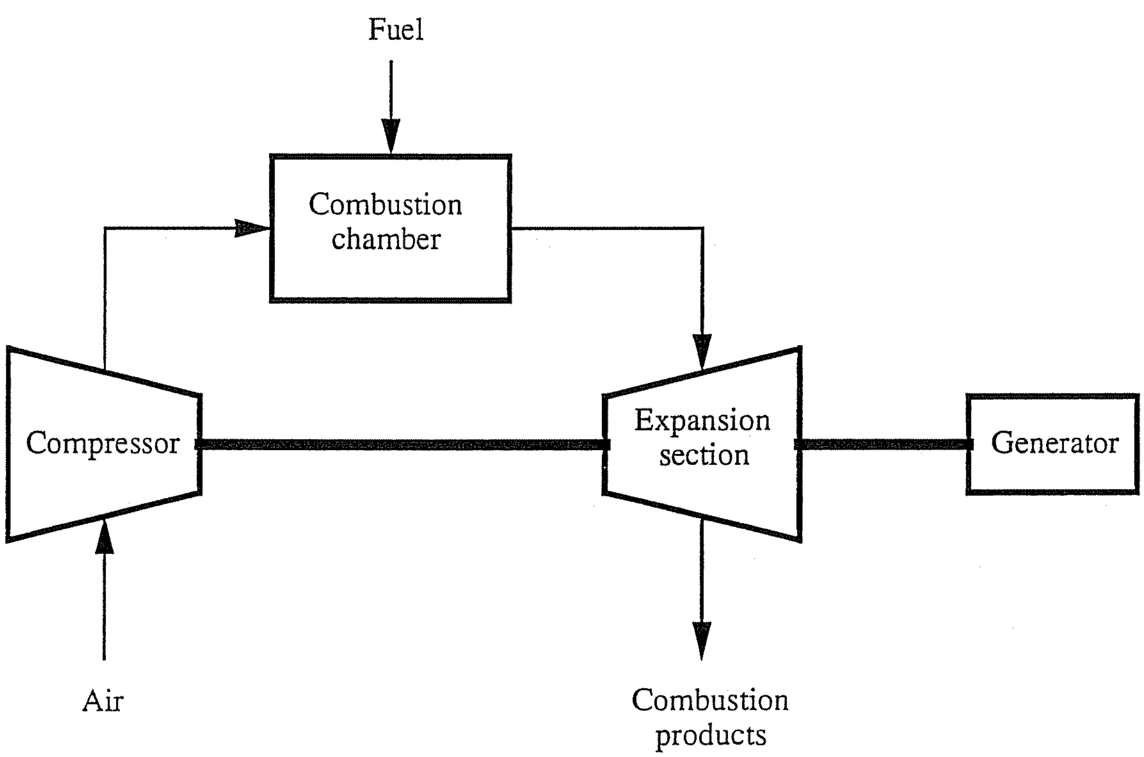


Figure 1.1.1: Simple Combustion Turbine Diagram

Air is drawn continuously into the compressor, and its pressure is increased substantially. The air then enters the combustion chamber, where fuel is added and combustion occurs. The hot gases then enter the expansion section of the power plant, where they cause the turbine to rotate at fixed speed before being discharged to the surroundings. The rotating shaft is connected to the compressor and the generator, which produces electricity (Moran and Shapiro 1992, 373).

The power developed at the shaft is proportional to the mass flow rate of combustion gases through the expansion section of the turbine. Single shaft, heavy duty combustion turbines are characterized by very nearly constant volumetric flow rates at each point in the cycle over a wide range of inlet temperatures. The volumetric flow rates for multiple shaft aircraft derivative turbines, on the other hand, decrease as the inlet temperature increases. The mass flow rate can be increased for both turbine types by decreasing the dry bulb temperature at the compressor inlet, which causes the air density to increase. This increase in mass flow rate is the basis of the generating capacity increase brought about by air inlet cooling. Furthermore, an increase in mass flow rate leads to an increase in the pressure difference between the inlet and outlet of the expansion section. This increase in pressure ratio leads to an increase in cycle efficiency (Kitchen 1994). The overall conversion efficiency, which includes the energy requirement of the refrigeration equipment, is usually higher for a combustion turbine power plant with inlet air cooling than for one without this feature. An increase in conversion efficiency is possible because inlet air cooling is a cycle improvement, somewhat akin to installing a more efficient compressor.

Combustion turbine power plants installed to meet summer peak loads are typically designed to operate at ambient dry bulb temperatures of 90° - 100° F. By cooling the inlet air stream to 40° F, the generating capacity can be increased by 20% to 30%, and the cycle efficiency can be increased by up to 5% (Andrepont 1994, Ebeling et al. 1994).

A second means of increasing power plant capacity by increasing the mass flow rate is to inject water into the combustion chamber along with the fuel. Water injection also decreases the firing temperature (the temperature of the hot gases in the combustion chamber) and thus helps limit the production of oxides of nitrogen (NO_X). However, water injection has the disadvantage of decreasing cycle efficiency.

This project is based on the performance of a single shaft turbine operated by a utility in the upper mid-Western United States. The turbine has a capacity at International Standards Organization (ISO) inlet air conditions (59° F, 14.7 psia, 60% relative humidity) of roughly 86 MW, and its capacity increases nearly linearly as the inlet dry bulb temperature decreases. The minimum air inlet temperature is 40° F. The turbine relies on water injection to control NO_X production, and may be operated in the "power augmentation mode" by further increasing the water mass flow rate. Many of the specific conclusions reached by this study will not apply to other combustion turbines having operating characteristics that differ significantly from the one modeled.

1.2 Literature Review

J. Ebeling (1994) wrote that utilities have historically sought to maintain combustion turbine capacity in hot weather by installing evaporative coolers or "on-line chillers" to cool the inlet air stream. These solutions have not been entirely satisfactory, however. Evaporative coolers are incapable of cooling inlet air significantly since they are limited to a finite approach to the ambient wet bulb temperature, and thus function particularly poorly in humid areas. On-line chillers produce cold water by means of either a mechanical compressor or absorption cycle only while the combustion turbine is operating. Although on-line chillers can lower the air temperature to about 46° F, their installed cost of $400-600 per additional kilowatt of generating capacity is nearly as high as the unit cost of installing another combustion turbine. On-line chilled water systems based on a mechanical vapor compression cycle also create significant parasitic power requirements.

By using a thermal storage medium such as ice or chilled water, the size of the refrigeration equipment and the on-peak parasitic power requirement can both be decreased significantly. This is because the ice or chilled water can be produced overnight with off-peak electricity. The on-peak parasitic power requirement can thus be limited to that required to operate the cooling coil water pumps. The Burns and McDonnell Engineering Company designed the ice-based inlet air cooling system installed at the Rokeby Power Station, which was the "brainchild" of Lincoln Electric System personnel. Burns and McDonnell also designed the system installed at Fayetteville's Butler-Warner Generating Plant. These projects are described in detail by R. Balsbaugh (1994) and J. Ebeling and his co-workers (1994). A third ice-based inlet cooling system for a California cogeneration plant is described by A. Hall and his co-workers (1994).

None of the authors listed in the previous paragraph presented a careful examination of the possibility of using chilled water as the storage medium. Ebeling (1994) simply stated that the cost of a chilled water based inlet air cooling system would be greater than that for an ice-based system "designed to similar parameters". Hall pointed out that ice storage requires less space than chilled water storage. The additional space requirement seems to have been the reason that water was rejected as a thermal storage medium for the facility he described.

J. Andrepont (1994) found that the additional capacity achievable with an ice-based cooling system did not justify its added expense. In an analysis for a site with six combustion turbines, he estimated that the unit costs for ice-based and chilled water-based systems would be $302/kW and $239/kW, respectively. These designs were based on daily combustion turbine operation of six hours, and fuel oil tanks were available for conversion to thermal storage tanks. The "incremental cost" of ice vs. chilled water storage (for the slight increase in capacity associated with ice storage) was $750 - $800 per kilowatt. Andrepont thus concluded that ice-based systems are not economically competitive with water-based systems.

Andrepont further pointed out that the optimal design cycle for a system based on ice storage is one week, while the optimal design cycle for a system based on chilled water storage is one day. The size of the ice harvester can be decreased significantly if it can be operated over the weekend while the combustion turbine is not in use, but no such size reduction is possible if chilled water is the storage medium. However, the ice storage tank size must be larger in a system that operates over the weekend than in a system that only operates on week days. Thus Andrepont claimed that the required storage tank size for an optimally designed ice storage-based system is nearly identical to that for an optimally designed chilled water storage-based system.

1.3 Scope of Study

D. Knebel of the Thermal Storage Applications Research Center suggested that the use of both ice and chilled water as storage media would likely result in the lowest unit costs for combustion turbine inlet air cooling systems. He reasoned that such designs could exploit both the relatively low costs associated with the refrigeration equipment used to chill water and the lower air stream temperatures (and thus higher power outputs) associated with ice storage. That suggestion provided the starting point for this study.

In order to test Knebel's hypothesis, a computer model of a combustion turbine inlet cooling system based on both ice and chilled water thermal storage was written using the TRNSYS simulation program (Klein et al. 1994). That system, which includes an optional evaporative cooling unit, is diagrammed in Figure 1.3.1 on the following page. Its operation is explained in detail in the first section of Chapter 4. By making minor program changes, the model can also be used to simulate the performance of systems based on either water or

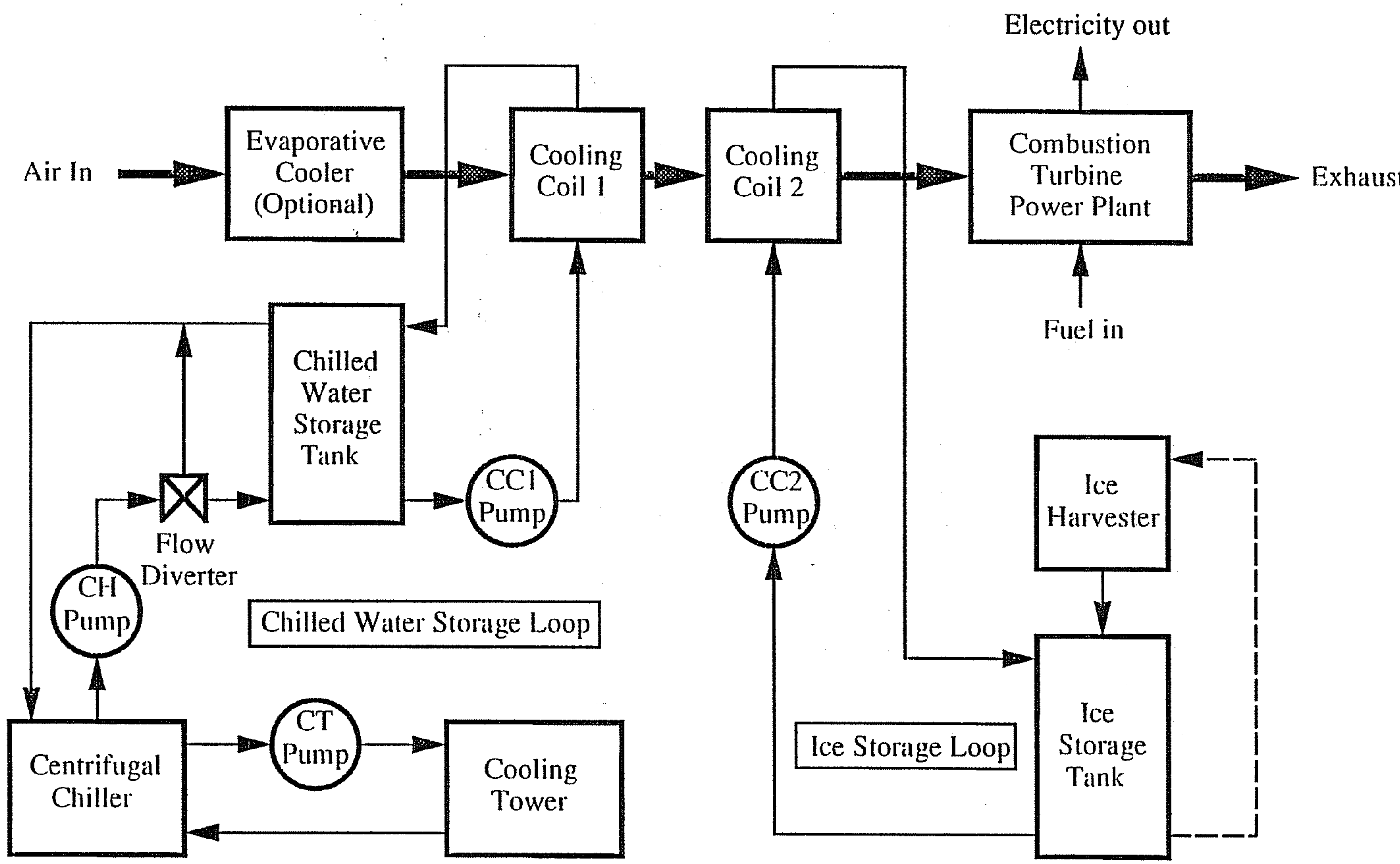


Figure 1.3.1: Combustion Turbine Inlet Air Cooling System Diagram

ice storage alone. Four additional models of system sections, or "loops", were written using a simultaneous equation solver, EES (Klein and Alvarado 1993). These section models describe the performance of the chilled water storage loop, the ice storage loop, and cooling systems based on either of these storage media separately.

The goals of this study were threefold: to determine the optimum "capacity split" between chilled water and ice storage that would result in the maximum power plant capacity enhancement for the lowest inlet air cooling system first costs; to compare first costs for systems based on chilled water, ice, and a combination of these two storage media for several different combustion turbine load profiles; and to compare all systems on the basis of the anticipated life cycle benefit to the utility. Performance curves and test data were obtained for the combustion turbine described in section 1.2 in order to model typical power plant behavior. Inlet air cooling systems were designed using the appropriate EES and TRNSYS programs interactively on the basis of a "design week" consisting of seven "design days" having the same dry and wet bulb temperature profiles. Annual cooling season simulations were performed in order to carry out a life cycle analysis based on a determined sequence of design days.

Chapters 2 and 3 describe individual component models used in the overall combustion turbine inlet air cooling system model. The system model is elaborated upon in Chapter 4. Chapter 5 describes the process of cooling system design and the results of that effort, while Chapter 6 explains the life cycle analysis used to evaluate the 24 cooling systems designed. Finally, conclusions and recommendations for further research are presented in Chapter 7.

CHAPTER 2: COMBUSTION TURBINE MODEL

Two different models were developed to describe the performance of the combustion turbine: an EES program and a TRNSYS subroutine. Although the two models are based on the same key set of equations, they played different roles in the course of this project. The EES model was written first in order to study the behavior of the combustion turbine apart from the inlet air cooling system. During the cooling system design process, the EES model was used to provide several parameters for the TRNSYS inlet air cooling system simulation model. The TRNSYS subroutine was incorporated into that simulation model. Its outputs include the dry air mass flow rate, the electric power output, and the fuel mass flow rate. The TRNSYS subroutine is capable of comparing combustion turbine performance with and without the inlet air cooling system in a single simulation run.

This chapter describes the bases on which the two models calculate the dry air mass flow rate, the electric power output, and the fuel mass flow rate. Since the air volumetric flow rate at the combustion turbine inlet is assumed to be constant (as long as the rotational speed is constant), determination of the dry air mass flow rate is a relatively straightforward matter. Two sets of test data were used to determine the air volumetric flow rate at the turbine inlet. Calculations of the electric power output and fuel mass flow rate are based on performance curves and two additional sets of test data for the power plant considered. The performance curves give the dependence of the dimensionless combustion turbine capacity and dimensionless conversion efficiency on five variables: air inlet dry bulb temperature, part load factor, inlet pressure drop, exhaust pressure drop, and water injection flow rate. These variables are assumed to operate independently of each other unless otherwise noted. The test data include all relevant flow rates, temperatures, pressure drops, and power outputs for two different operating conditions: "base mode" and "power augmentation mode".

2.1 Determination of Dry Air Mass Flow Rate

It is important that the TRNSYS combustion turbine model calculate the dry air mass flow rate because this variable has a significant influence on the cooling system load at each simulation time step. The constant inlet air volumetric flow rate was first determined from turbine test data. The instantaneous dry air mass flow rate depends on this parameter, the inlet pressure, inlet temperature, and inlet humidity ratio.

The two sets of test data used to determine the volumetric flow rate at the turbine inlet consist of the dry bulb temperature, ambient pressure, relative humidity, fuel mass flow rate, injected water mass flow rate, and exhaust mass flow rate. The dry air mass flow rate, specific volume, and inlet volumetric flow rate were calculated based on this information. Averaging the results of both data sets gave an inlet volumetric flow rate of 520,270 cfm. For details of this calculation, refer to EES program AVFR.2 in Appendix A. Both combustion turbine models treat the air and water vapor at the inlet as ideal gases to calculate the specific volume of the moist air as a function of inlet temperature and pressure. The instantaneous dry air mass flow rate can then be determined from this information together with the inlet volumetric flow rate. For details of this calculation, refer to the TRNSYS power plant model in Appendix C.

2.2 Effect of Inlet Air Temperature and Part Load Factor on Capacity and Efficiency

As discussed in Chapter 1, both the combustion turbine power plant capacity and conversion efficiency increase as the air inlet temperature decreases. The efficiency also increases as part load factor increases. The part load factor is defined as the actual power output divided by the capacity, or maximum power output, for a given set of operating conditions. Maximum power output occurs at the maximum permissible firing temperature, which is set by material limitations. The firing temperature can be decreased by lowering the fuel mass flow rate, which in turn decreases the output of the power plant.

For the combustion turbine studied, the dependence of the relative power output, "RPO", on air inlet temperature is shown in Figure 2.2.1 for a firing temperature of 1,880° F. The power plant capacity is proportional to the relative power output. The statistical analysis program Minitab (Ryan et al. 1985) was used to find the curve fit parameters in the following quadratic equation:

$$RPO = 1.158 - 2.477e\text{-}3*EDB - 3.73e\text{-}6*EDB^2 \quad (2.2.1)$$

where "EDB" is the entering dry bulb temperature at the compressor stage.

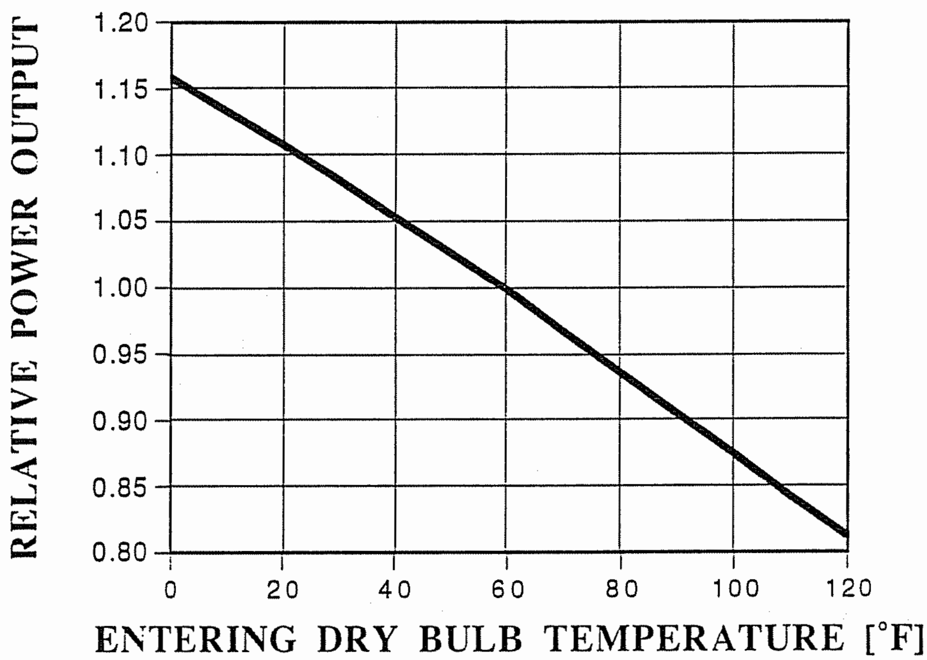


Figure 2.2.1: Relative Power Output vs. Inlet Dry Bulb Temperature

The relative efficiency of the combustion turbine is given as a set of curves (Figure 2.2.2) that depend on both the inlet air dry bulb temperature and part load factor. Minitab was used to find the following eight parameter equation to represent these curves:

$$\begin{aligned} n_{rel} = {} & 0.1777 + 2.340*PLF - 9.764e\text{-}4*EDB + 8.181e\text{-}4*PLF*EDB \\ & - 2.401*PLF^2 - 1.82e\text{-}6*EDB^2 - 1.95e\text{-}6*PLF*EDB^2 \\ & + 0.904*PLF^3 \end{aligned} \quad (2.2.2)$$

where "n_{rel}" is the relative efficiency and "PLF" is the part load factor. The conversion efficiency of the combustion turbine is proportional to the relative efficiency.

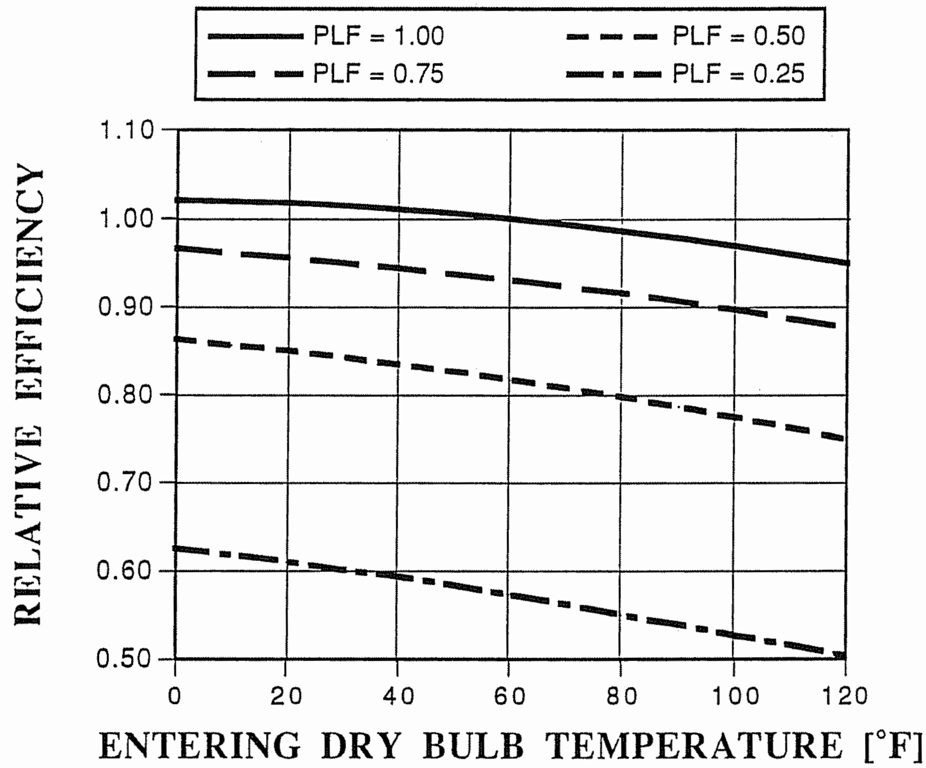


Figure 2.2.2: Dependence of Relative Efficiency on Inlet Dry Bulb Temperature and Part Load Factor

The instantaneous fuel mass flow rate depends on the instantaneous power output and the conversion efficiency. The following section describes additional factors that affect both turbine capacity and conversion efficiency.

2.3 Effect of Other Variables on Capacity and Efficiency

Three other variables affect combustion turbine capacity and conversion efficiency (and hence fuel mass flow rate): the inlet pressure drop, exhaust pressure drop, and the injected water mass flow rate. Evaporative coolers, cooling coils, and similar components create inlet pressure losses, which decrease both turbine capacity and conversion efficiency. Pressure losses in the exhaust system also decrease turbine capacity and conversion efficiency. As discussed in section 1.1, water can be added to the combustion chamber of a combustion turbine power plant for two reasons: to decrease the firing temperature and hence to decrease the production of oxides of nitrogen, and to increase the mass flow rate through the expansion section and hence increase plant capacity. However, water injection leads to a decrease in conversion efficiency

The dependence of the relative power output and relative efficiency on the inlet system pressure loss, "dPin", are shown in Figure 2.3.1. Both relationships are linear, and are described by Equations 2.3.1 and 2.3.2 below:

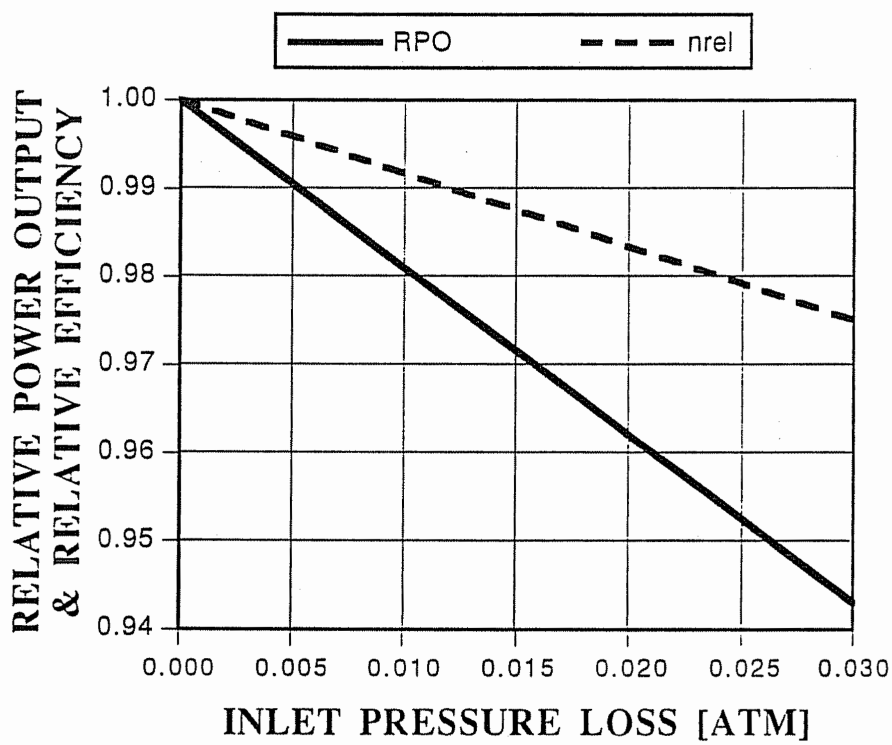


Figure 2.3.1: Dependence of Relative Power Output and Relative Efficiency on Inlet System Pressure Losses

IPLCM = 1.00 - 1.900*dPin (2.3.1)

IPLEM = 1.00 - 0.848*dPin (2.3.2)

where "IPLCM" is the inlet pressure loss capacity multiplier, "IPLEM" is the inlet pressure loss efficiency multiplier, and "dPin" is measured in atmospheres. The capacity of the combustion turbine is proportional to "IPLCM"; the conversion efficiency is proportional to "IPLEM".

The relative power output and relative efficiency both have the same dependence on the pressure loss in the exhaust system, "dPout", as shown in Figure 2.3.2. This relationship is given by Equation 2.3.3 below:

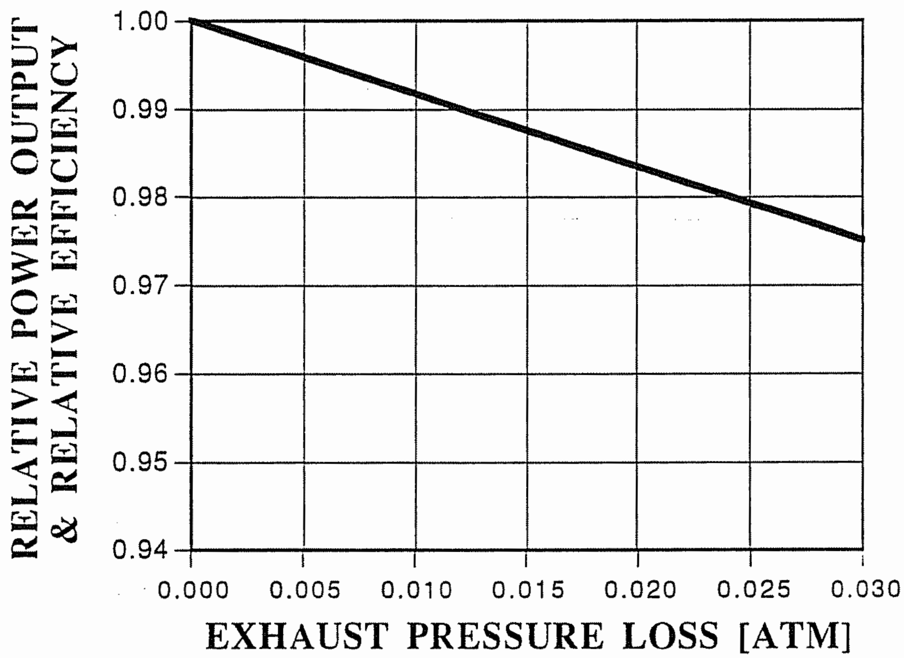


Figure 2.3.2: Dependence of Relative Power Output and Relative Efficiency on Exhaust System Pressure Losses

$$OPLCEM = 1.00 - 0.848*dPout \quad (2.3.3)$$

where "OPLCEM" is the outlet pressure loss capacity and efficiency multiplier, and "dPout" is again measured in atmospheres. Both the combustion turbine capacity and conversion efficiency are proportional to "OPLCEM".

Finally, the relative power output and relative efficiency both depend on the ratio of the water and fuel mass flow rates, as shown in Figure 2.3.3. These relationships differ

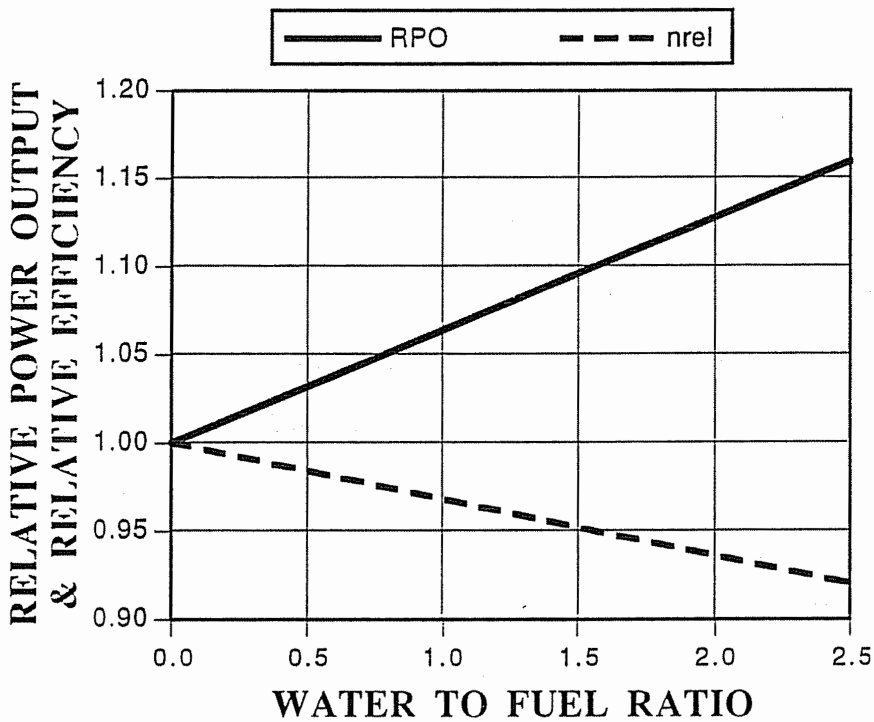


Figure 2.3.3: Dependence of Relative Power Output and Relative Efficiency on the Water-Fuel Mass Flow Ratio

slightly for different types of fuel; those shown in Figure 2.3.3 are for natural gas. Again, the relationships are linear:

$$WFRCM = 1.00 + 0.0642*WFR \quad (2.3.4)$$

$$WFREM = 1.00 - 0.0321*WFR \quad (2.3.5)$$

where "WFRCM" is the water-fuel ratio capacity multiplier, "WFREM" is the water-fuel ratio efficiency multiplier, and "WFR" is the ratio of the water and fuel mass flow rates. The capacity of the combustion turbine is proportional to "WFRCM"; the conversion efficiency is proportional to "WFREM". When the water to fuel ratio exceeds that necessary to control the production of oxides of nitrogen (about 1.8) and the firing temperature has attained its maximum value, the combustion turbine is said to operate in the "power augmentation mode". It is otherwise said to operate in the "base mode". The injection of water into the combustion chamber increases power output at the cost of decreasing the conversion efficiency.

2.4 Determination of Capacity and Fuel Mass Flow Rate

The base capacity, "BEP", is defined as the maximum power output of the combustion turbine for an inlet air temperature of 59° F, an ambient pressure of 14.2 psia, with all pressure losses and the water injection flow rate equal to zero. This ambient pressure was reported along with all other combustion turbine test data. The actual capacity, "MEP", is related to the base capacity and quantities defined in sections 2.1.2 and 2.1.3 by :

$$MEP = IPLCM*OPLCEM*WFRCM*RPO*BEP. \qquad (2.3.6)$$

The base efficiency, "n_{base}", is defined in a similar manner as the efficiency of the combustion turbine for an inlet air temperature of 59° F, at an ambient pressure of 14.2 psia, with no pressure losses and no water injection. The actual conversion efficiency, "n_{HHV}", is related to the base efficiency and quantities defined above by:

$$n_{HHV} = IPLEM*OPLCEM*WFREM*n_{rel}*n_{base} \qquad (2.3.7)$$

The conversion efficiency is based on the higher heating value of the fuel used. Thus the fuel mass flow rate, "FMFR", is given by:

$$FMFR = (EP*3412)/(n_{HHV}*HHV) \qquad (2.3.8)$$

Here "EP" is the electric power output in kilowatts and "HHV" is the higher heating value of the fuel in BTU/lb. The fuel mass flow rate is thus given in lb/hr.

Two detailed sets of test data were used to determine the base capacity and base efficiency. The first data set is for the combustion turbine operating in base mode; the second is for the combustion turbine operating in the power augmentation mode. The base capacity was found to equal approximately 80.5 MW; the base efficiency was found to equal roughly 0.290. For details of these calculations, refer to EES program BEP.1 in Appendix A.

As noted in the introduction to this chapter, the TRNSYS model can be used to compare combustion turbine performance with and without inlet air cooling simultaneously. A normalized desired power output must be provided at each simulation time step. The model first determines whether the desired power output can be met with and without inlet air cooling. If the desired output cannot be met, the actual output is set equal to the combustion turbine's capacity for the given inlet condition. The model then calculates the instantaneous fuel mass flow rate necessary to meet the desired (or maximum) power output both with and without inlet cooling. The TRNSYS model appears in Appendix C.

In order to calculate the cost of capacity enhancement due to the inlet cooling system in dollars per kilowatt, the TRNSYS model requires the maximum net power plant output both with and without inlet cooling at design conditions. The maximum net power plant output is defined as the combustion turbine capacity minus the cooling coil pump power requirement. The EES model was used during the system design process to calculate the gross and net power plant electric outputs. That model, BBPPmod.5, appears in Appendix A.

CHAPTER 3: COOLING SYSTEM COMPONENTS

Eight new TRNSYS components were written in order to simulate the performance of the proposed combustion turbine inlet air cooling systems. These component models include an ice harvester, a centrifugal chiller, an ice storage tank, a controller for the cooling coil water pumps, three specialized deadband controllers, and an evaporative cooler. Other standard components from the TRNSYS 14 Library used in the system simulations include a cooling tower, pumps, pipes, a plug flow chilled water storage tank, and a slightly modified cooling coil. A "cost calculator" component was also written to determine first costs and the cost of the electric power produced with inlet air cooling based on a specified discounted payback period.

The process of writing a new component typically involved two key steps. First, an EES program was written in order to check the independent behavior of the model. Second, the model was re-written in FORTRAN as a TRNSYS component. In most cases, only minor changes were required in order to effect this transformation. However, in the case of the ice harvester, the EES model differs significantly from the TRNSYS model. The EES ice harvester is a detailed model used to generate performance curves for the TRNSYS ice harvester. A detailed FORTRAN ice harvester model simply caused the system simulation to run too slowly on the computer used for system design.

3.1 Ice Harvester Models

The ice harvester uses a rotary screw compressor with ammonia as the refrigerant. The evaporator consists of large vertically oriented plates through which a liquid-vapor mixture of ammonia is circulated by a secondary pump. The plates are divided into several sections which are defrosted sequentially. Ice is generated by pumping a thin film of water over the outer surface of the evaporator plates. While ice is being formed, ammonia vapor discharged from the compressor is passed through an evaporative condenser unit. After the ice has built up to a thickness of roughly 3/8", the hot ammonia vapor is temporarily routed through one section of evaporator plates to remove the ice. The time required to build a 3/8" thick sheet of ice is on the order of half an hour, while approximately 50 seconds are required to defrost each section of the evaporator. There is no net formation of ice while any section of evaporator plates is in the defrost mode. The ice sheets fall directly into a storage tank located below the evaporator plates. as illustrated in Figure 3.1.1 (Knebel 1994a, Knebel 1991).

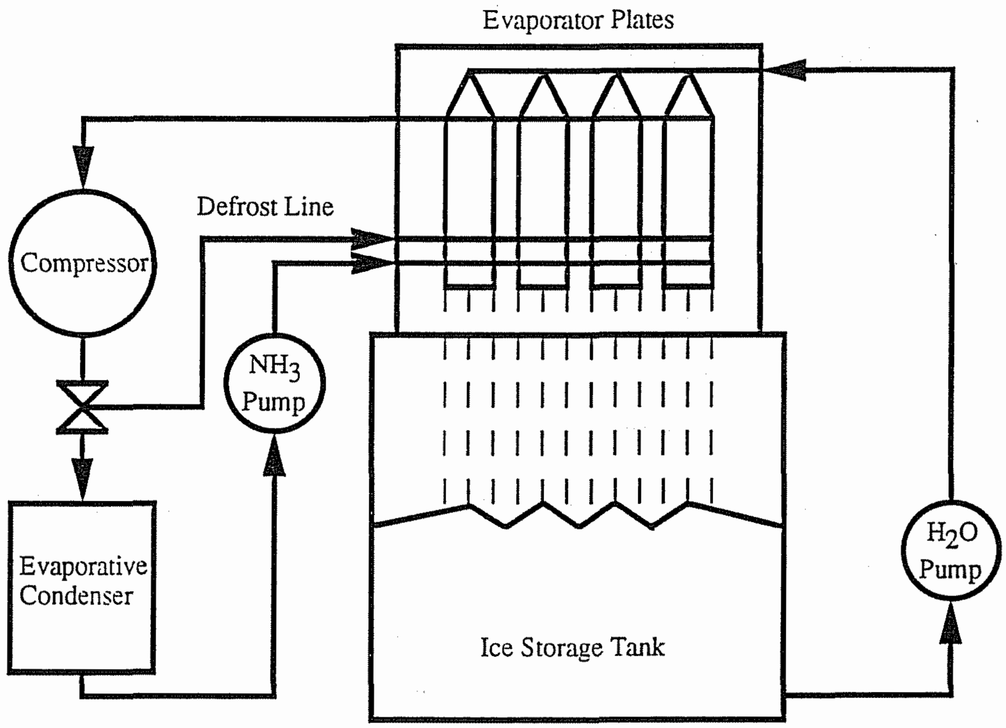


Figure 3.1.1: Ice Harvester and Storage Tank

The TRNSYS ice harvester model determines the net capacity, ice generation rate, and net power requirement of the ice harvester at each simulation time step based on the ambient wet bulb temperature and a control variable. The ice generation rate is approximated as the net capacity divided by the latent heat of fusion of water. The net capacity and net power requirement differ from the nominal values for the compressor for two reasons. First, the ability of the evaporative condensers to reject heat depends on the wet bulb temperature, and second, compressor capacity is required to defrost each group of evaporator plates. The difference between the net capacity and the nominal capacity ranges between 9% and 10% for the ice harvester modeled.

The detailed EES ice harvester model is based on performance maps for the Frick RWB-II 60E and RWB-II 177E rotary screw compressors operating with a flash economizer (Frick 1991). These maps give the refrigeration capacity in tons, "RefCap", and brake horsepower, "RefPow", as functions of the saturated condensing temperature and the saturated suction temperature. Minitab was used to find regression equations for "RefCap" and "RefPow" in the following form:

$$\text{RefCap} = C1 + C2*SST + C3*SST^2 + C4*SDT + C5*SDT^2 + C6*SST*SDT \qquad (3.1.1)$$

$$\text{RefPow} = P1 + P2*SST + P3*SST^2 + P4*SDT + P5*SDT^2 + P6*SST*SDT \qquad (3.1.2)$$

where "SST" is the saturated suction temperature of the compressor and "SDT" is the saturated discharge temperature, both in degrees Fahrenheit. For a saturated suction temperature of 20° F and a saturated discharge temperature of 95° F, the RWB-II 60E compressor has a nominal capacity of 135 tons; the RWB-II 177E has a nominal capacity of 410 tons. The equations for the capacity and brake horsepower derived for the RWB-II 60E are scaled to model the performance of compressors having a nominal capacity less than 250 tons, while equations derived for the RWB-II 177E are scaled to model the performance of compressors nominally rated at 400 tons and above.

The amount of heat rejected by the evaporative condenser is equal to the nominal rating of the unit divided by the heat rejection correction factor, "HRCF". This dimensionless quantity is a function of the ambient wet bulb temperature and the saturated condensing temperature. Minitab was used to find an equation for the heat rejection correction factor as a function of these two variables based on data for evaporative condensers with ammonia as the condensing fluid (Imeco n.d.):

$$\begin{aligned} HRCF = {} & 2.271 - 2.212e\text{-}2*SCT + 4.671e\text{-}5*AWB^2 \\ & - 8.043e\text{-}7*SCT*AWB^3 + 5.617e\text{-}9*SCT^2*AWB^3 \\ & + 3.742e\text{-}9*AWB^5 - 5.494e\text{-}9*SCT*AWB^4. \end{aligned} \tag{3.1.3}$$

Here "AWB" is the ambient wet bulb temperature and "SCT" is the saturated condensing temperature, both in degrees Fahrenheit.

By performing system energy balances for both the build and defrost modes, the net capacity and net power requirement of the ice harvester can be calculated. The model accounts for the relatively small refrigerant pump power requirement. The detailed EES model for an ice harvester based on the RWB-II 60E compressor appears in Appendix A. The EES models were used to create parametric tables for the net capacity and net power requirement as functions of the design wet bulb temperature, the ambient wet bulb temperature, and the nominal compressor capacity. The table for the RWB-II 60E compressor is included with the EES model. Finally, Minitab was used to generate equations describing the ice harvester's performance in terms of these three variables of the form:

$$Ncap = A1 + A2*NomCap + A3*NomCap*AWB + A4*NomCap*DWB \tag{3.1.4}$$

$$\begin{aligned} Npower = {} & B1 + B2*NomCap + B3*NomCap*AWB \\ & + B4*NomCap*DWB + B5*DWB^2 \end{aligned} \tag{3.1.5}$$

where "Ncap" is the net ice harvester capacity in tons, "NomCap" is the nominal compressor capacity in tons, "NPower" is the net ice harvester power requirement in kilowatts, and "DWB" is the design wet bulb temperature in degrees Fahrenheit. The TRNSYS model is based on Equations 3.1.4 and 3.1.5, and is shown in Appendix C.

3.2 Centrifugal Chiller Model

Although a TRNSYS chiller model has been in existence for a number of years, a new chiller model was written for use in this project. The established TRNSYS model requires an external performance data file, and one of the parameters required (the ratio of the temperature difference between the condenser water outlet and the evaporator water outlet relative to a design temperature difference) is not readily available from chiller manufacturers (Klein et al. 1994, 4.6.9-1 - 5). The centrifugal chiller model written for this project does not require an external data file. It is based on a five parameter equation relating the dimensionless power requirement to the dimensionless load and deviations from the design entering condenser and leaving evaporator water temperatures. That equation, which is used by the Trane Company in its simulation program "TRACE", is

$$P_{req}/P_{des} = [0.140 + 0.544*(Q_{load}/Q_{des}) + 0.316*(Q_{load}/Q_{des})^2\,]$$
$$*[\,1 + 0.012*(ECWT - DECWT) - 0.015*(LEWT - DLEWT)] \tag{3.2.1}$$

where "P_{req}" and "P_{des}" are the actual and design power requirements, "Q_{load}" and "Q_{des}" are the actual and design loads, "ECWT" and "DECWT" are the actual and design entering condenser water temperatures, and "LEWT" and "DLEWT" are the actual and design leaving evaporator water temperatures (Pawelski 1994) All temperatures are in degrees Fahrenheit. The design entering condenser water temperature and the design leaving evaporator water temperature are 85° F and 44° F, respectively.

In order to determine the relationship between the design load and the design power requirement, a simple ammonia-based refrigeration cycle model was written using EES. For a saturated condensing temperature of 35° F and a saturated evaporating temperature of 90° F, the coefficient of performance is 5.29. The refrigeration cycle model is embedded in the

EES chilled water storage loop model, which appears in Appendix B. The TRNSYS chiller model appears in Appendix C.

3.3 Ice Storage Tank Model

The ice sheets generated by the ice harvester fall and break into pieces in the ice storage tank. When the combustion turbine requires inlet cooling, water is sprayed over the top of the ice and drawn out of the bottom of the tank before being circulated through a cooling coil which is exposed to the turbine inlet air flow stream. If the instantaneous ice inventory exceeds 20% of the total storage capacity, the temperature of the water leaving the tank is 32° F. As the ice inventory drops below 20% of total storage capacity, the leaving water temperature approaches the entering water temperature (Stewart 1994).

A simple effectiveness model is used to describe the performance of the ice storage tank. The effectiveness depends on the "discharge fraction", or fraction of the tank's storage capacity that has been melted or "burned". For discharge fractions of less than 0.80, the effectiveness is unity. For discharge fractions between 0.80 and 1.00, the effectiveness is assumed to drop linearly from one to zero. The heat transfer rate in BTU's per hour to the circulating stream of water is given by:

$$q_{water} = eff*WMFR*C_{pw}*(EWT - 32) \tag{3.3.1}$$

where "eff" the instantaneous tank effectiveness, "WMFR" is the water mass flow rate in lb/hr, "C_{pw}" is the heat capacity of water at constant pressure in BTU/lb-°F, and "EWT" is the entering water temperature in degrees Fahrenheit.

The ice storage tank model uses an overall heat transfer loss coefficient to calculate environmental losses in addition to calculating losses to the circulating water stream. The remaining ice inventory is determined at the end of each time step. This value is then used as the mass of ~~ice~~ present at the beginning of the following time step. The TRNSYS ice storage tank model is included in Appendix C.

3.4 Controller Models

As shown in Figure 1.3.1, two cooling coils are present in the complete combustion turbine inlet air cooling system model: the first fed by the stratified chilled water storage tank, the second fed by water circulating through the ice storage tank. The water pumps associated with these coils must be activated when the combustion turbine is incapable of generating the desired electric power output with air at the ambient dry bulb temperature. Once these pumps have been activated, the water mass flow rate through each coil must be adjusted to give the desired power plant electric output at each simulation time step. The cooling coil pump controller sets the control variables for these pumps, "CV1" and "CV2", based on the desired electric power output using an iterative procedure at each simulation time step.

Each cooling coil pump control variable is a number between zero and unity. The instantaneous water mass flow rate is the product of this control variable and the maximum water mass flow rate through the pump. For a given ambient dry bulb temperature and desired power output a relationship between the sum of the control variables, "ΣCV", and the difference between the desired and actual power plant output, "ΔEP", exists as shown schematically in Figure 3.4.1. The controller subroutine finds the intersection between the curve and the abscissa, that is, the value of the sum of cooling coil pump control variables that makes the actual electric power output equal to the desired electric power output. Note that for values of the abscissa less than unity, the second control variable is zero, while for values of the abscissa greater than or equal to unity, the first control variable is equal to one.

Since the equation of the curve in Figure 3.4.1 is unknown, an iterative method must be used to find the value of "ΣCV" for which the difference between the desired and actual electric power outputs vanishes. The regula-falsi method works well for this calculation (*International Dictionary of Applied Mathematics* 1960, 761). This procedure involves determining the coordinates of points A, B and the point at which the chord between them intersects the abscissa. This point of intersection is taken as the first approximation to "ΣCV". The point on the curve corresponding to this approximate value, C, is used as one endpoint of the next chord. The intersection between chord CB and the abscissa is taken as the second approximation to "ΣCV". This process is repeated within each simulation time step until the intersection of the curve and the abscissa is determined to a sufficient degree of accuracy. The TRNSYS model of the cooling coil pump control variable appears in Appendix C.

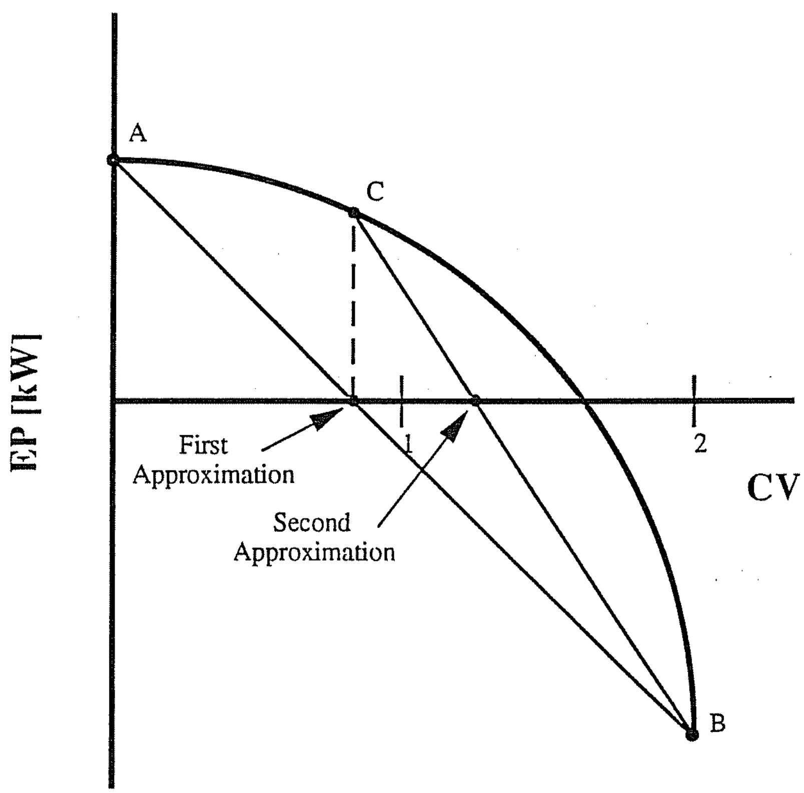


Figure 3.4.1 Illustration of Regula-Falsi Procedure

An on/off differential controller was written to ensure that the ice harvester would never fill the ice storage tank beyond its capacity. A second on/off differential controller was written to ensure that the chiller and cooling tower would not be turned on in the event that the entering water temperature from the storage tank is below a specified value. These differential controllers operate in conjunction with refrigeration equipment schedules as discussed in section 4.1. The third and final differential controller performs a somewhat more sophisticated function. Water from the chilled water storage tank must not enter the chiller evaporator if its temperature exceeds some maximum value in order to avoid overloading the chiller. A fraction of the water leaving the chiller evaporator is thus diverted back to the evaporator inlet and mixed with water from the storage tank if necessary, as shown in Figure 1.3.1. The temperature of the water stream entering the chiller evaporator is thus maintained below a specified maximum value, "MLWT". The diverted fraction of the flow stream leaving the chiller evaporator outlet, "divf", is given by:

$$\text{divf} = (\text{LWT} - \text{MLWT})/(\text{LWT} - \text{SPT}) \qquad (3.4.1)$$

where "LWT" is the storage tank leaving water temperature and "SPT" is the chiller set-point temperature. If "LWT" is less than "MLWT", then "divf" is set equal to zero. A deadband ensures controller stability. The three controller models described in this paragraph appear in Appendix C.

3.5 Evaporative Cooler Model

An evaporative cooler lowers the dry bulb temperature of an entering air stream by adding water to it. The wet bulb temperature stays constant in this process while the psychrometric state approaches the saturation curve as shown in Figure 3.5.1. The entering air state is identified as "E"; the leaving air state is identified as "L". The degree to which the dry bulb temperature approaches the wet bulb temperature is characterized by an effectiveness, "eff". Hence the leaving dry bulb temperature, "LDB", is given by

$$\text{LDB} = \text{EDB} - \text{eff}*(\text{EDB} - \text{EWB}) \qquad (3.5.1)$$

where "EDB" and "EWB" are the entering dry and wet bulb temperatures, respectively (Sauer and Howell 1992, 17.5).

An evaporative cooling unit is currently present in the inlet flow stream of the combustion turbine being considered. Performance measurements of the installed evaporative cooler indicate that its effectiveness is 0.89. In order to model an inlet air cooling system without an evaporative cooler, the effectiveness can simply be set equal to zero. The TRNSYS evaporative cooler model appears in Appendix C.

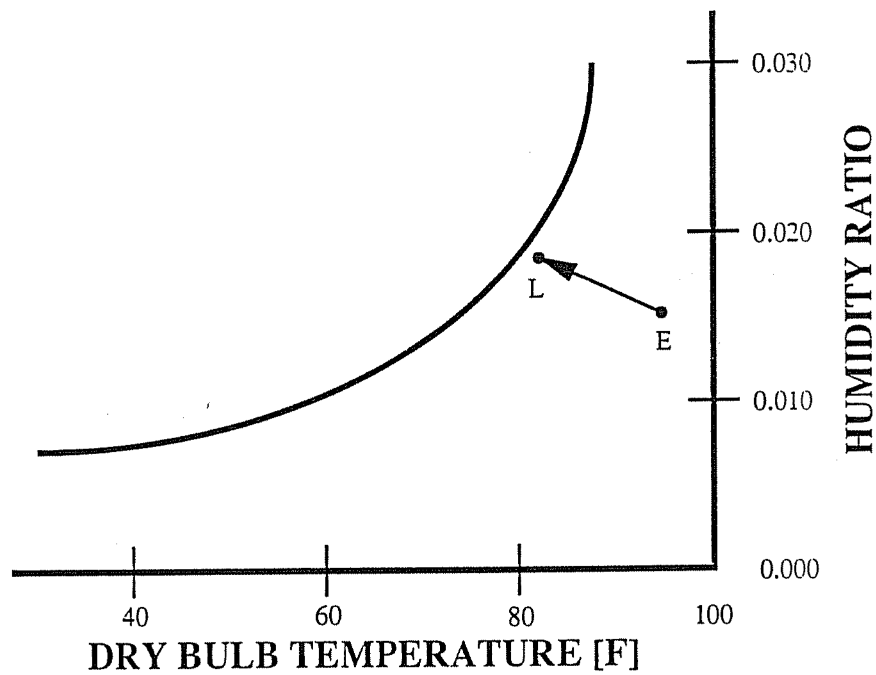


Figure 3.5.1: Psychrometric Diagram for Evaporative Cooling Process

3.6 Other Cooling System Components

Five other TRNSYS components are used in the complete inlet air cooling system model. Three of these, the cooling tower, the pump, and the pipe subroutines, are used in exactly the form in which they appear in the TRNSYS 14 Library. A minor repair was made to the plug flow chilled water tank, and a slight modification was made to the cooling coil subroutine in order to model coils having less than four rows.

The cooling tower model can be operated in two modes: one requires that the user enter coefficients for the mass transfer correlation, and the other requires the user to enter overall tower performance data (Klein et al. 1994, 4.6.7-1 - 8). The cooling tower is operated in the first of these two modes. The model calculates the number of transfer units for the tower heat exchange process using Equation 3.6.1:

$$Ntu = c^*(WMFR/AMFR)^{1+n} \quad (3.6.1)$$

where "WMFR" is the water mass flow rate and "AMFR" is the air mass flow rate. The coefficients for this correlation, "c" and "n", are 2 and -0.63, respectively. (Braun 1988, 68 - 72).

The pump and pipe subroutines are relatively simple models. The mass flow rate through a pump is given by its maximum mass flow rate multiplied by a control variable between zero and unity. The parameter f_{par} is the fraction of pump power that results in a water temperature rise (Klein et al. 1994, 4.5.1-1 - 3). This parameter was set equal to 0.65 for all pumps used in the system model. An overall heat transfer loss coefficient, U, can be set in the pipe model, which also leads to a water temperature rise (Klein et al. 1994, 4.5.4-1 - 3). The overall heat transfer loss coefficient for all pipes was set equal to 0.073 Btu/hr-ft^2-°F.

The plug flow chilled water storage tank model operates on the assumption that a high degree of stratification is present. It can also be operated in two modes: the first has fixed inlet positions, the second has variable inlet positions. The model is operated in mode 1, which corrects temperature inversions by mixing appropriate segments below the inlet position. The temperature distribution at the beginning of the simulation can be specified by making use of the parameters "T_I" and "T_{set}". The first parameter is the initial temperature of the lower portion of the tank; "T_{set}" is the initial temperature of the upper portion of the tank. A third parameter, "H_a", is used to set the initial position of the thermocline, or separation region between the cold lower water layer and warmer upper water layer (Klein et al. 1994, 4.3.3-1 - 6).

In order to avoid convergence problems. the plug flow chilled water tank amalgamates volume elements smaller than 1% of the total tank volume. Unfortunately, this limit leads to serious inaccuracies for small values of "VCOL", the volumetric flow rate relative to the tank volume. A simple TRNSYS model was written to allow changes in the flow rate and tank size to be made quickly. By experimenting with this model, it was possible to study the effect that the lower limit on volume element size has on the plug flow tank model. Setting this quantity equal to 0.25% of the total tank size gives good results for the tank sizes and flow rates considered in this project.

The cooling coil model in the TRNSYS 14 Library is only valid for cooling coils having more than about four rows. This is because the model approximates the multipass cross flow geometry characteristic of standard cooling coils with a counterflow geometry. The model can then apply a combined wet and dry analysis using modified definitions for the number of transfer units and the capacitance rate ratio (Klein et al. 1994, 4.6.8-1 - 8). However, many inlet cooling system designs call for cooling coils with four rows or less.

By defining an effectiveness for each tube pass, it is possible to write an exact expression for the total effectiveness of a multipass overall counterflow geometry with the fluids mixed between passes. Assuming that the water inside the cooling coil tubes is the maximum fluid, the effectiveness for a single tube pass with a dry outer surface is given by :

$$eps_{pass} = 1/Cstar*[1 - e^{-(gamma*Cstar)}] \qquad (3.6.2)$$

where "Cstar" is the ratio of the minimum to the maximum capacitance rate, "gamma" is given by:

$$gamma = 1 - e^{-Ntup} \qquad (3.6.3)$$

and "Ntup" is the number of transfer units per pass. given by the overall number of transfer units divided by the number of tube passes. The total cooling coil effectiveness (still assuming that the exterior surfaces of the tubes are dry) is given by

$$eps = (delta^n - 1)/(delta^n - Cstar) \qquad (3.6.4)$$

where

$$\text{delta} = (1 - \text{eps}_{\text{pass}}*\text{Cstar})/(1 - \text{eps}_{\text{pass}}) \qquad (3.6.5)$$

(Kays and London 1964, 19 - 20).

The total effectiveness of wet tubes can be calculated in a similar manner. The capacitance rate ratio and the number of transfer units per pass are replaced with modified quantities based on the change in enthalpy of saturated air with respect to the change in temperature over the temperature range of interest. The remainder of the modified cooling coil model used in this project is identical to that in the TRNSYS 14 Library. The use of the combined wet and dry analysis mode ensures the maximum level of accuracy afforded by the model.

3.7 Cost Calculator Subroutine

The cost calculator plays a key role in determining the optimum capacity split for combustion turbine inlet cooling systems based on a combination of chilled water and ice storage. It also provides the basis for comparing different system designs both in terms of first costs and life cycle benefit to the utility. Since all economic parameters are passed from the simulation program to the cost calculator as parameters, it is a very flexible component.

The cost calculator computes the installed cost of each component or component grouping (such as water pumps and pipes) used in the inlet cooling system based on thermal or physical size as appropriate. It then determines the cost of the installed system and the capacity enhancement cost, which is simply the system cost divided by the increase in combustion turbine capacity at design conditions due to the inlet air cooling system. Finally, the cost calculator computes the cost of the incremental electric power generated with inlet air cooling for a specified discounted payback period for the cooling system. This last quantity is based on first costs as well as on the discount rate, inflation rate, the increase in power plant output, the increase in power plant fuel consumption, and the increase in cooling system off-peak electricity consumption for the simulation period. The incremental electric power cost provides a measure of the benefit of installing an inlet air cooling system in terms of the resulting increase in power plant output over the system's useful life.

First costs are calculated on the basis of three sources: the 1992 *Means Facilities Cost Data* Catalogue, information from D. Knebel of the Thermal Storage Applications Research Center regarding ice harvester costs, and information from J. Ebeling of the Burns and McDonnell Engineering Company regarding the cost of custom ordered cooling coils for the Butler-Warner combustion turbine power plant operated by the city of Fayetteville, North Carolina. The data analysis and graphics program Kaleidagraph (Abelbeck n.d.) was used to derive curve fit parameters for quadratic or cubic equations relating the free on board (FOB) or installed cost of centrifugal chillers, cooling towers, pre-stressed concrete tanks, and welded steel schedule 40 pipe to the relevant thermal or physical size. Pump, ice harvester, and cooling coil FOB costs are related to size using linear equations. All first costs discussed below (Equations 3.7.1 - 3.7.8) are given in U.S. dollars.

The quadratic equation relating the capacity, "Cap", of a centrifugal chiller in tons to its FOB cost is:

$$Cost_{chiller} = 52{,}933 + 74.65*Cap + 0.04618*Cap^2 \qquad (3.7.1)$$

Equation 3.7.1 is based on data in the *Means Facilities Cost Data* Catalogue for centrifugal chillers ranging in size from 200 to 2,000 tons (Waier et al. 1992, 550). The cost per ton actually increases as the capacity increases beyond about 1000 tons.

The *Means Facilities Cost Data* Catalogue provides the cost per ton for induced air, double flow, gear drive cooling towers for sizes ranging from 125 to 840 tons (Waier et al. 1992, 557). Kaleidagraph yields Equation 3.7.2 for the FOB cooling tower cost as a function of its capacity, "Cap", in tons:

$$Cost_{tower} = 67.71*Cap - 6.134e\text{-}2*Cap^2 + 3.952e\text{-}5*Cap^3 \qquad (3.7.2)$$

In this case, the cost per ton decreases monotonically as the capacity increases.

The cost of pre-stressed concrete tanks used for chilled water and ice storage increases almost linearly with size. The *Means* Catalogue gives installed tank costs (which include labor, overhead, and profit) for sizes ranging from 500,000 to 2,000,000 gallons (Waier et al. 1992, 356). Based on these data, the cost of an installed tank is given by:

$$Cost_{tank} = 156{,}700 + 0.3543*Cap - 2.925e\text{-}8*Cap^2 \qquad (3.7.3)$$

Here "Cap" is measured in gallons.

The *Means* Catalogue provides the FOB cost per foot of welded steel schedule 40 pipe with inner diameters ranging from 4 to 30 inches (Waier et al. 1992, 420 - 21). Kaleidagraph yields the following equation for the FOB cost of a pipe section of length "L" (measured in feet) and inner diameter "D" (measured in inches):

$$Cost_{pipe} = -10.02*L + 3.610*L*D - 7.178e\text{-}4*L*D^2 \qquad (3.7.4)$$

A linear relationship is used to determine the costs of the water pumps, since these costs represent such a small fraction of total system costs. Based on data in the *Means* Catalogue for a 15 horsepower pump with a capacity of 1,000 gallons per minute, the relationship between the FOB pump cost and its capacity, "Cap" (in gallons per minute) is given by:

$$Cost_{pump} = 1.51*Cap \qquad (3.7.5)$$

(Waier et al. 1992, 480).

According to D. Knebel (1994b), the FOB cost of an ice harvester having a net capacity of 100 to 300 tons is

$$Cost_{harvester} = 8{,}000 + 1{,}365*NCap \qquad (3.7.6)$$

Equation 3.7.6 includes the cost of controls, the evaporative condenser unit, and the refrigerant pump. For ice harvesters having capacities greater than 300 tons, the FOB cost is

$$Cost_{harvester} = 10{,}000 + 1{,}202*NCap \qquad (3.7.7)$$

An "economy of scale" applies here that does not apply in the case of centrifugal chillers. However, the cost per ton of an ice harvester is between four to eight times greater than that for a centrifugal chiller.

J. Ebeling (1994b) reported a custom ordered FOB cooling coil cost of \$14.14 per row per square foot of face area based on quotes for the Butler-Warner Generating Plant. The FOB cooling coil cost is thus given as

$$Cost_{coil} = 14.14 * A * Nrows \tag{3.7.8}$$

where "A" is the face area in square feet and "Nrows" is the number of cooling coil rows. The cooling coils used in the Fayetteville project are made of stainless steel tubing and epoxy coated carbon steel plate fins. They are similar to those modeled in the present project, except that the fins are made of aluminum in the latter case. This difference should not have a significant effect on the calculated cooling coil cost.

The free on board cost does not include shipping, labor, contractor overhead or contractor profit. All FOB costs are thus multiplied by a factor of 2.5 to account for these expenses in order to determine the installed cost of each component or component grouping. The individual costs are summed to give the installed cost of the entire cooling system. Dividing system cost by the capacity increase in kilowatts due to the cooling system at design conditions gives the capacity enhancement cost in dollars per kilowatt.

Finally, the cost calculator performs a life cycle analysis of the inlet cooling system based on the concept of a discounted payback period. The discounted payback period is the number of years required for the discounted annual savings associated with operating a system to sum up to the initial investment. Setting the sum of discounted annual savings equal to the initial investment gives

$$Cost_{system} = [1/(d - i)] * \{1 - [(1 + i)/(1 + d)]^{NP}\} * Savings_{ann} \tag{3.7.9}$$

where "d" is the discount rate, "i" is the fuel inflation rate, and "N_P" is the payback period in years (Duffie and Beckman 1991, 471 - 472). Based on the assumption that a demand exists for all the energy that the power plant is capable of generating with inlet air cooling during the hours the plant operates in the cooling season, and that the utility operating the power plant must therefore purchase electricity from another utility in the absence of such a cooling system, the annual savings is given by

$$\mathrm{Savings}_{ann} = C_{PE}*\Delta EE_{NC} - C_F*\Delta \mathrm{Fuel} - C_{OPE}*EE_{OP} \qquad (3.7.10)$$

Here "C_{PE}" is the wholesale cost of on-peak electricity in dollars per kilowatt-hour, "ΔEE_{NC}" is the annual incremental electrical energy produced due to the inlet air cooling system in kilowatt-hours, "C_F" is the cost of the fuel in dollars per pound, "ΔFuel" is the annual excess fuel consumed by the power plant due to the operation of the inlet cooling system in pounds, "C_{OPE}" is the cost of off-peak energy used to charge the chilled water storage tank and/or ice storage tank in dollars per kilowatt-hour, and "EE_{OP}" is the amount of electric energy consumed annually by the cooling system in kilowatt-hours.

Solving Equations 3.7.9 and 3.7.10 for "C_{PE}" yields the wholesale cost of on-peak electricity that would result in a cooling system discounted payback period of "N_P" years. This quantity is equivalent to the the cost of the incremental electric power produced with inlet air cooling based on the same discounted system payback period. The cost calculator computes this quantity based on the results of a seasonal simulation and the economic parameters discussed in section 6.2. The TRNSYS cost calculator component appears in Appendix C.

CHAPTER 4: COMBUSTION TURBINE INLET AIR COOLING SYSTEM MODELS

The individual TRNSYS component models described in the previous two chapters were used to assemble a TRNSYS model of the combustion turbine inlet air cooling system represented in Figure 1.3.1. Additionally, four EES programs were written to model the sections comprising the overall system as discussed briefly in section 1.3. The TRNSYS system model, the EES section models, and the EES combustion turbine model were then used interactively to design inlet air cooling systems for a variety of power plant operating conditions, inlet configurations, and storage capacity splits. Finally, the TRNSYS model was used to perform the annual cooling season simulations upon which system life cycle analyses are based.

The TRNSYS and EES cooling system models are described in the first two sections of this chapter. The refrigeration equipment schedules are the same for all 24 final cooling system designs and are provided in section 4.1, which is devoted to the TRNSYS model. Details concerning the cooling coil surface used for all system designs appear in section 4.2, which is devoted to the EES programs. Weather conditions are also the same for all system designs, and are described in section 4.3 of this chapter.

4.1 The TRNSYS System Model

Figure 1.3.1 shows the combustion turbine inlet air cooling system modeled by TRNSYS. The system consists of two sections, or loops: one based on chilled water storage, the other based on ice storage. Air drawn into the combustion turbine passes through three separate cooling units: an evaporative cooler, a cooling coil fed by water being circulated through the chilled water storage tank, and a cooling coil fed by water being circulated through the ice storage tank. The optional evaporative cooler can be removed from the system by setting its effectiveness equal to zero. The first and second cooling coils and their associated refrigeration equipment can be removed by making minor changes to the simulation deck, as discussed at the end of this section.

The chilled water storage loop operates on the basis of a daily full storage strategy. Water drawn from the top of the stratified chilled water storage tank is cooled to a set point temperature of 40° F by the chiller each day the combustion turbine is in use, and then is returned to the bottom of the storage tank. One of the differential controllers described in section 3.4 turns the chiller off if the temperature at the top of the tank is lower than a specified value. The chiller operates during off-peak hours for a maximum of 15 hours per day as needed to charge the storage tank. A TRNSYS Time Dependent Forcing Function (Klein et al. 1994, 4.1.2-1 - 3) is used to set limits on the hours the chiller operates. These limits are 9:00 p.m. until 12:00 p.m. of the following day, Sunday through Thursday. The chiller does not operate over the weekend because the combustion turbine does not operate over the weekend, and stored chilled water temperature rises due to environmental losses over the entire weekend are very small (typically less than 0.1° F). The chiller and the combustion turbine are never on simultaneously. Water from the cooling tower is circulated around the chiller condenser while the chiller is in operation. The cooling tower, cooling tower pump, and chiller pump are all turned on and off by the same differential controller that determines when the chiller is in operation.

A flow diverter placed between the chiller evaporator outlet and the chilled water storage tank and a tempering valve that mixes the diverted stream of chilled water with the water entering the evaporator ensure that the chiller evaporator inlet temperature does not exceed a specified maximum value. This is necessary in order to avoid overloading the chiller, as discussed in section 3.4. The flow diverter is controlled by the differential controller based on Equation 3.4.1. The flow diverter and the tempering valve are both modeled by TRNSYS with equations, rather than with individual component subroutines.

The ice storage loop operates on the basis of a weekly full storage strategy. Water from the bottom of the ice storage tank is pumped over the evaporator plates of the ice harvester to generate ice as described in section 3.1. The ice harvester is also limited by a second Time Dependent Forcing Function to operating between 9:00 p.m. and 12:00 p.m. of the following weekday as needed, and never operates at the same time as the combustion turbine. It also can operate over the weekend, between 9:00 p.m. on Friday until 12:00 p.m. on Monday. The ice harvester can thus function for a total of 123 hours per week: 60 off-peak hours between Monday evening and Friday at noon, and 63 hours between Friday evening and Monday at noon. One of the other differential controllers discussed in section 3.4 turns the ice harvester off in the event that the ice storage tank is filled to capacity. The ice harvester is cooled by an evaporative condenser unit rather than by a water cooled condenser and cooling tower. The evaporative condenser is incorporated into the TRNSYS ice harvester model.

The water mass flow rate through each cooling coil depends on the desired power plant electric output and on the ambient air state at each simulation time step. By comparing the desired power output to the actual power output at each iteration within the simulation time step, the cooling coil pump controller determines those flow rates as explained in section 3.4. Design weather conditions are discussed in section 4.3 below. Power plant load profiles (the desired electric outputs) are discussed in Chapter 5. Time Dependent Forcing Functions are used to provide the ambient dry bulb temperature, wet bulb temperature, and desired power plant electric output at each simulation time step.

All pipes represented with solid lines in Figure 1.3.1 are modeled by TRNSYS in order to determine the temperature rise due to environmental losses. The pipes running between the chiller and the water storage tank and between the chiller and cooling tower are 100 feet long in all system designs. The pipes running between the water storage tank and the first cooling coil and between the ice storage tank and the second cooling coil are 300 feet long in all system designs. Since the pipe running from the bottom of the ice storage tank and the ice harvester is comparatively short, it is not modeled.

The TRNSYS simulation deck was assembled using TRNSED, a "front-end" program that allows preparation of an "input file" that hides all details of the TRNSYS program not needed by the user. Both the simulation deck and the input file appear in Appendix D. The user enters selected parameters, inputs, and initial conditions using the TRNSED input file before running a simulation. Most of these parameters, inputs, and initial conditions are generated by the EES section models, which are discussed in section 4.2 below.

The TRNSYS simulation generates detailed output files for use in system design and analysis. Some data are recorded at ten minute intervals (twice the simulation time step of five minutes), while other data are recorded only at the end of the simulation period. Data recorded every ten minutes include: ambient dry and wet bulb temperatures, the entering chiller condenser water temperature, the average chilled water storage tank temperature, the entering and leaving water temperatures for each cooling coil, the leaving air dry bulb temperatures for each cooling coil, water mass flow rates for the chiller evaporator and condenser, the ice generation rate, the ice storage tank inventory, the water mass flow rates for each cooling coil, the air mass flow rate into the combustion turbine, the net power plant electric output, the desired power plant electric output, and the electric power output for the combustion turbine in the absence of an inlet air cooling system. TRNSHELL, the environment program in which TRNSYS is housed, allows the user to prepare graphs of all data listed above quickly and easily. Integrated data recorded only at the end of the simulation period include: the electric energy consumed by the chiller, the electric energy consumed by the ice harvester, the sum of the electric energy consumed by all other refrigeration system components, the energy transferred from the inlet air stream to each cooling coil. the change in internal energy for each storage tank, the net electric energy produced by the power plant both with and without inlet air cooling, the mass of fuel consumed by the power plant both with and without inlet air cooling, and the overall conversion efficiency of the power plant both with and without inlet cooling. Cost data are also supplied at the end of the simulation, which include: the combined chiller and cooling tower cost, the ice harvester cost, the combined cooling coil cost, the cost of each storage tank, the total cost of all system pumps and pipes, the total cost of each storage loop, the total cooling system cost, the capacity enhancement cost, the cost of the excess fuel consumed by the power plant due to inlet air cooling, the cost of off-peak electricity consumed by the refrigeration equipment, and the cost of the incremental on-peak electricity produced with inlet air cooling based on the specified payback period. Data recorded at the end of the simulation time step can be viewed in a TRNSHELL "window" immediately after the simulation ends.

In order to model cooling systems based on only one storage medium, it is necessary to make minor changes in the TRNSYS simulation deck. To model a system based on chilled water storage alone, the inputs for the combustion turbine component model are changed from the second to the first cooling coil, and the ice harvester size is set to an arbitrarily small value less than 0.9 ton. To model a system based on ice storage alone, the order of the cooling coil pump control variables (supplied by the cooling coil pump controller to the two cooling coil pumps) is reversed and the chiller size is set to an arbitrarily small value less than 0.9 ton. No other changes are required to "disable" either of the two storage loops.

4.2 EES Cooling System Section Models

The EES cooling system section models were written and used to provide "guess values" for selected parameters, inputs, and initial conditions required by the TRNSYS system model. A total of four programs were written: CWSO.size, ISO.size, CWSL.size, and ISL.size. They model the performance of a cooling system based on chilled water storage alone, a cooling system based on ice storage alone, the chilled water loop in a cooling system based on both chilled water and ice storage, and the ice storage loop in a cooling system based on both chilled water and ice storage, respectively. All EES section models appear in Appendix B.

Each cooling system section model consists of a set of simultaneous equations that specify thermal characteristics, physical sizes, and other parameters relating to the components which comprise the system. The key component in all the section models is the cooling coil. Given the inlet air mass flow rate, the inlet and outlet air states, the inlet water temperature, and the number of rows, the EES model calculates the dimensions of the cooling coil and the required water mass flow rate for use in the TRNSYS simulation. These calculations are based on an effectiveness-NTU analysis for either completely wet or completely dry outer tube surfaces that is very similar to the analysis contained in the TRNSYS cooling coil model discussed in section 3.6. The EES model is not as sophisticated as the TRNSYS cooling coil model, which performs a combined wet and dry analysis. Nevertheless, the cooling coil loads calculated by the EES section models are generally within 7% of the values calculated by the more accurate TRNSYS system model.

The EES model also determines the cooling coil air and water side pressure drops. The air side pressure drop, "ΔP_{air}", is given by :

$$\Delta P_{air} = (G^2 * v_1/g_c)*[(K_c + 1 - s^2) + 2*(v_2/v_1 - 1) + f_{air}*v_m/(s*v_1) - (1 - s^2 - K_e)*v_2/v_1] \qquad 4.2.1$$

Here "G" is the air mass flux, "v_1" is the specific air volume at the entrance, "g_c" is the force - mass conversion factor for English units. "K_c" is the entrance pressure loss coefficient, "s" is the ratio of the free flow area to the frontal area of the cooling coil, "v_2" is the specific air volume at the exit, "f_{air}" is the friction factor for the existing flow conditions, "v_m" is the average specific air volume in the coil, and "K_e" is the exit pressure loss coefficient (Kays and London 1964, 33). Based on values for the air side pressure drop reported by D. Bantam

(1994) for a cooling coil surface similar to that used in this study, the presence of water on the cooling coil surface increases the air side pressure drop by a factor of approximately two. The equation presented by Kays and London gives the air side pressure drop for dry tubes; the right hand side of that equation has thus been multiplied by a factor of two to describe wet tubes in Equation 4.2.1. The coefficients "K_c" and "K_e" are represented graphically as functions of "s" and the Reynolds number on page 93 of *Compact Heat Exchangers.* The terms in Equation 4.2.1 that contain "K_c" and "K_e" account for coil inlet and exit pressure drops, respectively. Since there is no "expansion section" between the two cooling coils, the EES section models do not count entrance and exit pressure drops twice for cooling systems based on both water and ice storage: CWSL.size does not include the exit pressure drop term, and ISL.size does not include the entrance pressure drop term.

The water side pressure drop, "ΔP_{wat}", is given by Equation 4.2.2:

$$\Delta P_{wat} = 1/2 * f_{wat} * rho * CCWV^2 * L * D \qquad (4.2.2)$$

where "f_{wat}" is the Moody friction factor, "rho" is the density of water, "CCWV" is the water velocity in the tubes, "L" is the overall tube length, and "D" is the tube inner diameter. The Moody friction factor for a smooth surface is given by:

$$f_{wat} = 0.316 * Re_D^{-0.25} \qquad (Re_D \leq 20{,}000) \qquad (4.2.3a)$$

$$f_{wat} = 0.184 * Re_D^{-0.20} \qquad (Re_D > 20{,}000) \qquad (4.2.3b)$$

where "Re_D" is the Reynolds number based on the inner diameter of the tube (Incropera and DeWitt 1990, 472 - 474). Water properties are evaluated at the cooling coil entering water temperature.

The air side pressure drop is used by the EES combustion turbine model, BBPPmod.5, to determine the gross maximum electric power output with inlet air cooling as described in sections 2.3 and 2.4. The water side pressure drop is used by the EES section model to determine the cooling coil pump power requirement, which is in turn used by BBPPmod.5 to calculate the net maximum electric power output with inlet air cooling. The air side pressure drop and both the gross and net maximum power plant electric outputs are used as parameters in the TRNSED input file.

Fixed cooling coil parameters are based roughly on values reported by Ebeling and his co-workers (1994) for the inlet air cooling system installed at the Butler-Warner Generating Plant in Fayetteville, North Carolina. Several of these values were changed slightly in order to use heat transfer data reported by Kays and London (1964, 224) for "surface 8.0-3/8T" in the EES section models. These data, which include the air side friction factor and the product of the Stanton and Prandtl numbers as functions of the Reynolds number, are presented in the "Look-up Table" immediately following the first EES section model in Appendix B. Following Ebeling, the tube and fin materials were specified as stainless steel and aluminum, respectively. The use of copper was avoided due to the corrosion hazard posed by the ammonia refrigerant. Fixed cooling coil parameters for all systems designed are summarized in Table 4.2.1 below.

Parameter	**Value**	**Units**
Air face velocity	400	feet/minute
Water tube velocity	10	feet/second
Fin pitch	8	per inch
Fin thickness	0.013	in
Tube outside diameter	0.402	inch
Tube wall thickness	0.035	inch
Tube spacing in each row	1.00	inch
Spacing between rows	0.866	inch
Fin material conductivity	102.3	BTU/hr-ft-°F
Tube material conductivity	7.74	BTU/hr-ft-°F

Table 4.2.1: Fixed Cooling Coil Parameters

The cooling coil component is "linked" in each EES section model to other cooling system components. In the case of the chilled water storage loops, these include the water storage tank, the chiller, the cooling tower, and the connecting pipes. In the case of the ice storage loops, additional components include the ice harvester, the ice storage tank, and the connecting pipes. The thermal and physical sizes of these components depend on the calculated cooling coil loads.

Since stratification is not perfect in a chilled water storage tank, the usable storage volume is somewhat less than the total tank volume. The water storage tank is thus sized 5% larger than the volume of liquid circulated through the cooling coils each day (Mackie and Reeves 1988, 2-8). All the water in the storage tank must be cooled to the chilled water set point of 40° F in a time period of 15 hours for all system designs. The temperature difference between the top and bottom of the discharged stratified chilled water storage tank ranges between about 13° F and 16° F for the step power plant load profiles described in section 5.1. For the peaked power plant load profile, the temperature difference between the top and bottom of the discharged chilled water storage tank ranges up to 45° F.

The EES section models treat the chiller as a simple ammonia-based refrigeration cycle with an isentropic compressor efficiency of 0.67 and a motor efficiency of 0.94. The design condenser temperature is 90° F; the design evaporator temperature is 35° F. The design chiller load and power requirement are determined and used in the TRNSYS cooling system model. That model also requires a value for the minimum chiller load, which is set equal to 15% of the design load (Pawelski 1994).

A selection procedure devised by the Marley Corporation is used to size the cooling tower (Marley n.d.). A hot water temperature of 92° F at the cooling tower inlet and a cold water temperature of 85° F at the tower outlet are taken as design conditions. The required cooling tower water mass flow rate can be determined from the amount of heat rejected by the chiller and these two temperatures. The Marley selection procedure is then used to find the cooling tower fan brake horsepower. Assuming a fan efficiency of 0.80, the EES section models calculate the fan power requirement for the TRNSYS simulation. The air mass flow rate is also needed by the TRNSYS cooling tower component; it is set equal to 80% of the water mass flow rate (Stoecker and Jones 1982, 367 - 372).

The EES section models used for determining the size of ice storage loop components both feature an ice harvester component nearly identical to that described in section 3.1 above. The section model for cooling systems based on ice storage alone uses performance curves derived for the 410 ton capacity rotary screw compressor, while the section model for the ice storage loop in systems based on both storage media uses performance curves derived for the 135 ton capacity rotary screw compressor. The ice harvester is assumed to operate 123 hours per week for all system designs. The EES section models compute the ice harvester net capacity at design operating conditions based on this weekly period of operation and the weekly ice requirement, which is in turn based on the cooling coil load.

In order to ensure an ice storage tank discharge fraction of roughly 0.80 when the combustion turbine shuts down on Friday, the capacity of the tank is set equal to 1.2 times the mass of ice that can be generated over the weekend. The volume of the ice storage tank is computed as twice the value needed to store the computed mass of solid ice in order to account for the void and water volumes. The ice mass at the beginning of the simulation period (Monday at 12:00 a.m.) is also determined. These three values are required by the TRNSYS system model.

Finally, the EES section models calculate the pipe diameters and pump power requirements for the TRNSYS simulation. Pipe diameters depend on the water velocities and mass flow rates. The water velocity between the storage tanks and cooling coils is set equal to ten feet per second; the water velocity between the chiller and the cooling tower and between the chiller and the water storage tank is six feet per second for all system designs. Pipe lengths are also fixed for all system designs as discussed in section 4.1. The pressure drops in the pipes are calculated using Equations 4.2.2 and 4.2.3, and the pump power requirements are determined assuming overall pump efficiencies of 65%. Power requirements are on the order of one kilowatt for the chiller and cooling tower pumps, but are significantly larger for the cooling coil pumps.

Quantities calculated by the EES section models for use in the TRNSYS simulation are boxed on the "solutions worksheets", which facilitates the transfer of the calculated quantities to the TRNSED input file. All values required by the TRNSYS simulation appear with the corresponding EES programs in Appendix B for the three representative cases that are discussed in detail in Chapter 5.

4.3 Design Weather Conditions

The design weather conditions specified by the utility for power plant cooling system operation are a 95° F dry bulb temperature, a coincident 76° F wet bulb temperature, and an ambient pressure of 14.4 psia. The corresponding relative humidity is 43%. These temperatures appear in the EES section models in order to size the cooling coils. The EES section models further assume a 90° F design dry bulb temperature and a 77° F design wet bulb temperature for sizing the cooling tower, which only operates between the hours of 9:00 p.m. and 12:00 p.m. the following day. The same wet bulb temperature is used to size the evaporative condenser unit for the ice harvester. Finally, a daily average dry bulb temperature of 80° F is assumed for estimating storage tank losses.

Since the TRNSYS model performs transient simulations, it requires weather data at each simulation time step. Climatological data for an airport located within a 75 mile radius of the combustion turbine power station were used to generate the design day temperature profiles shown in Figure 4.3.1. The maximum dry bulb temperature of 95° F occurs at 5:00 p.m. daylight saving time with a coincident wet bulb temperature of 76° F. The maximum wet bulb temperature of 77° F occurs at 1:00 p.m. daylight saving time. The TRNSYS cooling system model assumes a constant ambient pressure of 14.4 psia. The TRNSYS combustion turbine model assumes a slightly lower ambient pressure of 14.2 psia, however, in determining the capacity and fuel mass flow rate at each simulation time step. Neither the incremental power produced by the combustion turbine nor any of the economic measures used to compare different cooling systems are affected by this small discrepancy.

All inlet air cooling systems were designed on the basis of a "design week", which is composed of 7 days having temperature profiles identical to those shown in Figure 4.3.1. One week was chosen because the ice harvester operates on the basis of a weekly full storage strategy to minimize the ice harvester refrigeration capacity. The ice storage tank is only filled to capacity at 12:00 p.m. on Monday. The chiller, in contrast, operates on the basis of a daily full storage strategy: the chiller re-charges the chilled water storage tank completely overnight and does not operate over the weekend, as discussed in section 4.1.

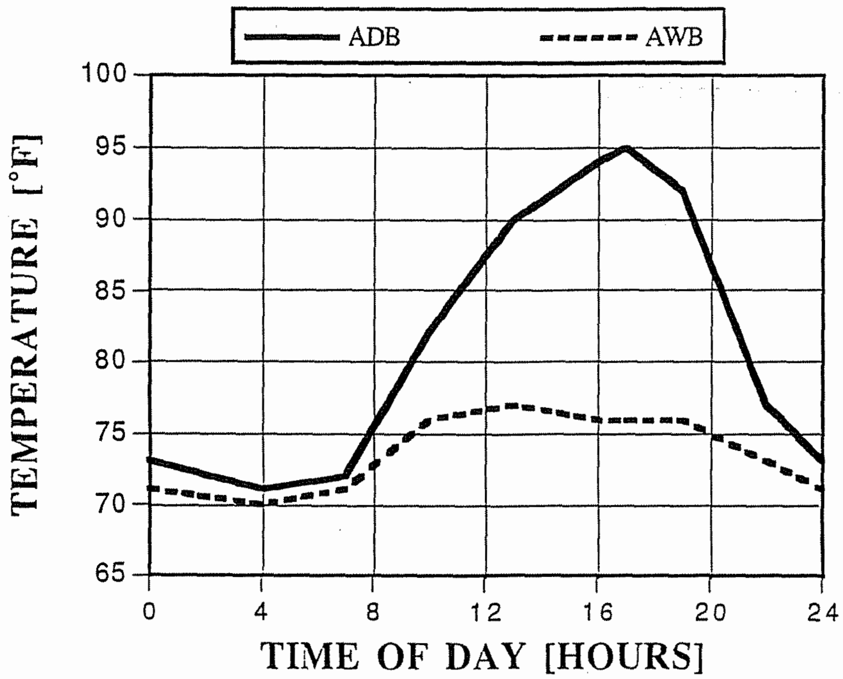


Figure 4.3.1: Design Day Ambient Dry and Wet Bulb Temperature Profiles

Depending on the power plant load profile, the cooling coils operate up to eight hours per day, between 1:00 and 9:00 p.m. For systems that lower the dry and wet bulb temperatures of the inlet air stream to 40° F, the cooling load varies by up to 6% from its average value during that time period. The load at 1:00 p.m. is 25.2 Btu/lb of dry air, which is actually slightly higher than the load at the "design conditions" used by the EES section models, 24.1 Btu/lb of dry air. The cooling load at 9:00 p.m. is 22.4 Btu/lb of dry air. By adjusting the cooling coil water mass flow rates, the TRNSYS model is able to match the cooling coil load corresponding to the desired power plant electric output closely at each simulation time step.

CHAPTER 5: INLET AIR COOLING SYSTEM DESIGN

The EES section models and the TRNSYS model were used together with the EES combustion turbine model to design a variety of alternative inlet air cooling systems. Two general cases were considered: 1) the power plant operating in the base mode without evaporative cooling and 2) the power plant operating in the power augmentation mode with an evaporative cooler in the inlet air flow stream. The capacity split between ice and chilled water storage for hybrid cooling systems yielding the lowest capacity enhancement cost was determined for each general case. Additionally, inlet air cooling systems for each general case based on the optimized combination of ice and chilled water storage, chilled water storage alone, and ice storage alone were designed for four different power plant load profiles. Weather conditions are identical for each case. A total of 24 different cases were thus considered. This chapter describes the defining characteristics of the different system designs, explains the design process using three representative examples, and presents the results of that process.

5.1 Description of Inlet Air Cooling Systems

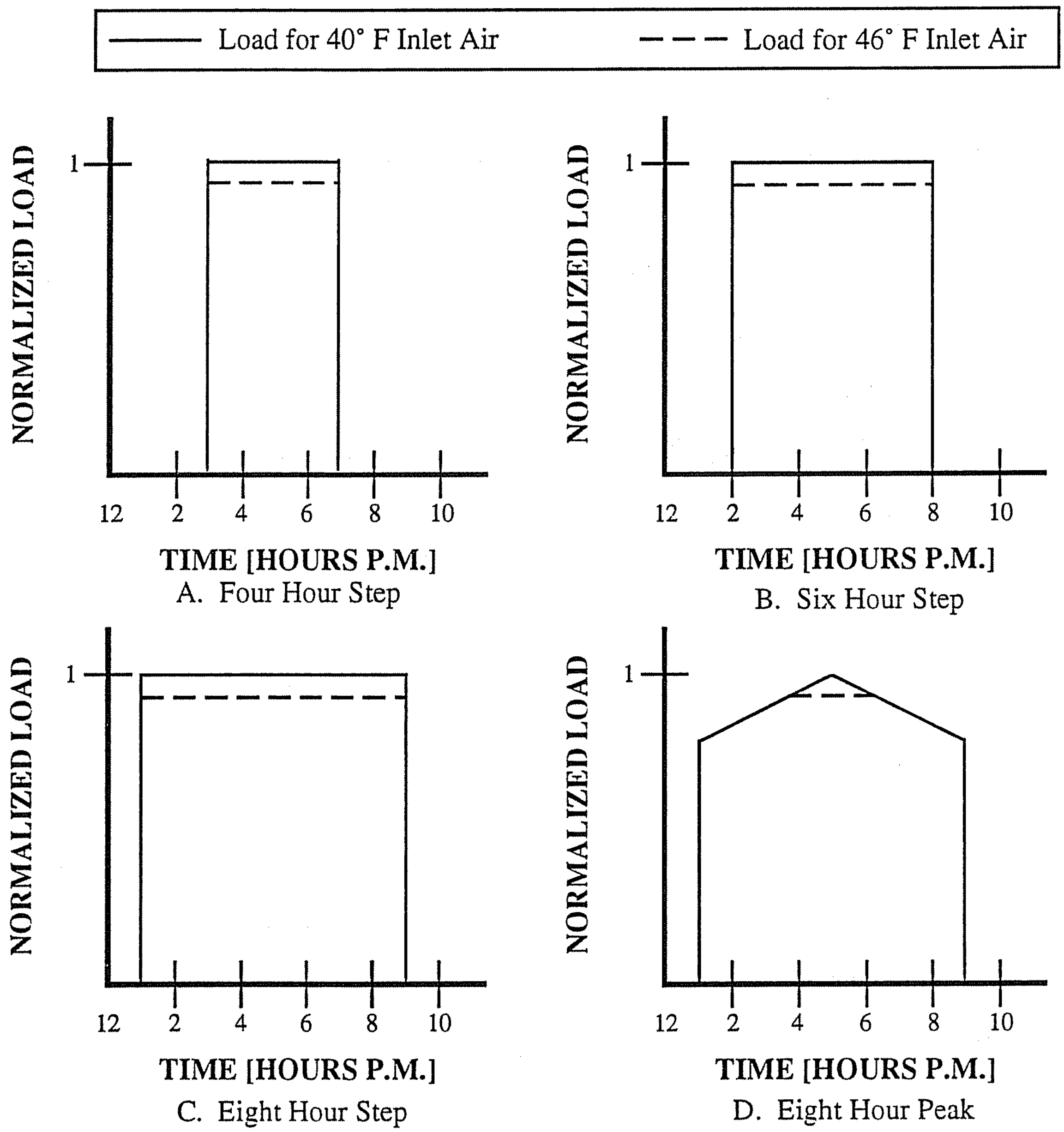
Not all combustion turbines are equipped with evaporative cooling units in their inlet air flow streams, and many cannot be operated in the power augmentation mode and must instead operate in the base mode (see section 2.3). This combination of conditions - base mode operation and no evaporative cooler - was chosen as the first "general case". The combustion turbine used as the basis for the power plant model currently does have an evaporative cooler in the inlet air stream. It is typically operated in the power augmentation mode in order to achieve maximum power output. This second combination of operating conditions was therefore chosen as the second general case. The evaporative cooling unit simply drives the air dry bulb temperature towards the ambient wet bulb temperature as discussed in section 3.5. The specific enthalpy of the air increases slightly during this process, so the evaporative cooler actually increases the total cooling coil load marginally. The additional moisture in the air improves the performance of the cooling coils, however, so the thermal storage based cooling system cost is generally slightly lower than it would be in the absence of the evaporative cooler. The combustion turbine model assumes no interaction between the water injection flow rate and the inlet air temperature. Therefore, the choice of combustion turbine operating mode has no influence on either the capacity or efficiency increase due to inlet cooling, or on the cooling system size.

The minimum compressor stage inlet air temperature for the turbine is 40° F. Lower air temperatures can lead to the formation of ice particles in the compressor section, a situation to be avoided. For the design weather conditions considered, the minimum compressor stage inlet air temperature can be achieved by cooling systems based either entirely or partially on ice storage, but not by cooling systems based on chilled water alone. Three different storage capacity splits were evaluated: one based on ice alone, one based on chilled water alone, and one based on the combination of these two media that yields the lowest power plant capacity enhancement cost, as defined in section 3.7. It was initially expected that systems based on ice storage alone would lead to the highest capacity enhancement costs and that systems based on a combination of chilled water and ice would lead to the lowest capacity enhancement costs for the three storage capacity splits considered. As it turned out, the systems based on water storage alone always yield even lower capacity enhancement costs than the systems based on a combination of storage media. This result, however, is sensitive to the cooling system operating strategy, combustion turbine performance characteristics, and power plant site layout, none of which were varied in this study.

Four daily power plant load profiles were considered for each general case and storage capacity split. These include four, six, and eight hours of full capacity electric power generation with a 40° F inlet dry bulb temperature and an eight hour symmetrically peaked profile. The symmetrically peaked profile increases from the power plant capacity at design conditions without inlet air cooling to the full power plant capacity with 40° F inlet air, and then decreases back to the original value. The "peaked profile" results in approximately the same amount of electric energy produced with inlet air cooling as the four hour "step profile". All load profiles are centered around 5:00 p.m. daylight saving time, the hour at which the ambient dry bulb temperature attains its maximum value (see section 4.3). Since systems based on chilled water storage alone cannot achieve inlet air temperatures of 40° F, the corresponding normalized daily power plant load profiles are "clipped" as represented in Figures 5.1.1a - d below.

In summary, two general cases were considered: base mode without evaporative cooling and power augmentation mode with evaporative cooling. Three storage capacity splits were considered: ice alone, chilled water alone, and a combination of these media. Four power plant load profiles were used: three "step loads" of varying duration and one "peaked load". Hence a grand total of 24 systems were designed based on all permutations of the above qualifiers.

The following five sections describe the cooling system design process. Three systems are described in detail. Each was designed for the combustion turbine operating in the base mode with no evaporative cooling upstream of the cooling coils. The first system uses ice storage alone and was designed for the four hour daily step power plant load profile, the second uses chilled water storage alone and was designed for the same four hour step load profile, and the third uses both chilled water and ice storage and was designed for the eight hour peaked power plant load profile. Additionally, the method used to determine the optimum capacity split for systems based on both storage media is outlined in section 5.4. The entire design process is summarized in section 5.6.



Figures 5.1.1: Normalized Daily Power Plant Load Profiles

5.2 The Cooling System Design Process: First Example

The EES program ISO.size was used to design the cooling system based on ice storage alone for a four hour step power plant load profile. Both the "equations worksheet" and the "solution sheet" for this system appear in Appendix B. The cooling coil leaving dry bulb temperature was specified to be 40° F. Following the Burns and McDonnell Engineering Company's design for the plant in Fayetteville, North Carolina, the number of cooling coil rows was specified to be ten (Ebeling et al. 1994). Thus, only two parameters could be varied: "num", the ratio of the cooling coil duct width to the duct height, and "CChrs", the effective number of hours that the cooling coil operates per week. The parameter "num" determines the cooling coil water mass flow rate in the EES model. The parameter "CChrs" typically differs slightly from the actual number of hours of cooling coil operation, due to the fact that the average cooling coil load is less than the design load.

First, a value of "num" was chosen such that the value computed for the cooling coil load based on the heat transfer to the water equaled the value for the heat transfer from the air stream. The value of "CChrs" was initially set equal to 20, since the cooling coil operates four hours per day, five days per week. The calculated air side pressure drop and the cooling coil pump power requirement were entered into the EES combustion turbine power plant model, which was used to determine the gross and net electric output with inlet air cooled to 40° F, as well as the net power plant output without cooling at the design temperature, 95° F. The boxed parameters on the EES section model and combustion turbine model solution sheets were then entered into a TRNSED input file similar to the one shown in Appendix D, and a daily simulation was run.

Graphs of the cooling coil leaving dry bulb temperature and the power plant net electric output were prepared for the simulation period using TRNSHELL. Based on the initial guess values from the EES section model, the cooling coil leaving dry bulb temperature was always slightly higher than the desired value of 40° F, and the net electric power output with inlet air cooling was consequently always lower than the value computed by the EES model. Thus a new value of "num" was chosen in order to increase the water mass flow rate through the cooling coil, and the process described in this and the previous paragraph was repeated. After several further iterations, the cooling coil leaving dry bulb temperature and the net electric power output computed by TRNSYS were equal to the desired values. Figure 5.2.1 shows the leaving dry bulb temperature for a design day. The leaving dry bulb temperature is equal to the ambient dry bulb temperature when the cooling coil is not in operation. Figure 5.2.2 shows the net electric power output with and without inlet air cooling, "NEP" and "EPNC" respectively, for the same time period.

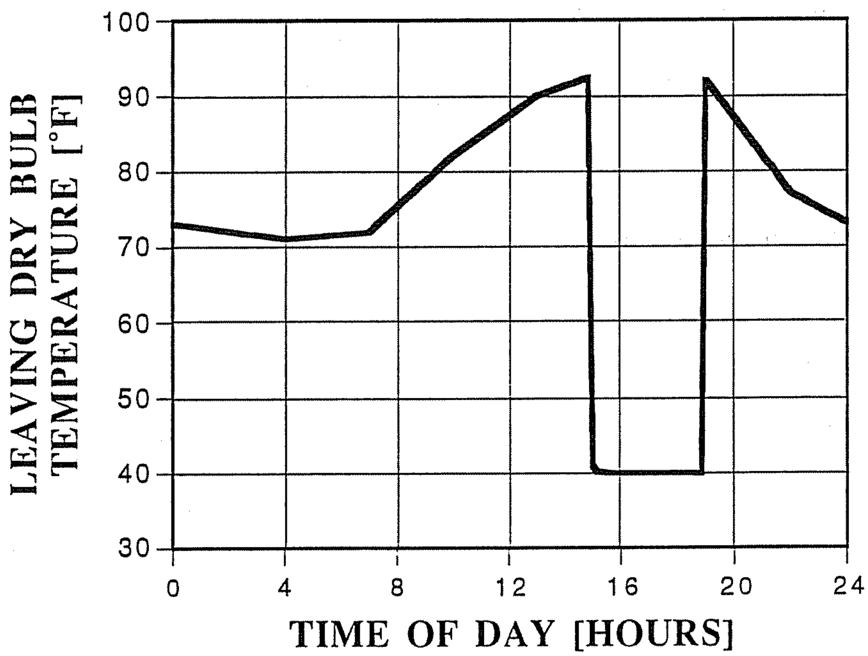


Figure 5.2.1: Design Day Cooling Coil Leaving Dry Bulb Temperature

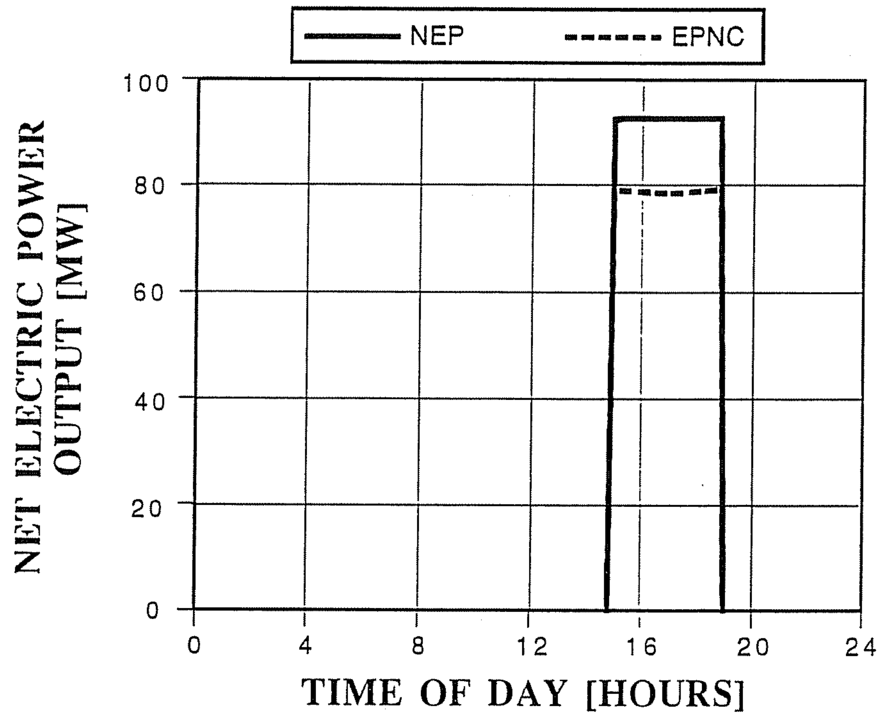


Figure 5.2.2: Design Day Net Electric Power Output with and without Inlet Air Cooling

After finishing the cooling coil design, it was necessary to refine the ice harvester and ice storage tank sizes. This was done by adjusting the parameter "CChrs" in the EES section model and running weekly TRNSYS simulations based on the EES section model results. For the initial value for "CChrs" of 20 hours, a weekly plot of the ice storage tank inventory indicated that the discharge fraction was less than 0.80 when the cooling coil was shut down on Friday afternoon. A lower value of "CChrs" was thus used in the EES section model, the new equipment sizes entered into the TRNSED input file, and a new TRNSYS simulation was performed. These steps were repeated until the sizes of the ice harvester and storage tank had been reduced as much as possible while still meeting the weekly cooling coil load. The ice inventory in the storage tank is shown in Figure 5.2.3 for the final design. The maximum discharge fraction is 0.80, and the ice inventories at the beginning and end of the design week are roughly equal. This concludes the discussion of the first inlet air cooling system design example.

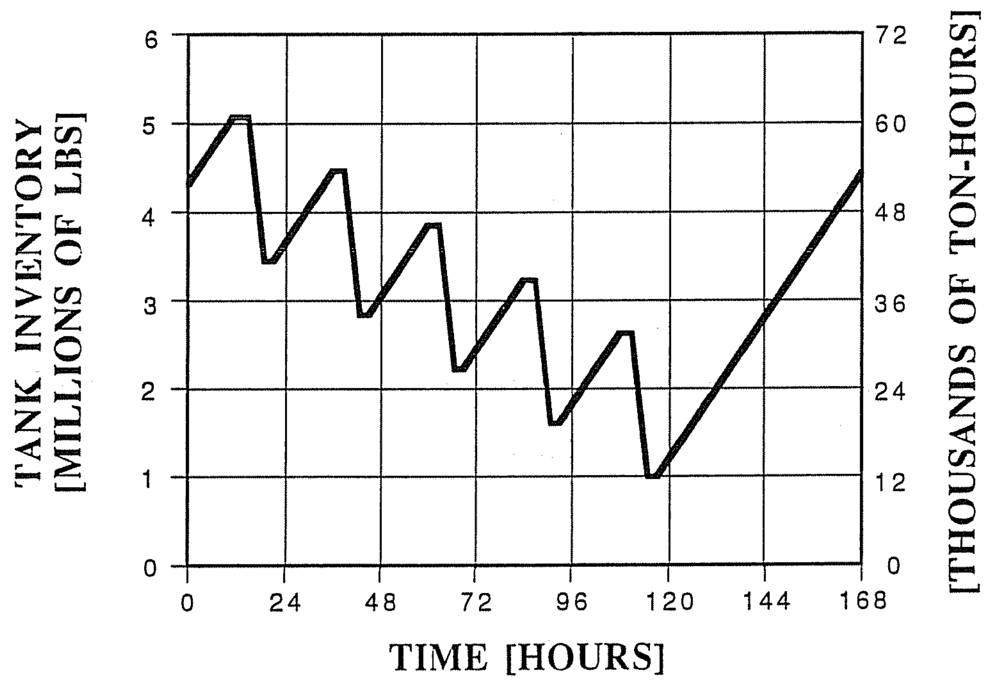


Figure 5.2.3: Ice Storage Tank Inventory for Design Week

5.3 The Cooling System Design Process: Second Example

The second cooling system example was designed for the same power plant load profile as the first, but is based on chilled water storage alone. The EES section model CWSO.size was used for this system, and is shown in Appendix B together with its solution sheet. Andrepont (1994) claimed that 46° F is about the lowest leaving dry bulb temperature achievable with 40° F chilled water storage for ambient dry bulb temperatures of 90° F - 100° F. Hence the cooling coil design leaving dry bulb temperature was specified to be 46° F. In this case, there is no distinction between the actual and effective number of hours of cooling coil operation, because the flow rate out of the bottom of the chilled water storage tank is very nearly constant. Thus "Dhrs", the daily hours of cooling coil operation, was simply set equal to four.

In this case, three EES section model parameters could be varied: "LWT", the cooling coil leaving water temperature, "Nrows", the number of cooling coil rows, and "MLWT", the maximum chiller evaporator inlet temperature. The parameters "LWT" and "Nrows" control the cooling coil water mass flow rate; "MLWT" determines the sizes of the chiller and cooling tower (in conjunction with the constant "Dhrs").

In designing the cooling coil for this system, several different choices for "Nrows" were attempted. Setting the number of cooling coil rows equal to ten resulted in water flow rates similar to that found in the design based on ice storage alone, and therefore seemed like a reasonable choice. In this case, "LWT" was adjusted until the air and water side cooling coil loads were equal. Next, the maximum chiller evaporator inlet temperature was set several degrees higher than the cooling coil leaving water temperature to ensure that the chiller would be sufficiently large to meet the daily cooling coil load, and all other parameters were calculated by CWSO.size. Again, the cooling coil leaving dry bulb temperature, air side pressure drop, and cooling coil pump power requirement were entered into the EES combustion turbine model, which was used to determine new power plant parameters for the TRNSED input file.

A daily TRNSYS simulation was run based on the values computed by CWSO.size and BBPPmod.5. As before, plots prepared using TRNSHELL showed that the leaving dry bulb temperature was always too high and the net electric power produced by the combustion turbine was always too low, indicating that it was necessary to increase the cooling coil water mass flow rate. This was done by decreasing the value of "LWT" in the EES section model and repeating the process just described until both the cooling coil leaving dry bulb temperature and the net combustion turbine electric power output calculated by TRNSYS reached their desired values of 46° F and 91,150 kW, respectively.

Once the cooling coil design had been completed, it was a simple matter to size the chiller by adjusting "MLWT". The maximum chiller evaporator inlet temperature was set 0.5° F above the minimum cooling coil leaving water temperature calculated by the TRNSYS model. The cooling coil leaving water temperature for a design day is shown in Figure 5.3.1 below. For the four hour period during which water flows through the cooling coil, the minimum leaving water temperature is 54.4° F. Hence "MLWT" was set equal to 54.9° F in both the EES and TRNSYS models. Final chiller capacity, cooling tower capacity, cooling tower fan power requirement, and pipe sizes were calculated by CWSO.size and entered into the TRNSED input file, and another daily simulation was run. Since "MLWT" is greater than the maximum cooling coil leaving water temperature, there is no need for the TRNSYS model to divert any water from the leaving chiller evaporator flow stream back to the chiller evaporator inlet.

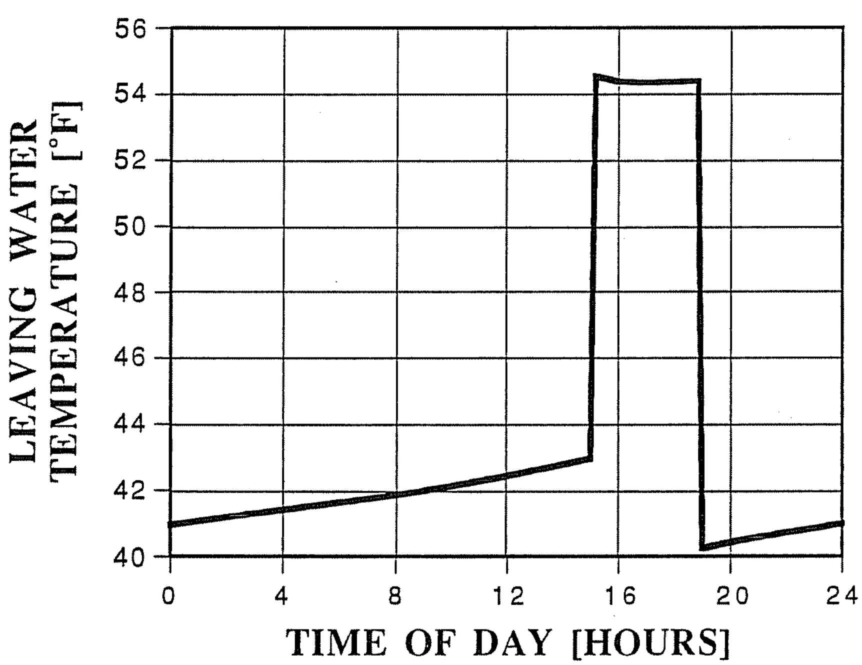


Figure 5.3.1: Design Day Cooling Coil Leaving Water Temperature

Finally, a weekly simulation was performed in order to compare the average storage tank water temperature at the beginning and end of the design week. These two temperatures were not equal to each other, indicating a need to adjust the initial temperature of the upper portion of the tank. The initial temperature of the upper 80% of the tank had originally been set to the value of "LWT" used in the EES section model; the initial temperature of the lower 20% of the tank was fixed at 40° F, the chiller set point temperature. By determining the upper tank temperature at the end of the week, setting the initial upper tank temperature equal to that value, and running a second weekly simulation, it was possible to ensure that the energy of the chilled water storage tank was very nearly the same at the end of the design period as at the beginning. The average chilled water storage tank temperature for the design week is shown in Figure 5.3.2. This concludes the discussion of the second combustion turbine inlet air cooling system design example.

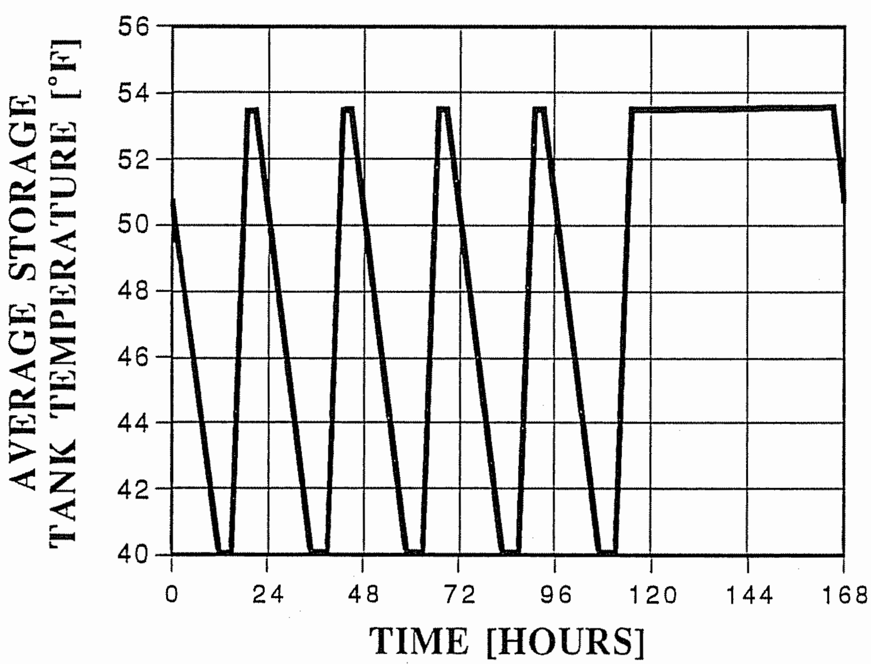


Figure 5.3.2: Average Chilled Water Storage Tank Temperature for Design Week

5.4 Optimum Capacity Split for Hybrid Cooling Systems

This section outlines the process for determining the "optimum capacity split" for hybrid cooling systems featuring both chilled water and ice storage. This optimum is expressed as the leaving dry bulb temperature from the first cooling coil that results in the lowest combustion turbine capacity enhancement cost. The first cooling coil is fed by the chilled water storage tank. As discussed in section 3.7, the capacity enhancement cost is defined as the total cost of the inlet air cooling system (exclusive of the evaporative cooler) divided by the increase in power plant capacity due to inlet air cooling. The leaving dry bulb temperature from the second cooling coil, fed by water circulating through the ice storage tank, is always equal to 40° F.

The EES section models CWSL.size and ISL.size were used to determine the optimum capacity splits for each general case based on the four hour step power plant load profile only. These determinations were made by varying the number of rows for the first cooling coil in order to vary its leaving dry bulb temperature. The first leaving dry bulb temperature was "fine tuned" by varying the cooling coil water mass flow rate in order to arrive at an air state that could be cooled to 40° F by the second cooling coil. The number of rows and the water mass flow rate for the second cooling coil were also varied. This cooling coil design process is similar to that described for the systems based on only one storage media type: the EES section models and the TRNSYS system model were used interactively to converge on two desired leaving dry bulb temperatures rather than only one, as was done in the two cases described above. Other component sizes were also determined according to the guidelines described above in order to calculate the capacity enhancement cost.

The optimum capacity split was found to be nearly the same for both general cases. In the first general case, a leaving dry bulb temperature of 47.2° F from the first cooling coil results in the lowest capacity enhancement cost; in the second general case, the optimum leaving dry bulb temperature from the first cooling coil is 47.5° F. These temperatures are achieved with nine cooling coil rows in the first general case and with eight cooling coil rows in the second general case. The presence of the evaporative cooling unit in the second general case improves the cooling coil performance slightly, as discussed in section 5.1.

Key characteristics of the inlet air cooling systems designed to determine the optimum capacity split between the chilled water storage loop and the ice storage loop for each general case are summarized in Tables 5.4.1 and 5.4.2 below. The two optimum cooling coil designs are highlighted. The cooling coil water volumetric flow rate in gallons per minute is represented by "WVFR", the leaving dry bulb temperature in degrees Fahrenheit by "LDB", and the capacity enhancement cost in dollars per kilowatt by "CEC". The leaving dry bulb temperature from the first cooling coil was varied between 52.8° F and 46.0° F in the first general case and between 49.0° F and 45.3° F in the second general case. The optimum capacity split is not strongly dependent on the number of rows in the first cooling coil in either general case. The optimum is defined somewhat more sharply in the first general case than in the second general case, which may simply be attributable to the use of a slightly more refined system design process for determining the optimum in the second case than in the first.

First Cooling Coil			**Second Cooling Coil**			**System**
Number of Rows	**WVFR [gal/min]**	**LDB [° F]**	**Number of Rows**	**WVFR [gal/min]**	**LDB [° F]**	**CEC [$/kW]**
6	6,749	52.8	4	5,596	40.0	222
7	8,630	50.1	3	8,394	40.0	223
8	7,302	48.6	3	4,846	40.0	215
9	**7,086**	**47.2**	**3**	**2,935**	**40.0**	**211**
10	7,214	46.0	2	11,192	40.0	217

Table 5.4.1: Selected Characteristics of Systems Designed to Determine Optimum Capacity Split for Base Mode Power Plant Operation without Evaporative Cooling

First Cooling Coil			Second Cooling Coil			System
Number of Rows	**WVFR [gal/min]**	**LDB [° F]**	**Number of Rows**	**WVFR [gal/min]**	**LDB [° F]**	**CEC [$/kW]**
7	7,469	49.0	3	6,770	40.0	312.70
8	**7,655**	**47.5**	**3**	**3,385**	**40.0**	**310.60**
9	7,658	46.3	3	2,279	40.0	310.80
10	7,674	45.3	2	5,527	40.0	311.20

Table 5.4.2: Selected Characteristics of Systems Designed to Determine Optimum Capacity Split for Power Augmentation Mode Power Plant Operation with Evaporative Cooling

5.5 The Cooling System Design Process: Third Example

The EES section models CWSL.size and ISL.size were used to design all systems based on a combination of chilled water and ice storage. The same optimized cooling coil configuration was used for all such systems within each general case, regardless of power plant daily load profile. The third inlet air cooling system to be discussed in detail relies on both storage media, and was designed for the eight hour duration peaked power plant load profile. The dimensions and maximum water volumetric flow rate for each cooling coil were thus set equal to the values corresponding to the optimum capacity split shown in Table 5.4.1. Since the desired electric power output increases linearly from 1:00 p.m. to 5:00 p.m. and then decreases linearly from 5:00 p.m. until 9:00 p.m., the total water mass flow rate through the two cooling coils must also increase and decrease over the same time intervals. Systems designed for peaked power plant load profiles differ in this regard from systems designed for step load profiles, which do not exhibit pronounced variations in cooling coil water mass flow rates.

Although the dimensions and maximum water mass flow rate for the first cooling coil had already been specified, it was necessary to consider an additional parameter in the

TRNSYS model that determines the instantaneous water mass flow rate: "CV1min", the minimum value of the first cooling coil pump control variable. The parameter "CV1min" needed to be set to the minimum value that would ensure that the minimum water capacitance rate would be greater than the air capacitance rate, as required by the TRNSYS cooling coil model. As "CV1min" decreases, the size of the chiller and water storage tank decrease. The EES section model CWSL.size was used to determine the ratio between the minimum and maximum water mass flow rates through the cooling coil, "g1min", to be 0.17. "CV1min" was set equal to 0.20 in the TRNSED input file to allow for a small margin of safety.

The dimensions and maximum water mass flow rate for the second cooling coil were likewise specified at the outset in accordance with Table 5.4.1, but it was necessary to consider another additional TRNSYS parameter: "CV2min", the minimum value of the second cooling coil pump control variable. This parameter also needed to be set to the smallest value that would ensure that the minimum water capacitance rate in the second cooling coil would be greater than the air capacitance rate. The EES program ISL.size calculated "g2min", the ratio between the minimum and maximum water mass flow rates for the second cooling coil, to be 0.41. "CV2min" was set equal to 0.50 to allow for a small margin of safety. This parameter has much less influence on the size of the ice harvester than "CV1min" has on the size of the chiller and water storage tank.

The parameter "Dhrs" was initially set equal to six in the EES chilled water storage loop model. In this case, there is a difference between the actual and effective number of hours of daily cooling coil operation. The effective number of hours of daily cooling coil operation is defined as the total amount of water that would have to enter the cooling coil at the chiller set point temperature and leave the cooling coil at the minimum leaving water temperature calculated by TRNSYS that would result in the actual integrated daily cooling coil load, divided by the maximum water mass flow rate. For this power plant load profile, both the instantaneous water mass flow rate out of the bottom of the storage tank and the cooling coil leaving water temperature vary significantly in order to meet the varying cooling load. The parameter "CChrs" was set equal to ten in the EES ice storage loop model. The effective number of weekly hours of operation for the second cooling coil is much smaller than for the first cooling coil, since the ice storage loop is only needed for the uppermost portion of the power plant load profile.

Values calculated by the EES section models were entered into the TRNSED input file, and a daily TRNSYS simulation was run. Graphs showing the water mass flow rate through both cooling coils and the leaving water temperature from the first cooling coil were prepared using TRNSHELL. The water mass flow rates for both cooling coils, "CCMF1" and "CCMF2", are shown in Figure 5.5.1 for a design day. The leaving water temperature for the first cooling coil is shown in Figure 5.5.2 for the same time period. In this example, the leaving water temperature for the first cooling coil varies significantly due to the pronounced variations in the water mass flow rate.

Plots were also prepared of the leaving dry bulb temperatures for both cooling coils, "LDB1" and "LDB2"; these are shown in Figure 5.5.3 for the design day. As expected, the minimum leaving dry bulb temperature from the first cooling coil is 47.2° F, while the minimum leaving dry bulb temperature from the second cooling coil is 40.0° F. Finally, plots of the net electric power output both with and without inlet air cooling, "NEP" and "EPNC" respectively, were generated and are shown in Figure 5.5.4. The actual net electric power output with inlet air cooling for the design day is equal to the desired net electric power output at each simulation time step.

After determining the appropriate values of "CV1min" and "CV2min" and generating Figures 5.5.1 - 5.5.4, it was necessary to size the remaining components of the chilled water storage and ice storage loops. The chiller, cooling tower, ice harvester, pump. and pipe sizes were found as described previously in sections 5.2 and 5.3 by adjusting the values of "Dhrs"

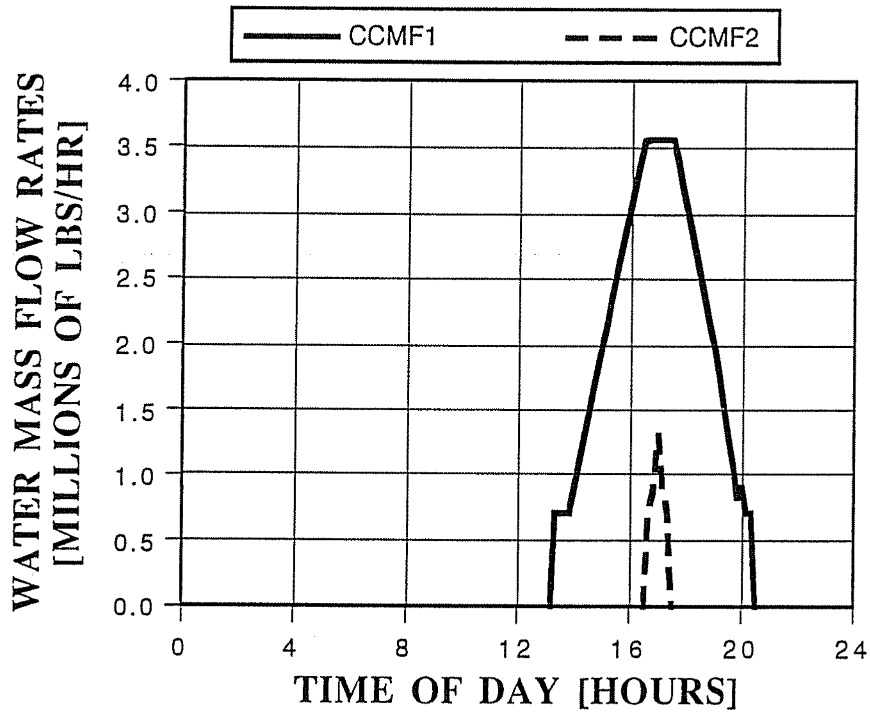


Figure 5.5.1: Design Day Cooling Coil Water Mass Flow Rates

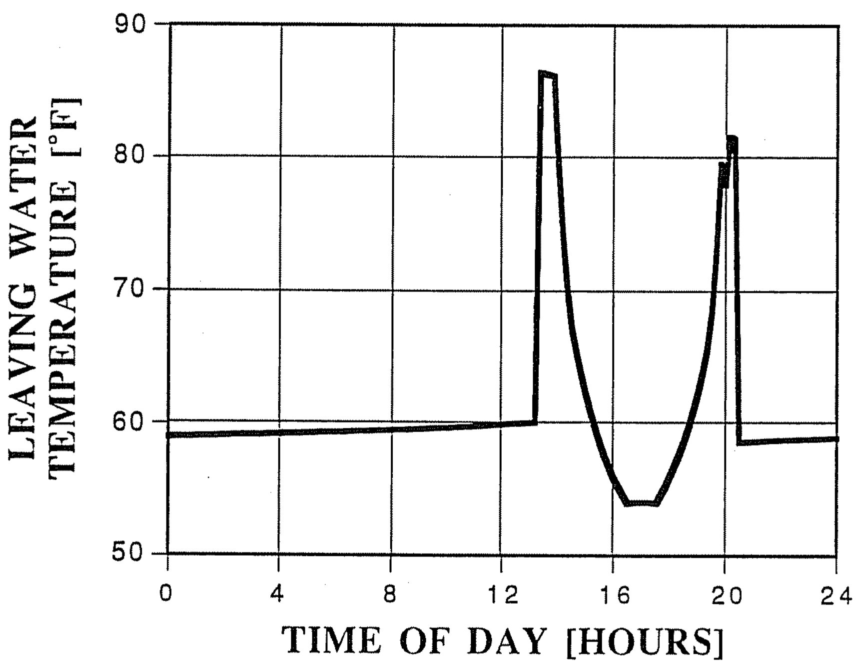


Figure 5.5.2: Design Day Leaving Water Temperature for First Cooling Coil

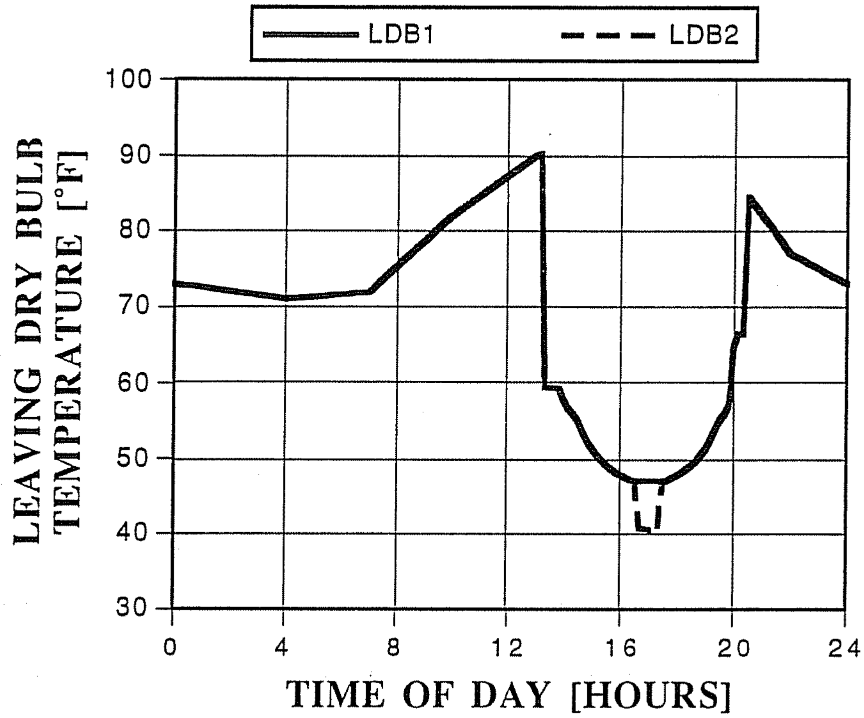


Figure 5.5.3: Design Day Cooling Coil Leaving Dry Bulb Temperatures

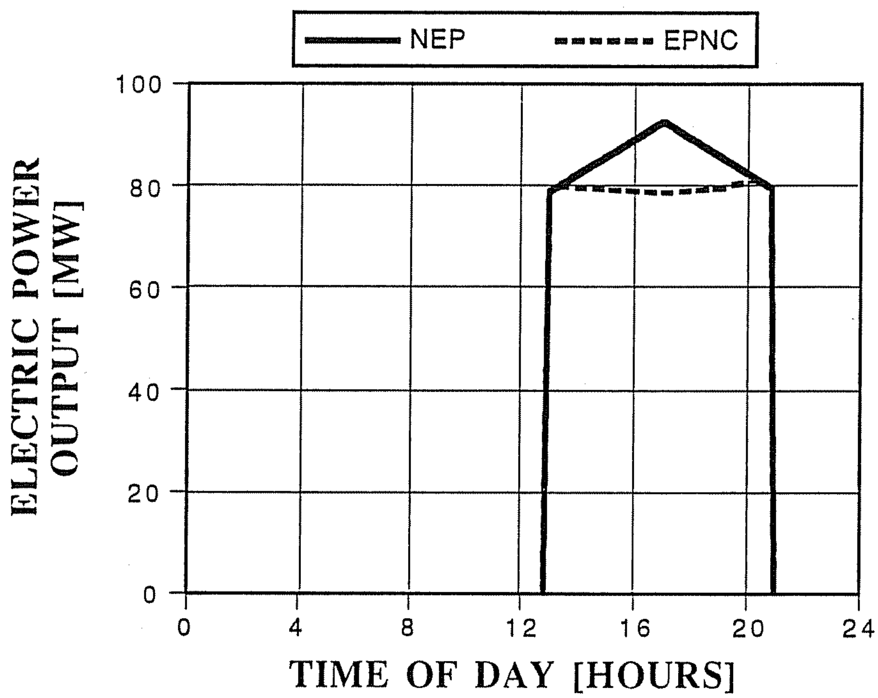


Figure 5.5.4: Design Day Electric Power Output with and without Inlet Air Cooling

and "CChrs" in the EES section models and running weekly TRNSYS simulations. The maximum chiller evaporator inlet temperature, "MLWT", was set 0.5° F above the minimum leaving water temperature from the first cooling coil of 54.1° F, as indicated by Figure 5.5.2. The EES chilled water storage section model assumes that the quantity of water passing through the cooling coil in the period "Dhrs" exits at a temperature less than or equal to "MLWT". Since the cooling coil leaving water temperature and hence the temperature at the top of the discharged chilled water storage tank are significantly higher than "MLWT" for the peaked power plant load profile, a fraction of the water from the chiller evaporator outlet must be diverted back to the evaporator inlet (rather than re-entering the storage tank) to maintain the entering evaporator water temperature equal to "MLWT". TRNSYS calculates this diverted fraction at each simulation time step as described in section 4.1. Consequently, the chilled water storage tank size calculated by the EES program based on "Dhrs" is always too high. Careful examination of Figures 5.5.1 and 5.5.2 indicates that the EES program "oversizes" the chilled water storage tank by a factor of 1.32. Thus 76% of the value for the chilled water storage tank capacity calculated by CWSL.size was entered into the TRNSED input file, along with the remaining parameters from the solution sheets of CWSL.size and ISL.size.

A value of "Dhrs" in the EES program that was too small resulted in the complete discharge of the chilled water storage tank in less than eight hours, as indicated by a plot of the cooling coil leaving water temperature. To ensure a total storage tank volume 5% larger than the usable storage tank volume, "Dhrs" was set to a value 5% larger than the minimum required for complete discharge at the end of the eight hour power plant load duration. The final value of "Dhrs" was found to be 5.3; the final value of "CChrs" resulting in the desired maximum ice storage tank discharge fraction was found to be 4.4. After determining the appropriate initial temperature of the upper portion of the chilled water storage tank, final graphs of both the average chilled water storage temperature and the ice inventory were prepared for the design week. These were similar to Figures 5.3.2 and 5.2.3 respectively, and are not reproduced here. The EES section models CWSL.size and ISL.size appear in Appendix B with their respective solution sheets. The TRNSED input file for this system appears in Appendix D. This concludes the discussion of the third and final example of combustion turbine inlet air cooling system design.

5.6 Summary of the Inlet Air Cooling System Design Process

Since cooling coil loads do not vary significantly between 1:00 and 9:00 p.m., it was found that the cooling coil dimensions and water mass flow rates are not affected by the duration or shape of the power plant load profile. It was thus possible to use the same cooling coil designs as those discussed in the three examples presented above for the nine remaining inlet air cooling systems for the first general case. It was only necessary to size the balance of each cooling system properly by adjusting "CChrs", "Dhrs", "CV1min", "CV2min", and the cooling tower fan power requirement in the EES section models as appropriate and using the resulting system parameters in weekly TRNSYS simulations. The initial temperature of the upper portion of the chilled water storage tank was always set equal to the final value at the end of the design week, and "CChrs" was always adjusted so that the maximum ice storage tank discharge fraction equaled 0.80.

It was necessary to redesign all cooling coils for the second general case, since the evaporative cooling unit both increases the total cooling coil load slightly and results in enhanced cooling coil performance. This was done using the same procedures outlined in the above three examples: "num", "LWT", and "Nrows" were varied in the EES section models to determine the dimensions and water mass flow rates for the cooling coils. Ten cooling coil rows are used for systems based on only one thermal storage medium; a total of eleven cooling coil rows are used for systems based on both storage media as discussed in section 5.4. The final leaving dry bulb temperatures are the same as those in the first general case.

After determining an acceptable cooling coil design for each cooling system in the second general case, the balance of each system was sized following the same procedures used for the first general case. Key component sizes and costs for each of the 24 final cooling system designs are presented in the next two sections.

5.7 Results of the Cooling System Design Process: Component Sizes

Table 5.7.1 summarizes the refrigeration equipment and storage tank sizes for the first general case. The cooling coil face area does not differ significantly for different systems; it varies between 1,460 and 1,481 square feet due to slight differences in the air mass flow rate for different compressor stage inlet air temperatures. Systems based on a single thermal storage medium have ten cooling coil rows; systems based on both thermal storage media have a total of twelve cooling coil rows in accordance with Table 5.4.1.

Storage Media	Load Profile	LDB2 [° F]	Chiller Size [Tons]	Ice Harvester Size [Tons]	Water Storage Tank Size [gallons]	Ice Storage Tank Size [gallons]
Water	4 hr step	46	1,241	---	1,775,000	---
Ice	4 hr step	40	---	812	---	1,332,000
Water/Ice	4 hr step	40	1.220	115	1,780,000	188,000
Water	6 hr step	46	1.861	---	2,663,000	---
Ice	6 hr step	40	---	1,213	---	1,989,000
Water/Ice	6 hr step	40	1.829	172	2,671,000	282,000
Water	8 hr step	46	2,480	---	3,551,000	---
Ice	8 hr step	40	---	1,616	---	2,650,000
Water/Ice	8 hr step	40	2,438	228	3,561,000	374,000
Water	8 hr peaked	46 (min)	1.644	---	1,788,000	---
Ice	8 hr peaked	40 (min)	---	1,192	---	1,955,000
Water/Ice	8 hr peaked	40 (min)	1677	24	1,861,000	40,000

Table 5.7.1: Selected Cooling System Component Sizes for First General Case

The sizes of the chiller, ice harvester, and storage tanks all increase linearly as the duration of the step power plant load increases. The total refrigeration capacities for the three storage capacity splits are shown as functions of power plant load duration in Figure 5.7.1. Refrigeration capacities of systems designed for the eight hour peaked power plant load profile are not plotted.

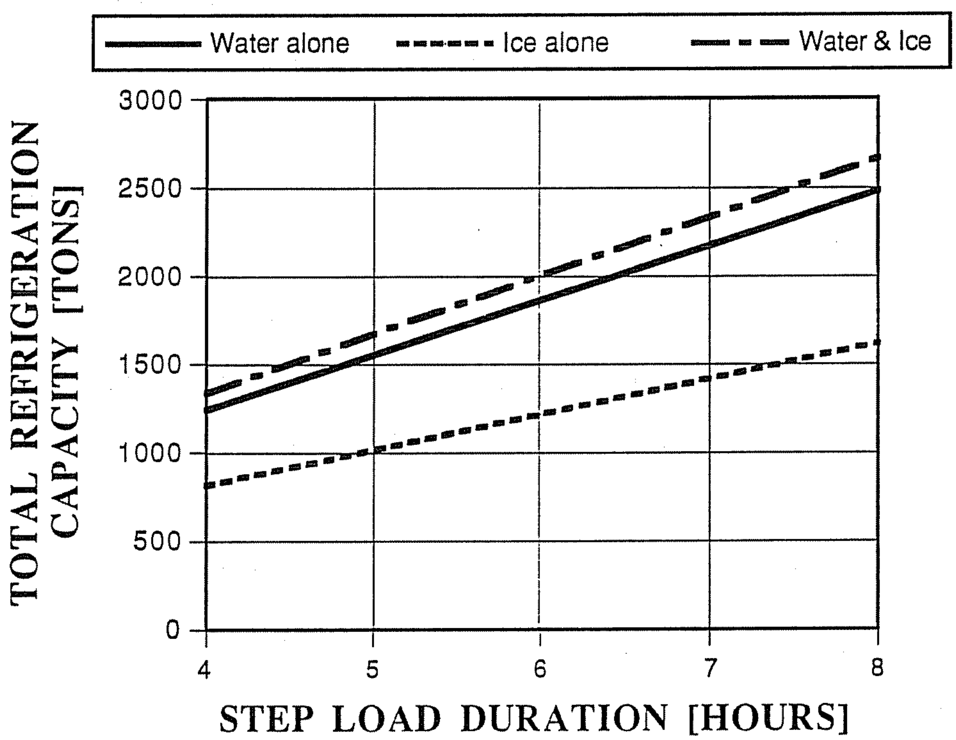


Figure 5.7.1: Total Refrigeration Capacities for Step Load Profiles: First General Case

As Table 5.7.1 and Figure 5.7.1 clearly indicate, the total refrigeration capacity for systems based on ice storage alone is always less than the total refrigeration capacity for the systems based either entirely or partially on chilled water storage. This is because the ice harvester is designed to operate 123 hours per week, while the chiller only operates 75 hours per week. For the equipment operating schedules considered, an ice harvester need only have roughly 60% of the capacity of a chiller in order to meet the same cooling coil load. The fact that the ice harvester has a higher duty cycle than the chiller partially offsets the higher cost of the ice harvester per ton of refrigeration capacity.

The total refrigeration capacity for systems based on a combination of ice and chilled water storage is greater than that for systems based on chilled water storage alone for all load profiles considered. This is because systems based on the former storage capacity split cool the inlet air stream to 40° F, while systems based on the latter storage capacity split only cool the inlet air stream to 46° F. Systems based on both storage media of course result in greater power plant capacity enhancement than do systems based on chilled water storage alone.

Although the water storage tank must only be large enough to store sufficient chilled water for one design day, the ice storage tank must be large enough to store over half of the ice needed for the entire design week. This difference in storage requirement partially offsets the advantage that ice has over water in terms of the stored energy to volume ratio. As can be seen from Table 5.7.1, the water storage tank is only about 34% larger than the ice storage tank for systems designed for the three step power plant load profiles and based on only one storage medium. In the case of systems based on both thermal storage media, the combined storage tank volume is roughly 48% larger than the storage tank required by the system based on ice alone.

For systems based on both chilled water and ice storage, the ratio of the ice harvester capacity to the chiller capacity for the three step power plant load profiles is 0.094; however, for the eight hour peaked power plant load profile, this ratio drops to 0.017. For the peaked load profile, the cooling coil fed by water circulating through the ice storage tank only needs to be turned on for about one hour each day, as indicated by Figure 5.5.1. The ice harvester thus needs to meet a much smaller fraction of the total daily cooling load in this case than for the step power plant load profiles.

The key distinction between the two general cases is the presence of the evaporative cooler in the inlet air flow stream in the second general case. As noted in section 5.1, the evaporative cooler increases both the total cooling coil load and the cooling coil effectiveness slightly. Refrigeration equipment and storage tank sizes for the second general case are summarized in Table 5.7.2. As in the first general case, cooling coil face areas do not differ widely for these cooling systems; they vary between 1,426 and 1,445 square feet. Systems based on a single thermal storage medium have ten cooling coil rows; systems based on both thermal storage media have eleven cooling coil rows in accordance with Table 5.4.2. Systems based on both storage media require one less cooling coil row in the second general case than in the first due to the increase in cooling coil effectiveness associated with the evaporative cooler.

Storage Media	Load Profile	LDB2 [°F]	Chiller Size [Tons]	Ice Harvester Size [Tons]	Water Storage Tank Size [gallons]	Ice Storage Tank Size [gallons]
Water	4 hr step	46	1,221	---	1,616,000	---
Ice	4 hr step	40	---	805	---	1,320,000
Water/Ice	4 hr step	40	1,192	123	1,923,000	202,000
Water	6 hr step	46	1,830	---	2,424,000	---
Ice	6 hr step	40	---	1,207	---	1,979,000
Water/Ice	6 hr step	40	1,787	184	2,885,000	302,000
Water	8 hr step	46	2,439	---	3,232,000	---
Ice	8 hr step	40	---	1,609	---	2,639,000
Water/Ice	8 hr step	40	2,382	244	3,847,000	400,000
Water	8 hr peaked	46 (min)	1,678	---	1,689,000	---
Ice	8 hr peaked	40 (min)	---	1,286	---	2,109,000
Water/Ice	8 hr peaked	40 (min)	1,757	42	2,156,000	69,000

Table 5.7.2: Selected Cooling System Component Sizes for Second General Case

The total refrigeration capacity for the first nine systems listed is shown as a function of power plant load duration in Figure 5.7.2 below. As in the first general case, the total refrigeration capacity for systems based on ice storage alone is always lower than the total refrigeration capacity for the systems based the other two storage capacity splits, and systems based on both storage media have the highest total refrigeration capacities. Since the cooling coil load for each system in the second general case is roughly equivalent to the cooling coil load for the corresponding system in the first general case, corresponding total refrigeration capacities are also roughly equivalent.

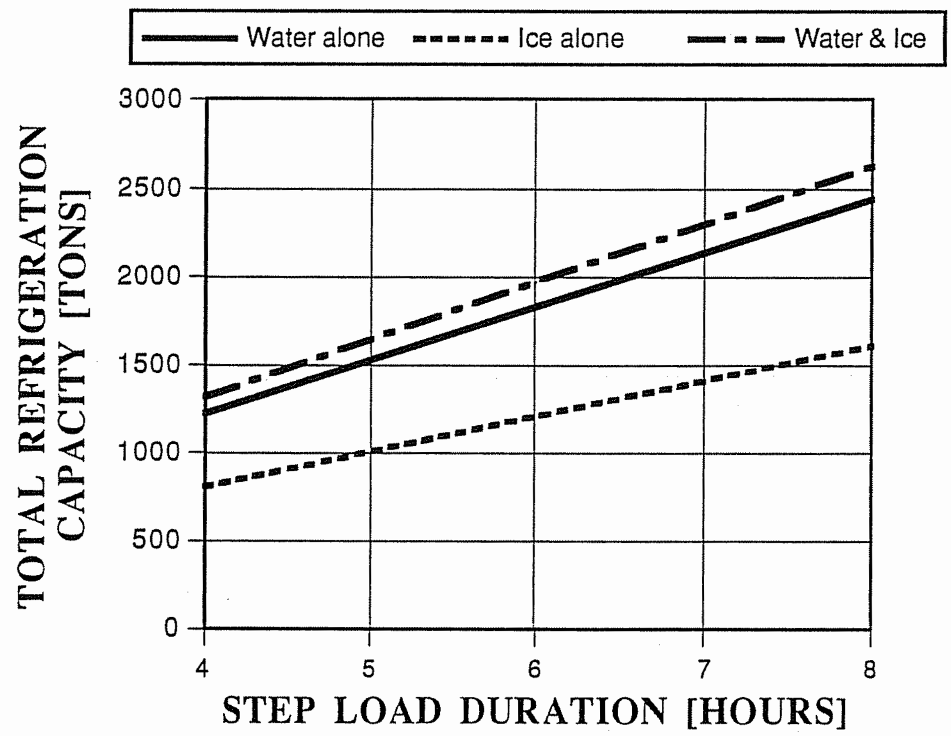


Figure 5.7.2: Total Refrigeration Capacities for Step Load Profiles : Second General Case

Since the cooling coil effectiveness is higher in the second general case than in the first, and since the number of cooling coil rows is the same for systems based on chilled water storage alone, the cooling coil leaving water temperature is higher in the second general case than in the first for systems based only on chilled water storage. Therefore, the storage tank volume is smaller for these systems in the second general case than in the first. For the three step power plant load profiles, the chilled water storage tank is only 22% larger than the ice storage tank for systems based on a single storage medium. Systems based on both ice and water storage, on the other hand, have one less cooling coil row in the chilled water storage loop in the second general case than in the first. Hence the leaving water temperature is slightly lower and the chilled water storage tank volumes are slightly higher in the second general case than in the first for this capacity split. For the three step power plant load profiles, the combined storage tank volume of systems based on both storage media exceeds the storage tank volume of systems based on ice alone by 61%.

For systems based on both chilled water and ice storage in the second general case, the ratio of the ice harvester capacity to the chiller capacity for the three step power plant load profiles is 0.103. For the eight hour peaked power plant load profile, this ratio is 0.024. As in the first general case, the ice harvester contributes significantly less to the total daily cooling coil load for the peaked power plant load profile than for the step load profiles.

5.8 Results of the Cooling System Design Process: Costs

Cost data were obtained for each finalized system design. The installed system cost can be broken down into four categories: refrigeration equipment (chiller, cooling tower, and/or ice harvester) cost, storage volume cost, cooling coil cost, and the cost of pumps and pipes. The cost breakdown for each of the three storage capacity splits is shown in Figure 5.8.1 for the four hour step power plant load profile in the first general case. Cooling coil costs are roughly equivalent for all three storage capacity splits, and range from 14% to 24% of the installed system cost for the four hour step power plant load profile. Cooling coil costs remain constant as the power plant load duration increases; all other costs increase as load duration increases. For systems based on ice storage alone, the refrigeration equipment cost dominates all other costs. For systems based either partly or entirely on water storage, the refrigeration equipment costs and the storage volume costs are roughly comparable. Pump and pipe costs are higher for systems based partly or entirely on water storage than for systems based on ice storage alone. That component group represents less than 8% of the system cost for all systems shown in Figure 5.8.1, and accounts for a decreasing fraction of the cost of each system as the power plant load duration increases. However, for cooling systems requiring pipe lengths significantly longer than those specified in this study, pump and pipe costs will represent a significantly larger share of total system costs.

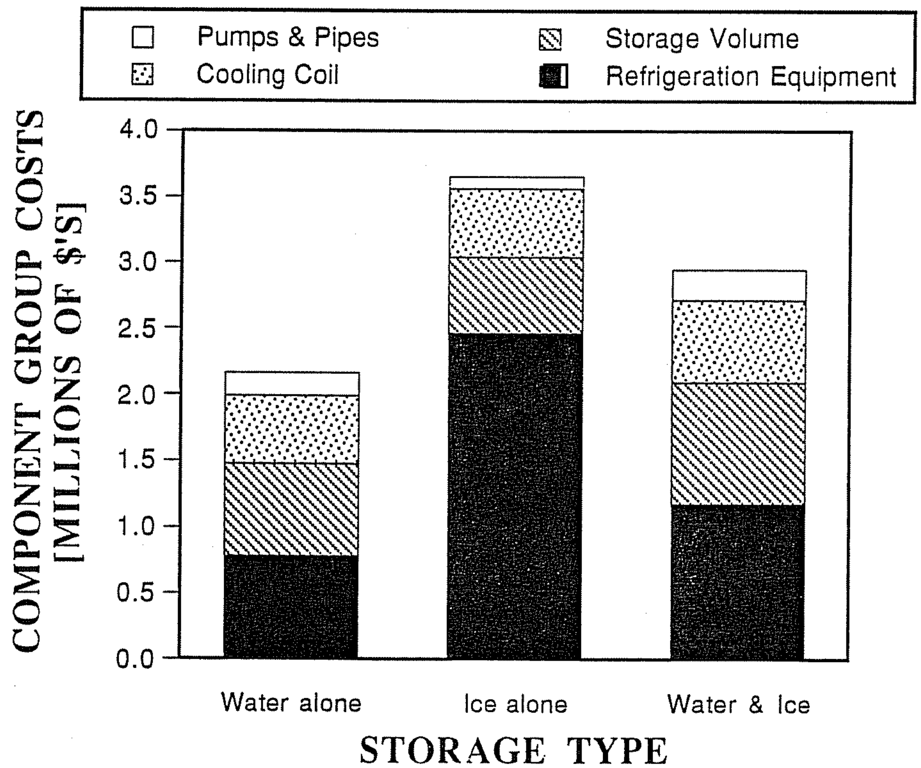


Figure 5.8.1: Component Costs for Four Hour Step Load Profile: First General Case

Table 5.8.1 shows the installed cost, the power plant capacity increase, and the capacity enhancement cost, "CEC", associated with each inlet air cooling system for the first general case. As discussed in section 3.7, the capacity enhancement cost is defined as the installed cost of the cooling system divided by the power plant capacity increase in kilowatts due to inlet air cooling. Table 5.8.1 also shows the peak capacity enhancement cost, "PCEC", which is defined as the cost difference between the cooling system considered and one based on chilled water storage alone for the same load profile, divided by the difference in the capacity enhancement for those two systems in kilowatts. This quantity is a measure of the cost of the incremental power plant capacity enhancement associated with cooling systems based either entirely or partially on ice storage relative to systems based on chilled water storage alone. The percent capacity increase due to inlet air cooling is 16.0% for systems based on chilled water storage alone, 17.9% for systems based on ice storage alone, and 17.8% for systems based on both storage media.

Storage Media	**Load Profile**	**System Cost [$]**	**Cap. Inc. [kW]**	**CEC [$/kW]**	**PCEC [$/kW]**
Water	4 hr step	2,165,000	12,578	172	---
Ice	4 hr step	3,655,000	14,051	260	1012
Water/Ice	4 hr step	2,948,000	13,964	211	565
Water	6 hr step	3,204,000	12,578	255	---
Ice	6 hr step	5,030,000	14,051	358	1,240
Water/Ice	6 hr step	4,178,000	13,964	299	703
Water	8 hr step	4,848,000	12,578	385	---
Ice	8 hr step	6,386,000	14,051	454	1,044
Water/Ice	8 hr step	5,973,000	13,964	428	812
Water	8 hr peaked	2,652,000	12,578	211	---
Ice	8 hr peaked	4,958,000	14,051	353	1,566
Water/Ice	8 hr peaked	3,160,000	13,964	226	366

Table 5.8.1: Cooling System Costs for First General Case

Despite the fact that the total refrigeration capacity and total storage tank size increase linearly with power plant load duration for all but the peaked load profile, installed cooling system costs do not increase linearly. This non-linearity is most pronounced in the case of systems based on chilled water storage alone. In general, component costs do not increase linearly with size, and chiller costs exhibit a fairly substantial "diseconomy of scale", varying between \$174/ton for the smallest unit and \$210/ton for the largest unit (see Equation 3.7.1 and Table 5.7.1). The chiller is the most costly individual component in all systems based either partially or entirely on chilled water storage, and thus has a significant influence on the total system cost. System costs are shown in Figure 5.8.2 as functions of power plant load duration for the three step load profiles.

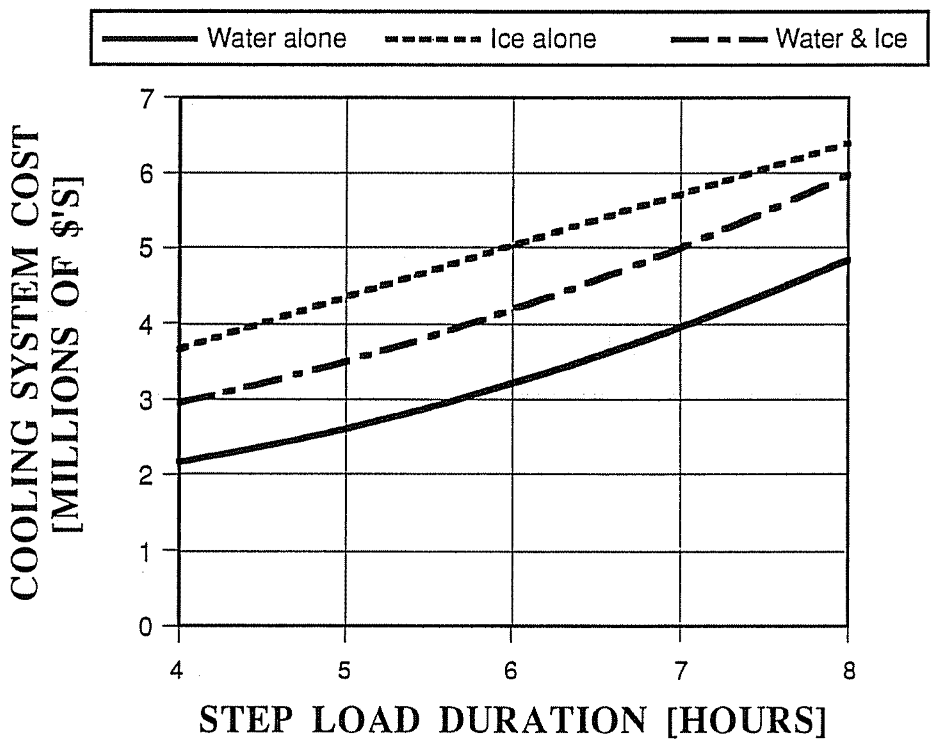


Figure 5.8.2: Cooling System Costs for Step Load Profiles: First General Case

Table 5.8.1 shows that inlet air cooling systems based on ice storage alone result in a power plant capacity increase nearly 12% greater than that provided by systems based on chilled water storage alone. Cooling systems based on both storage media provide slightly less capacity enhancement than those featuring ice storage alone due to the higher cooling coil pump power requirements for the former system type. The increased capacity enhancement associated with systems including an ice storage component partially offsets their increased cost with respect to systems based on chilled water storage alone. Nevertheless, systems based on chilled water storage alone yield the lowest capacity enhancement cost for each power plant load profile considered. In the case of the eight hour peaked power plant load profile, the capacity enhancement cost of the system based on both ice and water storage closely approaches that of the system based on chilled water storage alone. For all load profiles, both the capacity enhancement cost and the peak capacity enhancement cost are significantly lower for systems using the two storage media than for systems using ice storage alone. Capacity enhancement costs are shown for the three step power plant load profiles as functions of the load duration in Figure 5.8.3.

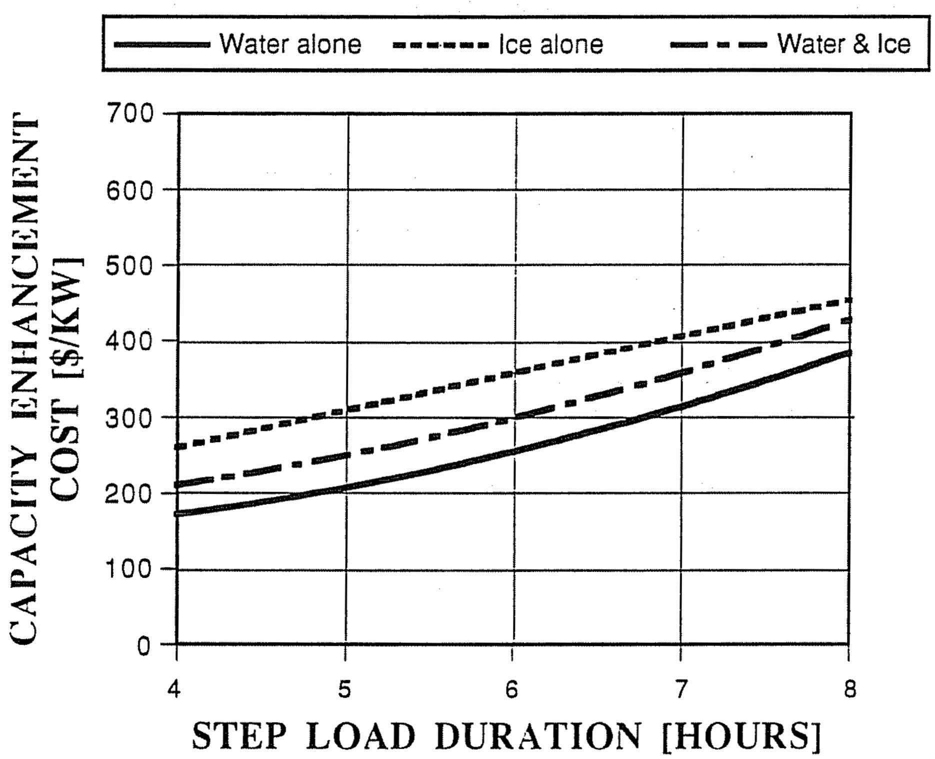


Figure 5.8.3: Capacity Enhancement Costs for Step Load Profiles: First General Case

The ratios between the costs of the different component groupings and the installed system cost for a given power plant load profile are roughly equivalent for the two general cases. Cost data for the second general case are shown in Table 5.8.2. The evaporative cooling unit is part of the original power plant, and its cost is therefore not included in the data presented below. System costs are slightly lower in the second than in the first general case. However, the associated power plant capacity increases are also lower, since power plant performance with the complete inlet air cooling system is compared to power plant performance with the evaporative cooler, rather than to power plant performance with no inlet cooling equipment. Here the percent capacity increase due to inlet air cooling is only 9.5% for systems based on chilled water storage alone, 11.3% for systems based on ice storage alone, and 11.2% for systems based on both storage media. Therefore, capacity enhancement costs are higher in the second general case than in the first for corresponding storage capacity splits and power plant load profiles. It follows that there is no good reason

Storage Media	Load Profile	System Cost [$]	Cap. Inc. [kW]	CEC [$/kW]	PCEC [$/kW]
Water	4 hr step	2,085,000	8,015	260	---
Ice	4 hr step	3,616,000	9,508	380	1,025
Water/Ice	4 hr step	2,937,000	9,458	311	590
Water	6 hr step	3,080,000	8,015	384	---
Ice	6 hr step	4,995,000	9,508	525	1,283
Water/Ice	6 hr step	4,142,000	9,458	438	736
Water	8 hr step	4,654,000	8,015	581	---
Ice	8 hr step	6,348,000	9,508	668	1,135
Water/Ice	8 hr step	5,862,000	9,458	620	837
Water	8 hr peaked	2,658,000	8,015	332	---
Ice	8 hr peaked	5,263,000	9,508	554	1,745
Water/Ice	8 hr peaked	3,377,000	9,458	357	498

Table 5.8.2: Cooling System Costs for Second General Case

to install both an evaporative cooler and a thermal storage based inlet air cooling system.

As in the first general case, system costs do not increase linearly with the power plant load duration, even though total refrigeration capacities and total storage tank capacities do increase linearly. System costs are shown in Figure 5.8.4 as functions of power plant load duration for the three step load profiles. Inlet air cooling systems based either partially or entirely on ice storage result in power plant capacity increases that are 18 - 19% higher than those provided by systems based only on chilled water storage, which is significantly greater than in the first general case. For that reason, the percent differences between the capacity enhancement costs for systems based either partially or entirely on ice storage and the capacity enhancement cost for systems based on chilled water storage alone are slightly lower in the second general case than in the first. Capacity enhancement costs are shown as functions of power plant load duration for the three step profiles in Figure 5.8.5.

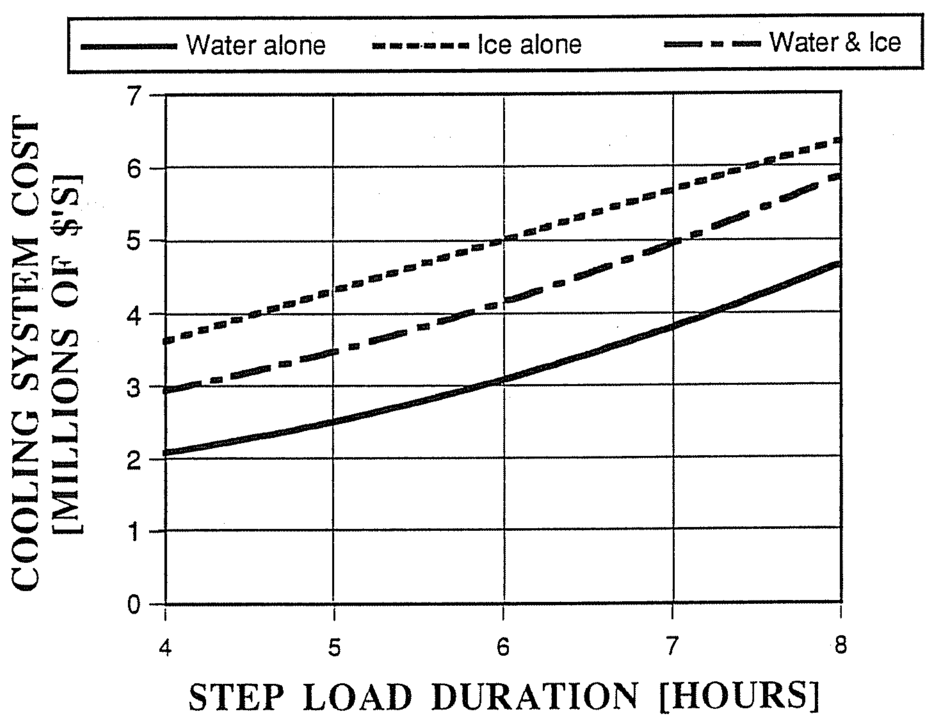


Figure 5.8.4: Cooling System Costs for Step Load Profiles: Second General Case

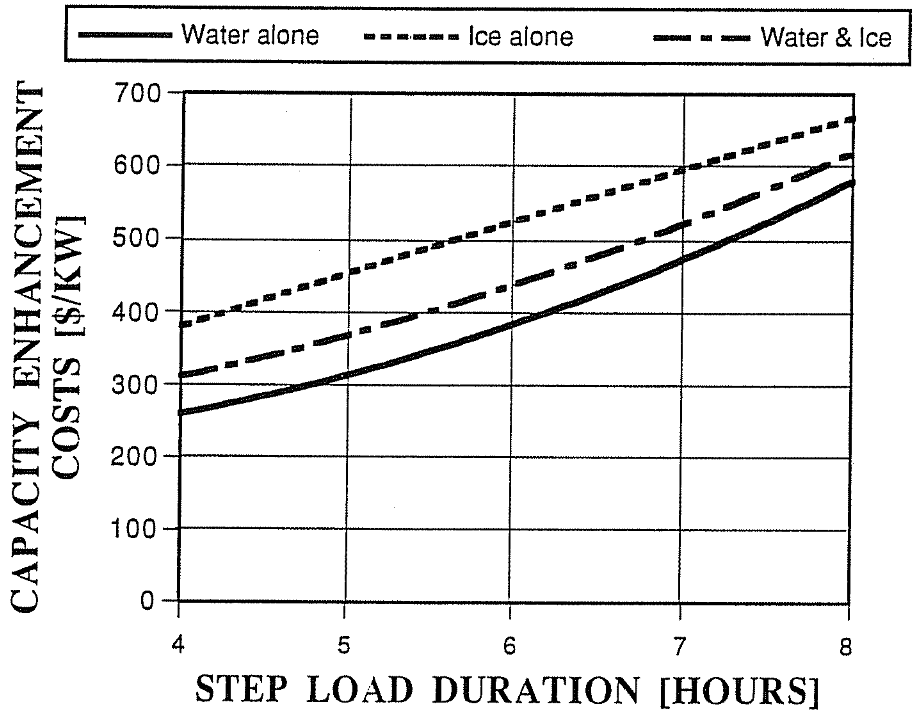


Figure 5.8.5: Capacity Enhancement Costs for Step Load Profiles: Second General Case

Despite the greater percent benefit associated with the use of at least some ice storage in the second general case, systems based on chilled water storage alone still yield lower capacity enhancement costs than the other two storage capacity splits. Peak capacity enhancement costs for systems based partially or entirely on ice storage are slightly higher in the second than in the first general case, and are 1.4 to 3.7 times greater for systems based on ice storage alone than for systems based on a combination of storage media.

The capacity enhancement cost and the peak capacity enhancement cost are not the only economic parameters of importance, however. A life cycle analysis results in a parameter that is useful in deciding whether the installation of any inlet cooling system is warranted. That parameter is the cost of the incremental electric power produced with inlet air cooling based on a specified system payback period, and is examined in Chapter 6.

CHAPTER 6: COOLING SYSTEM SEASONAL SIMULATIONS

Seasonal simulations were performed for the 24 combustion turbine inlet air cooling systems discussed in the previous chapter. Based on a generation forecast for the power plant considered, each system was assumed to cool the inlet air only 32 hours per year on average. The results of each seasonal simulation were used to calculate the cost of the incremental electric power produced with inlet air cooling based on a cooling system payback period of 20 years. This chapter describes the assumptions underlying the inlet air cooling system life cycle analysis, and presents the results of that analysis for all 24 cooling systems designed.

6.1 Cooling Season Characteristics

The 1995 annual forecasted electric output for the power plant considered is 6,115 Megawatt-hours. Of that total, 2,306 Megawatt-hours are to be generated in the period during which air inlet cooling would likely be required: July, August, and September. Assuming that the combustion turbine is to be run at its full capacity of 94 Megawatts with power augmentation and inlet air cooling, the predicted generation requirement implies that the cooling coils would only need to operate for a total of 24.5 hours in 1995.

The period of economic analysis is 20 years, during which time the demand for electricity is expected to increase 2.5% annually. Inlet air cooling is thus expected to be necessary for a total of 40.4 hours in the year 2014. The average annual requirement for inlet air cooling over the 20 year period of analysis is thus roughly 32 hours at full capacity. This average annual expected requirement for inlet air cooling is considerably lower than those for the combustion turbine power plants in either Lincoln, Nebraska or Fayetteville, North Carolina, which are 60 hours and 80 hours, respectively (Beaty 1994).

The combustion turbine was assumed to operate for 32 hours during each cooling season regardless of the duration or shape of the daily power plant load profile. Thus, cooling systems designed for a four hour step power plant load profile were assumed to operate eight days per year, systems designed for a six hour step load profile were assumed to operate five and one-third days per year, and systems designed for both the eight hour step and eight hour peaked load profile were assumed to operate four days per year. The amount of electric energy produced during the cooling season is not the same in all cases, since it depends on the compressor inlet air temperature, the ratio of the water and fuel mass flow rates into the combustion chamber, and the shape of the power plant load profile.

Design day dry and wet bulb temperature profiles were used for all cooling season simulations (see Figure 4.3.1). Another option would have been to use actual weather data taken near the site of the power plant for the four to eight hottest days in a single summer. However, it was not clear how to select a summer that would be most representative of the 20 year period of economic analysis. The use of "Typical Meteorological Year" (TMY) weather data would not have been a satisfactory solution to this problem, since TMY weather data do not include the temperature extremes for which the inlet air cooling systems are designed. Although the use of design day temperature profiles for the cooling season simulations results in slightly over predicting the benefits associated with inlet air cooling, it enables comparisons between different system designs to be made on an equal basis.

6.2 Economic Assumptions and Parameters

The cooling system life cycle economic analysis is based on two key assumptions as discussed briefly in section 3.7. First, it is assumed that a demand exists for all of the energy that could be produced by the power plant with inlet air cooling for all load profiles during the 32 hours of power plant operation per cooling season. Second, it is assumed that the utility operating the combustion turbine can always purchase electricity at some peak price "C_{PE}", in dollars per kilowatt-hour, from another utility to meet that demand in the event that no inlet air cooling system is present.

Based on these two assumptions, the annual savings provided by an inlet air cooling system is given by Equation 3.7.10, which is repeated here for convenience:

$$Savings_{ann} = C_{PE}*\Delta EE_{NC} - C_F*\Delta Fuel - C_{OPE}*EE_{OP} \qquad (6.2.1)$$

Here "ΔEE_{NC}" is the annual difference in kilowatt-hours between the electric energy produced with inlet air cooling and the electric energy that could be produced without inlet air cooling, "C_F" is the cost of the fuel in dollars per pound, "ΔFuel" is the annual excess fuel consumed by the power plant due to the operation of the inlet cooling system in pounds, "C_{OPE}" is the cost of off-peak electricity used to charge the chilled water and/or ice storage tanks in dollars per kilowatt-hour, and "EE_{OP}" is the amount of electric energy consumed annually by the cooling system in kilowatt-hours. The term "ΔEE_{NC}" is calculated on the basis of the net electric power produced with inlet air cooling, which is equal to the gross combustion turbine electric output minus the cooling coil water pump power requirement.

The payback period for the inlet cooling system, "N_P", is defined as the number of years required for the discounted sum of the annual savings to equal the initial investment. Equation 3.7.9 gives the relationship between the initial investment, "$Cost_{system}$", the annual savings, the discount rate, "d", the fuel inflation rate, "i", and the payback period. That equation can be expressed in a more compact form as

$$Cost_{system} = PWF*Savings_{ann} \qquad (6.2.2)$$

where "PWF" is the "present worth factor", given by

$$PWF = [1/(d - i)]*\{1 - [(1 + i)/(1 + d)]^{N_P}\} \qquad (i \neq d) \qquad (6.2.3a)$$

$$PWF = N_P/(1 + i) \qquad (i = d) \qquad (6.2.3b)$$

Equations 6.2.1 and 6.2.2 can be solved for "C_{PE}", the wholesale price for on-peak electricity that results in a cooling system payback period "N_P". "C_{PE}" can also be identified with the cost of the incremental electric power produced with inlet air cooling based on a payback period "N_P". The result is Equation 6.2.4:

$$C_{PE} = (1/\Delta EE_{NC})*(Cost_{system}/PWF + C_F*\Delta Fuel + C_{OPE}*\Delta EE_{OP}) \qquad (6.2.4)$$

Since the TRNSYS combustion turbine component computes the electric output and fuel consumption rate both with and without inlet air cooling simultaneously, the cost calculator component can calculate "C_{PE}" directly. The TRNSYS simulation also provides ΔEE_{NC}, the cooling system cost, the cost of the excess fuel consumed by the power plant due to inlet cooling, and the cost of the off-peak electricity consumed by the refrigeration equipment. These values can be used to calculate "C_{PE}" separately.

The payback period was specified to be 20 years. Other economic parameters include: a discount rate of 10.17%, a fuel inflation rate of 5.50%, an off-peak electricity cost of $0.0124/kW-hr, and a fuel (natural gas) cost of $0.058/lb, or $2.55 per million BTU. The present worth factor, given by Equation 6.2.3a, is 12.41. A higher fuel inflation rate, a lower discount rate, or a longer payback period would all tend to increase the present worth factor. An increased present worth factor would decrease "C_{PE}", thereby making investment in an inlet air cooling system appear more favorable.

6.3 Seasonal Simulation Results

Table 6.3.1 presents the seasonal simulation results for the first general case (no evaporative cooling and base mode power plant operation), as well as the calculated value of "C_{PE}", the cost of the incremental electric power produced with inlet air cooling based on a 20 year payback period. The table shows the average annual difference between the electric energy produced with inlet air cooling and the electric energy that could be produced without inlet air cooling, "ΔEE_{NC}", the average annual excess fuel cost due to inlet air cooling, and the average annual cost of the off-peak electricity needed to operate the refrigeration equipment, "OPEC". Since neither the chiller nor the ice harvester fully re-charge their respective storage tanks in the seasonal simulation periods, it was necessary to estimate the annual off-peak electricity costs on the basis of the values calculated by TRNSYS for each simulation period. Table 6.3.1 corresponds to Table 5.8.1, which gives the capacity enhancement costs and peak capacity enhancement costs for the first general case.

Storage Media	**Load Profile**	ΔEE_{NC} **[kW-hr/yr]**	**Excess Fuel Cost [$/yr]**	**OPEC [$/yr]**	C_{PE} **[$/kW-hr]**
Water	4 hr step	380,000	9,880	1,210	0.49
Ice	4 hr step	438,000	11,410	1,710	0.70
Water/Ice	4 hr step	435,000	11,400	1,500	0.58
Water	6 hr step	383,000	9,940	1,190	0.70
Ice	6 hr step	429,000	11,120	1,580	0.97
Water/Ice	6 hr step	428,000	11,180	1,430	0.81
Water	8 hr step	371,000	9,670	1,260	1.08
Ice	8 hr step	418,000	10,910	1,730	1.26
Water/Ice	8 hr step	415,000	10,900	1,500	1.19
Water	8 hr peaked	197,000	5,290	940	1.12
Ice	8 hr peaked	201,000	5,440	1,270	2.02
Water/Ice	8 hr peaked	201,000	5,403	960	1.30

Table 6.3.1: Seasonal Simulation Results for First General Case

Investment in an inlet air cooling system becomes increasingly attractive as "C_{PE}" decreases below the peak wholesale electricity price. The values of "C_{PE}" reported in the above table are quite high in comparison to the current range of typical peak wholesale electricity prices for the mid-Western United States of \$0.075 - \$0.150 per kilowatt-hour. Assuming that electricity in that price range is always available to the utility operating the combustion turbine, the incremental electric energy produced with inlet air cooling is more expensive than electricity that could be purchased from another utility in all of the cases shown in Table 6.3.1. Installation of an inlet air cooling system would therefore not be warranted for any of the load profiles considered. Depending on the load profile, use of the combustion turbine during the cooling season would have to increase by a factor of 3.2 to 7.5 (to 100 to 240 hours per cooling season) in order to reduce "C_{PE}" to $0.15/kilowatt-hour for the most cost effective storage capacity split.

The cost of the incremental electric power produced with inlet air cooling based on a 20 year payback period depends strongly on the installed cost of the cooling system. Since the installed cooling system cost does not depend linearly on the power plant load duration, "C_{PE}" does not depend linearly on the power plant load duration either. "C_{PE}" is shown as a function of power plant load duration for the three step load profiles in Figure 6.3.1.

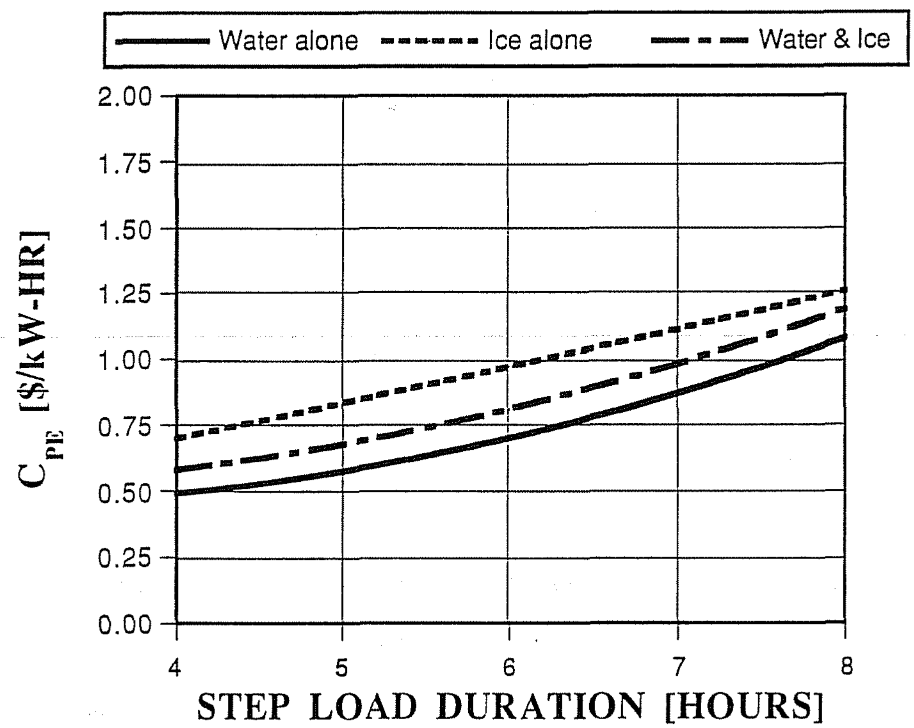


Figure 6.3.1: "C_{PE}" for Step Load Profiles: First General Case

Table 6.3.1 and Figure 6.3.1 show that cooling systems based on chilled water storage alone yield the lowest values of "C_{PE}" for the step power plant load profiles as well as for the peaked power plant load profile. Cooling systems based on ice storage alone yield significantly higher values of "C_{PE}" than do the other two storage capacity splits. For a given step power plant load profile, the ratios between the values of "C_{PE}" for different storage capacity splits are nearly identical to the ratios between the capacity enhancement costs.

For the eight hour peaked power plant load profile, the ratios between "C_{PE}" for the system based on chilled water storage alone and "C_{PE}" for either of the remaining storage capacity splits is much lower than the corresponding ratios of capacity enhancement costs. Despite the fact that the systems based either partially or entirely on ice storage increase the power plant capacity by 11% - 12% more than does the system based on chilled water storage alone, they only result in the production of 2% more electric energy per design day than does the system based on chilled water storage alone for the peaked power plant load profile. From the standpoint of energy production, operating the power plant at less than full capacity quickly cancels out any potential economic gains that might be realized by including an ice storage component.

The initial investment in the inlet air cooling system and the annual amount of electric energy generated with inlet cooling are by far the two most important factors in determining the value of "C_{PE}". However, "C_{PE}" also depends on the amount of excess fuel consumption due to inlet cooling and the amount of electricity required to operate the refrigeration equipment. Maintenance costs will also affect "C_{PE}", but are not considered in this analysis. As discussed in section 1.1, the overall conversion efficiency is generally higher with inlet air cooling than without inlet air cooling. Thus the total energy input, in terms of fuel and off-peak electricity, that is required to produce the incremental peak electric energy, "ΔEE_{NC}", is less than would be required to produce an equivalent increment without inlet cooling. In the first general case, the overall conversion efficiency for the three step power plant load profiles with inlet air cooling is approximately 0.267 for all storage capacity splits: for the eight hour peaked power plant load profile, the overall conversion efficiency with inlet air cooling is approximately 0.264 for all storage capacity splits. Without inlet air cooling, the overall conversion efficiency is 0.262. Hence, inlet air cooling increases the overall conversion efficiency by 0.8% - 1.9% in the first general case, depending on the shape of the power plant load profile.

Cooling season simulation results and values of "C_{PE}" for the second general case (evaporative cooling and power augmentation mode combustion turbine operation) are shown in Table 6.3.2. This table corresponds to Table 5.8.2, which provides cooling system installed cost, capacity enhancement cost, and peak capacity enhancement cost data for the second general case. As in Table 6.3.1, the total cost of off-peak electricity required to re-charge the storage tanks fully is estimated for all cooling systems based on output from the TRNSYS simulation.

Storage Media	Load Profile	ΔEE_{NC} [kW-hr/yr]	Excess Fuel Cost [$/yr]	OPEC [$/yr]	C_{PE} [$/kW-hr]
Water	4 hr step	254,000	6,890	1,240	0.69
Ice	4 hr step	303,000	8,200	1,700	0.99
Water/Ice	4 hr step	302,000	8,190	1,480	0.82
Water	6 hr step	254,000	6,880	1,160	1.01
Ice	6 hr step	303,000	8,180	1,570	1.36
Water/Ice	6 hr step	301,000	8,170	1,380	1.14
Water	8 hr step	252,000	6,820	1,260	1.52
Ice	8 hr step	300,000	8,130	1,720	1.74
Water/Ice	8 hr step	298,000	8,120	1,500	1.62
Water	8 hr peaked	135,000	3,790	1,000	1.62
Ice	8 hr peaked	147,000	4,190	1,370	2.92
Water/Ice	8 hr peaked	146,000	4,100	1,060	1.90

Table 6.3.2: Seasonal Simulation Results for Second General Case

The values of "C_{PE}" reported in Table 6.3.2 are even higher than those found in the first general case. Depending on the load profile, power plant use would have to increase by a factor of 4.6 to 10.8 (to 150 to 350 hours per cooling season) in order to reduce "C_{PE}" to \$0.15/kW-hr for the most cost-effective storage capacity split. As discussed in section 5.8, the reason why the thermal storage based inlet air cooling systems provide less economic benefit in the second than in the first general case is that combustion turbine performance with the complete inlet air cooling system is compared to combustion turbine performance with the evaporative cooler, rather than to combustion turbine performance with no inlet air cooling at all. The thermal storage based inlet air cooling systems provide less capacity enhancement and consequently less incremental electrical energy generation in the cooling season simulation period in the second than in the first general case.

Since cooling the inlet air stream from 46° F - 47.5° F to 40° F represents a greater percentage of the total cooling load (and results in a greater percentage of the total capacity enhancement) in the second general case than in the first, systems featuring ice storage become somewhat more economically competitive with systems based on chilled water storage alone. However, systems based on chilled water storage alone still yield the lowest values of "C_{PE}" for all power plant load profiles. Hybrid systems based on both storage media yield the next lowest "C_{PE}" values. For the three step power plant load profiles, the ratio of "C_{PE}" for systems based on chilled water storage alone to "C_{PE}" for systems based on both storage media ranges between 0.84 and 0.94. The cost of incremental power produced with inlet air cooling is shown for the three step power plant loads in Figure 6.3.2 below for all storage capacity splits.

As in the first general case, the system based on chilled water storage alone results in a much lower value of "C_{PE}" than do the systems based either partially or entirely on ice storage for the peaked power plant load profile. At full power plant capacity, systems featuring ice storage can enhance electrical energy production by 18% - 19% more than can systems based on chilled water storage alone. For the peaked power plant load profile, however, only 8% - 9% more electrical energy is produced per design day with systems based at least partially on ice storage than with systems based only on chilled water storage. Again, less than full utilization of the increased generating capacity made available by cooling systems based on ice storage reduces any potential advantage such systems have in relation to those based on chilled water storage alone

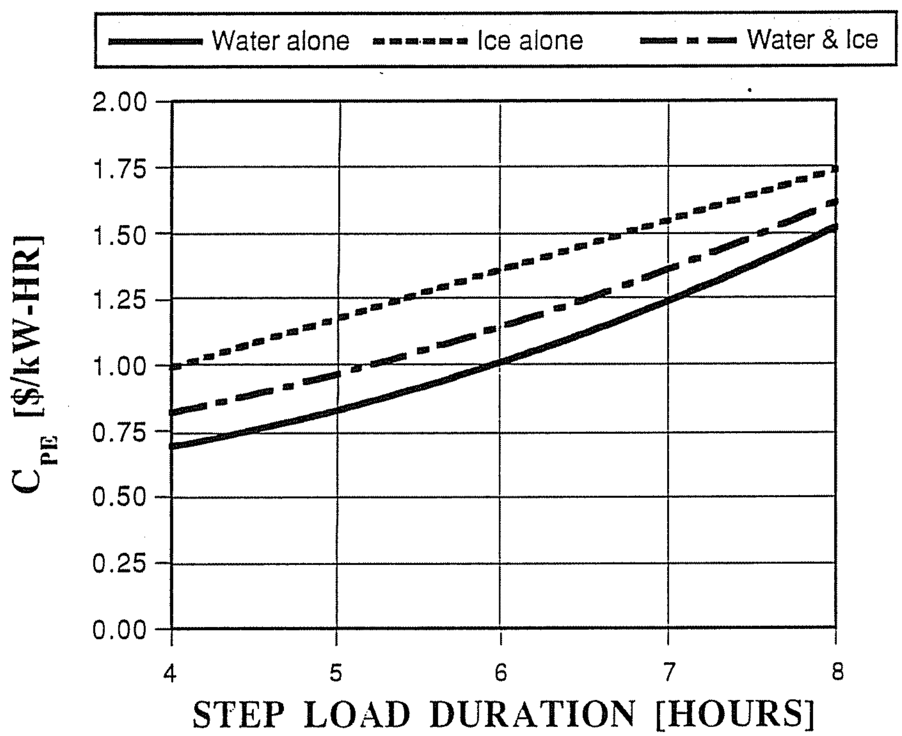


Figure 6.3.2: "C_{PE}" for Step Load Profiles: Second General Case

In the second general case, the overall conversion efficiencies do not improve as much with the addition of thermal storage media based inlet air cooling systems as in the first general case. For the three step power plant load profiles, the overall conversion efficiency with the complete inlet air cooling system is about 0.262 for all storage capacity splits; for the eight hour peaked power plant load profile, the overall conversion efficiency with the complete inlet air cooling system is about 0.260 for all storage capacity splits. With the evaporative cooler alone, the overall conversion efficiency is 0.260. Thus the addition of a thermal storage media based inlet air cooling system increases the overall conversion efficiency by 0.0% - 0.8% in the second general case, depending on power plant load profile shape.

This chapter shows that inlet air cooling systems based on chilled water storage alone yield the lowest values of "C_{PE}" for all situations considered. Hybrid cooling systems based on both chilled water and ice storage always result in lower values of "C_{PE}" than do cooling systems based on ice storage alone. However, in order for the cost of the incremental electric power generated with inlet air cooling to be less expensive than on-peak wholesale electricity prices typical of the mid-Western United States, it would be necessary to operate the combustion turbine for between 100 to 350 hours during the cooling season. depending on the power plant load profile and inlet configuration. These findings are specific to the inlet air cooling system design parameters outlined in Chapters 4 and 5, and on the economic parameters listed in section 6.2. General conclusions that can be derived from this study are presented in Chapter 7. Recommendations for further research are also included in that chapter.

CHAPTER 7: CONCLUSIONS AND RECOMMENDATIONS FOR FURTHER RESEARCH

Enough variation exists between different combustion turbines, sites, and usage patterns that different conclusions may be reached regarding the suitability of the three storage capacity splits considered above. Additional research is necessary to clarify further the situations in which different storage capacity splits between chilled water and ice would be most appropriate. Nevertheless, several general conclusions concerning combustion turbine inlet air cooling can be drawn from the research described in this thesis. This chapter presents those conclusions and suggests a number of areas for further investigation into combustion turbine inlet air cooling systems.

7.1 Conclusions

Before making the decision to install an inlet air cooling system, it is important to consider the number of hours the system is likely to operate each year, the availability of capacity from other sources, and the cost of electrical power from other sources. The concept of "C_{PE}", the cost of the incremental power produced with inlet cooling based on a cooling system payback period of "N_P" years, is useful for determining whether an investment in any type of cooling system is justified. The concept of capacity enhancement cost alone is not adequate for making this determination, especially if the inlet air cooling system will only be used for a small fraction of the time that the combustion turbine is to be operated each year. As the number of hours of anticipated cooling system use increases, "C_{PE}" decreases and investment in an inlet air cooling system becomes more favorable. Combined cycle power plants, which use both a combustion turbine and a steam driven turbine to meet utility base load power requirements, are thus likely to provide the best opportunity for investment in inlet air cooling, while combustion turbines installed to meet utility peak load power requirements for relatively short time periods in the cooling season are likely to offer the worst opportunity from an investment perspective.

Inlet air cooling systems based on ice storage alone do not result in the lowest capacity enhancement costs or "C_{PE}" values for any of the load profiles or inlet configurations considered in this research project. Systems based on a combination of ice and chilled water storage can provide inlet air temperatures as low as those provided by systems based on ice storage alone, and thus result in capacity increases very nearly as great as those resulting from systems based on ice storage alone, but at lower cost. Cooling systems based on ice alone do require somewhat less storage volume than those based on both thermal storage media, as discussed in section 5.7. Systems based on one storage medium are also inherently simpler than systems based on two storage media, and thus are likely to require less maintenance. Therefore, selecting an inlet air cooling system based on ice alone would be appropriate only if space were not available for other storage options or if the increased maintenance associated with using two storage media were deemed excessive.

Inlet air cooling systems based on chilled water storage alone yield the lowest capacity enhancement costs and the lowest incremental electric power production costs in all instances considered. Such cooling systems are also simpler and easier to maintain than systems that include an ice harvester. However, cooling systems based on chilled water storage alone cannot decrease the inlet air dry bulb temperature as much as can systems based on both chilled water and ice storage. The choice between chilled water alone and a combination of chilled water and ice storage will depend on the magnitude of the peak capacity enhancement cost for the latter storage option. If the peak capacity enhancement cost is lower than the unit cost for installing a second combustion turbine, the inclusion of an ice storage component in the inlet air cooling system may be justifiable.

The choice of storage capacity split will also depend, to some extent, on the shape of the power plant load profile. As discussed in section 6.3, designing a cooling system for a peaked power plant load profile tends to favor the selection of chilled water as the only storage medium based on anticipated electricity production costs. Each decision concerning the optimal storage capacity split for an inlet cooling system must be based on a wide range of considerations; no single capacity split will be the most favorable in all cases.

7.2 Recommendation for Further Research

The EES and TRNSYS combustion turbine models developed for this study should be adequate for simulating the performance of other single shaft combustion turbines. It would be necessary to derive curve fit parameters for the relative power output (Equation 2.2.1), the relative efficiency (Equation 2.2.2), the inlet pressure loss capacity and efficiency multipliers (Equations 2.3.1 and 2.3.2), the outlet pressure loss capacity and efficiency multipliers (Equation 2.3.3; in the present case these are one and the same), and the water-fuel ratio capacity and efficiency multipliers (Equation 2.3.4 and 2.3.5) for each different combustion turbine to be modeled. It would also be necessary to determine the compressor stage inlet air volumetric flow rate, the base efficiency, and the base capacity for each different combustion turbine, as discussed in sections 2.1 and 2.4 , based on data supplied by the manufacturer.

The compressor stage inlet air volumetric flow rate depends on the dry bulb temperature for multiple shaft aircraft derivative combustion turbines. Since the combustion turbine models used in this study assume a constant inlet air volumetric flow rate, some modifications to those models would be necessary in order to simulate the performance of aircraft derivative combustion turbines. It would be desirable to develop the capability to model both single and multiple shaft combustion turbines. It would be of interest to design inlet air cooling systems for a variety of combustion turbine power plant types and compare their capacity enhancement costs, peak capacity enhancement costs, and the cost of incremental power produced with inlet air cooling based on a specified payback period,

"C_{PE}". The calculation of "C_{PE}" should be modified to account for maintenance costs, which may vary significantly for different storage capacity splits.

All system designs are based on a "full storage strategy" in the present study. The ice storage loops are based on a weekly full storage strategy; the chilled water storage loops are based on a daily full storage strategy. The refrigeration equipment is constrained to operate between the hours of 9:00 p.m. and 12:00 p.m. on weekdays in all instances, as discussed in section 4.1. In future work, it would perhaps be more consistent to vary the hours of refrigeration equipment operation for different power plant load profiles so that the refrigeration equipment could operate whenever the power plant does not operate. By increasing the duty cycle of the refrigeration equipment, it would be possible to specify smaller refrigeration capacities, which would reduce system costs. It would also be interesting to consider a partial storage strategy for the cooling systems, which would require the chiller and/or ice harvester to operate continuously. All other things being equal, a partial storage strategy would reduce the cost of each inlet air cooling system, but would result in a greater on-peak parasitic power requirement. Depending on the magnitude of these changes, the capacity enhancement cost, peak capacity enhancement cost, and "C_{PE}" could increase, decrease, or stay the same. Further research could show whether a full or a partial storage strategy is most beneficial.

As discussed in section 7.1 above, inlet air cooling systems are likely to provide the largest life cycle economic benefits for combustion turbines that generate base load electricity, such as those in combined cycle power plants. It would thus be useful to have the capability to perform simulations lasting more than one or two weeks per year. In order to perform seasonal simulations for significantly longer time periods than those used in this study, two changes would be required. First, real weather data for the power plant site would have to be used in place of design day temperature profiles. Typical Meteorological Year data may be acceptable, since the hours of operation at or near design conditions would be only a small fraction of the total annual hours of cooling system operation. Second, the TRNSYS system model would have to be modified slightly to ensure that water from the first cooling coil does not re-enter the chilled water storage tank at temperatures substantially lower than the design leaving water temperature. This change would entail the addition of a flow diverter and a tempering valve in the cooling coil water flow stream that would allow water leaving the cooling coil to be recirculated in order to maintain the temperature of the water re-entering the chilled water storage tank above a specified minimum. This modification would prevent the possibility of under loading the chiller.

The number of cooling coil rows was not optimized in this research project for cooling systems based on either chilled water storage or ice storage alone. An increase in the number of cooling coil rows would raise the cost of the cooling coil, the on-peak parasitic pump power requirement, and the cooling coil leaving water temperature. An increase in the cooling coil leaving water temperature would raise the temperature difference between the top and bottom of the chilled water storage tank, which would decrease the required tank capacity. An increase in the number of cooling coil rows would also lower the leaving air dry bulb temperature for the cooling system based on chilled water storage alone slightly below the 46° F minimum cited by Andrepont (1994), leading to an increase in the combustion turbine capacity enhancement. It would be useful to optimize the number of cooling coil rows for systems based on chilled water or ice storage alone, just as the number of rows was optimized for the hybrid system based on a combination of the two storage media. It would also be useful to modify the TRNSYS component model to allow simulation of cooling coils in which the air capacitance rate exceeds the water capacitance rate.

Finally, it would be of interest to develop the capability to simulate the behavior of direct contact heat exchangers in addition to indirect contact cooling coils. Currently, many combustion turbines have evaporative coolers in their inlet air flow paths. As discussed in sections 5.7 and 5.8, the presence of an evaporative cooler does not significantly reduce the size or cost of a downstream thermal storage based cooling system. However, if it were possible to expose water from the chilled water or ice storage tank directly to the inlet air using an existing evaporative cooling unit, it might be possible to eliminate the need for the downstream cooling coils. Since the cooling coil cost can easily represent up to one quarter of the entire cooling system cost, modifying the evaporative cooling unit as suggested could result in significant savings. It would be necessary to develop a new TRNSYS component to simulate a direct contact air-water heat exchanger. Since the potential benefit of using existing evaporative coolers in place of cooling coils is so high, it is strongly recommended that the development of such a component and its use in inlet air cooling system designs be the subject of future research.

APPENDIX A: EES COMPONENT MODELS

- AVFR.2: Combustion Turbine Inlet Air Volumetric Flow Rate Calculation
- BEP.1: Combustion Turbine "Base" Capacity Calculation
- BBPPmod.5: Combustion Turbine Model
- IHmod.8: Ice Harvester Model (Based on Frick RWB-II 60E Compressor)

Determination of Volumetric Flow Rate
at Turbine Inlet

This EES deck determines the volumetric flow rate at the compressor
stage inlet, based on information provided by Doug Reindl.}

```
{Calculation of inlet air mass flow rate during test "A"}
EMFR[1] = 2415000 {exhaust mass flow rate; lb/hr}
FMFR[1] = 49462  {fuel mass flow rate; lb/hr}
WMFR[1] = 78915  {injected water mass flow rate; lb/hr}
w[1] = HumRat(AirH2O,T=T[1],P=P[1],R=RH[1])  {humidity ratio}
(1 + w[1])*AMFR[1] = EMFR[1] - (FMFR[1] + WMFR[1])  {dry air mass flow rate; lb/hr}

{Specific volume of air during test "A"}
v[1] = Volume(AirH2O,T=T[1],P=P[1],R=RH[1])  {specific volume; ft^3/lb dry air}
T[1] = 60 {F}
P[1] = 0.9662  {atm}
RH[1] = 0.90

{Volumetric flow rate of inlet air during test "A"}
AVFR[1] = AMFR[1]*v[1]/60  {cubic feet per minute}
AVFRm[1] = AVFR[1]*0.02832*60  {cubic meters per hour}

{Calculation of inlet air mass flow rate during test "B"}
EMFR[2] = 2463000 {exhaust mass flow rate; lb/hr}
FMFR[2] = 52080  {fuel mass flow rate; lb/hr}
WMFR[2] = 123350  {water mass flow rate; lb/hr}
w[2] = HumRat(AirH2O,T=T[2],P=P[2],R=RH[2])  {humidity ratio}
(1 + w[1])*AMFR[2] = EMFR[2] - (FMFR[2] + WMFR[2])  {dry air mass flow rate; lb/hr}

{Specific volume of air during test "B"}
v[2] = Volume(AirH2O,T=T[2],P=P[2],R=RH[2])  {specific volume; ft^3/lb dry air}
T[2] = 62 {F}
P[2] = 0.95654  {atm}
RH[2] = 0.87

{Volumetric flow rate of inlet air during test "B"}
AVFR[2] = AMFR[2]*v[1]/60  {cubic feet per minute}
AVFRm[2] = AVFR[2]*0.02832*60  {cubic meters per hour}

{Determination of average air volumetric flow rate at inlet}
AVFRbar = (AVFR[1] + AVFR[2])/2  {cubic feet per minute}
AVFRmbar = (AVFRm[1] + AVFRm[2])/2  {cubic meters per hour}
```

SOLUTION:

AVFRbar = 520268	[cfm]
AVFR[1] = 520160	[cfm]
AVFR[2] = 520376	[cfm]

****

Determination of BEP and nbase

This program calculates the base capacity and base efficiency of the combustion turbine power plant based on tests "1" and "2": base load and power augmentation mode with natural gas.}

```
{Base capacity for test 1}
NTEP[1] = 87294 {kW}
dPin[1] = 0.00651 {atm}
dPout[1] = 0.0119 {atm}
WFR[1] = 1.800
RPO[1] = 0.998
IPLCM[1] = 1 - 1.9*dPin[1]
OPLCEM[1] = 1 - 0.848*dPout[1]
WFRCM[1] = 1 + 0.064*WFR[1]
NTEP[1] = IPLCM[1]*OPLCEM[1]*WFRCM[1]*RPO[1]*BEP[1]

{Base efficiency for test 1}
nrel[1] = 1.000
IPLEM[1] = 1 - 0.848*dPin[1]
WFREM[1] = 1 - 0.032*WFR[1]
ntest[1] = 0.267
ntest[1] = IPLEM[1]*OPLCEM[1]*WFREM[1]*nrel[1]*nbase[1]

{Base capacity for test 2}
NTEP[2] = 89844 {kW}
dPin[2] = 0.00651 {atm}
dPout[2] = 0.0119 {atm}
WFR[2] = 2.252
RPO[2] = 0.993
IPLCM[2] = 1 - 1.9*dPin[2]
OPLCEM[2] = 1 - 0.848*dPout[2]
WFRCM[2] = 1 + 0.064*WFR[2]
NTEP[2] = IPLCM[2]*OPLCEM[2]*WFRCM[2]*RPO[2]*BEP[2]

{Base efficiency for test 2}
nrel[2] = 0.997
IPLEM[2] = 1 - 0.848*dPin[2]
WFREM[2] = 1 - 0.032*WFR[2]
ntest[2] = 0.267
ntest[2] = IPLEM[2]*OPLCEM[2]*WFREM[2]*nrel[2]*nbase[2]

{Average values for BEP and nbase}
BEPav = (BEP[1] + BEP[2])/2
nbaseav = (nbase[1] + nbase[2])/2
```

SOLUTION:

```
BEPav = 80556   [kW]
BEP[1] = 80225
BEP[2] = 80886
nbaseav = 0.290
nbase[1] = 0.288
nbase[2] = 0.293
```

****

"Black Box" Gas Turbine Power Plant Model

```
  This EES deck calculates the air mass flow rate, the electric power output, the fuel mass flow
rate, and the conversion efficiency of a combustion turbine power plant based on performance
curves and other data provided by the manufacturer.}

{The maximum value for the part load factor is 1.0.}
Procedure AdjPLF(PLF:APLF)  {Adjusted part load factor cannot exceed 1.0}
   If (PLF < 1.0) Then APLF := PLF Else APLF := 1.0
END

{The electric power output is limited to the base power output times the relative power
output (which is a function of temperature) for a part load factor of 1.}
Procedure AdjEP(PLF,RPO,BEP,EP,IPLCM,OPLCEM,WFRCM:AEP)
                                   {Adjusted electric power output}
   If (PLF < 1.0) Then AEP := EP Else AEP = IPLCM*OPLCEM*WFRCM*BEP*RPO
END

{Program inputs and selected parameters}
ADB = 95 {ambient dry bulb temperature; F}
ST1 = 40.0  {temperature of air leaving the cooling coil: F}
EDB = ST1 {entering dry bulb temperature (can be either ADB or ST1); F}
Patm = 0.977  {atmospheric pressure; atm}
w1 = HumRat(AirH2O,T=EDB,P=Patm,R=RH)
PLF = 1  {Part load factor}
RH = 0.92
BEP = 80039 {kW; approximate base net electric power output with no inlet pressure drop, no outlet
                  presure drop, and no water injection at EDB = 59 F}
nbase = 0.288  {approximate base conversion efficiency at the conditions described above}
HHV = 22760  {Btu/lb; higher heating value of natural gas}
WFR = 1.80  {water fuel mass ratio}
dPincc = 0.00135  {inlet pressure loss due to cooling coil; atm}
dPinex = 0.00000 {inlet pressure loss due to other sources, atm}
dPin = dPincc + dPinex  {total inlet pressure loss, atm}
dPout = 0.0119  {outlet pressure loss; atm}
VFR1 = 31216080  {inlet volumetric flow rate; ft^3/hr}
VFR1pm = VFR1/60  {inlet volumetric flow rate, cfm}

{Dry air flow rate depends on inlet temperature, ambient pressure, and humidity ratio}
AMFR = MFR1/(1 + w1)  {dry air mass flow rate: lb/hr}
MFR1 = VFR1/v1  {moist air mass flow rate; lb/hr}
v1 = 1/(1/va1 + 1/vw1)  {ft^3/lb; specific volume of moist air}
va1 =0.02519*(EDB + 459.7)/Pa1  {specific volume of dry air; ft^3/lb}
vw1 = 0.04050*(EDB + 459.7)/Pw1  {specific volume of water vapor; ft^3/lb}
Pw1 = y1*P1  {partial pressure of water vapor; atm}
y1 = 29/18*w1/(1 + w1)  {mole fraction of water vapor}
P1 = Patm - dPin  {total inlet pressure; atm}
Pa1 = P1 - Pw1  {partial pressure of air at inlet; atm}

{The turbine electric power output is simply the peak power output multiplied by the control
variable, gamma}
EPpeak = IPLCM*OPLCEM*WFRCM*BEP*RPOmax  {peak power out at ST1; kW}
IPLCM = 1.0 - 1.90*dPin  {Inlet pressure loss capacity multiplier}
OPLCEM = 1.0 -0.848*dPout  {Outlet pressure loss capacity and efficiency multiplier}
WFRCM = 1.0 + 0.0642*WFR  {Water-fuel ratio capacity multiplier}
RPOmax = r1 + r2*ST1 + r3*ST1^2  {Maximum relative power output for EDB = ST1}
EP = gamma*EPpeak  {kW}
```

r1 = 1.158; r2 = -2.478e-3; r3 = -3.73e-6

{The part load factor is the electric power output divided by the product of the plant's electric power output at test conditions and the "Relative Power Output" (cf. Annex 1 of Performance Test Report).}
PLF = EP/(IPLCM*OPLCEM*WFRCM*BEP*RPO) {Load Factor}
RPO = r1 + r2*EDB + r3*EDB^2 {Relative Power output for PLF = 1}

{The efficiency (which is based on the higher heating value) is a function of both inlet temperature and load factor (cf. Annex 10 of performance test report).}
nHHV = IPLEM*OPLCEM*WFREM*nbase*nrel {conversion efficiency based on higher heating value}
IPLEM = 1.00 - 0.848*dPin {Inlet pressure loss efficiency multiplier}
WFREM = 1.0 - 0.0321*WFR {Water-fuel ratio efficiency multiplier}
Call AdjPLF(PLF:APLF)
nrel = a1 + a2*PLF + a3*EDB + a4*PLF*EDB + a5*PLF^2 + a6*EDB^2 + a7*PLF*EDB^2 + a8*PLF^3
{relative conversion efficiency of power plant operation}
a1 = 0.1777; a2 = 2.341; a3 = -9.764e-4; a4 = 8.181e-4; a5 = -2.401; a6 = -1.82e-6; a7 = -1.95e-6; a8 = 0.9040

{The fuel flow rate is simply the electric power output divided by the product of the conversion efficiency and the Higher Heating Value.}
Call AdjEP(PLF,RPO,BEP,EP,IPLCM,OPLCEM,WFRCM:AEP)
FMFR = AEP/(nHHV*HHV)*3412 {lb/hr: fuel mass flow rate}

{The maximum outlet power with no cooling coil inlet pressure loss at the design dry bulb temperature is found below}
EPmin = IPLCMmin*OPLCEM*WFRCM*BEP*RPOmin {kW}
IPLCMmin = 1.00 - 1.90*dPinex {inlet pressure loss capacity multiplier w/o cooling coils. atm}
RPOmin = r1 + r2*ADB + r3*ADB^2 {minimum relative power output for EDB = ADB}

{The maximum electric power that can be produced with inlet cooling is AEP minus the cooling coil pump power found in the water storage and ice storage loop sizing programs.}
EPmax = AEP - P3pow - P4pow {kW}
P3pow = 212 {kW}
P4pow = 79 {kW}

SOLUTION:

BEP = 80039	[kW]
dPin = 0.00135	[atm]
dPincc = 0.00135	[atm]
dPinex = 0.00000	[atm]
dPout = 0.01190	[atm]
EPmax = 92534	[kW]
EPmin = 78570	[kW]
EPpeak = 92825	[kW]
HHV = 22760	[Btu/lb]
nbase = 0.2880	
VFR1pm = 520268	[cfm]
WFR = 1.80	

****

Basic Ice Harvester Model
(Based on D. Knebel's Model)

This EES deck models the behavior of an Ice Harvester using performance curves derived for the Frick RWB-II 60E rotary screw compressor, which has a reference capacity of 135.0 tons at SST = 20 F and SDT = 95 F. The perfomance curve fit parameters are scaled by the factor "mult" which is just the nominal capacity (at the above SST and SDT) divided by the reference capacity. The refrigerant used is ammonia. The model calculates the net tons of refrigeration, ice generation rate, power requirement, and cycle time as functions of wet bulb temperature, design wet bulb temperature, and the nominal capacity. The power required to operate the refrigerant pump is taken into account in calculating the net power requirement.}

{Determination of compressor, evaporative condensor, and evaporator sizes: these depend on design evaporator, condensor, discharge, suction, and wet bulb temperatures. The performance of the ice harvester is a function of wet bulb temperature only.}
RefCap = 135.0 {reference capacity; tons}
NomCap = qevapbdes {nominal compressor capacity; tons}
NCC = qconddes*HRCFdes {evaporative condensor capacity; tons}
qconddes = qevapbdes + CPbdes*2545/12000 {design condensor heat rejection; tons}
qevapbdes = mult*(C1 + C2*SSTdes + C3*SSTdes^2 + C4*SDTdes + C5*SDTdes^2 + C6*SSTdes*SDTdes)
{design refrigeration capacity during pure build period; tons}
CPbdes = mult*(P1 + P2*SSTdes + P3*SSTdes^2 + P4*SDTdes + P5*SDTdes^2 + P6*SSTdes*SDTdes)
{design compressor brake horse power during pure build period; hp}
HRCFdes = E1 + E2*SCTdes + E3*WBdes^3 + E4*WBdes^3*SCTdes + E5*WBdes^3*SCTdes^2 + E6*WBdes^5 + E7*WBdes^4*SCTdes
{design heat rejection correction factor}
Nplates = qevapbdes*12000/(Ubarb*Parea*(PEWT - SETdes)) {number of plates}
Ubarb = 51 {average evaporator U-value during build period; Btu/hr-ft^2-F}
Parea = 3.833*6.833*2 {plate area; ft^2}
PEWT = 32.0 {plate entering water temperature; F}
SSTdes = 20 {design saturated suction temperature (should have been 16); F}
SDTdes = 95 {design saturated discharge temperature; F}
SCTdes = 93 {design saturated condensing temperature; F}
SETdes = 24 {design saturated evaporator temperature (should have been 20); F}

{Evaporative condensor: the refrigerant is cooled by means of an evaporative condensor unit. The capacity of this unit is its nominal capacity divided by the "heat rejection correction factor", HRCF. This factor is calculated by means of a six parameter equation derived from data provided by "IMECO".}
qcond = NCC/HRCF {condensor heat rejection; tons}
HRCF = E1 + E2*SCT + E3*WB^3 + E4*WB^3*SCT + E5*WB^3*SCT^2 + E6*WB^5 + E7*WB^4*SCT
{heat rejection correction factor}

{Pure ice-making mode: here we determine the power requirement and refrigeration capacity of the ice harvester when ice is being built on all plate sections.}
CPb = mult*(P1 + P2*SST + P3*SST^2 + P4*SDT + P5*SDT^2 + P6*SST*SDT)
{compressor brake horse power during pure build period; hp}
qevapb = mult*(C1 + C2*SST + C3*SST^2 + C4*SDT + C5*SDT^2 + C6*SST*SDT)
{refrigeration capacity during pure build period; tons}
SST = SET - SLL {saturated suction temperature; F}
SLL = 4.0 {suction line losses (subcooling); F}
SDT = SCT + DLL {saturated discharge temperature; F}
DLL = 2.0 {discharge line losses (superheat); F}
qcond = qevapb + CPb*2545/12000 {condensor heat rejection; tons}
qevapb = Ubarb*Nplates*Parea*(PEWT - SET)/12000 {tons}

{Defrost mode: in defrost mode, one section of evaporator plates is defrosted by re-routing the hot gas from the condensor to that section. Ice continues to build on the remaining plate sections during this time. The compressor power requirement and capacity are determined below.}
CPd = mult*(P1 + P2*DSST + P3*DSST^2 + P4*DSDT + P5*DSDT^2 + P6*DSST*DSDT)
{compressor brake horsepower during defrost period: hp}
qevapd = mult*(C1 + C2*DSST + C3*DSST^2 + C4*DSDT + C5*DSDT^2 + C6*DSST*DSDT)
{refrigeration capacity during defrost period; tons}
DSST = DSET - SLL {saturated evaporator temperature during defrost mode; F}
DSDT = DSCT + DDLL {saturated condensor temperature during defrost mode; F}
DDLL = 5 {discharge line losses (superheat) during defrost mode; F}
qevapd = Ubarb*(Nsect-1)/Nsect*Nplates*Parea*(PEWT - DSET)/12000 {tons}
Nsect = 4 {number of sequentially defrosted evaporator sections}
qrejd = qevapd + CPd*2545/12000 {heat rejected at defrosting evaporator plate; tons}

{Determination of saturated condensor temperature during defrost mode}
qrejd = DSWMFR*Cpw*effdef*(DSCT - PEWT)/12000 {tons}
DSWMFR = PWMFR*Nplates/Nsect {defrost section water mass flow rate; lb/hr}
PWMFR = 10*0.13368*62.41*60 {water mass flow rate per plate, lb/hr}
effdef = 1 - exp(-NTUdef) {evaporator effectiveness in condensor mode during defrost period}
NTUdef = (Udefp*Nplates*Parea/Nsect)/(DSWMFR*Cpw) {number of transfer units}
Udefp = 64.5 {average evaporator U-value for defrosting section: Btu/hr-ft^2-F}
Cpw = 1.0 {Heat capacity of water, Btu/lb-F}

{Determination of ice generation rate: ice is built up on each plate during PBtime (while the harvester is operating in the pure ice building mode) and DBtime (while other evaporator sections are defrosting). The total cycle time is the sum of these last two periods and the period during which each plate is defrosting.}
IGR = Icemass/Ctime {ice generation rate; lbs/hr}
Icemass = xmax*Nplates*Parea*rhoice {lbs}
xmax = 0.375/12 {maximum ice thickness; ft}
rhoice = 57.5 {density of ice; lb /ft^3}
TIHC = Ncap*Ctime {Total heat capacity of ice built per cycle(Ncap is the net cap): ton-hrs}
TIHC = Icemass*LHF/12000 {ton-hrs}
LHF = 143.5 {latent heat of fusion of water; Btu/lb}
Ctime = PBtime + DBtime + Dtime {cycle time: hours}
Dtime = 50/3600 {period during which section is defrosted; hrs}
DBtime = (Nsect - 1)*Dtime {build time during defrost mode: hrs}

{Calculation of refrigerant pump brake horsepower (the liquid-vapor ratio at the evaporator outlet is roughly 3:1)}
SEP = Pressure(Ammonia,T=SET,x=1) {saturated evaporator pressure;psia}
SCP = Pressure(Ammonia,T=SCT,x=1) {saturated condensor pressure;psia}
scool = 2.0 {amount of subcooling; F}
sheat = 10 {amount of superheat; F}
hcondo = Enthalpy(Ammonia,T=SCT-scool,P=SCP) {outlet condensor Enthalpy; Btu/lb}
hevapi = hcondo {inlet evaporator enthalpy; Btu/lb}
hevapo = Enthalpy(Ammonia,T=SET+sheat,P=SEP) {outlet evaporator Enthalpy; Btu/lb}
RMFR = qevapb*12000/(hevapo - hevapi) {refrigerant mass flow rate; lb/hr}
vam = Volume(Ammonia,T=SET,x=0.0) {refrigerant specific Volume, ft^3/lb}
RVFR = 3*RMFR*vam {refrigerant volumetric flow rate; ft^3/hr}
RPhead = 30 {refrigerant pump head: psi}
RPP = RPhead*RVFR/13750 {refrigerant pump brake horsepower: hp}

{Calculation of net electric power consumption and net refrigeration effect}
NPower = (CPb/ncomp*(1 - XDF) + CPd/ncomp*XDF + RPP/npump)*0.746
{net electric power requirement: kW}

ncomp = 0.95 {overall efficiency of compressor}
npump = 0.65 {overall efficiency of refrigerant pump}
Ncap = qevapb - XDF*qrejd {net refrigeration capacity; tons}
XDF = (DBtime + Dtime)/Ctime {defrost fraction of cycle time}

{Curve fit parameters for compressor and evaporative condensor performance}
E1 = 2.271; E2 = -2.212e-2; E3 = 4.671e-5; E4 = -8.043e-7
E5 = 5.617e-9; E6 = 3.742e-9; E7 = -5.494e-9
P1 = 22.86; P2 = -1.11299; P3 = -0.0075875; P4 = 0.5892; P5 = 0.006632; P6 = 0.0184585
C1 = 94.204; C2 = 2.10578; C3 = 0.0158157; C4 = -0.02908; C5 = -0.0003119; C6 = -0.0007221

PARAMETRIC TABLE: NCAP AND NPOWER AS FUNCTIONS OF WBdes, WB, AND NomCap:

Run	WBdes [F]	WB [F]	Nom. Cap. [tons]	NCap [tons]	NPow. [kW]	Run	WBdes [F]	WB [F]	Nom. Cap. [tons]	NCap [tons]	NPow. [kW]
1	65	50	40.0	36.6	29.2	33	75	50	40.0	36.8	24.3
2	65	50	80.0	73.2	58.5	34	75	50	80.0	73.7	48.6
3	65	50	120.0	109.8	87.7	35	75	50	120.0	110.5	72.9
4	65	50	160.0	146.4	116.9	36	75	50	160.0	147.3	97.2
5	65	60	40.0	36.4	32.4	37	75	60	40.0	36.6	28.3
6	65	60	80.0	72.9	64.7	38	75	60	80.0	73.3	56.5
7	65	60	120.0	109.3	97.1	39	75	60	120.0	109.9	84.8
8	65	60	160.0	145.7	129.5	40	75	60	160.0	146.6	113.1
9	65	70	40.0	36.3	35.4	41	75	70	40.0	36.5	31.9
10	65	70	80.0	72.6	70.7	42	75	70	80.0	72.9	63.9
11	65	70	120.0	108.8	106.1	43	75	70	120.0	109.4	95.8
12	65	70	160.0	145.1	141.5	44	75	70	160.0	145.8	127.8
13	65	80	40.0	36.1	39.1	45	75	80	40.0	36.3	35.9
14	65	80	80.0	72.2	78.2	46	75	80	80.0	72.5	71.8
15	65	80	120.0	108.3	117.4	47	75	80	120.0	108.8	107.7
16	65	80	160.0	144.4	156.5	48	75	80	160.0	145.0	143.7
17	70	50	40.0	36.7	27.1	49	80	50	40.0	37.0	20.9
18	70	50	80.0	73.4	54.1	50	80	50	80.0	74.0	41.9
19	70	50	120.0	110.1	81.2	51	80	50	120.0	111.0	62.8
20	70	50	160.0	146.8	108.2	52	80	50	160.0	148.0	83.8
21	70	60	40.0	36.5	30.5	53	80	60	40.0	36.8	25.5
22	70	60	80.0	73.0	61.1	54	80	60	80.0	73.6	51.0
23	70	60	120.0	109.6	91.6	55	80	60	120.0	110.3	76.5
24	70	60	160.0	146.1	122.2	56	80	60	160.0	147.1	102.0
25	70	70	40.0	36.4	33.8	57	80	70	40.0	36.6	29.6
26	70	70	80.0	72.7	67.7	58	80	70	80.0	73.1	59.2
27	70	70	120.0	109.1	101.5	59	80	70	120.0	109.7	88.9
28	70	70	160.0	145.4	135.4	60	80	70	160.0	146.3	118.5
29	70	80	40.0	36.2	37.7	61	80	80	40.0	33.8	33.8
30	70	80	80.0	72.3	75.3	62	80	80	80.0	67.7	67.7
31	70	80	120.0	108.5	113.0	63	80	80	120.0	101.5	101.5
32	70	80	160.0	144.7	150.6	64	80	80	160.0	135.4	135.4

APPENDIX B: EES SYSTEM MODELS

- ISO.size: Model of Cooling System Based on Ice Storage Alone
- CWSO.size: Model of Cooling System Based on Chilled Water Storage Alone
- CWSL.size: Model of Ice Storage Loop for Hybrid Cooling System
- ISL.size: Model of Chilled Water Storage Loop for Hybrid Cooling System

Ice Storage Based System Sizing Program
Version 1 }

```
{This EES deck determines the parameters required by TRNSYS for all components of
the ice storage loop. These components include: the cooling coil, the cooling coil pump,
the ice harvester, and the ice storage tank.

The dimensions of the cooling coil, the water mass flow rate through the cooling
coil, and the temperature of the water as it leaves the cooling coil are calculated first.
Design conditions for the cooling coil are: EDB = 95 F, EWB = 76 F, LDB = 40 F, LWB = 40 F
(saturated air), EWT = 32.5 F, an air face velocity of 400 fpm, a water flow velocity of
ten fps and an outlet air volumetric flow rate of 525,560 cfm. The coil has 10 rows.}

{Determination of dry air mass flow rate and cooling coil load associated with the
design entering air state and the desired outlet air state}
AMFR = AVFR/Avolout  {air mass flow rate;lb/hr}
AVFR = 525560*60  {air volumetric flow rate at outlet;ft^3/hr}
Avolout = Volume(AirH2O,T=LDB,P=Patm,R=1)  {air specific volume at outlet; ft^3/lb}
LDB = 40  {leaving dry bulb temperature; F}
Patm = 14.7  {ambient pressure: psia (should have been 14.4 psia)}
CCLoad = AMFR*(ENTHin - ENTHout)/12000  {cooling coil load; tons}
ENTHin = Enthalpy(AirH2O,T=EDB,P=Patm,w=EHR)  {entering enthalpy; Btu/lb}
ENTHout = Enthalpy(AirH2O,T=LDB,P=Patm,R=1)  {leaving enthalpy; Btu/lb}
EDB = 95  {entering dry bulb temperature; F}
EHR = 0.0150  {entering humidity ratio}

{Determination of core area, frontal area, duct width, duct height, volume, and effective
area of the cooling coil based on the design air flow velocity and the selection of surface
8.0 - 3/8T on page 224 of "Compact Heat Exchangers" (CHE's)}
AVin = 400*60  {air face velocity at inlet: feet/hr}
Afr = AMFR*Avolin/AVin  {frontal area of cooling coil: ft^2}
Avolin = Volume(AirH2O,T=EDB,P=Patm,w=EHR)  {air specific volume at inlet; ft^3/lb}
Ac = sigma*Afr  {core area of cooling coil, ft^2}
sigma = 0.534  {ratio from CHE's}
DHt= Afr/DWt  {duct height: ft}
DWt = num*DHt  {duct width; ft}
num = 20.75 {ratio of duct width to duct height}
V = Nrows*L*Afr  {cooling coil volume; ft^3}
Nrows = 10 {number of rows}
L = 0.866/12  {row spacing from CHE's; ft}
A = V*alpha  {heat transfer area: ft^2}
alpha = 179  {ratio from CHE's}

{Calculation of the number of transfer units for the air side based on material found
in Chapters 3 and 11, "Fundamentals of Heat and Mass Transfer" (FHMT) by Incropera
and DeWitt, and in class notes from John Mitchell}
NTUa = UAa/(AMFR*Cpa)  {equation 19.26, Mitchell}
Cpa = SpecHeat(AirH2O,T=Tav,P=Patm,w=wav)  {average specific heat of air-water; Btu/lb-F}
Tav = (EDB + LDB)/2  {average air temperature in cooling coil; F}
wav = (EHR + LHR)/2  {average humidity ratio in cooling coil; F}
LHR = HumRat(AirH2O,T=LDB,P=Patm,R=1)  {leaving humidity ratio}
1/UAa = 1/(no*ha*A)  {equation 11.1, FHMT; modified}
no = 1 - AfA*(1 - nf)  {overall efficiency of finned surface, equation 11.3, FHMT}
AfA = 0.913  {ratio of fin area to total area, from CHE's}
Lc = Lf + tf/2  {corrected fin length: ft}
Lf = (FD - TOD)/2  {fin length: ft}
```

```
FD = sqrt(4*L*S/pi) {equivalent fin diameter for the rectangular plate fin
                        (Threlkeld, eq. 12.25); ft}
S = 1.00/12 {distance between centers of tubes in a row from CHE's; ft}
TOD = 0.402/12 {tube outside diameter; ft}
tf = 0.013/12 {fin thickness; ft}
k = 177/1.731 {thermal conductivity of aluminum fins; Btu-ft/hr-ft^2-F}
Ap = Lc*tf {corrected fin profile area; ft^2}
x = (Lc^1.5)*(ha/(k*Ap))^0.5 {abscissa used in graph of fin efficiency}
nf = lookup(lookuprow(1,x),2) {fin efficiency from figure 3.19, FHMT; r2c/r1 = 3}
ReD = Arho*AVcore*DH/mua {Air Reynolds number}
AVcore = AVin/sigma {air velocity inside the cooling coil; ft/hr}
Arho = 1/Volume(AirH2O,T=Tav,P=Patm,w=wav) {average air density in coil; lb/ft^3}
DH = 0.01192 {hydraulic diameter from CHE's; ft}
mua = Viscosity(AirH2O,T=Tav,P=Patm,w=wav) {air viscosity; lb/ft-hr}
G = AMFR/(sigma*Afr) {mass flux; lb/ft^2-hr}
z = ha/(G*Cpa)*Pr^0.667 {ordinate for correlation with ReD, CHE's}
Pr = 0.71 {Air Prandtl number, which is roughly constant in the range of interest}
z = lookup(lookuprow(3,ReD),4) {entries based on fig 10.83, CHE's}

{Calculation of the number of transfer units for the water side based on the Dittus-Boelter
equation, the design water flow velocity, and the as-yet undetermined leaving water
temperature}
NTUw = UAw/(CCWMFR*Cpw) {Equation 19.27, Mitchell}
CCWMFR = Wrho*Ntubes*Atube*CCWV {cooling coil water mass flow rate; lbs/hr}
Wrho = 1/Volume(Water,T=EWT+1,P=Patm) {density of water; lb/ft^3}
EWT = 32.5 {entering water temperature; F}
Ntubes = DWt/S {number of tubes per row}
Atube = pi*TID^2/4 {inside area of tube; ft^2}
TID = TOD - 2*ttw {inside diameter of tube; ft}
ttw = 0.035/12 {thickness of tube wall; ft}
CCWV = 10*3600 {water velocity in tubes; ft/hr}
Cpw = SpecHeat(Water,T=EWT+1,P=Patm) {specific heat of water; Btu/lb-F}
UAw = hw*A {UA product for water (based on air side area!); Btu/hr-F}
hw = kw*Nuw/TID {Btu/ft^2-hr-F}
kw = Conductivity(Water,T=EWT+1,P=Patm) {Btu/hr-ft-F}
Nuw = 0.023*ReDw^0.8*Prw^0.4 {Dittus-Boelter equation 8.60, FMHT}
ReDw = Wrho*CCWV*TID/muw {Reynolds number for water}
muw = Viscosity(Water,T=EWT+1,P=Patm) {lb/ft-hr}
Prw = 12.9 {Prandtl number of water at EWT, Table A-6, FHMT}

{Determination of the number of transfer units for the wet tubes, and the effectiveness
assuming either completely wet or completely dry tubes from Compact Heat Exchangers,
Chapter 2.}
NTUwet = NTUa/(1 + mstar*NTUa/NTUw) {equation 19.29, Mitchell}
mstar = AMFR*Cs/(CCWMFR*Cpw) {equation 19.21, Mitchell}
Cs = (Enthalpy(AirH2O,T=LWT,P=Patm,R=1) - Enthalpy(AirH2O,T=EWT,P=Patm,R=1))/(LWT-EWT)
                                    {effective specific heat (cf. p 19-12, Mitchell); Btu/lb-F}
effwp = 1/mstar*(1 - exp(-gammaw*mstar)) {effectiveness of wet coils per pass}
gammaw = 1 - exp(-NTUwet/Nrows)
CCeffw = (((1 - effwp*mstar)/(1 - effwp))^Nrows - 1)/(((1 - effwp*mstar)/(1 - effwp))^Nrows - mstar)
                                    {cooling coil effectiveness for completely wet tubes}
effdp = 1/Cr*(1 - exp(-gammad*Cr)) {effectiveness of dry coils per pass}
gammad = 1 - exp(-NTUa/Nrows)
CCeffd = (((1 - effdp*Cr)/(1 - effdp))^Nrows - 1)/(((1 - effdp*Cr)/(1 - effdp))^Nrows - Cr)
                                    {cooling coil effectiveness for completely dry tubes}
Cwat = CCWMFR*Cpw {water heat capacity rate; Btu/hr-F}
Cair = AMFR*Cpa {air-water heat capacity rate, Btu/hr-F}
```

Cr = Cair/Cwat {heat capacity ratio}
gmin2 = Cr {minimum value of cooling coil pump control function}

{Determination of the leaving water temperature and check for consistency using the effectivenesses found above.}
CCLoad = CCWMFR*Cpw*(LWT - EWT)/12000 {cooling coil load based on temperature rise of water;tons}
CCLoadw = AMFR*CCeffw*(ENTHin - ENTHsat)/12000 {cooling coil load assuming completely wet tubes (equation 19.33, Mitchell)}
CCLoadd = AMFR*CCeffd*(ENTHin - ENTHsat)/12000 {cooling coil load assuming completely dry tubes, equation 19.33, Mitchell}
ENTHsat = Enthalpy(AirH2O,T=EWT,P=Patm,R=1) {enthalpy of saturated air-water at water inlet temp}

{Determination of the air and water side pressure drops for the cooling coil}
dPair = FF*(G^2*Avolin/(2*32.2*3600^2))*((Kc + 1 - sigma^2) + 2*(Avolout/Avolin - 1) + f*A*Avolav/(Ac*Avolin) - (1 - sigma^2 - Ke)*Avolout/Avolin)/(144*14.7)
{Air pressure drop (equation 2-26a, CHE's);atm}
FF = 2 {fudge factor for air side pressure drop to account for the fact that tubes are wet}
Avolav = (Avolout + Avolin)/2 {average air specific volume in cooling coil; ft^3/lb}
f = lookup(lookuprow(3,ReD),5) {friction factor from CHE's}
Kc = 0.67 {from figure 5-2 in CHE's; 4*(L/D)/ReD = 0.01; laminar flow}
Ke = -0.03 {from figure 5-2 in CHE's; 4*(L/D)/ReD = 0.01; laminar flow}
dPwatCC = fwat*Nrows*DHt*CCWV^2*Wrho/(2*TID*3600^2*32.2*144)
{water pressure drop, eq. 8-16 FHMT; psi}
fwat = 0.316*ReDw^(-0.25) {Moody friction factor, eq. 8.20 FHMT}

{The size of the ice harvester is calculated next. The ice harvester operates between 9:00 p.m. and 12:00 p.m on weekdays, and all weekend (from 9:00 p.m. on Friday until 12:00 p.m. on Monday). The ice produced must meet the weekly cooling coil energy requirement. The cooling coil operates a total of CChrs per week (Monday - Friday). The ice harvester is modeled using performance curves derived for the Frick RWB-II 177E rotary screw compressor using ammonia as the refrigerant. The model calculates the evaporative condensor capacity, the number of plates, the nominal refrigeration capacity, the compressor brake horsepower, the refrigerant pump brake horsepower, and the average power requirement of the ice harvester at a saturated discharge temperature of 95 F and a saturated suction temperature of 20 F.}

{Determination of required ice generation rate}
CCenergy = CCLoad*CChrs {weekly cooling coil energy requirement: ton-hours}
CChrs = 19.75 {weekly hours of operation for cooling coil}
TWIR = 1.01*CCenergy*12000/LHF {total weekly ice requirement: lbs}
LHF = 143.5 {latent heat of fusion of water; Btu/lb}
IGR = TWIR/IHhrs {ice generation rate: lb/hr}
IHhrs = WEhrs + 4*WDhrs {weekly hours of operation for ice harvester}
WEhrs = 63 {weekend hours of operation for the ice harvester}
WDhrs = 15 {weekday hours of operation for the ice harvester}

{Evaporative condensor: the refrigerant is cooled by means of an evaporative condensor unit. The capacity of this unit is its nominal capacity divided by the "heat rejection correction factor", HRCF. This factor is calculated by means of a seven parameter equation derived from data provided by "IMECO".}
qcond = NCC/HRCF {condensor heat rejection: tons}
HRCF = E1 + E2*SCT + E3*WB^3 + E4*WB^3*SCT + E5*WB^3*SCT^2 + E6*WB^5 + E7*WB^4*SCT
{heat rejection correction factor}
WB = 77 {design wet bulb temperature for hours of operation: F}

{Pure ice-making mode: here the power requirement and refrigeration capacity of the ice harvester is calculated when ice is being built on all plate sections.}

CPb = mult*(P1 + P2*SST + P3*SST^2 + P4*SDT + P5*SDT^2 + P6*SST*SDT)
{compressor brake horse power during pure build period: hp}
qevapb = mult*(C1 + C2*SST + C3*SST^2 + C4*SDT + C5*SDT^2 + C6*SST*SDT)
{refrigeration capacity during pure build period: tons}
SST = SET - SLL {saturated suction temperature: F}
SST = 20 {saturated suction temperature (should have been 16); F}
SLL = 4.0 {suction line losses (subcooling): F}
SDT = SCT + DLL {saturated discharge temperature: F}
SDT = 95 {saturated discharge temperature: F}
DLL = 2.0 {discharge line losses (superheat): F}
qcond = qevapb + CPb*2545/12000 {condensor heat rejection: tons}

{Determination of number of evaporator plates required}
qevapb = Ubarb*Nplates*Parea*(PEWT - SET)/12000 {tons}
Ubarb = 51 {average evaporator U-value during build period: Btu/hr-ft^2-F}
Parea = 3.833*6.833*2 {plate area: ft^2}
PEWT = 32.0 {plate entering water temperature: F}

{Defrost mode: in defrost mode, one section of evaporator plates is defrosted by re-routing the hot gas from the condensor to that section. Ice continues to build on the remaining plate sections during this time. The compressor power requirement and capacity are determined below.}

CPd = mult*(P1 + P2*DSST + P3*DSST^2 + P4*DSDT + P5*DSDT^2 + P6*DSST*DSDT)
{compressor brake horsepower during defrost period: hp}
qevapd = mult*(C1 + C2*DSST + C3*DSST^2 + C4*DSDT + C5*DSDT^2 + C6*DSST*DSDT)
{refrigeration capacity during defrost period; tons}
DSST = DSET - SLL {saturated evaporator temperature during defrost mode; F}
DSDT = DSCT + DDLL {saturated condensor temperature during defrost mode; F}
DDLL = 5 {discharge line losses (superheat) during defrost mode; F}
qevapd = Ubarb*(Nsect-1)/Nsect*Nplates*Parea*(PEWT - DSET)/12000 {tons}
Nsect = 4 {number of sequentially defrosted evaporator sections}
qrejd = qevapd + CPd*2545/12000 {heat rejected at defrosting evaporator plate; tons}

{Determination of saturated condensor temperature during defrost mode and mass flow rate for evaporator water pump}

qrejd = DSWMFR*Cpw*effdef*(DSCT - PEWT)/12000 {tons}
DSWMFR = EWMFR/Nsect {defrost section water mass flow rate: lb/hr}
EWMFR = PWMFR*Nplates {evaporator water mass flow rate; lb/hr}
PWMFR = 10*0.13368*62.41*60 {water mass flow rate per plate; lb/hr}
effdef = 1 - exp(-NTUdef) {evaporator effectiveness in condensor mode during defrost period}
NTUdef = (Udefp*Nplates*Parea/Nsect)/(DSWMFR*Cpw) {number of transfer units}
Udefp = 64.5 {evaporator U-value for defrosting section; Btu/hr-ft^2-F}

{Relationship between ice generation rate and refrigeration capacities during pure build and defrost mode: ice is built up on each plate during PBtime (while the harvester is operating in the pure ice building mode) and DBtime (while other evaporator sections are defrosting). The total cycle time is the sum of these last two periods and the period during which each plate is defrosting.}

IGR = Icemass/Ctime {ice generation rate: lbs/hr}
Icemass = xmax*Nplates*Parea*rhoice {lbs}
xmax = 0.375/12 {maximum ice thickness: ft}
rhoice = 57.5 {density of ice; lb /ft^3}
TIHC = Ncap*Ctime {Total heat capacity of ice built per cycle; ton-hrs}
TIHC = Icemass*LHF/12000 {ton-hrs}
Ctime = PBtime + DBtime + Dtime {cycle time: hours}

```
Dtime = 50/3600  {period during which section is defrosted; hrs}
DBtime = (Nsect - 1)*Dtime  {build time during defrost mode; hrs}

{Calculation of refrigerant pump brake horsepower (the liquid-vapor ratio at the evaporator
outlet is roughly 3:1)}
SEP = Pressure(Ammonia,T=SET,x=1)  {saturated evaporator pressure; psia}
SCP = Pressure(Ammonia,T=SCT,x=1)  {saturated condensor pressure; psia}
scool = 2.0  {amount of subcooling; F}
sheat = 10  {amount of superheat; F}
hcondo = Enthalpy(Ammonia,T=SCT-scool,P=SCP)  {outlet condensor Enthalpy; Btu/lb}
hevapi = hcondo  {inlet evaporator enthalpy; Btu/lb}
hevapo = Enthalpy(Ammonia,T=SET+sheat,P=SEP)  {outlet evaporator Enthalpy; Btu/lb}
RMFR = qevapb*12000/(hevapo - hevapi)  {refrigerant mass flow rate; lb/hr}
vam = Volume(Ammonia,T=SET,x=0.0)  {refrigerant specific Volume, ft^3/lb}
RVFR = 3*RMFR*vam  {refrigerant volumetric flow rate; ft^3/hr}
RPhead = 30  {refrigerant pump head; psi}
RPP = RPhead*RVFR/13750  {refrigerant pump brake horsepower; hp}

{Calculation of net electric power consumption and net refrigeration effect}
NPower = (CPb/ncomp*(1 - XDF) + CPd/ncomp*XDF + RPP/npump)*0.746
                {net electric power requirement; kW}
ncomp = 0.95  {overall efficiency of compressor}
npump = 0.65  {overall efficiency of refrigerant pump}
Ncap = qevapb - XDF*qrejd {net refrigeration capacity; tons}
XDF = (DBtime + Dtime)/Ctime  {defrost fraction of cycle time}

{Curve fit parameters for ice harvester compressor performance}
E1 = 2.271; E2 = -2.212e-2; E3 = 4.671e-5; E4 = -8.043e-7
E5 = 5.617e-9; E6 = 3.742e-9; E7 = -5.494e-9
P1 = 44.59; P2 = -3.4540; P3 = -0.023402; P4 =2.1514; P5 = 0.01560; P6 = 0.05615
C1 = 276.4; C2 = 6.1214; C3 = 0.04615; C4 = 0.0898; C5 = -0.001640; C6 =0.0001408

{The dimensions of the ice storage tank and the ice mass at the beginning of the simulation
(Monday at 12:00 a.m.) are calculated next. The ice storage tank must hold 20% more ice than
is generated over the weekend, in order to ensure that the effectiveness of the tank does not
fall below 1.0 on Friday afternoon. The tank height is 20 feet; the void fraction of the ice is
assumed to be 0.50.}
Ctnk = 1.20*IGR*WEhrs  {capacity of ice storage tank; lbs}
BIM = Ctnk - 12*IGR  {beginning ice mass; lbs}
Vtnk = Ctnk/(rhoice*0.50)  {tank volume; ft^3}
BA = Vtnk/ht  {tank base area; ft^2}
ht = 40  {tank height, ft}
rtnk = sqrt(BA/pi)  {tank radius; ft}

{Determination of required pipe size and pump size: water must be pumped between the
the ice storage tank and the cooling coil. This pipe run is assumed to be 300 feet; the water
velocity is assumed to be 10 feet per second.}
PL4 = 300  {pipe length; ft}
Dpipe4 = sqrt(4*CCWMFR/(pi*Wrho*CCWV))  {diameter of pipe running between ice tank and
                                        cooling coil; ft}
dPpipe4 = fwat4*2*PL4*CCWV^2*Wrho/(2*Dpipe4*3600^2*32.2*144)
                              {water pressure drop in pipe run 4, eq. 8-16 FHMT; psi}
fwat4 =  0.184*ReDw4^(-0.20)  {Moody friction factor for pipe 4, eq. 8.21 FHMT}
ReDw4 = Wrho*CCWV*Dpipe4/muw  {Reynolds number for water in pipe 4}
P4bhp = CCWMFR/Wrho*(dPpipe4 + dPwatCC)/13750  {brake horsepower for water pump 4, hp}
P4POW = P4bhp/nwpum*0.746  {power requirement for pump 4, kW}
nwpum = 0.65  {mechanical efficiency of water pumps}
```

{Miscellaneous quantities for TRNSED input file}
NomCap =qevapb {nominal capacity of ice harvester, tons}
Vtnkg = Vtnk*7.48055 {volume of storage tank, gal}
CCWVFR = CCWMFR/501.3 {cooling coil volumetric flow rate, gpm}
Dpipe4i = Dpipe4*12 {diameter of pipe 4, in}
TODi = TOD*12 {tube outside diameter, in}
TIDi = TID*12 {tube inside diameter, in}
tfi = tf*12 {thickness of fin, in}
FS = 1/8 {fin spacing, in}
Nfins = DHt*12/FS {number of fins}
Si = S*12 {distance between centers of tubes in a row; in }
Li = L*12 {row spacing; in }

LOOKUP TABLE:

Row	x	nf	ReD	z	f
1	0.00	1.00	500	0.0140	0.0310
2	0.50	0.78	1000	0.0105	0.0290
3	0.90	0.55	1500	0.0090	0.0250
4	1.10	0.45	2000	0.0080	0.0230
5	1.40	0.36	3000	0.0068	0.0220
6	1.80	0.27	4000	0.0060	0.0210
7	2.20	0.20	5000	0.0056	0.0205

SOLUTION:

BIM = 4305185 [lb]
CCLoad = 4991 [tons]
CCLoadd = 5244 [tons]
CCLoadw = 5029 [tons]
CCWVFR = 5700 [gpm]
Ctnk = 5117484 [lbs]
DHt = 8.45 [ft]
dPair = 0.00115 [atm]
Dpipe4i = 15.2 [in]
DWt = 175.32 [ft]
FS = 0.125 [in]
gmin2 = 0.21
ht = 40 [ft]
k = 102.3 [Btu/hr-ft-F]
Li = 0.866 [in]
Ncap = 809 [tons]
NCC = 1877 [tons]
Nfins = 811
NomCap = 889 [tons]
Nrows = 10
Ntubes = 2104
P4POW = 239 [kW]
PL4 = 300 [ft]
Si = 1.000 [in]
tfi = 0.0130 [in]
TIDi = 0.332 [in]
TODi = 0.402 [in]
Vtnkg = 1331534 [gal]

****

Chilled Water Storage Based System Sizing Program
Version 1 }

{This EES deck determines the parameters required by TRNSYS for all components of the chilled water storage loop. These components include: the cooling coil, the cooling coil pump, the storage tank, the chiller, the chiller pump, the cooling tower, and the cooling tower pump.

The dimensions of the cooling coil, the water mass flow rate through the cooling coil, and the temperature of the water as it leaves the cooling coil are calculated first. Design conditions for the cooling coil are: EDB = 95 F, EWB = 76 F, Nrows = 10, EWT = 40.5 F, an air face velocity of 400 fpm, a water flow velocity of ten fps and an outlet air volumetric flow rate of 525,560 cfm. The leaving dry bulb temperature is 46 F; the air is saturated when it leaves.}

```
{Determination of the cooling coil load associated with the design entering air state
and the desired outlet air state}
AMFR = AVFR/Avolout {air mass flow rate:lb/hr}
AVFR = 525560*60  {air volumetric flow rate at outlet: ft^3/hr}
Avolout = Volume(AirH2O,T=LDB,P=Patm,R=1)  {air specific volume at outlet; ft^3/lb}
Patm = 14.7  {ambient pressure; psia (should have been 14.4 psia)}
CCLoad = AMFR*(ENTHin - ENTHout)/12000  {cooling coil load; tons}
ENTHin =  Enthalpy(AirH2O,T=EDB,P=Patm,w=EHR)  {entering enthalpy; Btu/lb}
ENTHout =  Enthalpy(AirH2O,T=LDB,P=Patm,R=1)  {leaving enthalpy; Btu/lb}
EDB = 95.0  {entering dry bulb temperature; F}
EHR = 0.0150 {entering humidity ratio}
LDB = 46  {leaving dry bulb temperature; F}
LWT = 55.1 {leaving water temperature: F}

{Determination of core area, frontal area, duct width, duct height, volume, and effective
area of the cooling coil based on the design air flow velocity and the selection of surface
8.0 - 3/8T on page 224 of "Compact Heat Exchangers" (CHE's)}
AVin = 400*60  {air face velocity at inlet: feet/hr}
Afr = AMFR*Avolin/AVin  {frontal area of cooling coil; ft^2}
Avolin = Volume(AirH2O,T=EDB,P=Patm,w=EHR)  {air specific volume at inlet; ft^3/lb}
Ac = sigma*Afr  {core area of cooling coil; ft^2}
sigma = 0.534  {ratio from CHE's}
DHt= Afr/DWt  {duct height: ft}
DWt = num*DHt  {duct width}
V = Nrows*L*Afr  {cooling coil volume: ft^3}
Nrows = 10 {number of rows}
L = 0.866/12  {row spacing from CHE's: ft}
A = V*alpha  {heat transfer area; ft^2}
alpha = 179  {ratio from CHE's}

{Calculation of the number of transfer units for the air side based on material found
in  Chapters 3 and 11, "Fundamentals of Heat and Mass Transfer" (FHMT) by Incropera
and DeWitt, and in class notes from John Mitchell}
NTUa = UAa/(AMFR*Cpa)  {equation 19.26, Mitchell}
Cpa = SpecHeat(AirH2O,T=Tav,P=Patm,w=wav)  {average specific heat of air-water; Btu/lb-F}
Tav = (EDB + LDB)/2  {average air temperature in cooling coil: F}
wav = (EHR + LHR)/2  {average humidity ratio in cooling coil: F}
LHR =  HumRat(AirH2O,T=LDB,P=Patm,R=1)  {leaving humidity ratio (we assume air is saturated)}
1/UAa = 1/(no*ha*A)  {equation 11.1, FHMT: modified}
no = 1 - AfA*(1 - nf)  {overall efficiency of finned surface, equation 11.3, FHMT}
AfA = 0.913  {ratio of fin area to total area, from CHE's}
Lc = Lf + tf/2  {corrected fin length: ft}
```

```
Lf = (FD - TOD)/2  {fin length: ft}
FD = sqrt(4*L*S/pi)  {equivalent fin diameter for the rectangular plate fin
                    (Threlkeld, eq. 12.25): ft}
S = 1.00/12  {distance between centers of tubes in a row from CHE's; ft}
TOD = 0.402/12  {tube outside diameter: ft}
tf = 0.013/12  {fin thickness; ft}
k = 177/1.731  {thermal conductivity of aluminum fins: Btu-ft/hr-ft^2-F}
Ap = Lc*tf  {corrected fin profile area: ft^2}
x = (Lc^1.5)*(ha/(k*Ap))^0.5  {abscissa used in graph of fin efficiency}
nf = lookup(lookuprow(1,x),2)  {fin efficiency from figure 3.19, FHMT; r2c/r1 = 3}
ReD = Arho*AVcore*DH/mua  {Air Reynolds number}
AVcore = AVin/sigma  {air velocity inside the cooling coil; ft/hr}
Arho = 1/Volume(AirH2O,T=Tav,P=Patm,w=wav)  {average air density in coil: lb/ft^3}
DH = 0.01192  {hydraulic diameter from CHE's; ft}
mua = Viscosity(AirH2O,T=Tav,P=Patm,w=wav)  {air viscosity: lb/ft-hr}
G = AMFR/(sigma*Afr)  {mass flux; lb/ft^2-hr}
z = ha/(G*Cpa)*Pr^0.667  {ordinate for correlation with ReD, CHE's}
Pr = 0.71  {Air Prandtl number, which is roughly constant in the range of interest}
z = lookup(lookuprow(3,ReD),4)  {entries based on fig 10.83, CHE's}

{Calculation of the number of transfer units for the water side based on the Dittus-Boelter
equation, the design water flow velocity, and the as-yet undetermined leaving water
temperature}
NTUw = UAw/(CCWMFR*Cpw)  {Equation 19.27, Mitchell}
CCWMFR = Wrho*Ntubes*Atube*CCWV  {cooling coil water mass flow rate; lbs/hr}
Wrho = 1/Volume(Water,T=EWT,P=Patm)  {density of water; lb/ft^3}
EWT = 40.5 {entering water temperature: F}
Ntubes = DWt/S  {number of tubes per row}
Atube = pi*TID^2/4  {inside area of tube: ft^2}
TID = TOD - 2*ttw  {inside diameter of tube; ft}
ttw = 0.035/12  {thickness of tube wall; ft}
CCWV = 10*3600  {water velocity in tubes: ft/hr}
Cpw = SpecHeat(Water,T=EWT,P=Patm)  {specific heat of water: Btu/lb-F}
UAw = hw*A  {UA product for water (based on air side area!); Btu/hr-F}
hw = kw*Nuw/TID  {Btu/ft^2-hr-F}
kw = Conductivity(Water,T=EWT,P=Patm)  {Btu/hr-ft-F}
Nuw = 0.023*ReDw^0.8*Prw^0.4  {Dittus-Boelter equation 8.60, FMHT}
ReDw = Wrho*CCWV*TID/muw  {Reynolds number for water}
muw = Viscosity(Water,T=EWT,P=Patm)  {lb/ft-hr}
Prw = 10.26  {Prandtl number of water, Table A-6, FHMT}

{Determination of the number of transfer units for the wet tubes, and the effectiveness
assuming either completely wet or completely dry tubes from Compact Heat Exchangers,
Chapter 2.}
NTUwet = NTUa/(1 + mstar*NTUa/NTUw)  {equation 19.29, Mitchell}
mstar = AMFR*Cs/(CCWMFR*Cpw)  {equation 19.21, Mitchell}
Cs = (Enthalpy(AirH2O,T=LWT,P=Patm,R=1) - Enthalpy(AirH2O,T=EWT,P=Patm,R=1))/(LWT-EWT)
                    {effective specific heat (cf. p 19-12, Mitchell); Btu/lb-F}
effwp = 1/mstar*(1 - exp(-gammaw*mstar))  {effectiveness of wet coils per pass}
gammaw = 1 - exp(-NTUwet/Nrows)
CCeffw = (((1 - effwp*mstar)/(1 - effwp))^Nrows - 1)/(((1 - effwp*mstar)/(1 - effwp))^Nrows - mstar)
                    {cooling coil effectiveness for completely wet tubes}
effdp = 1/Cr*(1 - exp(-gammad*Cr))  {effectiveness of dry coils per pass}
gammad = 1 - exp(-NTUa/Nrows)
CCeffd = (((1 - effdp*Cr)/(1 - effdp))^Nrows - 1)/(((1 - effdp*Cr)/(1 - effdp))^Nrows - Cr)
                    {cooling coil effectiveness for completely dry tubes}
Cwat = CCWMFR*Cpw  {water heat capacity rate: Btu/hr-F}
```

Cair = AMFR*Cpa {air-water heat capacity rate. Btu/hr-F}
Cr = Cair/Cwat {heat capacity ratio}
g1min = Cr {minimum value of cooling coil pump control variable}

{Check for consistency using the effectivenesses found above.}
CCLoad = CCWMFR*Cpw*(LWT - EWT)/12000 {cooling coil load based on temperature rise of water:tons}
CCLoadw = AMFR*CCeffw*(ENTHin - ENTHsat)/12000 {cooling coil load assuming completely wet tubes (equation 19.33, Mitchell)}
CCLoadd = AMFR*CCeffd*(ENTHin - ENTHsat)/12000 {cooling coil load assuming completely dry tubes. equation 19.33, Mitchell}
ENTHsat = Enthalpy(AirH2O,T=EWT,P=Patm,R=1) {enthalpy of saturated air-water at water inlet temp}

{Determination of the air and water side pressure drops for the cooling coil}
dPair = FF*(G^2*Avolin/(2*32.2*3600^2))*((Kc + 1 - sigma^2) + 2*(Avolout/Avolin - 1) + f*A*Avolav/(Ac*Avolin) - (1 - sigma^2 - Ke)*Avolout/Avolin)/(144*14.7)
{Air pressure drop (equation 2-26a, CHE's);atm}
FF = 2 {fudge factor for air side pressure drop to account for the fact that the tubes are wet}
Avolav = (Avolout + Avolin)/2 {average air specific volume in cooling coil; ft^3/lb}
f = lookup(lookuprow(3,ReD),5) {friction factor from CHE's}
Kc = 0.67 {from figure 5-2 in CHE's; 4*(L/D)/ReD = 0.02: laminar flow}
Ke = -0.03 {from figure 5-2 in CHE's; 4*(L/D)/ReD = 0.02: laminar flow}
dPwatCC = fwat*Nrows*DHt*CCWV^2*Wrho/(2*TID*3600^2*32.2*144)
{water pressure drop. eq. 8-16 FHMT; psi}
fwat = 0.316*ReDw^(-0.25) {Moody friction factor, eq. 8.20 FHMT}

{The dimensions of the chilled water storage tank are calculated next. The tank must hold 5% more than enough water to meet the design water flow rate found above for the number of hours of daily cooling coil operation. The tank height is 50 feet.}
Vtnk = 1.05*(Dhrs*CCWMFR/Wrho) {volume of storage tank; ft^3}
Dhrs = 4.0 {daily hours of cooling coil operation}
Atnk = Vtnk/Htnk {area of tank's footprint; ft^2}
Htnk = 50 {tank height; ft}
rtnk = sqrt(Atnk/pi) {tank radius: ft}

{The design load and power requirement for the chiller are calculated below. The chiller must cool the water in the storage tank from MLWT to the chilled water set point, 40.0 F, in a period of 15 hours, since it operates between 9:00 p.m. and 12:00 p.m of the following day. It must also make up for losses through the storage tank walls, which take place 24 hours per day at an average (August) temperature of 80 F. The chiller is modelled as a ammonia vapor compression cycle operating at an average evaporator temperature of 35 F and an average condensor temperature of 90 F. The isentropic efficiency of the compressor is 0.67; the motor efficiency is 0.94}

{Determination of design chiller load}
CHWMFR = Vtnk*Wrho/15 {water mass flow rate through chiller; lb/hr}
WLoad = CHWMFR*Cpw*(MLWT - SPT)/12000 {chiller load due to cooling coil; tons}
MLWT = 54.90 {maximum leaving water temperature, F}
SPT = 40.0 {chiller set point temperature; F}
TLoad = UAtnk*(ADB - AWT)*24/(15*12000) {chiller load due to tank losses, tons}
UAtnk = Utnk*SAtnk {UA product for storage tank. Btu/hr-F}
Utnk = 0.0734 {tank loss coefficient; Btu/hr-ft^2-F}
SAtnk = 2*pi*rtnk*Htnk + Atnk {exposed surface area of tank, ft^2}
ADB = 80 {average dry bulb temperature. F}
AWT = (SPT + LWT)/2 {average tank water temperature: F}

```
CHLoad = WLoad + TLoad {total chiller load, tons}
CHLoadmin = 0.15*CHLoad {minimum chiller load, tons}
MEEWT = SPT + 0.15*(MLWT - SPT) {minimum entering evaporator water temperature, F}

{Refrigeration cycle states}
Pevap = Pressure(Ammonia,T=Tevap,x=1.0) {evaporator pressure; psia}
Tevap = 35 {evaporator temperature, F}
h1 = Enthalpy(Ammonia,T=Tevap,x=1.0) {Btu/lb}
Tsh = Tevap + 10 {superheated refrigerant temperature; F}
h1sh = Enthalpy(Ammonia,T=Tsh,P=Pevap) {Btu/lb}
s1sh = Entropy(Ammonia,T=Tsh,P=Pevap) {Btu/lb-F}
Pcon = Pressure(Ammonia,T=Tcon,x=1.0) {condensor pressure; psia}
Tcon = 90 {condensor temperature; F}
h2s = Enthalpy(Ammonia,P=Pcon,s=s1sh) {Btu/lb}
h2 = h1sh + (h2s - h1sh)/nc {Btu/lb}
nc = 0.67 {isentropic compressor efficiency}
Tsc = Tcon - 10 {subcooled refrigerant temperature; F}
h3 = Enthalpy(Ammonia,T=Tsc,P=Pcon) {Btu/lb}
h4 = h3

{Determination of cycle COP, power requirement, and heat rejection at condensor}
CHLoad = mfr*(h1 - h4)/12000 {Chiller load: tons}
Qcond = mfr*(h2 - h3)/12000 {heat rejection at condensor; tons}
Wcomp = mfr*(h2 - h1sh)/(3600*nmech)*1.055 {compressor power requirement; kW}
nmech = 0.94
COP = (h1 - h4)/(h2 - h1sh) {cycle COP}

{The next component to be sized is the cooling tower. The Marley selection procedure is
used to find the fan power. The design wet bulb temperature for the hours of operation
is 77 F, the design hot water temperature is 92 F, and the design cold water temperature
is 85 F. These values result in a "tower selection factor" of 6.3. The design dry bulb
temperature for hours of tower operation is 90 F.}

{Determination of water flow rate through cooling tower}
Qcond = CTWMFR*Cpw*(HWT - CWT)/12000 {heat rejection in cooling tower; tons}
HWT = 92 {hot water temperature from chiller condensor circuit; F}
CWT = 85 {cold water temperature leaving cooling tower; F}
CTWVFR = CTWMFR*7.48055/(Wrho*60) {volumetric water flow rate through cooling tower; GPM}
CTbhp = 60 {cooling tower brake horse power from Marley selection chart}
CTpow = CTbhp/nfan*0.746 {fan power requirement; kW}
nfan = 0.80

{Determination of air volumetric flow rate and sump volume: the air mass flow
rate is assumed to be 80% of the water mass flow rate. The contents of the sump
are assumed to be replaced every 15 minutes.}
CTAMFR = 0.8*CTWMFR {cooling tower air mass flow rate; lb/hr}
CTAVFR = CTAMFR*CTAvol {air volumetric flow rate through cooling tower; ft^3/hr}
CTAvol = Volume(AirH2O,T=TTow,P=Patm,R=1) {air specific volume at tower outlet: ft^3/lb}
TTow = 90 {Design tower air temperature: F}
Vsump = CTWVFR*15 {sump volume; gallons}

{Determination of required pipe sizes and pump sizes: water must be pumped between the
chiller and the cooling tower, between the chiller and the water storage tank, and between
the water storage tank and the cooling coil. The first pipe run is 100 feet, the second is 100
feet, and the third is 300 feet. The cooling coil water velocity is 10 feet per second; all other
water velocities are assumed to be 6 feet per second.}
PL1 = 100 {first pipe run: feet}
```

```
Dpipe1 = sqrt(4*CTWMFR/(pi*Wrho*WV))  {diameter of pipe running between chiller
                                        and cooling tower; ft}
WV = 6*3600  {water velocity between chiller, cooling tower, and storage tank; ft/hr}
dPpipe1 = fwat1*2*PL1*WV^2*Wrho/(2*Dpipe1*3600^2*32.2*144)
                              {water pressure drop in pipe 1, eq. 8-16 FHMT: psi}
fwat1 =  0.184*ReDw1^(-0.20)  {Moody friction factor for pipe 1, eq. 8.21 FHMT}
ReDw1 = Wrho*WV*Dpipe1/muw1  {Reynolds number for water in pipe 1}
muw1 = Viscosity(Water,T=85,P=Patm)  {viscosity of water in pipe 1; lb/ft-hr}
P1bhp = CTWMFR/Wrho*dPpipe1/13750  {brake horsepower for water pump 1, hp}
P1pow = P1bhp/nwpum*0.746  {power requirement for pump 1, kW}
PL2 = 100  {second pipe run; ft}
Dpipe2 = sqrt(4*CHWMFR/(pi*Wrho*WV))  {diameter of pipe running between chiller
                                        and water storage tank; ft}
dPpipe2 = fwat2*2*PL2*WV^2*Wrho/(2*Dpipe2*3600^2*32.2*144)
                              {water pressure drop in pipe 2, eq. 8-16 FHMT: psi}
fwat2 =  0.184*ReDw2^(-0.20)  {Moody friction factor for pipe 2, eq. 8.21 FHMT}
ReDw2= Wrho*WV*Dpipe2/muw  {Reynolds number for water in pipe 2}
P2bhp = CHWMFR/Wrho*dPpipe2/13750  {brake horsepower for water pump 2, hp}
P2pow = P2bhp/nwpum*0.746  {power requirement for pump 2, kW}
PL3 = 300  {third pipe run; ft}
Dpipe3 = sqrt(4*CCWMFR/(pi*Wrho*CCWV))  {diameter of pipe running between water storage
                                        tank and cooling coil; ft}
dPpipe3 = fwat3*2*PL3*CCWV^2*Wrho/(2*Dpipe3*3600^2*32.2*144)
                              {water pressure drop in pipe 3, eq. 8-16 FHMT: psi}
fwat3 =  0.184*ReDw3^(-0.20)  {Moody friction factor for pipe 3, eq. 8.21 FHMT}
ReDw3 = Wrho*CCWV*Dpipe3/muw  {Reynolds number for water in pipe 3}
P3bhp = CCWMFR/Wrho*(dPpipe3 + dPwatCC)/13750  {brake horsepower for water pump 3, hp}
P3pow = P3bhp/nwpum*0.746  {power requirement for pump 3, kW}
nwpum = 0.65  {mechanical efficiency of water pumps}

{Miscellaneous quantities required by TRNSED input file}
Dpipe1i = Dpipe1*12  {diameter of pipe 1; in}
Dpipe2i = Dpipe2*12  {diameter of pipe 2; in}
Dpipe3i = Dpipe3*12  {diameter of pipe 3; in}
CTAVFR2 = CTAVFR/60  {cooling tower air volumetric flow rate, cfm}
CHWVFR = CHWMFR/501.3  {evaporator water volumetric flow rate, gpm}
Vtnkg = Vtnk*7.48055  {volume of storage tank, gal}
CCWVFR = CCWMFR/501.3  {cooling coil volumetric flow rate, gpm}
TODi = TOD*12  {tube outside diameter, in}
TIDi = TID*12  {tube inside diameter, in}
tfi = tf*12  {thickness of fin, in}
FS = 1/8  {fin spacing, in}
Nfins = DHt*12/FS  {number of fins per tube pass}
Si = S*12  {distance between centers of tubes in a row; in }
Li = L*12  {row spacing; in }
```

SOLUTION:

```
CCLoad = 4325
CCLoadd = 4545
CCLoadw = 4382
CCWVFR = 7066          [gpm]
CHLoad = 1241   [tons]
CHLoadmin = 186        [tons]
CHWVFR = 1978          [gpm]
CTAVFR2 = 494348       [cfm]
```

CTpow = 56.0 [kW]
CTWVFR = 5067 [GPM]
DHt = 6.71 [ft]
dPair = 0.00116 [atm]
Dpipe1i = 18.6 [in]
Dpipe2i = 11.6 [in]
Dpipe3i = 17.0 [in]
DWt = 217.57 [ft]
FS = 0.125 [i n]
g1min = 0.17
Htnk = 50 [ft]
k = 102.3 [Btu/hr-ft-F]
Li = 0.866 [in]
MEEWT = 42.24 [F]
MLWT = 54.90 [F]
Nfins = 644
Nrows = 10
Ntubes = 2611
P1pow = 1.2 [kW]
P2pow = 0.9 [kW]
P3pow = 230 [kW]
PL1 = 100 [ft]
PL2 = 100 [ft]
PL3 = 300 [ft]
Qcond = 1491 [tons]
Si = 1.000 [in]
SPT = 40.00 [F]
tfi = 0.0130 [in]
TIDi = 0.332 [in]
TODi = 0.402 [in]
Vsump = 76005 [gal]
Vtnkg = 1775283 [gal]
Wcomp = 878 [kW]

****

Chilled Water Storage Loop Sizing Program }

{This EES deck determines the parameters required by TRNSYS for all components of the chilled water storage loop. These components include: the cooling coil, the cooling coil pump, the storage tank, the chiller, the chiller pump, the cooling tower, and the cooling tower pump.

The dimensions of the cooling coil, the water mass flow rate through the cooling coil, and the temperature of the water as it leaves the cooling coil are calculated first. Design conditions for the cooling coil are: EDB = 95 F, EWB = 76 F, LDB = 47.2 F, LWB = 47.2 F (saturated air), EWT = 40.5 F, an air face velocity of 400 fpm, a water flow velocity of ten fps and an air mass flow rate of 2,481,840 lb/hr.}

{Determination of the cooling coil load associated with the design entering air state and the desired outlet air state}
AMFR = 2481840 {air mass flow rate;lb/hr}
LDB = 47.2 {leaving dry bulb temperature; F}
Patm = 14.7 {ambient pressure;psia (should have been 14.4 psia)}
CCLoad = AMFR*(ENTHin - ENTHout)/12000 {cooling coil load; tons}
ENTHin = Enthalpy(AirH2O,T=EDB,P=Patm,w=EHR) {entering enthalpy; Btu/lb}
ENTHout = Enthalpy(AirH2O,T=LDB,P=Patm,R=1) {leaving enthalpy; Btu/lb}
EDB = 95 {entering dry bulb temperature; F}
EHR = 0.0150 {entering humidity ratio}
LWT = 54.83 {leaving water temperature; F}

{Determination of core area, frontal area, duct width, duct height, volume, and effective area of the cooling coil based on the design air flow velocity and the selection of surface 8.0 - 3/8T on page 224 of "Compact Heat Exchangers" (CHE's)}
AVin = 400*60 {air face velocity at inlet; feet/hr}
Afr = AMFR*Avolin/AVin {frontal area of cooling coil; ft^2}
Avolin = Volume(AirH2O,T=EDB,P=Patm,w=EHR) {air specific volume at inlet; ft^3/lb}
Ac = sigma*Afr {core area of cooling coil; ft^2}
sigma = 0.534 {ratio from CHE's}
DHt= Afr/DWt {duct height; ft}
DWt = num*DHt {duct width (num is the duct "aspect ratio"); ft}
V = Nrows*L*Afr {cooling coil volume; ft^3}
Nrows = 9 {number of rows}
L = 0.866/12 {row spacing from CHE's; ft}
A = V*alpha {heat transfer area; ft^2}
alpha = 179 {ratio from CHE's}

{Calculation of the number of transfer units for the air side based on material found in Chapters 3 and 11, "Fundamentals of Heat and Mass Transfer" (FHMT) by Incropera and DeWitt, and in class notes from John Mitchell}
NTUa = UAa/(AMFR*Cpa) {equation 19.26, Mitchell}
Cpa = SpecHeat(AirH2O,T=Tav,P=Patm,w=wav) {average specific heat of air-water; Btu/lb-F}
Tav = (EDB + LDB)/2 {average air temperature in cooling coil; F}
wav = (EHR + LHR)/2 {average humidity ratio in cooling coil; F}
LHR = HumRat(AirH2O,T=LDB,P=Patm,R=1) {leaving humidity ratio}
1/UAa = 1/(no*ha*A) {equation 11.1, FHMT; modified}
no = 1 - AfA*(1 - nf) {overall efficiency of finned surface, equation 11.3, FHMT}
AfA = 0.913 {ratio of fin area to total area, from CHE's}
Lc = Lf + tf/2 {corrected fin length; ft}
Lf = (FD - TOD)/2 {fin length; ft}
FD = sqrt(4*L*S/pi) {equivalent fin diameter for the rectangular plate fin (Threlkeld, eq. 12.25); ft}
S = 1.00/12 {distance between centers of tubes in a row from CHE's; ft}
TOD = 0.402/12 {tube outside diameter; ft}
tf = 0.013/12 {fin thickness; ft}
k = 177/1.731 {thermal conductivity of aluminum fins; Btu-ft/hr-ft^2-F}
Ap = Lc*tf {corrected fin profile area; ft^2}
x = (Lc^1.5)*(ha/(k*Ap))^0.5 {abscissa used in graph of fin efficiency}
nf = lookup(lookuprow(1,x),2) {fin efficiency from figure 3.19, FHMT; r2c/r1 = 3}
ReD = Arho*AVcore*DH/mua {Air Reynolds number}
AVcore = AVin/sigma {air velocity inside the cooling coil; ft/hr}
Arho = 1/Volume(AirH2O,T=Tav,P=Patm,w=wav) {average air density in coil; lb/ft^3}
DH = 0.01192 {hydraulic diameter from CHE's; ft}
mua = Viscosity(AirH2O,T=Tav,P=Patm,w=wav) {air viscosity; lb/ft-hr}
G = AMFR/(sigma*Afr) {mass flux; lb/ft^2-hr}

```
z = ha/(G*Cpa)*Pr^0.667  {ordinate for correlation with ReD, CHE's}
Pr = 0.71  {Air Prandtl number, which is roughly constant in the range of interest}
z = lookup(lookuprow(3,ReD),4)  {entries based on fig 10.83, CHE's}

{Calculation of the number of transfer units for the water side based on the Dittus-Boelter
equation, the design water flow velocity, and the as-yet undetermined leaving water
temperature}
NTUw = UAw/(CCWMFR*Cpw)  {Equation 19.27, Mitchell}
CCWMFR = Wrho*Ntubes*Atube*CCWV  {cooling coil water mass flow rate; lbs/hr}
Wrho = 1/Volume(Water,T=EWT,P=Patm)  {density of water; lb/ft^3}
EWT = 40.5 {entering water temperature; F}
Ntubes = DWt/S  {number of tubes per row}
Atube = pi*TID^2/4  {inside area of tube; ft^2}
TID = TOD - 2*ttw  {inside diameter of tube; ft}
ttw = 0.035/12  {thickness of tube wall: ft}
CCWV = 10*3600  {water velocity in tubes: ft/hr}
Cpw = SpecHeat(Water,T=EWT,P=Patm)  {specific heat of water; Btu/lb-F}
UAw = hw*A  {UA product for water (based on air side area!); Btu/hr-F}
hw = kw*Nuw/TID  {Btu/ft^2-hr-F}
kw = Conductivity(Water,T=EWT,P=Patm)  {Btu/hr-ft-F}
Nuw = 0.023*ReDw^0.8*Prw^0.4  {Dittus-Boelter equation 8.60, FMHT}
ReDw = Wrho*CCWV*TID/muw  {Reynolds number for water}
muw = Viscosity(Water,T=EWT,P=Patm)  {lb/ft-hr}
Prw = 10.26  {Prandtl number of water. Table A-6, FHMT}

{Determination of the number of transfer units for the wet tubes, and the effectiveness
assuming either completely wet or completely dry tubes from Compact Heat Exchangers,
Chapter 2.}
NTUwet = NTUa/(1 + mstar*NTUa/NTUw)  {equation 19.29, Mitchell}
mstar = AMFR*Cs/(CCWMFR*Cpw)  {equation 19.21, Mitchell}
Cs = (Enthalpy(AirH2O,T=LWT,P=Patm,R=1) - Enthalpy(AirH2O,T=EWT,P=Patm,R=1))/(LWT-EWT)
                                   {effective specific heat (cf. p 19-12, Mitchell); Btu/lb-F}
effwp = 1/mstar*(1 - exp(-gammaw*mstar))  {effectiveness of wet coils per pass}
gammaw = 1 - exp(-NTUwet/Nrows)
CCeffw = (((1 - effwp*mstar)/(1 - effwp))^Nrows - 1)/(((1 - effwp*mstar)/(1 - effwp))^Nrows - mstar)
                                   {cooling coil effectiveness for completely wet tubes}
effdp = 1/Cr*(1 - exp(-gammad*Cr))  {effectiveness of dry coils per pass}
gammad = 1 - exp(-NTUa/Nrows)
CCeffd = (((1 - effdp*Cr)/(1 - effdp))^Nrows - 1)/(((1 - effdp*Cr)/(1 - effdp))^Nrows - Cr)
                                   {cooling coil effectiveness for completely dry tubes}
Cwat = CCWMFR*Cpw  {water heat capacity rate; Btu/hr-F}
Cair = AMFR*Cpa  {air-water heat capacity rate, Btu/hr-F}
Cr = Cair/Cwat  {heat capacity ratio}

{Check for consistency using the effectivenesses found above.}
CCLoad = CCWMFR*Cpw*(LWT - EWT)/12000  {cooling coil load based on temperature rise
                                   of water:tons}
CCLoadw = AMFR*CCeffw*(ENTHin - ENTHsat)/12000  {cooling coil load assuming
                                   completely wet tubes (equation 19.33, Mitchell)}
CCLoadd = AMFR*CCeffd*(ENTHin - ENTHsat)/12000  {cooling coil load assuming
                                   completely dry tubes, equation 19.33, Mitchell}
ENTHsat = Enthalpy(AirH2O,T=EWT,P=Patm,R=1)  {enthalpy of saturated air-water at water
                                   inlet temp}
g1min = Cair/Cwat  {minimum value of combined control variable for cooling coil water}

{Determination of the air and water side pressure drops for the cooling coil}
```

```
dPair = FF*(G^2*Avolin/(2*32.2*3600^2))*((Kc + 1 - sigma^2) + 2*(Avolout/Avolin - 1) +
f*A*Avolav/(Ac*Avolin) )/(144*14.7)
                                        {Air pressure drop (equation 2-26a (No exit effect), CHE's);atm}
FF = 2  {fudge factor for air side pressure drop to account for the fact that the tubes are wet}
Avolout = Volume(AirH2O,T=LDB,P=Patm,w=LHR)   {air specific volume at outlet; ft^3/lb}
Avolav = (Avolout + Avolin)/2  {average air specific volume in cooling coil; ft^3/lb}
f = lookup(lookuprow(3,ReD),5)  {friction factor from CHE's}
Kc = 0.67  {from figure 5-2 in CHE's; 4*(L/D)/ReD = 0.05; laminar flow}
dPwatCC = fwat*Nrows*DHt*CCWV^2*Wrho/(2*TID*3600^2*32.2*144)
                              {water pressure drop, eq. 8-16 FHMT; psi}
fwat = 0.316*ReDw^(-0.25)  {Moody friction factor, eq. 8.20 FHMT}

{The dimensions of the chilled water storage tank are calculated next. The tank must hold
5% more than enough water to meet the design water flow rate found above for the number of
hours the coil is in operation per day. The tank height is 50 feet.}
Vtnk = 1.05*(DHours*CCWMFR/Wrho)  {volume of storage tank; ft^3}
DHours = 5.5  {Daily hours of cooling coil operation}
Atnk = Vtnk/Htnk  {area of tank's footprint; ft^2}
Htnk = 50  {tank height; ft}
rtnk = sqrt(Atnk/pi)  {tank radius; ft}

{The design load and power requirement for the chiller are calculated below. The chiller
must cool the water in the storage tank from the maximum leaving water temperature (found
from TRNSYS) to the chilled water set point, 40.0 F, in a period of 15 hours, since it operates
between 9:00 p.m. and 12:00 p.m of the following day. It must also make up for losses through
the storage tank walls, which take place 24 hours per day at an average (August) temperature
of 80 F. The chiller is modelled as a ammonia vapor compression cycle operating at an average
evaporator temperature of 35 F and an average condensor temperature of 90 F. The isentropic
efficiency of the compressor is 0.67; the motor efficiency is 0.94}

{Determination of design chiller load}
CHWMFR = Vtnk*Wrho/15  {water mass flow rate through chiller; lb/hr}
WLoad = CHWMFR*Cpw*(MLWT - SPT)/12000  {chiller load due to cooling coil: tons}
MLWT = 54.60 {maximum leaving water temperature}
SPT = 40.0  {chiller set point temperature; F}
TLoad = UAtnk*(ADB - AWT)*24/(15*12000)  {chiller load due to tank losses, tons}
UAtnk = Utnk*SAtnk  {UA product for storage tank, Btu/hr-F}
Utnk = 0.0734  {tank loss coefficient; Btu/hr-ft^2-F}
SAtnk = 2*pi*rtnk*Htnk + Atnk  {exposed surface area of tank, ft^2}
ADB = 80  {average dry bulb temperature, F}
AWT = (SPT + MLWT)/2  {average tank water temperature; F}
CHLoad = WLoad + TLoad  {total chiller load, tons}
CHLoadmin = 0.15*CHLoad  {minimum chiller load, tons}
MEEWT = SPT + 0.15*(MLWT - SPT)  {minimum entering evaporator water temperature, C}

{Refrigeration cycle states}
Pevap = Pressure(Ammonia,T=Tevap,x=1.0)  {evaporator pressure; psia}
Tevap = 35  {evaporator temperature, F}
h1 = Enthalpy(Ammonia,T=Tevap,x=1.0)  {Btu/lb}
Tsh = Tevap + 10  {superheated refrigerant temperature; F}
h1sh = Enthalpy(Ammonia,T=Tsh,P=Pevap)  {Btu/lb}
s1sh = Entropy(Ammonia,T=Tsh,P=Pevap)  {Btu/lb-F}
Pcon =  Pressure(Ammonia,T=Tcon,x=1.0)  {condensor pressure; psia}
Tcon = 90  {condensor temperature: F}
h2s = Enthalpy(Ammonia,P=Pcon,s=s1sh)  {Btu/lb}
h2 = h1sh + (h2s - h1sh)/nc  {Btu/lb}
nc = 0.67  {isentropic compressor efficiency}
```

```
Tsc = Tcon - 10 {subcooled refrigerant temperature: F}
h3 = Enthalpy(Ammonia,T=Tsc,P=Pcon) {Btu/lb}
h4 = h3

{Determination of cycle COP, power requirement, and heat rejection at condensor}
CHLoad = mfr*(h1 - h4)/12000 {Chiller load; tons}
Qcond = mfr*(h2 - h3)/12000 {heat rejection at condensor: tons}
Wcomp = mfr*(h2 - h1sh)/(3600*nmech)*1.055 {compressor power requirement; kW}
nmech = 0.94
COP = (h1 - h4)/(h2 - h1sh) {cycle COP}

{The next component to be sized is the cooling tower. The Marley selection procedure is
used to find the fan power. The design wet bulb temperature for the hours of operation
is 77 F, the design hot water temperature is 92 F, and the design cold water temperature
is 85 F. These values result in a "tower selection factor" of 6.3. The design dry bulb
temperature for hours of tower operation is 90 F.}

{Determination of water flow rate through cooling tower}
Qcond = CTWMFR*Cpw*(HWT - CWT)/12000 {heat rejection in cooling tower; tons}
HWT = 92 {hot water temperature from chiller condensor circuit; F}
CWT = 85 {cold water temperature leaving cooling tower: F}
CTWVFR = CTWMFR*7.48055/(Wrho*60) {volumetric water flow rate through cooling tower; GPM}
CTbhp = 60 {cooling tower brake horse power from Marley selection chart}
CTpow = CTbhp/nfan*0.746 {fan power requirement: kW}
nfan = 0.80

{Determination of air volumetric flow rate and sump volume: the air mass flow
rate is assumed to be 80% of the water mass flow rate. The contents of the sump
are assumed to be replaced every 15 minutes.}
CTAMFR = 0.8*CTWMFR {cooling tower air mass flow rate; lb/hr}
CTAVFR = CTAMFR*CTAvol {air volumetric flow rate through cooling tower; ft^3/hr}
CTAvol = Volume(AirH2O,T=TTow,P=Patm,R=1) {air specific volume at tower outlet: ft^3/lb}
TTow = 90 {Design tower air temperature: F}
Vsump = CTWVFR*15 {sump volume: gallons}

{Determination of required pipe sizes and pump sizes: water must be pumped between the
chiller and the cooling tower, between the chiller and the water storage tank, and between
the water storage tank and the cooling coil. The first pipe run is 100 feet, the second is 100
feet, and the third is 300 feet. All water velocities are assumed to be 6 feet per second.}
PL1 = 100 {first pipe run: feet}
Dpipe1 = sqrt(4*CTWMFR/(pi*Wrho*WV)) {diameter of pipe running between chiller
                                        and cooling tower: ft}
WV = 6*3600 {water velocity between chiller, cooling tower, and storage tank; ft/hr}
dPpipe1 = fwat1*2*PL1*WV^2*Wrho/(2*Dpipe1*3600^2*32.2*144)
                              {water pressure drop in pipe 1, eq. 8-16 FHMT; psi}
fwat1 = 0.184*ReDw1^(-0.20) {Moody friction factor for pipe 1, eq. 8.21 FHMT}
ReDw1 = Wrho*WV*Dpipe1/muw1 {Reynolds number for water in pipe 1}
muw1 = Viscosity(Water,T=85,P=Patm) {viscosity of water in pipe 1; lb/ft-hr}
P1bhp = CTWMFR/Wrho*dPpipe1/13750 {brake horsepower for water pump 1, hp}
P1pow = P1bhp/nwpum*0.746 {power requirement for pump 1, kW}
PL2 = 100 {second pipe run: ft}
Dpipe2 = sqrt(4*CHWMFR/(pi*Wrho*WV)) {diameter of pipe running between chiller
                                        and water storage tank: ft}
dPpipe2 = fwat2*2*PL2*WV^2*Wrho/(2*Dpipe2*3600^2*32.2*144)
                              {water pressure drop in pipe 2, eq. 8-16 FHMT; psi}
fwat2 = 0.184*ReDw2^(-0.20) {Moody friction factor for pipe 2, eq. 8.21 FHMT}
ReDw2= Wrho*WV*Dpipe2/muw {Reynolds number for water in pipe 2}
```

P2bhp = CHWMFR/Wrho*dPpipe2/13750 {brake horsepower for water pump 2, hp}
P2pow = P2bhp/nwpum*0.746 {power requirement for pump 2, kW}
PL3 = 300 {third pipe run: ft}
Dpipe3 = sqrt(4*CCWMFR/(pi*Wrho*CCWV)) {diameter of pipe running between water storage tank and cooling coil; ft}
dPpipe3 = fwat3*2*PL3*CCWV^2*Wrho/(2*Dpipe3*3600^2*32.2*144) {water pressure drop in pipe 3, eq. 8-16 FHMT; psi}
fwat3 = 0.184*ReDw3^(-0.20) {Moody friction factor for pipe 3, eq. 8.21 FHMT}
ReDw3 = Wrho*CCWV*Dpipe3/muw {Reynolds number for water in pipe 3}
P3bhp = CCWMFR/Wrho*(dPpipe3 + dPwatCC)/13750 {brake horsepower for water pump 3, hp}
P3pow = P3bhp/nwpum*0.746 {power requirement for pump 3, kW}
nwpum = 0.65 {mechanical efficiency of water pumps}

{Miscellaneous quantities for TRNSED input file}
Dpipe1i = Dpipe1*12 {diameter of pipe 1; in}
Dpipe2i = Dpipe2*12 {diameter of pipe 2; in}
Dpipe3i = Dpipe3*12 {diameter of pipe 3; in}
CTAVFR2 = CTAVFR/60 {cooling tower air volumetric flow rate, cfm}
CHWVFR = CHWMFR/501.3 {evaporator water volumetric flow rate, gpm}
Vtnkg = Vtnk*7.48055 {volume of storage tank, gal}
CCWVFR = CCWMFR/501.3 {cooling coil volumetric flow rate, gpm}
TODi = TOD*12 {tube outside diameter, in}
TIDi = TID*12 {tube inside diameter, in}
tfi = tf*12 {thickness of fin, in}
FS = 1/8 {fin spacing, in}
Nfins = DHt*12/FS {number of fins per tube pass}
Si = S*12 {distance between centers of tubes in a row; in }
Li = L*12 {row spacing: in }

SOLUTION:

CCLoad = 4257
CCLoadd = 4512
CCLoadw = 4331
CCWVFR = 7086 [gpm]
CHLoad = 1677 [tons]
CHLoadmin = 252 [tons]
CHWVFR = 2728 [gpm]
CTAVFR2 = 667743 [cfm]
CTpow = 56.0 [kW]
DHt = 6.79 [ft]
dPair = 0.001163 [atm]
Dpipe1i = 21.6 [in]
Dpipe2i = 13.6 [in]
Dpipe3i = 17.0 [in]
FS = 0.125 [in]
g1min = 0.17
Htnk = 50.00 [ft]
k = 102.253 [Btu/hr-ft-F]
Li = 0.866 [in]
MEEWT = 42.19 [F]
MLWT = 54.60 [F]
Nfins = 652
Nrows = 9
Ntubes = 2618
P1pow = 1.4 [kW]

P2pow = 1.1	[kW]
P3pow = 212	[kW]
PL1 = 100.0	[ft]
PL2 = 100.0	[ft]
PL3 = 300.0	[ft]
Qcond = 2015	[tons]
Si = 1.000	[in]
SPT = 40.00	[F]
tfi = 0.0130	[in]
TIDi = 0.332	[in]
TODi = 0.402	[i n]
Vsump = 102664	[gal]
Vtnkg = 2448110	[gal]
Wcomp = 1186	[kW]

****

Ice Storage Loop Sizing Program }

{This EES deck determines the parameters required by TRNSYS for all components of the ice storage loop. These components include: the cooling coil, the cooling coil pump, the ice harvester, the ice storage tank, the cooling tower, and the cooling tower pump.

The dimensions of the cooling coil, the water mass flow rate through the cooling coil, the number of rows required, and the temperature of the water as it leaves the cooling coil are calculated first. Design conditions for the cooling coil are: EDB = 47.2 F, EWB = 47.2 F (saturated air), LDB = 40 F, LWB = 40 F (also saturated), EWT = 32.5 F, an air face velocity of 400 fpm, a water flow velocity of ten fps and an outlet air volumetric flow rate of 525,560 cfm.}

{Determination of dry air mass flow rate and cooling coil load associated with the design entering air state and the desired outlet air state}
AMFR = AVFR/Avolout {air mass flow rate;lb/hr}
AVFR = 525560*60 {air volumetric flow rate at outlet;ft^3/hr}
Avolout = Volume(AirH2O,T=LDB,P=Patm,R=1) {air specific volume at outlet; ft^3/lb}
LDB = 40 {leaving dry bulb temperature; F}
Patm = 14.7 {ambient pressure; psia (should have been 14.4 psia)}
CCLoad = AMFR*(ENTHin - ENTHout)/12000 {cooling coil load: tons}
ENTHin = Enthalpy(AirH2O,T=EDB,P=Patm,R=1) {entering enthalpy; Btu/lb}
ENTHout = Enthalpy(AirH2O,T=LDB,P=Patm,R=1) {leaving enthalpy; Btu/lb}
EDB = 47.2 {entering dry bulb temperature; F}

{Determination of core area, frontal area, duct width, duct height, volume, and effective area of the cooling coil based on the design air flow velocity and the selection of surface 8.0 - 3/8T on page 224 of "Compact Heat Exchangers" (CHE's)}
AVin = 400*60 {air face velocity at inlet (water storage loop cooling coil); feet/hr}
Afr = AMFR*Avolam/AVin {frontal area of cooling coil: ft^2}

```
Avolam = Volume(AirH2O,T=ADB,P=Patm,w=AHR)  {air specific volume at water storage loop
                                             inlet; ft^3/lb}
ADB = 95  {ambient dry bulb temperature; F}
AHR = 0.0150  {ambient humidity ratio}
Ac = sigma*Afr  {core area of cooling coil, ft^2}
sigma = 0.534  {ratio from CHE's}
DHt= Afr/DWt  {duct height; ft}
DWt = num*DHt  {duct width; ft}
num = 5.5 {ratio of duct width to duct height}
V = Nrows*L*Afr  {cooling coil volume; ft^3}
Nrows = 3 {number of rows}
L = 0.866/12  {row spacing from CHE's; ft}
A = V*alpha  {heat transfer area; ft^2}
alpha = 179  {ratio from CHE's}

{Calculation of the number of transfer units for the air side based on material found
in  Chapters 3 and 11, "Fundamentals of Heat and Mass Transfer" (FHMT) by Incropera
and DeWitt, and in class notes from John Mitchell}
NTUa = UAa/(AMFR*Cpa)  {equation 19.26, Mitchell}
Cpa = SpecHeat(AirH2O,T=Tav,P=Patm,w=wav)  {average specific heat of air-water; Btu/lb-F}
Tav = (EDB + LDB)/2  {average air temperature in cooling coil; F}
wav = (EHR + LHR)/2  {average humidity ratio in cooling coil; F}
LHR = HumRat(AirH2O,T=LDB,P=Patm,R=1)  {leaving humidity ratio}
EHR = HumRat(AirH2O,T=EDB,P=Patm,R=1)  {entering humidity ratio}
1/UAa = 1/(no*ha*A)  {equation 11.1, FHMT; modified}
no = 1 - AfA*(1 - nf)  {overall efficiency of finned surface, equation 11.3, FHMT}
AfA = 0.913  {ratio of fin area to total area, from CHE's}
Lc = Lf + tf/2  {corrected fin length; ft}
Lf = (FD - TOD)/2  {fin length; ft}
FD = sqrt(4*L*S/pi)  {equivalent fin diameter for the rectangular plate fin
                     (Threlkeld, eq. 12.25); ft}
S = 1.00/12  {distance between centers of tubes in a row from CHE's; ft}
TOD = 0.402/12  {tube outside diameter; ft}
tf = 0.013/12  {fin thickness; ft}
k = 177/1.731  {thermal conductivity of aluminum fins; Btu-ft/hr-ft^2-F}
Ap = Lc*tf  {corrected fin profile area; ft^2}
x = (Lc^1.5)*(ha/(k*Ap))^0.5  {abscissa used in graph of fin efficiency}
nf = lookup(lookuprow(1,x),2)  {fin efficiency from figure 3.19, FHMT; r2c/r1 = 3}
ReD = Arho*AVcore*DH/mua  {Air Reynolds number}
AVcore = AVin/sigma  {air velocity inside the cooling coil; ft/hr}
Arho = 1/Volume(AirH2O,T=Tav,P=Patm,w=wav)  {average air density in coil; lb/ft^3}
DH = 0.01192  {hydraulic diameter from CHE's; ft}
mua = Viscosity(AirH2O,T=Tav,P=Patm,w=wav)  {air viscosity; lb/ft-hr}
G = AMFR/(sigma*Afr)  {mass flux; lb/ft^2-hr}
z = ha/(G*Cpa)*Pr^0.667  {ordinate for correlation with ReD, CHE's}
Pr = 0.71  {Air Prandtl number, which is roughly constant in the range of interest}
z = lookup(lookuprow(3,ReD),4)  {entries based on fig 10.83, CHE's}

{Calculation of the number of transfer units for the water side based on the Dittus-Boelter
equation, the design water flow velocity, and the as-yet undetermined leaving water
temperature}
NTUw = UAw/(CCWMFR*Cpw)  {Equation 19.27, Mitchell}
CCWMFR = Wrho*Ntubes*Atube*CCWV  {cooling coil water mass flow rate; lbs/hr}
Wrho = 1/Volume(Water,T=EWT+1,P=Patm)  {density of water; lb/ft^3}
EWT = 32.5 {entering water temperature; F}
Ntubes = DWt/S  {number of tubes per row}
Atube = pi*TID^2/4  {inside area of tube; ft^2}
```

```
TID = TOD - 2*ttw {inside diameter of tube; ft}
ttw = 0.035/12 {thickness of tube wall; ft}
CCWV = 10*3600 {water velocity in tubes; ft/hr}
Cpw = SpecHeat(Water,T=EWT+1,P=Patm) {specific heat of water; Btu/lb-F}
UAw = hw*A {UA product for water (based on air side area!); Btu/hr-F}
hw = kw*Nuw/TID {Btu/ft^2-hr-F}
kw = Conductivity(Water,T=EWT+1,P=Patm) {Btu/hr-ft-F}
Nuw = 0.023*ReDw^0.8*Prw^0.4 {Dittus-Boelter equation 8.60, FMHT}
ReDw = Wrho*CCWV*TID/muw {Reynolds number for water}
muw = Viscosity(Water,T=EWT+1,P=Patm) {lb/ft-hr}
Prw = 12.9 {Prandtl number of water at EWT , Table A-6, FHMT}

{Determination of the number of transfer units for the wet tubes, and the effectiveness
assuming completely dry tubes or completely wet tubes from Compact Heat Exchangers,
Chapter 2.}
NTUwet = NTUa/(1 + mstar*NTUa/NTUw) {equation 19.29, Mitchell}
mstar = AMFR*Cs/(CCWMFR*Cpw) {equation 19.21, Mitchell}
Cs = (Enthalpy(AirH2O,T=LWT,P=Patm,R=1) - Enthalpy(AirH2O,T=EWT,P=Patm,R=1))/(LWT-EWT)
                                        {effective specific heat (cf. p 19-12, Mitchell); Btu/lb-F}
effwp = 1/mstar*(1 - exp(-gammaw*mstar)) {effectiveness of wet coils per pass}
gammaw = 1 - exp(-NTUwet/Nrows)
CCeffw = (((1 - effwp*mstar)/(1 - effwp))^Nrows - 1)/(((1 - effwp*mstar)/(1 - effwp))^Nrows - mstar)
                                        {cooling coil effectiveness for completely wet tubes}
effdp = 1/Cr*(1 - exp(-gammad*Cr)) {effectiveness of dry coils per pass}
gammad = 1 - exp(-NTUa/Nrows)
CCeffd = (((1 - effdp*Cr)/(1 - effdp))^Nrows - 1)/(((1 - effdp*Cr)/(1 - effdp))^Nrows - Cr)
                                        {cooling coil effectiveness for completely dry tubes}
Cwat = CCWMFR*Cpw {water heat capacity rate; Btu/hr-F}
Cair = AMFR*Cpa {air-water heat capacity rate, Btu/hr-F}
Cr = Cair/Cwat {heat capacity ratio}
gmin2 = Cair/Cwat {minimum value of cooling coil control variable}

{Determination of the leaving water temperature and check for consistency using the
effectivenesses found above.}
CCLoad = CCWMFR*Cpw*(LWT - EWT)/12000 {cooling coil load based on temperature rise
                                                    of water;tons}
CCLoadw = AMFR*CCeffw*(ENTHin - ENTHsat)/12000 {cooling coil load assuming
                                          completely wet tubes (equation 19.33, Mitchell)}
CCLoadd = AMFR*CCeffd*(ENTHin - ENTHsat)/12000 {cooling coil load assuming
                                          completely dry tubes, equation 19.33, Mitchell}
ENTHsat = Enthalpy(AirH2O,T=EWT,P=Patm,R=1) {enthalpy of saturated air-water at water
                                                    inlet temp}

{Determination of the air and water side pressure drops for the cooling coil}
dPair = FF*(G^2*Avolin/(2*32.2*3600^2))*(2*(Avolout/Avolin - 1) + f*A*Avolav/(Ac*Avolin) - (1 -
sigma^2 - Ke)*Avolout/Avolin)/(144*14.7)
                              {Air pressure drop, no entrance effect (equation 2-26a, CHE's);atm}
FF = 2 {fudge factor for air side pressure drop to account for the fact that tubes are wet}
Avolin = Volume(AirH2O,T=EDB,P=Patm,R=1) {air specific volume at inlet; ft^3/lb}
Avolav = (Avolout + Avolin)/2 {average air specific volume in cooling coil: ft^3/lb}
f = lookup(lookuprow(3,ReD),5) {friction factor from CHE's}
Ke = -0.03 {from figure 5-2 in CHE's: 4*(L/D)/ReD = 0.01; laminar flow}
dPwatCC = fwat*Nrows*DHt*CCWV^2*Wrho/(2*TID*3600^2*32.2*144)
                              {water pressure drop, eq. 8-16 FHMT: psi}
fwat = 0.316*ReDw^(-0.25) {Moody friction factor, eq. 8.20 FHMT}
```

```
{The size of the ice harvester is calculated next. The ice harvester operates between
9:00 p.m. and 12:00 p.m on weekdays, and all weekend (from 9:00 p.m. on Friday until 12:00
p.m. on Monday). The ice produced must meet the weekly cooling coil energy requirement. The
ice harvester is modeled using performance curves derived for the Frick RWB-II 60E
rotary screw compressor using ammonia as the refrigerant. The model calculates the
evaporative condensor capacity, the number of plates, the nominal refrigeration capacity,
the compressor brake horsepower, the refrigerant pump brake horsepower, and the average
power requirement of the ice harvester at a saturated discharge temperature of 95 F and a
saturated suction temperature of 20 F.}

{Determination of required ice generation rate}
CCenergy = CCLoad*CChrs {weekly cooling coil energy requirement; ton-hours}
CChrs = 4.0 {weekly hours of operation for cooling coil}
TWIR = 1.01*CCenergy*12000/LHF {total weekly ice requirement; lbs}
LHF = 143.5 {latent heat of fusion of water; Btu/lb}
IGR = TWIR/IHhrs {ice generation rate; lb/hr}
IHhrs = WEhrs + 4*WDhrs {weekly hours of operation for ice harvester}
WEhrs = 63 {weekend hours of operation for the ice harvester}
WDhrs = 15 {weekday hours of operation for the ice harvester}

{Evaporative condensor: the refrigerant is cooled by means of an evaporative condensor
unit. The capacity of this unit is its nominal capacity divided by the "heat rejection correction
factor", HRCF. This factor is calculated by means of a seven parameter equation derived
from data provided by "IMECO".}
qcond = NCC/HRCF {condensor heat rejection; tons}
HRCF = E1 + E2*SCT + E3*WB^3 + E4*WB^3*SCT + E5*WB^3*SCT^2 + E6*WB^5 + E7*WB^4*SCT
        {heat rejection correction factor}
WB = 77 {design wet bulb temperature for hours of operation; F}

{Pure ice-making mode: here the power requirement and refrigeration capacity
of the ice harvester is calculated when ice is being built on all plate sections.}
CPb = mult*(P1 + P2*SST + P3*SST^2 + P4*SDT + P5*SDT^2 + P6*SST*SDT)
        {compressor brake horse power during pure build period; hp}
qevapb = mult*(C1 + C2*SST + C3*SST^2 + C4*SDT + C5*SDT^2 + C6*SST*SDT)
        {refrigeration capacity during pure build period; tons}
SST = SET - SLL {saturated suction temperature; F}
SST = 20 {saturated suction temperature (should have been 16); F}
SLL = 4.0 {suction line losses (subcooling); F}
SDT = SCT + DLL {saturated discharge temperature; F}
SDT = 95 {saturated discharge temperature; F}
DLL = 2.0 {discharge line losses (superheat); F}
qcond = qevapb + CPb*2545/12000 {condensor heat rejection; tons}

{Determination of number of evaporator plates required}
qevapb = Ubarb*Nplates*Parea*(PEWT - SET)/12000 {tons}
Ubarb = 51 {average evaporator U-value during build period; Btu/hr-ft^2-F}
Parea = 3.833*6.833*2 {plate area; ft^2}
PEWT = 32.0 {plate entering water temperature; F}

{Defrost mode: in defrost mode, one section of evaporator plates is defrosted by re-routing
the hot gas from the condensor to that section. Ice continues to build on the remaining plate
sections during this time. The compressor power requirement and capacity are determined
below.}
CPd = mult*(P1 + P2*DSST + P3*DSST^2 + P4*DSDT + P5*DSDT^2 + P6*DSST*DSDT)
        {compressor brake horsepower during defrost period; hp}
qevapd = mult*(C1 + C2*DSST + C3*DSST^2 + C4*DSDT + C5*DSDT^2 + C6*DSST*DSDT)
        {refrigeration capacity during defrost period; tons}
```

DSST = DSET - SLL {saturated evaporator temperature during defrost mode: F}
DSDT = DSCT + DDLL {saturated condensor temperature during defrost mode; F}
DDLL = 5 {discharge line losses (superheat) during defrost mode; F}
qevapd = Ubarb*(Nsect-1)/Nsect*Nplates*Parea*(PEWT - DSET)/12000 {tons}
Nsect = 4 {number of sequentially defrosted evaporator sections}
qrejd = qevapd + CPd*2545/12000 {heat rejected at defrosting evaporator plate; tons}

{Determination of saturated condensor temperature during defrost mode and mass flow rate for evaporator water pump}
qrejd = DSWMFR*Cpw*effdef*(DSCT - PEWT)/12000 {tons}
DSWMFR = EWMFR/Nsect {defrost section water mass flow rate; lb/hr}
EWMFR = PWMFR*Nplates {evaporator water mass flow rate; lb/hr}
PWMFR = 10*0.13368*62.41*60 {water mass flow rate per plate; lb/hr}
effdef = 1 - exp(-NTUdef) {evaporator effectiveness in condensor mode during defrost period}
NTUdef = (Udefp*Nplates*Parea/Nsect)/(DSWMFR*Cpw) {number of transfer units}
Udefp = 64.5 {evaporator U-value for defrosting section; Btu/hr-ft^2-F}

{Relationship between ice generation rate and refrigeration capacities during pure build and defrost mode: ice is built up on each plate during PBtime (while the harvester is operating in the pure ice building mode) and DBtime (while other evaporator sections are defrosting). The total cycle time is the sum of these last two periods and the period during which each plate is defrosting.}
IGR = Icemass/Ctime {ice generation rate; lbs/hr}
Icemass = xmax*Nplates*Parea*rhoice {lbs}
xmax = 0.375/12 {maximum ice thickness; ft}
rhoice = 57.5 {density of ice; lb /ft^3}
TIHC = Ncap*Ctime {Total heat capacity of ice built per cycle; ton-hrs}
TIHC = Icemass*LHF/12000 {ton-hrs}
Ctime = PBtime + DBtime + Dtime {cycle time: hours}
Dtime = 50/3600 {period during which section is defrosted; hrs}
DBtime = (Nsect - 1)*Dtime {build time during defrost mode; hrs}

{Calculation of refrigerant pump brake horsepower (the liquid-vapor ratio at the evaporator outlet is roughly 3:1)}
SEP = Pressure(Ammonia,T=SET,x=1) {saturated evaporator pressure; psia}
SCP = Pressure(Ammonia,T=SCT,x=1) {saturated condensor pressure; psia}
scool = 2.0 {amount of subcooling; F}
sheat = 10 {amount of superheat; F}
hcondo = Enthalpy(Ammonia,T=SCT-scool,P=SCP) {outlet condensor Enthalpy; Btu/lb}
hevapi = hcondo {inlet evaporator enthalpy; Btu/lb}
hevapo = Enthalpy(Ammonia,T=SET+sheat,P=SEP) {outlet evaporator Enthalpy; Btu/lb}
RMFR = qevapb*12000/(hevapo - hevapi) {refrigerant mass flow rate; lb/hr}
vam = Volume(Ammonia,T=SET,x=0.0) {refrigerant specific Volume, ft^3/lb}
RVFR = 3*RMFR*vam {refrigerant volumetric flow rate; ft^3/hr}
RPhead = 30 {refrigerant pump head; psi}
RPP = RPhead*RVFR/13750 {refrigerant pump brake horsepower; hp}

{Calculation of net electric power consumption and net refrigeration effect}
NPower = (CPb/ncomp*(1 - XDF) + CPd/ncomp*XDF + RPP/npump)*0.746
{net electric power requirement: kW}
ncomp = 0.95 {overall efficiency of compressor}
npump = 0.65 {overall efficiency of refrigerant pump}
Ncap = qevapb - XDF*qrejd {net refrigeration capacity: tons}
XDF = (DBtime + Dtime)/Ctime {defrost fraction of cycle time}

{Curve fit parameters for ice harvester compressor performance}
E1 = 2.271; E2 = -2.212e-2; E3 = 4.671e-5; E4 = -8.043e-7

E5 = 5.617e-9; E6 = 3.742e-9; E7 = -5.494e-9
P1 = 22.86; P2 = -1.11299; P3 = -0.0075875; P4 = 0.5892; P5 = 0.006632; P6 = 0.0184585
C1 = 94.204; C2 = 2.10578; C3 = 0.0158157; C4 = -0.02908; C5 = -0.0003119; C6 =-0.0007221

{The dimensions of the ice storage tank and the ice mass at the beginning of the simulation (Monday at 12:00 a.m.) are calculated next. The ice storage tank must hold 20% more ice than is generated over the weekend. in order to ensure that the effectiveness of the tank does not fall below 1.0 on Friday afternoon. The tank height is 20 feet; the void fraction of the ice is assumed to be 0.50.}
Ctnk = 1.20*IGR*WEhrs {capacity of ice storage tank; lbs}
BIM = Ctnk - 12*IGR {beginning ice mass; lbs}
Vtnk = Ctnk/(rhoice*0.50) {tank volume; ft^3}
BA = Vtnk/ht {tank base area: ft^2}
ht = 20 {tank height, ft}
rtnk = sqrt(BA/pi) {tank radius: ft}

{Determination of required pipe size and pump size: water must be pumped between the the ice storage tank and the cooling coil. This pipe run is assumed to be 300 feet: the water velocity is assumed to be 6 feet per second.}
PL4 = 300 {pipe length; ft}
Dpipe4 = sqrt(4*CCWMFR/(pi*Wrho*CCWV)) {diameter of pipe running between ice tank and cooling coil; ft}
dPpipe4 = fwat4*2*PL4*CCWV^2*Wrho/(2*Dpipe4*3600^2*32.2*144) {water pressure drop in pipe run 4, eq. 8-16 FHMT: psi}
fwat4 = 0.184*ReDw4^(-0.20) {Moody friction factor for pipe 4, eq. 8.21 FHMT}
ReDw4 = Wrho*CCWV*Dpipe4/muw {Reynolds number for water in pipe 4}
P4bhp = CCWMFR/Wrho*(dPpipe4 + dPwatCC)/13750 {brake horsepower for water pump 4, hp}
P4POW = P4bhp/nwpum*0.746 {power requirement for pump 4, kW}
nwpum = 0.65 {mechanical efficiency of water pumps}

{Miscellaneous quantities for TRNSED input file}
NomCap =qevapb {nominal capacity of ice harvester, tons}
Vtnkg = Vtnk*7.48055 {volume of storage tank, gal}
CCWVFR = CCWMFR/501.3 {cooling coil volumetric flow rate, gpm}
Dpipe4i = Dpipe4*12 {diameter of pipe 4, in}
TODi = TOD*12 {tube outside diameter, in}
TIDi = TID*12 {tube inside diameter, in}
tfi = tf*12 {thickness of fin, in}
FS = 1/8 {fin spacing, in}
Nfins = DHt*12/FS {number of fins}
Si = S*12 {distance between centers of tubes in a row; in }
Li = L*12 {row spacing; in }

SOLUTION:

BIM = 128215 [lb]
CCLoad = 734 [tons]
CCLoadd = 744 [tons]
CCLoadw = 689 [tons]
CCWVFR = 2934 [gpm]
Ctnk = 152407 [lbs]
DHt = 16.41 [ft]
dPair = 0.000191 [atm]
Dpipe4i = 10.9 [in]
DWt = 90.26 [ft]
FS = 0.125 [in]

gmin2 = 0.41	
ht = 20.00	[ft]
k = 102.3	[Btu/hr-ft-F]
Li = 0.866	[in]
Ncap = 24	[tons]
NCC = 57	[tons]
Nfins = 1575	
NomCap = 27	[tons]
Nrows = 3	
Ntubes = 1083	
PL4 = 300	[ft]
Si = 1.000	[in]
tfi = 0.0130	[in]
TIDi = 0.332	[in]
TODi = 0.402	[in]
Vtnkg = 39655	[gal]

APPENDIX C: TRNSYS COMPONENT MODELS

- TYPE 68: Centrifugal Chiller Model
- TYPE 69: Combustion Turbine Power Plant Model
- TYPE 71: Ice Storage Tank Model
- TYPE 72: Ice Harvester Model
- TYPE 73: Cost Calculator
- TYPE 75: Cooling Coil Pump Controller
- TYPE 76: Chiller and Cooling Tower Controller
- TYPE 77: Ice Harvester Controller
- TYPE 78: Flow Diverter Controller
- TYPE 79: Evaporative Cooler Model

```
c             CENTRIFUGAL CHILLER MODEL  TYPE 68
c
c     This subroutine models the operation of a chiller based on
c   a five parameter equation relating the dimensionless power to the

c   dimensionless load and deviations from design entering condensor
c   and chilled water set point temperatures. It differs from "type 53"
c   in that it does not require an external data file. Given values for
c   the chilled water set point temperature, the evaporator water inlet
c   temperature and mass flow rate, and the condensor water inlet tempera-
c   ture and mass flow rate, the subroutine will return the evaporator
c   water outlet temperature (and mass flow rate), the condensor water
c   outlet temperature (and mass flow rate), the load, the power require-
c   ment, the condensor heat rejection, and the coefficient of performance.
c   A control variable allows the chiller to be shut off when it is not
c   needed.
c
      subroutine type68(time,xin,out,t,dtdt,par,info,icntrl,*)
c
c   Variable declaration module: variables from the main program
c
      implicit none
      real*8 out(8),xin(6)
      real*4 time,t(1),dtdt(1),par(9)
      integer*4 info(10)
      integer icntrl
c
c   Variable declaration module: variables used only in subroutine
c
      real*8 SPT,EEWT,EWMFR,ECWT,CWMFR,gam,Qmax,Qmin,Qdes,
     @     Pdes,a,b,c,d,e,LEWT,LCWT,Qload,Ptot,Qcond,COP,DLEWT,Cp,
     @     DECWT
      character*3 ycheck(6),ocheck(8)
c
c   TYPECK, YCHECK, OCHECK, and RCHECK subroutine calling module: this
c   program segment sets info(6) and info(9), and calls the subroutines
c   listed above.
c
      If (info(7) .eq. -1) then
        info(6) = 8
        info(9) = 0
        call typeck(1,info,6,9,0)
        data ycheck/'TE1','TE1','MF1','TE1','MF1','CF1'/
        data ocheck/'TE1','MF1','TE1','MF1','PW1','PW3','PW1','DM1'/
        call rcheck(info,ycheck,ocheck)
      endif
c
c   Constant module: this program segment converts inputs and parameters
c   into English units for use in the subroutine. The design entering
c   condensor water temperature, leaving evaporator water temperature,
c   and specific heat of water are set here as well.
c
c   Inputs from main program:
c
      SPT = xin(1)*1.8 + 32.0
      EEWT = xin(2)*1.8 + 32.0
      EWMFR = xin(3)*2.2046
```

```
      ECWT = xin(4)*1.8 + 32.0
      CWMFR = xin(5)*2.2046
      gam = xin(6)
c
c     Parameters
c
      Qmax = par(1)/12672
      Qmin = par(2)/12672
      Qdes = par(3)/12672
      Pdes = par(4)
      a = par(5)
      b = par(6)
      c = par(7)
      d = par(8)
      e = par(9)
c
c     Design temperatures and specific heat of water
c
      DLEWT = 44.0
      DECWT = 85.0
      Cp = 1.0
c
c     Shut down module: if the load is less than the minimum load specified
c     in the main program, or if the control variable, gam, is set equal to
c     0, then the subroutine sets the leaving condensor and evaporator
c     water temperatures equal to the corresponding entering temperatures,
c     and sets Qload, Ptot, Qcond, and COP equal to 0.
c
      Qload = EWMFR*Cp*(EEWT - SPT)/12000
      If ((gam .lt. 0.0001) .or. (Qload .lt. Qmin)) then
        LEWT = EEWT
        LCWT = ECWT
        Qload = 1e-6
        Ptot = 1e-6
        Qcond = 1e-6
        COP = 1e-6
c
c     Normal chiller operation module: this program segment calculates
c     remaining output values for Qmin < Qload < Qmax, in which case the
c     chilled water set point temperature remains unchanged.
c
      Elseif ((Qload .ge. Qmin) .and. (Qload .le. Qmax)) then
        LEWT = SPT
        Ptot = Pdes*(a + b*(Qload/Qdes) + c*(Qload/Qdes)**2)
     @        *(1 + d*(ECWT - DECWT) - e*(LEWT - DLEWT))
        Qcond = Qload + Ptot/3.52
        LCWT = ECWT + Qcond*12000/(CWMFR*Cp+1)
        COP = Qload*3.52/Ptot
c
c     "Overload" chiller operation module: if Qload is found to be
c     greater than Qmax, then Qload is set equal to Qmax and a new
c     leaving evaporator water temperature is determined. Remaining
c     output values are calculated as well.
c
      Elseif (Qload .gt. Qmax) then
        Qload = Qmax
        LEWT = EEWT - Qload*12000/(EWMFR*Cp+1)
```

```
      Ptot = Pdes*(a + b*(Qload/Qdes) + c*(Qload/Qdes)**2)
     @        *(1 + d*(ECWT - DECWT) - e*(LEWT - DLEWT))
      Qcond = Qload + Ptot/3.52
      LCWT = ECWT + Qcond*12000/(CWMFR*Cp+1)
      COP = Qload*3.52/Ptot
     Endif
c
c    Output array module: this program segment fills the array out(8)
c    with values calculated in the subroutine to be returned to the
c    main program.  English units are converted to metric units.
c
     out(1) = 0.5556*(LEWT - 32.0)
     out(2) = EWMFR*0.4536
     out(3) = 0.5556*(LCWT - 32.0)
     out(4) = CWMFR*0.4536
     out(5) = Qload*12672
     out(6) = Ptot
     out(7) = Qcond*12672
     out(8) = COP
c
c    Return module
c
     return 1
     end


*****************************************************************************************


c              BLACK BOX POWER PLANT MODEL   TYPE 69
c
c       This subroutine models the part load operation of a combustion
c    turbine power plant. For given ambient dry bulb temperature, entering
c    dry bulb temperature, humidity ratio, ambient pressure, normalized
c    power requirement, and cooling coil pump control variables it will
c    return the dry air mass flow rate, the net electric power generated,
c    the fuel mass flow rate, the electric power output if there were no
c    inlet air cooling, the desired electric  power output, and the fuel
c    mass flow rate if there were no inlet cooling.  If the normalized power
c    requirement is set equal to 0, all five outputs will be set equal to 0
c    as well.
c
     subroutine type69(time,xin,out,t,dtdt,par,info,icntrl,*)
c
c    Variable declaration module: variables from main program
c
     implicit none
     real*8 out(6),xin(7)
     real*4 time,t(1),dtdt(1),par(25)
     integer*4 info(10)
```

```
      integer icntrl
c
c     Variable declaration module: variables used only in subroutine
c
      real*8 ADB,EDB,Patm,w1,AMFR,EP,MEP,FMFR,nHHV,BEP,nbase,HHV,dPincc,
     @      dPinex,dPin,dPout,WFR,VFR1,EPpeak,r(3),a(8),RPO,nrel,MFR1,
     @      v1,va1,vw1,Pa1,Pw1,y1,P1,PLF,gamma,b,c,d,e,f,IPLCM,IPLEM,
     @      OPLCEM,WFRCM,WFREM,EPNC,DEP,CV1,CV2,FMFRNC,PLFNC
      character*3 ycheck(7),ocheck(6)
c
c     TYPECK, YCHECK, OCHECK, and RCHECK subroutine calling module: this
c     program segment sets info(6) amd info(9), and calls the subroutines
c     listed above.
c
      If (info(7) .eq. -1) then
        info(6) = 6
        info(9) = 1
        call typeck(1,info,7,25,0)
        data ycheck/'TE1','TE1','PR1','DM1','CF1','CF1','CF1'/
        data ocheck/'MF1','PW3','MF1','PW3','PW3','MF1'/
        call rcheck(info,ycheck,ocheck)
      endif
c
c     Constant module: this program segment sets subroutine constants
c     equal to inputs and parameters passed from the main program. Metric
c     units are converted to English units. The total inlet pressure drop
c     is the sum of the pressure drop due to the cooling coils and the
c     pressure drop from other sources.
c
      ADB = xin(1)*1.8 + 32.0
      EDB = xin(2)*1.8 + 32.0
      Patm = xin(3)
      w1 = xin(4)
      gamma = xin(5)
      CV1 = xin(6)
      CV2 = xin(7)
      BEP = par(1)
      nbase = par(2)
      HHV = par(3)*0.4299
      dPincc = par(4)
      dPinex = par(5)
      dPout = par(6)
      WFR = par(7)
      VFR1 = par(8)*35.31
      EPpeak = par(9)
      r(1) = par(10)
      r(2) = par(11)
      r(3) = par(12)
      a(1) = par(13)
      a(2) = par(14)
      a(3) = par(15)
      a(4) = par(16)
      a(5) = par(17)
      a(6) = par(18)
      a(7) = par(19)
      a(8) = par(20)
      b = par(21)
```

```
      c = par(22)
      d = par(23)
      e = par(24)
      f = par(25)
      dPin = dPincc + dPinex
c
c     Shut down module: this program segment sets the air mass flow rate,
c     the electric power generation, the fuel mass flow rate, the electric
c     power generation in the absence of inlet cooling, the desired electric
c     power, and the fuel mass flow rate in the absence of inlet cooling
c     equal to 0 when the normalized power requirement is equal to 0.
c
      If (gamma .lt. 0.0001) then
        AMFR = 1e-6
        EP = 1e-6
        FMFR = 1e-6
        EPNC = 1e-6
        DEP = 1e-6
        FMFRNC = 1e-6
c
c     Net power and part load factor calculation module: this program segment
c     calculates the desired electric power, the net power output, and the
c     part load factor. The desired electric power is simply the peak power
c     output multiplied by the normalized power requirement, gamma. The
c     maximum electric power output for the entering dry bulb temperature is
c     calculated using curve fits derived from data provided by the turbine
c     manufacturer. If the entering dry bulb temperature is equal to the
c     ambient dry bulb temperature, then the part load factor is calculated
c     directly. The part load factor is simply the electric power actually
c     produced divided by the maximum electric power that could be produced
c     at the entering dry bulb temperature. If the entering dry bulb tempe-
c     rature is less than the ambient dry bulb (i.e., if the inlet cooling
c     system is in use), then the electric power is set equal to the maximum
c     electric power , which corresponds to a part load factor of 1. However,
c     if either cooling coil pump control variable is set equal to its mini-
c     mum "on" value, the electric power is set equal to the desired electric
c     power.
c
      Elseif (gamma .gt. 0.0001) then
        DEP = gamma*EPpeak
        EP = DEP
        IPLCM = 1.0 + b*dPin
        OPLCEM = 1.0 + d*dPout
        WFRCM = 1.0 + e*WFR
        RPO = r(1) + r(2)*EDB + r(3)*EDB**2
        MEP = IPLCM*OPLCEM*WFRCM*BEP*RPO
        If (DEP .gt. MEP) EP = MEP
        If ((CV1 .gt. 0.9999) .and. (CV2 .lt. 1e-6)) EP = MEP
        If (CV2 .ge. 0.9999) EP = MEP
        PLF = EP/MEP
c
c     Electric power output in the absence of inlet cooling calculation
c     module: this program segment calculates the electric power that could
c     be produced by the power plant if there were no inlet air cooling
c     system, EPNC. If the desired electric power exceeds the maximum
c     electric power that could be produced using air at the ambient dry bulb
c     temperature, EPNC is set equal to that maximum. The part load factor
```

```
c     in the absence of inlet cooling is determined as well.
c
        EPNC = DEP
        RPO = r(1) + r(2)*ADB + r(3)*ADB**2
        IPLCM = 1.0 + b*dPinex
        MEP = IPLCM*OPLCEM*WFRCM*BEP*RPO
        If (EPNC .gt. MEP) then
          EPNC = MEP
        Endif
        PLFNC = EPNC/MEP
c
c     Fuel flow rate with inlet cooling module: this program segment
c     calculates the fuel mass flow rate as a function of the part load
c     factor and the entering dry bulb temperature. It uses curve fit
c     parameters derived from relative efficiency data provided by the
c     turbine manufacturer.
c
        IPLEM = 1.0 + c*dPin
        WFREM = 1.0 + f*WFR
        nrel = a(1) + a(2)*PLF + a(3)*EDB + a(4)*PLF*EDB + a(5)*PLF**2
     @        + a(6)*EDB**2 + a(7)*PLF*EDB**2 + a(8)*PLF**3
        nHHV = IPLEM*OPLCEM*WFREM*nbase*nrel
        FMFR = EP/(nHHV*HHV)*3412
c
c     Fuel flow rate in the absence of inlet cooling module: this program
c     segment calulates the fuel mass flow rate for the turbine in the event
c     that there is no inlet air cooling. The same curve fit parameters
c     are used as before, but PLFNC and ADB are substituted for PLF and EDB.
c     IPLEM is re-calculated.
c
        IPLEM = 1.0 + c*dPinex
        nrel = a(1) + a(2)*PLFNC + a(3)*ADB + a(4)*PLFNC*ADB
     @        + a(5)*PLFNC**2 + a(6)*ADB**2 + a(7)*PLFNC*ADB**2
     @        + a(8)*PLFNC**3
        nHHV = IPLEM*OPLCEM*WFREM*nbase*nrel
        FMFRNC = EPNC/(nHHV*HHV)*3412
c
c     Air mass flow rate calculation module: this program segment calculates
c     the dry air mass flow rate, AMFR, based on the assumption that the
c     volumetric flow rate at the compressor inlet is independent of the inlet
c     temperature.
c
        y1 = (29/18)*w1/(1 + w1)
        P1 = Patm - dPin
        Pw1 = y1*P1
        Pa1 = P1 - Pw1
        va1 = 0.02519*(EDB + 459.7)/Pa1
        vw1 = 0.04050*(EDB + 459.7)/Pw1
        v1 = 1/(1/va1 + 1/vw1)
        MFR1 = VFR1/v1
        AMFR = MFR1/(1 + w1)
      Endif
c
c     Output array module: this program segment fills the array out(6)
c     with values calculated in the subroutine to be returned to the main
c     program. English units are converted to metric units.
c
```

```
      out(1) = AMFR/2.2046
      out(2) = EP
      out(3) = FMFR/2.2046
      out(4) = EPNC
      out(5) = DEP
      out(6) = FMFRNC/2.2046
c
c     Return module
c
      return 1
      end
```

****

```
c              ICE STORAGE TANK MODEL  TYPE 71
c
c        This subroutine models the operation of an ice storage tank.
c     The tank is characterized by its capacity (in terms of pounds of
c     ice), volume, height, and overall loss coefficient. Inputs are:
c     entering water temperature, water mass flow rate, the ice generation
c     rate (from an ice harvester), and the temperature of the environment.
c     There is one derivative: the mass of ice in the tank at the beginning
c     of the simulation period. Outputs are: , leaving water temperature,
c     the water mass flow rate, ice mass at the end of the simulation time
c     step, the ice "burn rate", the rate of heat loss to the environment,
c     the rate of energy "input" to the tank via ice generation, and the
c     the rate of energy "supplied" to the water stream.
c
      subroutine type71(time,xin,out,t,dtdt,par,info,icntrl,*)
c
c     Variable declaration module: variables from main program
c
      implicit none
      real*8 out(7),xin(4)
      real*4 time,t(1),dtdt(1),par(4),time0,tfinal,delt
      integer*4 info(10),iwarn
      integer icntrl
c
c     Variable declaration module: variables used only in subroutine
c
      real*8 WMFR,EWT,BIM,IGR,Tenv,LWT,FIM,IBR,qenv,cap,vol,ht,
     @      BA,rad,ETA,Ttnk,LHF,qtot,qwat,DF,eff,Cp,pi,Ut,qsupp
      character*3 ycheck(4),ocheck(7)
c
c     Common module: the initial time, the final time, and the time
c     step are required by this subroutine.
c
      common/sim/time0,tfinal,delt,iwarn
```

```
c
c   TYPECK, YCHECK, OCHECK, and RCHECK subroutine calling module:
c   this program segment sets info(6) and info(9), and calls the
c   subroutineslisted above.
c
    If (info(7) .eq. -1) then
      info(6) = 5
      info(9) = 1
      call typeck(1,info,4,4,1)
      data ycheck/'TE1','MF1','MF1','TE1'/
      data ocheck/'TE1','MF1','MA1','MF1','PW1','PW1','PW1'/
      call rcheck(info,ycheck,ocheck)
    Endif
c
c   Constant module: this program segment converts inputs and para-
c   meters into English units for use in the subroutine. Property
c   values are also set here.
c
    EWT = xin(1)*1.8 + 32.0
    WMFR = xin(2)*2.2046
    IGR = xin(3)*2.2046
    Tenv = xin(4)*1.8 + 32.0
    If ((time - time0) .lt. 0.0001) then
      BIM = t(1)*2.2046
    Elseif (((time - time0) .ge. 0.0001) .and. info(7) .eq. 0) then
      BIM = FIM
    Endif
    cap = par(1)*2.2046
    vol = par(2)*35.311
    ht = par(3)*3.2808
    Ut = par(4)*0.04892
    Cp = 1.0
    LHF = 143.5
c
c   Environmental loss calculation module: this program segment
c   calculates the heat transfer rate from the interior of the ice
c   tank (assumed to be at 32 degrees F) to the environment.
c
    BA = vol/ht
    pi = 3.1416
    rad = sqrt(BA/pi)
    ETA = BA + 2*pi*rad*ht
    Ttnk = 32.0
    qenv = Ut*ETA*(Tenv - Ttnk)
c
c   Ice burn rate calculation module: this program segment determines
c   the rate at which is is "burned" due to water flow through the
c   tank and losses to the environment.
c
    DF = (cap - BIM)/cap
    If (DF .lt. 0.80) then
      eff = 1.0
    Elseif (DF .ge. 0.80) then
      eff = 1.0 - 5.0*(DF - 0.80)
    Endif
    qwat = eff*WMFR*Cp*(EWT - Ttnk)
    qtot = qwat + qenv
```

```
      IBR = qtot/LHF
c
c     Leaving water temperature calculation module: this program segment
c     calculates the temperature of the water leaving the ice storage
c     tank.  If water is not circulating through the tank, this temperature
c     is simply set equal to the inlet water temperature.
c
      If (WMFR .gt. 1.0) then
        LWT = EWT - qwat/(WMFR*Cp)
      Elseif (WMFR .le. 1.0) then
        LWT = Ttnk
      Endif
c
c     Final ice mass calculation module: this program segment determines
c     the mass of ice remaining in the tank at the end of the simulation
c     time step.  This value may not be less than 0.  LWT, IBR, and qwat
c     are also re-set if FIM is initially found to be less than 0.
c
      FIM = BIM + (IGR - IBR)*delt
      If (FIM .lt. 0.0) then
        LWT = EWT
        FIM = 0.0001
        IBR = BIM/delt
        qwat = IBR*LHF - qenv
      Endif
      qsupp = IGR*LHF
c
c     Output array module: this program segment fills the array out(5)
c     with values calculated in the subroutine.  English units are
c     converted to metric units.
c
      out(1) = (LWT - 32)*0.5556
      out(2) = WMFR*0.4536
      out(3) = FIM*0.4536
      out(4) = IBR*0.4536
      out(5) = qenv*1.055
      out(6) = qsupp*1.055
      out(7) = qwat*1.055
c
c     Return module
c
      return 1
      end
```

****

```
c            BLACK BOX ICE HARVESTER  TYPE 72
c
c       This subroutine models the operation of an ice harvester
c    using curve fit parameters derived from a detailed EES model.
c    Given the ambient wet bulb temperature, the subroutine will return
c    the net refrigeration effect, the ice generation rate, and the
c    net power requirement.  A control variable allows the ice harvester
c    to be shut down when it is not needed.
c
      subroutine type72(time,xin,out,t,dtdt,par,info,icntrl,*)
c
c    Variable declaration module: variables from main program
c
      implicit none
      real*8 out(3),xin(2)
      real*4 time,t(1),dtdt(1),par(12)
      integer*4 info(10)
      integer icntrl
c
c    Variable declaration module: variables used only in subroutine
c
      real*8 WBT,gamma,Ncap,IGR,Npower,NomCap,DWBT,NCC,C(4),P(5),LHF
      character*3 ycheck(2),ocheck(3)
c
c    TYPECK, YCHECK, OCHECK, and RCHECK subroutine calling module:
c    this program segment sets info(6) and info(9), and calls the
c    subroutines listed above.
c
      If (info(7) .eq. -1) then
        info(6) = 3
        info(9) = 0
        call typeck(1,info,2,12,0)
        data ycheck/'TE1','CF1'/
        data ocheck/'PW1','MF1','PW3'/
        call rcheck(info,ycheck,ocheck)
      endif
c
c    Constant module: this program segment converts inputs and para-
c    meters into English units for use in the subroutine.  The latent
c    heat of fusion of water is also set here.
c
      WBT = xin(1)*1.8 + 32.0
      gamma = xin(2)
      NomCap = par(1)/12672
      DWBT = par(2)*1.8 + 32.0
      NCC = par(3)/12672
      C(1) = par(4)
      C(2) = par(5)
      C(3) = par(6)
      C(4) = par(7)
      P(1) = par(8)
      P(2) = par(9)
      P(3) = par(10)
      P(4) = par(11)
      P(5) = par(12)
      LHF = 143.5
```

```
c
c   Shut down module: if the control variable, gamma, is set equal to
c   0, the subroutine returns values of 0 for the net refrigeration
c   effect, the ice generation rate, and the net power requirment.
c
      If (gamma .le. 0.0001) then
        Ncap = 1e-6
        IGR = 1e-6
        Npower = 1e-6
c
c   Calculation module: the net refrigeration effect, ice generation
c   rate, and net power requirement are calculated below if the control
c   variable is set equal to 1.
c
      Elseif (gamma .gt. 0.0001) then
        Ncap = C(1) + C(2)*NomCap + C(3)*WBT*NomCap + C(4)*DWBT*NomCap
        Npower = P(1) + P(2)*NomCap + P(3)*WBT*NomCap
     @          + P(4)*DWBT*NomCap + P(5)*(DWBT)**2
        IGR = Ncap*12000/LHF
      Endif
c
c   Output array module: this program segment fills the array out(3)
c   with values calculated in the subroutine to be returned to the
c   main program.  English units are converted to metric units.
c
      out(1) = Ncap*12672
      out(2) = IGR/2.2046
      out(3) = Npower
c
c   Return module
c
      return 1
      end
```

****

```
c              COST CALCULATOR  TYPE 73
c
c      This subroutine calculates all costs associated with the
c   combustion turbine inlet cooling system.  Given all system sizes,
c   the subroutine will return the cost of: the chiller/cooling tower,
c   the ice harvester, the cooling coils, each storage tank, the water
c   pumps and pipes, the chilled water storage loop, the ice storage loop,
c   and the combined system.  It also will return the cost per kilowatt of
c   additional power plant generating capacity, the annual excess fuel cost
c   and annual off-peak electricity cost when inlet cooling is used, and
c   the cost of electricity purchased from another utility (discounted to
c   the present) that will yield a cooling system payback period of a
c   specified number of years.
```

```
c
      subroutine type73(time,xin,out,t,dtdt,par,info,icntrl,*)
c
c     Variable declaration module: variables from main program
c
      implicit none
      real*8 out(13),xin(5)
      real*4 time,t(1),dtdt(1),par(46),time0,tfinal,delt
      integer*4 info(10),iwarn
      integer icntrl
c
c     Variable declaration module: variables used only in subroutine
c
      real*8 Qmax,NCap,Lduct(2),Wduct(2),Nrows(2),Qtow,VT(2),
     @     Mmax(4),PL(4),PD(4),EPmax,EPmin,C(3),H(2),CC1,W(3),K(3),
     @     P1,Q(3),CHcost,IHcost,CC1cost,CC2cost,CTcost,TNK1cost,
     @     TNK2cost,PMWLcost,PMILcost,PIPWLcost,PIPILcost,CHCTcost,
     @     CCcost,PMPIcost,CWSLcost,ISLcost,SYScost,CPKW,PPnrg,
     @     PPnrgNC,FM,FMNC,OPnrg,d,i,Np,Cf,Ceop,Cepur,deltnrg,
     @     deltaFM,PWF,EFC,OPEC
      character*3 ycheck
c
c     Common module: the initial time, the final time, and the time
c     step are passed to the subroutine from the main program.
c
      common/sim/time0,tfinal,delt,iwarn
c
c     TYPECK and YCHECK subroutine calling module: this program segment sets
c     info(6) and info(9), and calls the subroutines listed above.
c
      if (info(7) .eq. -1) then
        info(6) = 13
        info(9) = 1
        call typeck(1,info,5,46,0)
        data ycheck/'EN2','EN2','MA1','MA1','EN2'/
      endif
c
c     Constant module: this program segment converts the inputs and para-
c     meters into English units for use in the subroutine.
c
      PPnrg = xin(1)
      PPnrgNC = xin(2)
      FM = xin(3)*2.2046
      FMNC = xin(4)*2.2046
      OPnrg = xin(5)
      Qmax = par(1)/12672
      NCap = par(2)/12672
      Lduct(1) = par(3)*3.2808
      Wduct(1) = par(4)*3.2808
      Nrows(1) = par(5)
      Lduct(2) = par(6)*3.2808
      Wduct(2) = par(7)*3.2808
      Nrows(2) = par(8)
      Qtow = par(9)/12672
      VT(1) = par(10)*264.17
      VT(2) = par(11)*264.17
      Mmax(1) = par(12)/227.1
```

```
      Mmax(2) = par(13)/227.1
      Mmax(3) = par(14)/227.1
      Mmax(4) = par(15)/227.1
      PL(1) = par(16)*3.2808
      PD(1) = par(17)*39.37
      PL(2) = par(18)*3.2808
      PD(2) = par(19)*39.37
      PL(3) = par(20)*3.2808
      PD(3) = par(21)*39.37
      PL(4) = par(22)*3.2808
      PD(4) = par(23)*39.37
      EPmax = par(24)
      EPmin = par(25)
      C(1) = par(26)
      C(2) = par(27)
      C(3) = par(28)
      H(1) = par(29)
      H(2) = par(30)
      CC1 = par(31)
      W(1) = par(32)
      W(2) = par(33)
      W(3) = par(34)
      K(1) = par(35)
      K(2) = par(36)
      K(3) = par(37)
      P1 = par(38)
      Q(1) = par(39)
      Q(2) = par(40)
      Q(3) = par(41)
      d = par(42)
      i = par(43)
      Np = par(44)
      Cf = par(45)
      Ceop = par(46)
c
c     Component cost calculation module: the total costs of the chiller,
c     ice harvester, cooling coils, cooling tower, pumps, and pipes are
c     found by multilying bare materials costs by 2.5.  The total costs of
c     the storage tanks are based on total cost data in the Means catalogue.
c
      if (time .gt. tfinal - 0.01) then
        CHcost = 2.5*(C(1) + C(2)*Qmax + C(3)*Qmax**2)
        IHcost = 2.5*(NCap*H(1) +  H(2))
        CC1cost = 2.5*Lduct(1)*Wduct(1)*Nrows(1)*CC1
        CC2cost = 2.5*Lduct(2)*Wduct(2)*Nrows(2)*CC1
        CTcost = 2.5*Qtow*(W(1) + W(2)*Qtow + W(3)*Qtow**2)
        TNK1cost = K(1) + K(2)*VT(1) + K(3)*VT(1)**2
        TNK2cost = K(1) + K(2)*VT(2) + K(3)*VT(2)**2
        PMWLcost = 2.5*P1*(Mmax(1) + Mmax(2) + Mmax(3))
        PMILcost = 2.5*P1*Mmax(4)
        PIPWLcost = 2.5*2*(Q(1)*(PL(1) + PL(2) + PL(3))
     @            + Q(2)*(PL(1)*PD(1) + PL(2)*PD(2) + PL(3)*PD(3))
     @            + Q(3)*(PL(1)*PD(1)**2 + PL(2)*PD(2)**2
     @            + PL(3)*PD(3)**2))
        PIPILcost = 2.5*2*(Q(1)*PL(4) + Q(2)*PL(4)*PD(4)
     @            + Q(3)*PL(4)*PD(4)**2)
```

```
c
c   System cost calculation module: the costs of the chilled water
c   storage loop and the ice storage loop are calculated separately.
c   The total system cost and the cost per kilowatt of additional
c   power plant generating capacity are also determined.  If Qmax
c   or NCap are set equal to less than 0.9 tons in the main program, a
c   value of 0 will be returned for all costs associated with the chilled
c   water storage loop cost or with the ice storage loop cost, respectively.
c
        if (Qmax .lt. 0.9) then
          CHcost = 0.0
          CTcost = 0.0
          CC1cost = 0.0
          TNK1cost = 0.0
          PMWLcost = 0.0
          PIPWLcost = 0.0
        endif
        if (NCap .lt. 0.9) then
          IHcost = 0.0
          CC2cost = 0.0
          TNK2cost = 0.0
          PMILcost = 0.0
          PIPILcost = 0.0
        endif
        CHCTcost = CHcost + CTcost
        CCcost = CC1cost + CC2cost
        PMPIcost = PMWLcost + PIPWLcost + PMILcost + PIPILcost
        CWSLcost = CHcost + CC1cost + CTcost + TNK1cost + PMWLcost
     @            + PIPWLcost
        ISLcost = IHcost + CC2cost + TNK2cost + PMILcost + PIPILcost
        SYScost = CWSLcost + ISLcost
        CPKW = SYScost/(EPmax - EPmin)
c
c   Required electric energy purchase cost calculation module: this
c   program segment determines the annual excess fuel cost, the annual off-
c   peak electricity cost, and the cost of electricity purchased from
c   another utility (discounted to the present) that yields a payback
c   period for the cooling system of Np years.
c
        deltnrg = PPnrg - PPnrgNC
        deltaFM = FM - FMNC
        EFC = Cf*deltaFM
        OPEC = Ceop*OPnrg
        PWF = 1/(d - i)*(1 - ((1 + i)/(1 + d))**Np)
        Cepur = 1/deltnrg*(SYScost/PWF + EFC + OPEC)
      endif
c
c   Output array module: this program segment fills the array out(13)
c   with values calculated in the subroutine.  Units are dollars.
c
      out(1)  = CHCTcost
      out(2)  = IHcost
      out(3)  = CCcost
      out(4)  = TNK1cost
      out(5)  = TNK2cost
      out(6)  = PMPIcost
      out(7)  = CWSLcost
```

```
      out(8)  = ISLcost
      out(9)  = SYScost
      out(10) = CPKW
      out(11) = EFC
      out(12) = OPEC
      out(13) = Cepur
c
c     Return module
c
      return 1
      end
```

****

```
c             COOLING COIL PUMP CONTROLLER  TYPE 75
c
c        This subroutine sets the control variables for the cooling
c     coil pump fed by the chilled water storage loop and for the cooling
c     coil pump fed by the ice storage loop.  The controller attempts to
c     make the difference between the desired electric power and the electric
c     power actually produced nearly equal to 0.  If no inlet cooling is
c     required, both control variables will be set equal to 0.  If the
c     difference between the desired electric power and the power actually
c     produced with the maximum water flow through each cooling coil is
c     greater than 0, both control variables will be set equal to 1.  The
c     minimum values for each control variable (unless it is 0) is passed
c     as a parameter.
c
      subroutine type75(time,xin,out,t,dtdt,par,info,icntrl,*)
c
c     Variable declaration module: variables from main program
c
      implicit none
      real*8 out(2),xin(2)
      real*4 time,t(1),dtdt(1),par(2)
      integer*4 info(10)
      integer icntrl
c
c     Variable declaration module: variables used only in subroutine
c
      real*8 DEP,EP,del,CV1n,CV2n,CV1o,CV2o,delp,delm,CV1min,CV2min
      character*3 ycheck(2),ocheck(2)
c
c     TYPECK, YCHECK, OCHECK, and RCHECK subroutine calling module: this
c     program segment sets info(6) amd info(9), and calls the subroutines
c     listed above.
c
      If (info(7) .eq. -1) then
        info(6) = 2
```

```
        info(9) = 1
        call typeck(1,info,2,2,0)
        data ycheck/'PW3','PW3'/
        data ocheck/'CF1','CF1'/
        call rcheck(info,ycheck,ocheck)
      endif
c
c     Constant module: this program segment sets subroutine constants
c     equal to inputs and parameters passed from the main program.  The
c     difference between the desired electric power and electric power
c     actually produced is also determined.
c
      DEP = xin(1)
      EP = xin(2)
      CV1min = par(1)
      CV2min = par(2)
      del = DEP - EP
c
c     Initial guess module: Both control variables are set equal to 0 for
c     the first call to this unit in each timestep.
c
      If (info(7) .le. 0) then
        CV1n = 0
        CV2n = 0
        goto 100
      Endif
c
c     Second guess module: if del is not equal to 0 after the first
c     call to this unit, the first control variable is set equal to 1 while
c     the second remains set to 0.
c
      CV1o = CV1n
      CV2o = CV2n
      If (info(7) .ne. 1) goto 10
      If (abs(del) .le. 1) then
        CV1n = CV1o
        CV2n = CV2o
        goto 100
      Else
        delp = del
        CV1n = 1
        CV2n = 0
        goto 100
      Endif
c
c     Third guess module: If del is still not equal to 0, then the second
c     control variable is also set equal to 1.  Otherwise, a value for
c     the first control variable between 0 and 1 is calculated.
c
10    If (info(7) .ne. 2) goto 20
      If (abs(del) .le. 1) then
        CV1n = CV1o
        CV2n = CV2o
        goto 100
      Elseif (del .gt. 0) then
        delp = del
        CV1n = 1
```

```
         CV2n = 1
         goto 100
       Else
         delm = del
         CV1n = delp/(delp - delm)*CV1o
         CV2n = CV2o
         goto 100
       Endif
c
c      Fourth and subsequent guess module: if the second control variable
c      has been determined to be 0, then the unit is called until it converges
c      on a value for the first control variable. Otherwise, the unit is
c      called until it converges on the second control variable.
c
20     If (CV2o .gt. 0.0001) goto 30
       If (abs(del) .le. 1) then
         CV1n = CV1o
         CV2n = CV2o
         goto 100
       Elseif (del .gt. 0) then
         delp = del
         CV1n = delp/(delp - delm)*(CV1o - 1) + CV1o
         CV2n = CV2o
         goto 100
       Else
         delm = del
         CV1n = delp/(delp - delm)*CV1o
         CV2n = CV2o
       Endif
30     If (abs(del) .le. 1) then
         CV1n = CV1o
         CV2n = CV2o
         goto 100
       Endif
       If (info(7) .ne. 3) goto 40
       If (del .gt. 0) then
         CV1n = CV1o
         CV2n = CV2o
         goto 100
       Else
         delm = del
         CV2n = delp/(delp - delm)*CV2o
         CV1n = CV1o
         goto 100
       Endif
40     If (del .gt. 0) then
         delp = del
         CV2n = delp/(delp - delm)*(CV2o - 1) + CV2o
         CV1n = CV1o
         goto 100
       Else
         delm = del
         CV2n = delp/(delp - delm)*CV2o
         CV1n = CV1o
         goto 100
       Endif
c
```

```
c
c  Lower limit module: this program segment sets CV1 or CV2 equal to
c  a minimum value if it was calculated to be between 0 and that
c  minimum value above.
c
100  If ((CV1n .gt. 1e-6) .and. (CV1n .lt. CV1min))  CV1n = CV1min
     If ((CV2n .gt. 1e-6) .and. (CV2n .lt. CV2min))  CV2n = CV2min
c
c  Output array module: this program segment fills the array out(5)
c  with the control variables calculated above to be returned to the
c  main program.
c
c
     out(1) = CV1n
     out(2) = CV2n
c
c  Return module
c
     return 1
     end
```

****

```
c    CHILLER AND COOLING TOWER CONTROLLER  TYPE 76
c
c       This subroutine sets the control variable for the chiller, the
c   the chilled water pump, the cooling tower pump, and the cooling tower
c   fan. This value is determined based on the value of the evaporator
c   water inlet temperature, the chiller schedule, and the old value of
c   of the chiller control variable.
c
     subroutine type76(time,xin,out,t,dtdt,par,info,icntrl,*)
c
c   Variable declaration module: variables from main program
c
     implicit none
     real*8 out(1),xin(3)
     real*4 time,t(1),dtdt(1),par(1)
     integer*4 info(10)
     integer icntrl
c
c   Variable declaration module: variables used only in subroutine
c
     real*8 EEWT,sched1,gamma1n,gamma1o,MEEWT
     character*3 ycheck(3),ocheck(1)
c
c   TYPECK, YCHECK, OCHECK, and RCHECK subroutine calling module: this
c   program segment sets info(6) amd info(9), and calls the subroutines
c   listed above.  Additionally, gamma2o is set equal to 0.
```

```
c
    If (info(7) .eq. -1) then
      info(6) = 1
      info(9) = 1
      call typeck(1,info,3,1,0)
      data ycheck/'TE1','CF1','CF1'/
      data ocheck/'CF1'/
      call rcheck(info,ycheck,ocheck)
      xin(3) = 0
    endif
c
c   Constant module: this program segment sets subroutine constants
c   equal to inputs and parameters passed from the main program.  English
c   units are converted to metric units.
c
    EEWT = xin(1)*1.8 + 32.0
    sched1 = xin(2)
    gamma1o = xin(3)
    MEEWT = par(1)*1.8 + 32.0
c
c   Control variable determination module: if the entering evaporator
c   water temperature is less than the minimum entering evaporator water
c   temperature, the control variable is set equal to 0.  If the entering
c   evaporator water temperature is greater than the minimum entering
c   evaporator water temperature plus 0.25 degree F, then the control
c   variable is set equal to sched1.  A dead band ensures controller
c   stability.
c
    If (EEWT .lt. MEEWT) gamma1n = 0
    If ((EEWT .gt. MEEWT) .and. (gamma1o .eq. 1)) gamma1n = sched1
    If ((EEWT .gt. MEEWT) .and. (gamma1o .lt. 0.9999))  gamma1n = 0
    If (EEWT .gt. MEEWT+0.25)  gamma1n = sched1
c
c   Output array module: this program segment fills the array out(1)
c   with the control variables calculated above to be returned to the
c   main program.
c
    out(1) = gamma1n
c
c   Return module
c
    return 1
    end
```

****

```
c             ICE HARVESTER CONTROLLER  TYPE 77
c
c        This subroutine sets the control variable for the ice harvester.
c     This value is determined by the ice tank inventory, the ice tank
c     capacity, the ice harvester schedule, and the previous value of the
c     control variable.
c
      subroutine type77(time,xin,out,t,dtdt,par,info,icntrl,*)
c
c     Variable declaration module: variables from main program
c
      implicit none
      real*8 out(1),xin(3)
      real*4 time,t(1),dtdt(1),par(1)
      integer*4 info(10)
      integer icntrl
c
c     Variable declaration module: variables used only in subroutine
c
      real*8 FIM,sched2,Ctnk,gamma3n,gamma3o
      character*3 ycheck(3),ocheck(1)
c
c     TYPECK, YCHECK, OCHECK, and RCHECK subroutine calling module: this
c     program segment sets info(6) and info(9), and calls the subroutines
c     listed above.  The old value of the control variable is initialized
c     to 0.
c
      If (info(7) .eq. -1) then
        info(6) = 1
        info(9) = 1
        call typeck(1,info,3,1,0)
        data ycheck/'MA1','CF1','CF1'/
        data ocheck/'CF1'/
        call rcheck(info,ycheck,ocheck)
        xin(3) = 0
      endif
c
c     Constant module: this program segment sets subroutine constants
c     equal to inputs and parameters passed from the main program.  English
c     units are converted to metric units.
c
      FIM = xin(1)*2.2046
      sched2 = xin(2)
      gamma3o = xin(3)
      Ctnk = par(1)*2.2046
c
c     Control variable determination module: If the ice tank inventory is
c     greater than 99% of its maximum value, the control variable will be set
c     equal to 0.  If the ice tank inventory is less than 98.5% of its
c     maximum value, then the control variable is equal to sched2.  A dead
c     band ensures controller stability.
c
      If (FIM .gt. 0.99*Ctnk)  gamma3n = 0
      If ((FIM .lt. 0.99*Ctnk) .and. (gamma3o .eq. 1)) gamma3n = sched2
      If ((FIM .lt. 0.99*Ctnk) .and. (gamma3o .lt. 0.9999)) gamma3n = 0
      If (FIM .lt. 0.985*Ctnk)  gamma3n = sched2
```

```
c
c   Output array module: this program segment fills the array out(1)
c   with the control variable calculated above to be returned to the
c   main program.
c
      out(1) = gamma3n
c
c   Return module
c
      return 1
      end
```

****

```
c              DIVERTER CONTROLLER  TYPE 78
c
c       This subroutine sets the control variable for the diverter.
c   This value is determined by the temperature of the water leaving
c   the chilled water storage tank, the maximum allowable chiller
c   evaporator inlet temperature, the chiller set point temperature, and
c   the previous value of the control variable.
c
      subroutine type78(time,xin,out,t,dtdt,par,info,icntrl,*)
c
c   Variable declaration module: variables from main program
c
      implicit none
      real*8 out(1),xin(2)
      real*4 time,t(1),dtdt(1),par(2)
      integer*4 info(10)
      integer icntrl
c
c   Variable declaration module: variables used only in subroutine
c
      real*8 LWT,divfo,SPT,MLWT,divfn
      character*3 ycheck(2),ocheck(1)
c
c   TYPECK, YCHECK, OCHECK, and RCHECK subroutine calling module: this
c   program segment sets info(6) and info(9), and calls the subroutines
c   listed above.  The old value of the control variable is initialized
c   to 0.
c
      If (info(7) .eq. -1) then
        info(6) = 1
        info(9) = 1
        call typeck(1,info,2,2,0)
        data ycheck/'TE1','CF1'/
        data ocheck/'CF1'/
        call rcheck(info,ycheck,ocheck)
```

```
      xin(2) = 0
    endif
c
c   Constant module: this program segment sets subroutine constants
c   equal to inputs and parameters passed from the main program. English
c   units are converted to metric units.
c
    LWT = xin(1)*1.8 + 32.0
    divfo = xin(2)
    SPT = par(1)*1.8 + 32.0
    MLWT = par(2)*1.8 + 32.0
c
c   Control variable determination module: If the storage tank leaving
c   water temperature is lower than the desired chiller inlet temperature, the
c   control variable is set equal to 0. If the storage tank leaving water
c   temperature is slightly higher than the desired chiller inlet
c   temperature. the control variable is set to a value that ensures
c   a roughly constant chiller inlet temperature. A deadband ensure con-
c   troller stability.
c
    If (LWT .le. MLWT) divfn = 0
    If ((LWT .gt. MLWT) .and. (divfo .gt. 0.001))
   @   divfn = (LWT - MLWT)/(LWT - SPT)
    If ((LWT .gt. MLWT) .and. (divfo .le. 0.001)) divfn = 0
    If (LWT .gt. MLWT + 0.25) divfn = (LWT - MLWT)/(LWT - SPT)
c
c   Output array module: this program segment fills the array out(1)
c   with the control variable calculated above to be returned to the
c   main program.
c
    out(1) = divfn
c
c   Return module
c
    return 1
    end
```

****

```
c                 EVAPORATIVE COOLER  TYPE 79
c
c      This subroutine models the performance of an evaporative cooler.
c   Given the entering dry and wet bulb temperatures and the dry air mass
c   flow rate. it will return the leaving dry and wet bulb temperatures
c   based on the evaporative cooler efficiency, as well as the dry air mass
c   flow rate.
c
    subroutine type79(time.xin.out.t.dtdt.par.info.icntrl.*)
c
```

```
c
c   Variable declaration module: variables from main program
c
      implicit none
      real*8 out(3),xin(3)
      real*4 time,t(1),dtdt(1),par(1)
      integer*4 info(10)
      integer icntrl
c
c   Variable declaration module: variables used only in subroutine
c
      real*8 EDB,EWB,eff,LDB,LWB,AMFR
      character*3 ycheck(3),ocheck(3)
c
c   TYPECK, YCHECK, OCHECK, and RCHECK subroutine calling module: this
c   program segment sets info(6) and info(9), and calls the subroutines
c   listed above.
c
      If (info(7) .eq. -1) then
        info(6) = 3
        info(9) = 0
        call typeck(1,info,3,1,0)
        data ycheck/'TE1','TE1','MF1'/
        data ocheck/'TE1','TE1','MF1'/
        call rcheck(info,ycheck,ocheck)
      endif
c
c   Constant module: this program segment sets subroutine constants
c   equal to inputs and parameters passed from the main program.  English
c   units are converted to metric units.
c
      EDB = xin(1)*1.8 + 32.0
      EWB = xin(2)*1.8 + 32.0
      AMFR = xin(3)*2.2046
      eff = par(1)
c
c   Leaving dry bulb calculation module: this program segments determines
c   the leaving dry and wet bulb temperatures.  If the air mass flow rate
c   is 0, the leaving dry bulb temperature is set equal to the entering
c   dry bulb temperature.
c
      LDB = EDB - eff*(EDB - EWB)
      If (AMFR .lt. 1)  LDB = EDB
      LWB = EWB
c
c   Output array module: this program segment fills the array out(3)
c   with the values calculated above to be returned to the main program.
c   English units are converted back to metric units.
c
      out(1) = (LDB - 32)/1.8
      out(2) = (LWB - 32)/1.8
      out(3) = AMFR/2.2046
c
c   Return module
c
      return 1
      end
```

APPENDIX D: TRNSYS SIMULATION FILES

- TRNSED Input File
- TRNSYS Simulation Deck

TRNSED INPUT FILE:
COMBUSTION TURBINE INLET COOLING SYSTEM MODEL
VERSION 9E

PREPARED BY KEVIN CROSS
UNIVERSITY OF WISCONSIN
SOLAR ENERGY LABORATORY
OCTOBER 2ND, 1994

First hour of simulation	5616	
Final hour of simulation	5784	
Simulation time step	0.0833	hr

***** UNIT 28 TYPE 79 EVAPORATIVE COOLER *****

PARAMETER	VALUE	UNITS
Evaporative cooler efficiency	0.00	

***** UNIT 25 TYPE 76 CHILLER AND CHILLER PUMP CONTROLLER *****

PARAMETERS	VALUE	UNITS
Minimum entering evaporator temperature	42.19	F

***** UNIT 27 TYPE 78 FLOW DIVERTER CONTROLLER *****

PARAMETERS	VALUE	UNITS
Chilled water set temperature	40.00	F
Maximum chiller evaporator inlet temperature	54.60	F

***** UNIT 1 TYPE 68 CENTRIFUGAL CHILLER *****

PARAMETERS	VALUE	UNITS
Maximum chiller load	1677.0	tons
Minimum chiller load	252.0	tons
Design load to normalize data with	1677.0	tons
Power consumption for design conditions	1186.0	kW

***** UNIT 2 TYPE 3 PUMP *****

PARAMETERS	VALUE	UNITS
Maximum flow rate	6844.2	GPM
Maximum power consumption	1.40	kW

***** UNITS 31 AND 32 TYPE 31 PIPES *****

PARAMETERS	VALUE	UNITS
Chiller - cooling tower pipe run	100.0	ft
Pipe diameter	21.60	in
Pipe loss coefficient per unit area	0.073	Btu/hrft2F

***** UNIT 3 TYPE 51 COOLING TOWER *****

PARAMETERS	VALUE	UNITS
Maximum air vol flow rate for each cell	667767	CFM
Power requirement of each fan	56.00	kW
Sump volume	-102664.4	gal
Cooling tower capacity	2015.0	tons

INPUTS	VALUE	UNITS
Sump make-up water temperature	77.0	F

***** UNIT 4 TYPE 3 PUMP *****

PARAMETERS	VALUE	UNITS
Maximum flow rate	2728.0	GPM
Maximum power consumption	1.10	kW

***** UNITS 33 AND 34 TYPE 31 PIPES *****

PARAMETERS	VALUE	UNITS
Chiller - storage tank pipe run	100.0	ft
Pipe diameter	13.60	in
Pipe loss coefficient per unit area	0.073	Btu/hrft2F

***** UNIT 5 TYPE 38 ALGEBRAIC TANK

PARAMETERS	VALUE	UNITS
Tank volume	1860575.7	gal
Tank Height	50.00	ft
Tank loss coefficient per unit area	0.073	Btu/hrft2F
Initial temperature of lower 1/5 of tank	40.00	F
Initial temperature of upper 4/5 of tank	59.98	F

***** UNIT 24 TYPE 75 COOLING COIL PUMP CONTROLLER *****

PARAMETERS	VALUE	UNITS
Minimum non-zero value of first control variable	0.200	
Minimum non-zero value of second control variable	0.500	

***** UNIT 6 TYPE 3 PUMP *****

PARAMETERS	VALUE	UNITS
Maximum flow rate	7086.1	GPM
Maximum power consumption	212.00	kW

***** UNITS 35 AND 36 TYPE 31 PIPES *****

PARAMETERS	VALUE	UNITS
Water storage tank - cooling coil pipe run	300.0	ft
Pipe diameter	17.00	in
Pipe loss coefficient per unit area	0.073	Btu/hrft2F

***** UNIT 7 TYPE 67 MODIFIED COOLING COIL *****

PARAMETERS	VALUE	UNITS
Number of heat exchanger rows	9	
Number of parallel tubes in each row	2618	
Duct height parallel to the tubes	6.79	ft
Duct width	218.20	ft
Outside diameter of tube	0.4020	in
Inside diameter of tube	0.3320	in
Thermal conductivity of tube material	7.74	Btu/hr-ft-F
Thickness of individual fin	0.01300	in
Spacing between individual fins	0.1250	in
Number of fins	652	
Thermal conductivity of fin material	102.27	Btu/hr-ft-F
Distance between centers of tubes in a row	1.000	in
Distance between center lines of rows	0.866	in

***** UNIT 26 TYPE 77 ICE HARVESTER CONTROLLER *****

PARAMETERS	VALUE	UNITS
Ice storage tank capacity	152407	lbs

***** UNIT 51 TYPE 72 BLACK BOX ICE HARVESTER *****

PARAMETERS	VALUE	UNITS
Nominal (compressor) capacity	27.00	tons
Net ice harvester capacity	24.00	tons
Design wet bulb temperature	77.0	F
Nominal evaporative condensor capacity	57.0	tons

***** UNIT 55 TYPE 71 ICE STORAGE TANK *****

PARAMETERS	VALUE	UNITS
Tank volume	39654.6	gal
Tank height	20.00	ft
Tank loss coefficient per unit area	0.073	Btu/hrft2F

DERIVATIVE	VALUE	UNITS
Beginning ice mass	128214	lb

***** UNIT 56 TYPE 3 PUMP *****

PARAMETERS	VALUE	UNITS
Maximum flow rate	2934.0	GPM
Maximum power consumption	79.00	kW

***** UNITS 45 AND 46 TYPE 31 PIPES *****

PARAMETERS	VALUE	UNITS
Ice storage tank - cooling coil pipe run	300.0	ft
Pipe diameter	10.90	in
Pipe loss coefficient per unit area	0.073	Btu/hrft2F

***** UNIT 57 TYPE 67 MODIFIED COOLING COIL *****

PARAMETERS	VALUE	UNITS
Number of heat exchanger rows	3	
Number of parallel tubes in each row	1083	
Duct height parallel to the tubes	16.41	ft
Duct width	90.26	ft
Outside diameter of tube	0.4020	in
Inside diameter of tube	0.3320	in
Thermal conductivity of tube material	7.74	Btu/hr-ft-F
Thickness of individual fin	0.01300	in
Spacing between individual fins	0.1250	in
Number of fins	1575	
Thermal conductivity of fin material	102.27	Btu/hr-ft-F
Distance between centers of tubes in a row	1.000	in
Distance between center lines of rows	0.866	in

***** UNIT 8 TYPE 69 BLACK BOX POWER PLANT MODEL *****

PARAMETERS	VALUE	UNITS
Base Electric Power	80039	kW
Base efficiency	0.2880	
Higher Heating Value	22760	Btu/lb
Inlet pressure drop due to cooling coils	0.00135	atm
Inlet pressure drop due to other sources	0.00000	atm
Exhaust pressure drop	0.01190	atm
Water-fuel ratio	1.80	
Air volumetric flow rate	520269	CFM
Peak CT electric power	92825	kW
Peak power plant output	92534	kW
Peak power plant output w/o inlet cooling	78570	kW

***** UNIT 10 TYPE 14 DAILY POWER PLANT SCHEDULE *****

The power plant is assumed to operate in a window between 12:00 noon and 9:00 p.m., Monday through Friday. The following parameters are the "normalized power outputs", i.e. the ratios of the desired power output to the peak power output, for each hour of the power plant's operation.

PARAMETERS	VALUE	UNITS
Normalized power requirement at 1:00	0.000	
Normalized power requirement at 1:00	0.849	
Normalized power requirement at 2:00	0.887	
Normalized power requirement at 2:00	0.887	
Normalized power requirement at 3:00	0.924	
Normalized power requirement at 3:00	0.924	
Normalized power requirement at 5:00	1.000	
Normalized power requirement at 7:00	0.924	
Normalized power requirement at 7:00	0.924	
Normalized power requirement at 8:00	0.887	
Normalized power requirement at 8:00	0.887	
Normalized power requirement at 9:00	0.849	
Normalized power requirement at 9:00	0.000	

****

```
*TRNSED
*|*                    TRNSYS SIMULATION DECK
*|*         COMBUSTION TURBINE INLET COOLING SYSTEM MODEL
*|*                          VERSION 9E
*|*
*|*
*|*                                         PREPARED BY KEVIN CROSS
*|*                                         UNIVERSITY OF WISCONSIN
*|*                                         SOLAR ENERGY LABORATORY
*|*                                         OCTOBER 2ND, 1994
*|*

ASSIGN \TRNSYS14\KEVC\CTICSME.LST 6
ASSIGN \TRNSYS14\KEVC\CTICSME.PLT 21
ASSIGN \TRNSYS14\KEVC\CTICSME.OUT 22
ASSIGN \TRNSYS14\KEVC\CTICSMEM.PLT 23
ASSIGN \TRNSYS14\KEVC\CTICSMEM.OUT 24
ASSIGN \TRNSYS14\KEVC\CTICSMEL.OUT 25
LIMITS  50 50 50
*|*
CONSTANTS 3
BT =  5.6160E+0003
*|First hour of simulation          | | |0|1|0|8760|1000
ET =  5.7840E+0003
*|Final hour of simulation          | | |0|1|0|8760|1000
DELT =  8.3300E-0002
*|Simulation time step              |hr|hr|0|1|0|1.0000|1000
SIMULATION BT ET DELT
TOLERANCES 0.0001 0.0001
*
*
```

```
EQUATIONS 13
*Energy flows*
CHpow = [1,6]
IHpow = [51,3]
FPpow = [3.3] + ([2.3] + [4,3])/3600
OPpow = CHpow + IHpow + FPpow
qcoil1 = MAX(0,([7,6]))*0.9479
qcoil2 = MAX(0,([57.6]))*0.9479
delUtnk1 = -[5,7]*0.9479
delqtnk2 = ([55,6] - ([55,7] + [55,5]))*0.9479
NEP = [8.2] - ([6,3] + [56,3])/3600
DNEP = [8.5] - ([6.3] + [56,3])/3600
*Power plant control variable, total simulation time, plot interval*
PPCNT = [10.2]*[18.1]
TST = ET - BT
PINT = 2*DELT
*
*
UNIT 9   TYPE 14  TIME DEP. FORCING FUNCTION (DAILY CHILLER SCHEDULE)
PARAMETERS 12
0     1.0
12    1.0
12    0.0
21    0.0
21    1.0
24    1.0
*
*
UNIT 18 TYPE 14 TIME DEP. FORCING FUNCTION (WEEKLY POWER PLANT SCHEDULE)
*(DAILY POWER PLANT SCHEDULE APPEARS WITH THE POWER PLANT COMPONENT. UNIT 8.)
PARAMETERS 12
0     1.0
21    1.0
21    0.0
69    0.0
69    1.0
168   1.0
*
*
UNIT 20 TYPE 14 TIME DEP. FORCING FUNCTION (WEEKLY CHILLER SCHEDULE)
PARAMETERS 44
0     1.0
12    1.0
12    0.0
69    0.0
69    1.0
84    1.0
84    0.0
93    0.0
93    1.0
108   1.0
108   0.0
117   0.0
117   1.0
132   1.0
132   0.0
141   0.0
```

```
141   1.0
156   1.0
156   0.0
165   0.0
165   1.0
168   1.0
*
*
UNIT 68 TYPE 14 TIME DEP. FORCING FUNCTION  (WEEKLY ICE HARVESTER SCHEDULE)
PARAMETERS 44
0     1.0
12    1.0
12    0.0
21    0.0
21    1.0
84    1.0
84    0.0
93    0.0
93    1.0
108   1.0
108   0.0
117   0.0
117   1.0
132   1.0
132   0.0
141   0.0
141   1.0
156   1.0
156   0.0
165   0.0
165   1.0
168   1.0
*
*
UNIT 11 TYPE 14  TIME DEP. FORCING FUNCTION   (DRY BULB TEMPERATURE)
PARAMETERS 20
0    22.8
4    21.7
7    22.2
10   27.8
13   32.2
16   34.4
17   35.0
19   33.3
22   25.0
24   22.8
*
*
UNIT 19 TYPE 14  TIME DEP. FORCING FUNCTION  (WET BULB TEMPERATURE)
PARAMETERS 20
0    21.7
4    21.1
7    21.7
10   24.4
13   25.0
16   24.4
17   24.4
```

```
19    24.4
22    22.8
24    21.7
*
*
UNIT 15 TYPE 33  PSYCHROMETRICS (DETERMINES HUMIDITY RATIO AT CC1 INLET)
PARAMETERS 4
1 1 0 1
INPUTS 2
28,1 28,2
25 24
*
*
UNIT 12  TYPE 24  QUANTITY INTEGRATOR (ENERGY FLOW RATES)
PARAMETERS 1
9999
INPUTS 9
CHpow IHpow FPpow OPpow qcoil1 qcoil2 delqtnk2 NEP 8,4
0 0 0 0 0 0 0 0 0
*
*
UNIT 29  TYPE 24  QUANTITY INTEGRATOR (MASS FLOW RATES)
PARAMETERS 1
9999
INPUTS 2
8,3 8,6
0 0
*
*
UNIT 13  TYPE 25  PRINTER (ENERGY FLOWS FOR SIMULATION PERIOD)
PARAMETERS 5
TST BT ET 22 2
INPUTS 10
12,1 12,2 12,3 12,4 12,5 12,6 delUtnk1 12,7 12,8 12,9
*
CHnrg IHnrg FPnrg OPnrg Qcoil1 Qcoil2 delUT1 delUT2 PPnrg PnrgNC
*
*
UNIT 14  TYPE 25  PRINTER  (KEY SYSTEM TEMPERATURES)
PARAMETERS 5
PINT BT ET 21 2
INPUTS 10
23,1 23,2 23,3 23,4 23,5 23,6 23,7 23,8 23,9 23,10
*
ADB AWB ECWT1 TTNK1 EWT1 LWT1 LDB1 EWT2 LWT2 LDB2
*
*
UNIT 17  TYPE 25  PRINTER  (KEY SYSTEM MASS FLOWS AND POWER PLANT OUTPUT)
PARAMETERS 5
PINT BT ET 23 2
INPUTS 10
23,11 23,12 23,13 23,14 23,15 23,16 23,17 NEP 8,4 DNEP
*
CMF1 EMF1 CCMF1 IGR FIM CCMF2 AMFR NEP EPNC DNEP
*
*
```

```
UNIT 23 TYPE 57 UNIT CONVERSION ROUTINE
PARAMETERS 57
1 1 2 1 1 2 1 1 2 1 1 2 1 1 2 1 1 2 1 1 2 1 1 2 1 1 2 1 1 2
15 1 3 15 1 3 15 1 3 15 1 3 7 1 3 15 1 3 15 1 3 15 1 3 15 1 3
INPUTS 19
11,2 19,2 32,1 5,10 35,1 7,4 7,1 45,1 57,4 57,1
32,2 33,2 36,2 51,2 55,3 46,2 57,3 8,3 8,6
50 50 50 50 50 50 50 50 50 50 1000 1000 1000 1000 1000 1000 1000 1000 1000
*
*
*UNIT 21 TYPE 25 PRINTER (FIRST COSTS)
*PARAMETERS 5
*TST BT ET 24 2
*INPUTS 10
*73,1 73,2 73,3 73,4 73,5 73,6 73,7 73,8 73,9 73,10
*
*CHCTS IH$ CC$ TNK1$ TNK2$ PMPIS CWSL$ ISL$ TOTAL CPKW
*
*
*FUEL USE AND EFFICIENCIES*
EQUATIONS 4
FM = [29,1]*2.2046
FMNC = [29,2]*2.2046
EFFC = [12,8]/(FM*22760/3412 + [12,4] + 1)
EFFNC = [12,9]/(FMNC*22760/3412 + 1)
*
*
UNIT 30 TYPE 25 PRINTER (FUEL USE, EFFICIENCIES, AND COSTS)
PARAMETERS 5
TST BT ET 25 2
INPUTS 10
FM FMNC EFFC EFFNC 73,7 73,8 73,9 73,11 73,12 73,13
*
FM FMNC EFFC EFFNC CWSL$ ISL$ TOTAL EFC OPEC Cepur
*
*
UNIT 74 TYPE 65 ONLINE PRINTER (AIR AND TANK TEMPERATURES; ICE INVENTORY)
PARAMETERS 14
3 1 35 95 0 10000 1 1 3 1 10 1 2 0
*
EQUATIONS 1
mFIM = [23,15]/1000
*
INPUTS 4
23,1 23,10 23,4 mFIM
ADB LDB2 TTNK1 mFIM
LABELS 4
F MLBS
AIR & TANK TEMPERATURES
ICE TANK INVENTORY
*
*|*
*|*     ***** UNIT 28 TYPE 79 EVAPORATIVE COOLER *****
*|*
*|*         PARAMETER                    VALUE      UNITS
*|*
```

```
CONSTANTS 1
EFF = 0.0000E+0000
*|Evaporative cooler efficiency          ||0|1|0|1.00|1000
*
UNIT 28 TYPE 79 EVAPORATIVE COOLER
PARAMETERS 1
EFF
INPUTS 3
11,2 19,2 7,3
25 24 1000000
*
*|*
*|*   ***** UNIT 25 TYPE 76 CHILLER AND CHILLER PUMP CONTROLLER *****
*|*
*|*      PARAMETERS                    VALUE     UNITS
*|*
CONSTANTS 1
MEEWT = 5.6589E+0000
*|Minimum entering evaporator temperature  |C|F|17.78|1.8|0|100.00|1000
*
UNIT 25 TYPE 76 CHILLER AND CHILLER PUMP CONTROLLER
PARAMETERS 1
MEEWT
INPUTS 3
5,3 20,1 25,1
15 1 0
*
*|*
*|*   ***** UNIT 27 TYPE 78 FLOW DIVERTER CONTROLLER *****
*|*
*|*      PARAMETERS                    VALUE     UNITS
*|*
CONSTANTS 2
TCHWS = 4.4422E+0000
*|Chilled water set temperature               |C|F|17.78|1.8|0|100.00|1000
MLWT = 1.2553E+0001
*|Maximum chiller evaporator inlet temperature |C|F|17.78|1.8|0|100.00|1000
*
*
UNIT 27 TYPE 78 DIVERTER CONTROLLER
PARAMETERS 2
TCHWS MLWT
INPUTS 2
34,1 27,1
10 0
*
*
*THE FOLLOWING FIVE EQUATIONS REPRESENT A FLOW DIVERTER AND A TEE-PIECE
*IN THE CHILLER - CHILLED WATER STORAGE TANK FLOW STREAM. A PORTION
*OF THE WATER LEAVING THE EVAPORATOR OUTLET MAY BE ROUTED DIRECTLY BACK
*TO THE EVAPORATOR INLET TO BE MIXED WITH WATER LEAVING THE STORAGE TANK.
*THIS MAY BE DONE TO MAINTAIN A CONSTANT EVAPORATOR INLET TEMPERATURE.
*
*
EQUATIONS 5
divf = [27,1]
TFR = [4,2]*(1 - divf)
```

```
DIVFR = [4,2]*divf
CHEFR = MAX(1,(TFR + DIVFR))
CHETMP = (TFR*[34,1] + DIVFR*[4,1])/CHEFR
*
*|*
*|*    ***** UNIT 1  TYPE 68  CENTRIFUGAL CHILLER *****
*|*
*|*        PARAMETERS                    VALUE      UNITS
*|*
CONSTANTS 4
QMAX = 2.1252E+0007
*|Maximum chiller load                  |kJ/hr|tons|0|7.891E-5|0|90000000.0|1000
QMIN = 3.1935E+0006
*|Minimum chiller load                  |kJ/hr|tons|0|7.891E-5|0|90000000.0|1000
QDES = 2.1252E+0007
*|Design load to normalize data with    |kJ/hr|tons|0|7.891E-5|0|90000000.0|1000
PDES = 1.1860E+0003
*|Power consumption for design conditions  |kW|kW|0|1.00|0|100000.0|1000
*
*
UNIT 1  TYPE 68  CENTRIFUGAL CHILLER
PARAMETERS 9
QMAX QMIN QDES PDES 0.140 0.544 0.316 0.012 0.015
INPUTS 6
TCHWS CHETMP CHEFR 32,1 32,2 25,1
TCHWS 12 500000 25 500000 1
*
*|*
*|*    ***** UNIT 2  TYPE 3  PUMP *****
*|*
*|*        PARAMETERS                    VALUE      UNITS
*|*
CONSTANTS 2
MMAX1 = 1.5562E+0006
*|Maximum flow rate                     |kg/hr|GPM|0|4.398E-3|0|9000000.0|1000
PPMAX1 = 5.0396E+0003
*|Maximum power consumption             |kJ/hr|kW|0|2.778E-4|0|1000000.00|1000
*
UNIT 2  TYPE 3  PUMP
PARAMETERS 4
MMAX1 4.2 PPMAX1 0.65
INPUTS 3
1,3 0,0 25,1
30 MMAX1 1
*
*|*
*|*    *****  UNITS 31 AND 32  TYPE 31  PIPES *****
*|*
*|*        PARAMETERS                    VALUE      UNITS
*|*

CONSTANTS 3
PL1 = 3.0479E+0001
*|Chiller - cooling tower pipe run      |m|ft|0|3.281|0|1000.0|1000
PD1 = 5.4864E-0001
*|Pipe diameter                         |m|in|0|39.37|0|100.00|1000
UP1 = 1.4922E+0000
```

```
*|Pipe loss coefficient per unit area       |kJ/hrm2C|Btu/hrft2F|0|4.892E-2|0|1000.000|1000
*
UNIT 31 TYPE 31 PIPE
PARAMETERS 6
PD1 PL1 UP1 1000 4.2 30
INPUTS 3
2,1 2,2 11,2
30 MMAX1 25
*
*|*
*|*     ***** UNIT 3  TYPE 51  COOLING TOWER *****
*|*
*|*        PARAMETERS                   VALUE      UNITS
*|*
CONSTANTS 4
VAMAX = 1.1345E+0006
*|Maximum air vol flow rate for each cell   |m3/hr|CFM|0|0.5886|0|9000000000|1000
PMAX = 5.6000E+0001
*|Power requirement of each fan             |kW|kW|0|1.00|0|10000.00|1000
VS = -3.8863E+0002
*|Sump volume                               |m3|gal|0|264.17|-1000|100000.0|1000
QTOW = 2.5535E+0007
*|Cooling tower capacity                    |kJ/hr|tons|0|7.891E-5|0|90000000.0|1000
*|*
*|*        INPUTS                  VALUE      UNITS
*|*
CONSTANTS 1
TMAIN = 2.4998E+0001
*|Sump make-up water temperature            |C|F|17.78|1.8|0|100.0|1000
*
UNIT 3  TYPE 51  COOLING TOWER
PARAMETERS 11
1 1 1 VAMAX PMAX 400000 VS 25 2 -0.63 2
INPUTS 6
31,1 31,2 11,2 19,2 TMAIN 25,1
20 MMAX1 25 24 20 1
*
*
UNIT 32 TYPE 31 PIPE
PARAMETERS 6
PD1 PL1 UP1 1000 4.2 24
INPUTS 3
3,1 3,2 11,2
24 MMAX1 25
*
*|*
*|*     ***** UNIT 4  TYPE 3  PUMP *****
*|*
*|*        PARAMETERS                   VALUE      UNITS
*|*
CONSTANTS 2
MMAX2 = 6.2028E+0005
*|Maximum flow rate                         |kg/hr|GPM|0|4.398E-3|0|10000000.0|1000
PPMAX2 = 3.9597E+0003
*|Maximum power consumption                 |kJ/hr|kW|0|2.778E-4|0|1000000.0|1000
*
```

```
UNIT 4 TYPE 3 PUMP
PARAMETERS 4
MMAX2 4.2 PPMAX2 0.65
INPUTS 3
1.1 0,0 25,1
TCHWS MMAX2 1
*
*|*
*|*    ***** UNITS 33 AND 34 TYPE 31 PIPES *****
*|*
*|*       PARAMETERS                    VALUE      UNITS
*|*
CONSTANTS 3
PL2 = 3.0479E+0001
*|Chiller - storage tank pipe run       |m|ft|0|3.281|0|1000.0|1000
PD2 = 3.4544E-0001
*|Pipe diameter                         |m|in|0|39.37|0|100.00|1000
UP2 = 1.4922E+0000
*|Pipe loss coefficient per unit area   |kJ/hrm2C|Btu/hrft2F|0|4.892E-2|0|1000.000|1000
*
UNIT 33 TYPE 31 PIPE
PARAMETERS 6
PD2 PL2 UP2 1000 4.2 4.44
INPUTS 3
4.1 TFR 11.2
TCHWS MMAX2 25
*
*|*
*|*    ***** UNIT 5 TYPE 38 ALGEBRAIC TANK
*|*
*|*       PARAMETERS                    VALUE      UNITS
*|*
CONSTANTS 8
VT1 = 7.0431E+0003
*|Tank volume                            |m3|gal|0|264.17|0|100000.0|1000
HT1 = 1.5239E+0001
*|Tank Height                            |m|ft|0|3.281|-50|100.00|1000
UT1 = 1.4922E+0000
*|Tank loss coefficient per unit area    |kJ/hrm2C|Btu/hrft2F|0|4.892E-2|0|10000.000|1000
ITB = 4.4422E+0000
*|Initial temperature of lower 1/5 of tank  |C|F|17.78|1.8|0|100.00|1000
ITT = 1.5542E+0001
*|Initial temperature of upper 4/5 of tank  |C|F|17.78|1.8|0|100.00|1000
MODE = 1
CONFIG = 1
HTHERM = 0.20*HT1
EQUATIONS 1
UA1 = UT1*2*(3.14*VT1*HT1)**0.5
*
UNIT 5 TYPE 38 ALGEBRAIC TANK
PARAMETERS 17
MODE VT1 HT1 HT1 4.2 1000 0.067 CONFIG UA1 1 ITB 0 HTHERM HTHERM ITT 0 0
INPUTS 6
36.1 36.2 33.1 33.2 11,2 0.0
12 1000000 TCHWS MMAX2 20 0
*
*
```

```
UNIT 34 TYPE 31 PIPE
PARAMETERS 6
PD2 PL2 UP2 1000 4.2 12
INPUTS 3
5,3 5,4 11,2
TCHWS MMAX2 25
*
*|*
*|*     ***** UNIT 24  TYPE 75  COOLING COIL PUMP CONTROLLER *****
*|*
*|*        PARAMETERS                    VALUE      UNITS
*|*
CONSTANTS 2
CV1min = 2.0000E-0001
*|Minimum non-zero value of first control variable       | | |0|1|0|1.000|1000
CV2min = 5.0000E-0001
*|Minimum non-zero value of second control variable    | | |0|1|0|1.000|1000
*
UNIT 24 TYPE 75  COOLING COIL PUMP CONTROLLER
PARAMETERS 2
CV1min CV2min
INPUTS 2
8,5 8,2
0 0
*
*|*
*|*     ***** UNIT 6  TYPE 3  PUMP *****
*|*
*|*        PARAMETERS                    VALUE      UNITS
*|*
CONSTANTS 2
MMAX3 = 1.6112E+0006
*|Maximum flow rate                   |kg/hr|GPM|0|4.398E-3|0|10000000.0|1000
PPMAX3 = 7.6314E+0005
*|Maximum power consumption           |kJ/hr|kW|0|2.778E-4|0|1000000.00|1000
*
UNIT 6 TYPE 3 PUMP
PARAMETERS 4
MMAX3 4.2 PPMAX3 0.65
INPUTS 3
5,1 5,2 24,1
TCHWS 0 0
*
*|*
*|*     ***** UNITS 35 AND 36  TYPE 31  PIPES *****
*|*
*|*        PARAMETERS                    VALUE      UNITS
*|*
CONSTANTS 3
PL3 = 9.1436E+0001
*|Water storage tank - cooling coil pipe run |m|ft|0|3.281|0|1000.0|1000
PD3 = 4.3180E-0001
*|Pipe diameter                      |m|in|0|39.37|0|100.00|1000
UP3 = 1.4922E+0000
*|Pipe loss coefficient per unit area    |kJ/hrm2C|Btu/hrft2F|0|4.892E-2|0|1000.000|1000
*
```

```
UNIT 35 TYPE 31 PIPE
PARAMETERS 6
PD3 PL3 UP3 1000 4.2 8.0
INPUTS 3
6.1 6.2 11.2
TCHWS MMAX3 25
*
*|*
*|*    ***** UNIT 7  TYPE 67  MODIFIED COOLING COIL *****
*|*
*|*        PARAMETERS                    VALUE      UNITS
*|*
CONSTANTS 13
NROWS1 = 9.0000E+0000
*|Number of heat exchanger rows              | | |0|1|0|20|1000
TUBES1 = 2.6180E+0003
*|Number of parallel tubes in each row       | | |0|1|0|5000|1000
LDUCT1 = 2.0695E+0000
*|Duct height parallel to the tubes          |m|ft|0|3.281|0|100.0|1000
WDUCT1 = 6.6504E+0001
*|Duct width                                 |m|ft|0|3.281|0|500.0|1000
DO1 = 1.0211E-0002
*|Outside diameter of tube                   |m|in|0|39.37|0|5.000|1000
DI1 = 8.4328E-0003
*|Inside diameter of tube                    |m|in|0|39.37|0|5.000|1000
KTUBE1 = 4.8224E+0001
*|Thermal conductivity of tube material      |kJ/hr-m-C|Btu/hr-ft-F|0|0.1605|0|1000.0|1000
FT1 = 3.3020E-0004
*|Thickness of individual fin                |m|in|0|39.37|0|1.0000|1000
FS1 = 3.1750E-0003
*|Spacing between individual fins            |m|in|0|39.37|0|5.000|1000
NFIN1 = 6.5200E+0002
*|Number of fins                             |||0|1|0|10000|1000
KFIN1 = 6.3720E+0002
*|Thermal conductivity of fin material       |kJ/hr-m-C|Btu/hr-ft-F|0|0.1605|0|1000.0|1000
FD1 = 2.5400E-0002
*|Distance between centers of tubes in a row |m|in|0|39.37|0|5.00|1000
C1 = 2.1996E-0002
*|Distance between center lines of rows      |m|in|0|39.37|0|5.00|1000
*
*

UNIT 7  TYPE 67  MODIFIED COOLING COIL
PARAMETERS 15
2 NROWS1 TUBES1 LDUCT1 WDUCT1 DO1 DI1 KTUBE1 FT1 FS1
NFIN1 KFIN1 1 FD1 C1
INPUTS 5
28.1 15.1 57.3 35.1 35.2
25 0.005 1000000 TCHWS MMAX3
*
*
UNIT 36 TYPE 31 PIPE
PARAMETERS 6
PD3 PL3 UP3 1000 4.2 8
INPUTS 3
7.4 7.5 11.2
TCHWS MMAX3 25
```

```
*
*|*
*|*     ***** UNIT 26  TYPE 77  ICE HARVESTER CONTROLLER *****
*|*
*|*        PARAMETERS                    VALUE      UNITS
*|*
CONSTANTS  1
CAP =  6.9119E+0004
*|Ice storage tank capacity              |kg|lbs|0|2.205|0|9000000|1000
*
UNIT 26 TYPE 77  ICE HARVESTER CONTROLLER
PARAMETERS 1
CAP
INPUTS 3
55,3 68,1 26,1
0 0 1
*
*|*
*|*     ***** UNIT 51  TYPE 72  BLACK BOX ICE HARVESTER *****
*|*
*|*        PARAMETERS                    VALUE      UNITS
*|*
CONSTANTS 4
NomCap =  3.4216E+0005
*|Nominal (compressor) capacity          |kJ/hr|tons|0|7.891E-5|0|30000000.00|1000
NCap =  3.0414E+0005
*|Net ice harvester capacity             |kJ/hr|tons|0|7.891E-5|0|30000000.00|1000
DWBT =  2.4998E+0001
*|Design wet bulb temperature            |C|F|17.78|1.8|0|100.0|1000
NCC =  7.2234E+0005
*|Nominal evaporative condensor capacity |kJ/hr|tons|0|7.891E-5|0|50000000.0|1000
*
UNIT 51 TYPE 72  BLACK BOX ICE HARVESTER
PARAMETERS 12
NomCap DWBT NCC 0.0125 0.9035 -0.0004658 0.0005408 0.716 0.9469
0.009320 -0.01074 -0.0001389
INPUTS 2
19,2 26,1
20 1
*
*|*
*|*     ***** UNIT 55  TYPE 71  ICE STORAGE TANK *****
*|*
*|*        PARAMETERS               VALUE      UNITS
*|*
CONSTANTS 3
VT2 =  1.5011E+0002
*|Tank volume                            |m3|gal|0|264.17|0|90000.0|1000
HT2 =  6.0961E+0000
*|Tank height                            |m|ft|0|3.2808|0|1000.0|1000
UT2 =  1.4922E+0000
*|Tank loss coefficient per unit area    |kJ/hrm2C|Btu/hrft2F|0|4.892E-2|0|10000.000|1000
*|*
*|*        DERIVATIVE              VALUE      UNITS
*|*
CONSTANTS  1
BIM =  5.8147E+0004
```

```
*|Beginning ice mass                  |kg|lb|0|2.205|0|9000000|1000
*
UNIT 55 TYPE 71  ICE STORAGE TANK
PARAMETERS 4
CAP VT2 HT2 UT2
INPUTS 4
46,1 46,2 51,2 11,2
4 1000000 7000 25
DERIVATIVES 1
BIM
*
*|*
*|*    ***** UNIT 56  TYPE 3  PUMP *****
*|*
*|*       PARAMETERS                  VALUE     UNITS
*|*
CONSTANTS 2
MMAX4 =  6.6712E+0005
*|Maximum flow rate                   |kg/hr|GPM|0|4.398E-3|0|10000000.0|1000
PPMAX4 =  2.8438E+0005
*|Maximum power consumption           |kJ/hr|kW|0|2.778E-4|0|1000000.00|1000
*
UNIT 56 TYPE 3 PUMP
PARAMETERS 4
MMAX4 4.2 PPMAX4 0.65
INPUTS 3
55,1 0,0 24,2
4 MMAX4 0
*
*|*
*|*    ***** UNITS 45 AND 46  TYPE 31  PIPES *****
*|*
*|*       PARAMETERS                  VALUE     UNITS
*|*
CONSTANTS 3
PL4 =  9.1436E+0001
*|Ice storage tank - cooling coil pipe run  |m|ft|0|3.281|0|500.0|1000
PD4 =  2.7686E-0001
*|Pipe diameter                       |m|in|0|39.37|0|100.00|1000
UP4 =  1.4922E+0000
*|Pipe loss coefficient per unit area       |kJ/hrm2C|Btu/hrft2F|0|4.892E-2|0|1000.000|1000
*
UNIT 45 TYPE 31 PIPE
PARAMETERS 6
PD4 PL4 UP4 1000 4.2 5.1
INPUTS 3
56,1 56,2 11,2
4 MMAX4 20
*
*|*
*|*    ***** UNIT 57 TYPE 67 MODIFIED COOLING COIL *****
*|*
*|*       PARAMETERS                  VALUE     UNITS
*|*
CONSTANTS 13
NROWS2 =  3.0000E+0000
*|Number of heat exchanger rows       | | |0|1|0|20|1000
```

```
TUBES2 =  1.0830E+0003
*|Number of parallel tubes in each row       | | |0|1|0|10000|1000
LDUCT2 =  5.0015E+0000
*|Duct height parallel to the tubes          |m|ft|0|3.281|0|100.0|1000
WDUCT2 =  2.7510E+0001
*|Duct width                                 |m|ft|0|3.281|0|500.0|1000
DO2 =  1.0211E-0002
*|Outside diameter of tube                   |m|in|0|39.37|0|5.0000|1000
DI2 =  8.4328E-0003
*|Inside diameter of tube                    |m|in|0|39.37|0|5.0000|1000
KTUBE2 =  4.8224E+0001
*|Thermal conductivity of tube material      |kJ/hr-m-C|Btu/hr-ft-F|0|0.1605|0|1000.0|1000
FT2 =  3.3020E-0004
*|Thickness of individual fin                |m|in|0|39.37|0|1.0000|1000
FS2 =  3.1750E-0003
*|Spacing between individual fins            |m|in|0|39.37|0|5.0000|1000
NFIN2 =  1.5750E+0003
*|Number of fins                             |||0|1|0|10000|1000
KFIN2 =  6.3720E+0002
*|Thermal conductivity of fin material       |kJ/hr-m-C|Btu/hr-ft-F|0|0.1605|0|1000.0|1000
FD2 =  2.5400E-0002
*|Distance between centers of tubes in a row |m|in|0|39.37|0|5.000|1000
C2 =  2.1996E-0002
*|Distance between center lines of rows      |m|in|0|39.37|0|5.000|1000
*
*
UNIT 57 TYPE 67  MODIFIED COOLING COIL
PARAMETERS 15
2 NROWS2 TUBES2 LDUCT2 WDUCT2 DO2 DI2 KTUBE2 FT2 FS2
NFIN2 KFIN2 1 FD2 C2
INPUTS 5
7,1 7,2 8,1 45,1 45,2
7.78 0.005 1000000 4 MMAX4
*
*
UNIT 46 TYPE 31 PIPE
PARAMETERS 6
PD4 PL4 UP4 1000 4.2 4.8
INPUTS 3
57,4 57,5 11,2
5 MMAX4 25
*
*|*
*|*     ***** UNIT 8  TYPE 69  BLACK BOX POWER PLANT MODEL *****
*|*
*|*        PARAMETERS               VALUE     UNITS
*|*
CONSTANTS 11
BEP =  8.0039E+0004
*|Base Electric Power                        |kW|kW|0|1.00|0|500000|1000
nbase =  2.8800E-0001
*|Base efficiency                            | | |0|1|0|1.0000|1000
HHV =  5.2943E+0004
*|Higher Heating Value                       |kJ/kg|Btu/lb|0|0.4299|0|100000|1000
CIPL =  1.3500E-0003
*|Inlet pressure drop due to cooling coils   |atm|atm|0|1.00|0|1.0000|1000
EIPL =  0.0000E+0000
```

```
*|Inlet pressure drop due to other sources    |atm|atm|0|1.00|0|1.0000|1000
OPL = 1.1900E-0002
*|Exhaust pressure drop                       |atm|atm|0|1.00|0|1.0000|1000
WFR = 1.8000E+0000
*|Water-fuel ratio                            | | |0|1|0|10.0|1000
VFR1 = 8.8391E+0005
*|Air volumetric flow rate                    |m3/hr|CFM|0|0.5886|0|1000000|1000
EPpeak = 9.2825E+0004
*|Peak CT electric power                      |kW|kW|0|1.00|0|500000|1000
EPmax = 9.2534E+0004
*|Peak power plant output                     |kW|kW|0|1.00|0|500000|1000
EPmin = 7.8570E+0004
*|Peak power plant output w/o inlet cooling   |kW|kW|0|1.00|0|500000|1000
*
*
UNIT 8 TYPE 69  BLACK BOX POWER PLANT MODEL
PARAMETERS 25
BEP nbase HHV CIPL EIPL OPL WFR VFR1 EPpeak 1.158 -2.478E-3 -3.73E-6 0.1777
2.341 -9.764E-4 8.181E-4 -2.401 -1.82E-6 -1.95E-6 0.9040 -1.90 -0.848 -0.848
0.0642 -0.0321
INPUTS 7
28,1 57,1 0,0 57,2 PPCNT 24,1 24,2
25 25 0.977 0.006 0 0 0
*
*|*
*|*    ***** UNIT 10  TYPE 14  DAILY POWER PLANT SCHEDULE *****
*|*
*|*    The power plant is assumed to operate in a window between 12:00
*|*    noon and 9:00 p.m., Monday through Friday. The following
*|*    parameters are the "normalized power outputs", i.e. the ratios
*|*    of the desired power output to the peak power output, for each hour
*|*    of the power plant's operation.
*|*
*|*       PARAMETERS           VALUE     UNITS
*|*
CONSTANTS 13
G1 = 0.0000E+0000
*|Normalized power requirement at 1:00      | | |0|1|0|1.00|1000
G2 = 8.4900E-0001
*|Normalized power requirement at 1:00      | | |0|1|0|1.00|1000
G3 = 8.8700E-0001
*|Normalized power requirement at 2:00      | | |0|1|0|1.00|1000
G4 = 8.8700E-0001
*|Normalized power requirement at 2:00      | | |0|1|0|1.00|1000
G5 = 9.2400E-0001
*|Normalized power requirement at 3:00      | | |0|1|0|1.00|1000
G6 = 9.2400E-0001
*|Normalized power requirement at 3:00      | | |0|1|0|1.00|1000
G7 = 1.0000E+0000
*|Normalized power requirement at 5:00      | | |0|1|0|2.00|1000
G8 = 9.2400E-0001
*|Normalized power requirement at 7:00      | | |0|1|0|1.00|1000
G9 = 9.2400E-0001
*|Normalized power requirement at 7:00      | | |0|1|0|1.00|1000
G10 = 8.8700E-0001
*|Normalized power requirement at 8:00      | | |0|1|0|1.00|1000
G11 = 8.8700E-0001
```

```
*|Normalized power requirement at 8:00      | |0|1|0|1.000|1000
G12 =  8.4900E-0001
*|Normalized power requirement at 9:00      | |0|1|0|1.000|1000
G13 =  0.0000E+0000
*|Normalized power requirement at 9:00      | |0|1|0|1.000|1000
*
*
UNIT 10   TYPE 14  TIME DEP. FORCING FUNCTION  (DAILY POWER PLANT SCHEDULE)
PARAMS 32
0      0.000
12.0   0.000
13.0   G1
13.0   G2
14.0   G3
14.0   G4
15.0   G5
15.0   G6
17.0   G7
19.0   G8
19.0   G9
20.0   G10
20.0   G11
21.0   G12
21.0   G13
24.0   0.000
*|*
*
UNIT 73 TYPE 73  COST CALCULATOR
PARAMETERS 46
QMAX NCap LDUCT1 WDUCT1 NROWS1 LDUCT2 WDUCT2 NROWS2 QTOW VT1
VT2 MMAX1 MMAX2 MMAX3 MMAX4 PL1 PD1 PL2 PD2 PL3 PD3 PL4 PD4 EPMAX EPmin
52933 74.65 0.04618 1365 8000 14.14 67.71 -0.06134 3.952E-5
1.567E+5 0.3543 -2.925E-8 1.51 -10.02 3.610 -7.178E-4 0.1017 0.055 20
0.058 0.0124
INPUTS 5
12.8 12.9 29.1 29.2 12.4
1 1 1 1 1
*
*
END
```

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