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User's Manual for CTCOOL: A Computer Model for Evaluating Combustion Turbine Inlet Air Cooling

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Summary

Combustion turbine generating capacity is inversely related to inlet air temperature. Conversely, peak generating demands, driven by space cooling needs, are positively related to ambient air temperature. Thus, generating capacity is low when the demand for electricity is the highest. Inlet air cooling offers an effective method of increasing the generating capacity of combustion turbines.

Previously, the Pacific Northwest National Laboratory investigated several alternative inlet air cooling systems over a broad range of application conditions (Brown, Katipamula, and Konynenbelt 1996). The results of this study showed that inlet air cooling was more cost-effective than building additional uncooled combustion turbine power plants for all of the application conditions investigated. This included both aeroderivative and "industrial-frame" turbines in simple- and combined-cycle configurations. Cost-effectiveness was shown even for relatively mild climates such as found in San Francisco.

A computer model called CTCOOL was developed to assist with the investigation described above. CTCOOL is a collection of C++ Classes developed under Microsoft Visual C++ 5.0©. The model consists of several input dialogs and analysis modules. In general, the model is capable of assessing the economic feasibility of cooling the inlet air to combustion turbines for site-specific conditions by simulating the performance of the turbine, sizing necessary equipment, estimating the cost of the equipment, and calculating the monetary value of incremental power production and the net present value of the capital investment.

This report describes CTCOOL and presents instructions for its use. For further information and/or to receive a copy of the CTCOOL software, please contact Srinivas Katipamula by email at Srinivas.Katipamula@pnl.gov or by phone at (509) 372-4592.

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1. Introduction

Combustion turbine inlet air cooling (CTIAC) should be considered by the power generating industry when seeking attractive options for increasing peak generating capacity. Inlet air cooling increases the power production capacity and decreases the heat rate of a combustion turbine during hot weather, when the demand for electricity is generally the greatest. Off-peak ice generation and storage using electrically-driven chillers has proven to be cost-effective under peak shaving conditions where there is a large difference between the value of peak and off-peak power.

Previously, the Pacific Northwest National Laboratory[1] conducted an assessment of CTIAC options for the U.S. Department of Energy's Office of Utility Technologies as part of its Thermal Energy Storage Program. The principal objectives of this study were as follows:

1) Identify the preferred CTIAC technology as a function of application conditions from currently available cooling technology

2) Identify application conditions where currently available cooling technology is not cost-effective, i.e., where construction of additional power plant capacity without CTIAC would be preferred

3) Determine the potential attractiveness of an advanced ammoniated salt solid/vapor absorption cooling system

4) Determine the potential attractiveness of cooling combustion turbine inlet air to 0 °F (in addition to cooling inlet air to a temperature around 40 to 45 °F, which is considered standard practice for CTIAC.).

To address the objectives described above, a screening study was designed to evaluate the principal cooling system technology options over a broad range of application conditions. Application variables included power plant type, power plant operating schedule, cooling system operating schedule, design inlet air temperature, and climate. Cooling system technology options included evaporative or refrigerative cooling, refrigerative chiller type, the inclusion (or exclusion) of storage, and the type of storage (storage media type, daily or weekly cycle, and load-leveling or load-shifting design).

The results of this study were previously published in Brown, Katipamula, and Konynenbelt (1996). The document that follows provides descriptive information and instructions for using the model developed by PNNL (CTCOOL) to evaluate the CTIAC options.

The following section provides an overview of inlet air cooling for those less familiar with the concept. The subsequent two sections introduce CTCOOL and provide instructions for getting the software set up and running on your computer. Detailed descriptions of the input screens and output options are provided next. The final chapter presents a sample case. Additional description of the model structure, underlying equations, and weather file formatting requirements are presented in the appendices.

[1] Operated for the U.S. Department of Energy by Battelle Memorial Institute under Contract DE-AC06-76RLO 1830

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2. Inlet Air Cooling Overview

Combustion turbines are constant volume machines, i.e., air intake is limited to a fixed volume of air regardless of ambient air conditions. As air temperature rises, its density falls. Thus, although the volumetric flow rate remains constant, the mass flow rate is reduced as air temperature rises. Power output is also reduced as air temperature rises because power output is proportional to mass flow rate. The conversion efficiency of the gas turbine also falls as air temperature rises because more power is required to compress the warmer air.

The impact of compressor inlet air temperature on mass flow rate, power output, and conversion efficiency is shown in Figure 2.1 for a typical "industrial" type turbine. Per custom, conversion efficiency is reported as a heat rate, which is the amount of fuel energy consumed per kWh of electricity produced. Thus, a rise in the heat rate is consistent with a drop in conversion efficiency. In general, the relationships are linear with temperature, or nearly so.

The performance curves show the dramatic affect of temperature on turbine performance, and the opportunity to improve performance via inlet air cooling. Cooling the inlet air improves both the power output and the heat rate, but the impact on power output is greater. Furthermore, the positive impacts on power output and heat rate increase with higher ambient temperatures, when utilities typically experience the highest demand for power. Therefore, incremental power production is greatest at the time it is most valued. *Usually, the primary objective of gas turbine inlet air cooling is to increase peak power output. The heat rate improvement is a significant, but secondary benefit.*

The relative importance of the two impacts depends mostly on the value of incremental kW and kWh production, the number of operating hours per year, and ambient air temperatures during the operating period. A greater number of annual operating hours will increase the importance of the heat rate impact, while fewer hours will emphasize the increase in power output. Applications with a higher ratio of average to peak ambient operating temperature will also increase the importance of the heat rate impact, all else equal.

In theory, power output could be further increased by cooling below the temperature range indicated in Figure 2.1. In practice, all turbines are designed around a maximum thrust level that sets a useful limit on the inlet air temperature. In particular, the maximum power output of "aeroderivative" type turbines often occurs around the freezing point and may actually decline below this temperature.

A block diagram of a generic combustion turbine inlet air cooling system, with and without storage, is shown in Figure 2.2. The basic building blocks are the chiller, its cooling tower, the air coil, and interconnecting piping. Cold fluid from the chiller is pumped through the air coil, where the coolant is heated and returned to the chiller, while the inlet air is cooled prior to entering the compressor. The cooling tower provides cooling water to the chiller condenser. Alternatively, an evaporative condenser might be used with some types of chillers. Including storage and its associated piping loop increases the number of system components, but allows the chiller and cooling tower components to be down-sized, assuming that cooling is not conducted 24 hours per day. Storage also significantly reduces on-peak parasitic power consumption for electrically-driven chiller systems.

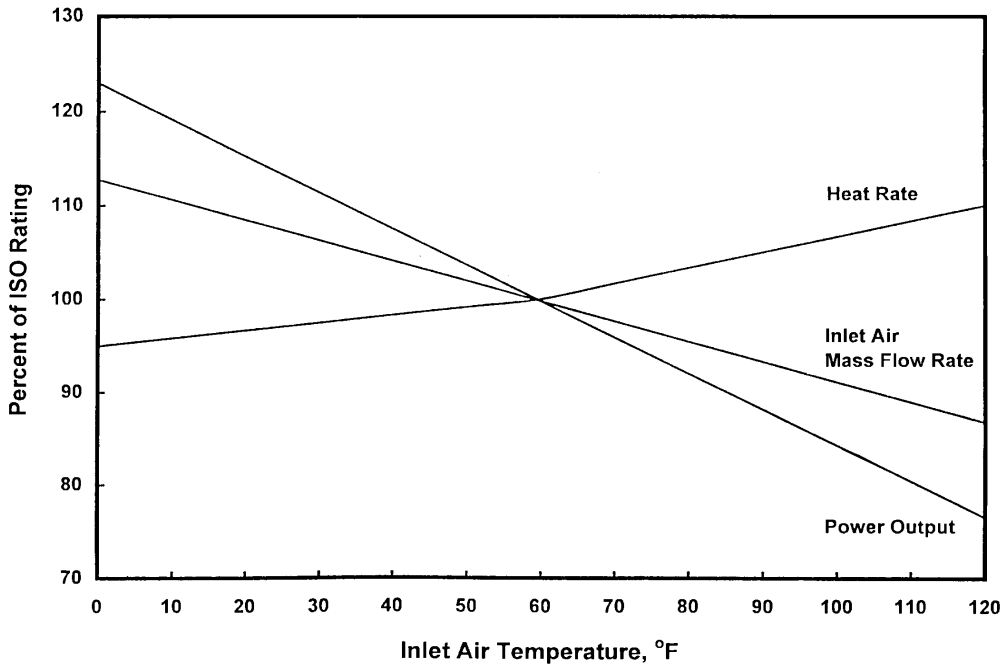

Figure 2.1 Typical Inlet Air Cooling Impacts on Combustion Turbine Performance

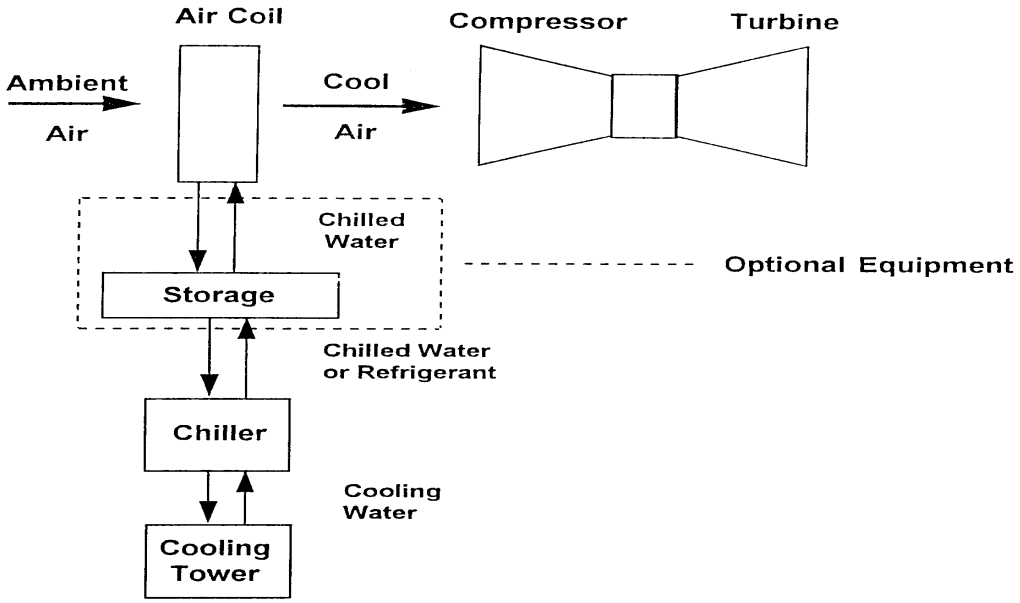

Figure 2.2 Generic Inlet Air Cooling System

The fundamental benefits of inlet air cooling were described above. The most obvious cost is the up-front investment and periodic maintenance of the inlet air cooling system hardware. The energy used to drive the chiller may also result in a significant expense, although the cost will vary significantly depending on whether the chiller is thermally or electrically-driven and whether the energy source is thermal or electric. Inclusion of an air cooling coil within the inlet duct to the compressor will incur an additional pressure loss, with negative consequences to power output and heat rate, but the impacts will generally be less than 0.5%.

The previous two paragraphs described the use of refrigerative systems for cooling turbine inlet air. A commonly employed alternative, besides not cooling the inlet air at all, is to use evaporative cooling. Direct contact evaporative cooling, accomplished by passing the inlet air through a wetted media, can be particularly effective in drier climates. Any consideration of refrigerative inlet air cooling should also consider evaporative cooling as an option. Note that evaporative and refrigerative approaches should be considered independently. Evaporative cooling followed by refrigerative cooling would not reduce the refrigerative cooling load. It would only substitute latent load for sensible load.

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3. Model Overview

The combustion turbine inlet air cooling performance evaluation model is a collection of C++ Classes developed under Microsoft Visual C++ 5.0©.

3.1. Model Structure

The model consists of several input dialogs and several analysis modules. The key analysis modules determine annual gas turbine performance, size cooling/storage equipment, estimate the cost of cooling/storage equipment, and calculate the net present value of a given configuration. Several TMY[2] weather files are provided with the software tool.

3.2. Model Capabilities

In general, the model is capable of assessing the economic feasibility of cooling the inlet air to combustion turbines for site-specific conditions by simulating the performance of the turbine with inlet air cooling, sizing necessary components, calculating the cost of the components, and calculating the monetary value of the incremental power and the net present value (NPV) of the investment. Sample output from the model is presented in Section 6.

The model is capable of evaluating the performance of cooling the inlet air to turbines in simple-cycle (industrial frame and aeroderivative type) or combined-cycle plant configurations. The air to the turbine can either be cooled refrigeratively or evaporatively. Thermal energy storage is an option for refrigeratively cooled systems. The model allows either daily or weekly ice or water storage with load-shifting or load-leveling storage designs. At the present time, the model can only handle three different inlet air temperatures: 1) 52 °F, 2) 42 °F and 3) 0 °F. For 0 °F cooling, the model cools the air in two stages with two different chillers: 1) from the ambient condition to 42 °F and 2) from 42 °F to 0 °F. The chillers can either be electrically-driven or thermally-activated; for thermally-activated chillers, the turbine exhaust gas (simple-cycle) or steam from the combined-cycle plant is used.

The performance analysis includes calculation of the power output and fuel consumption of the combustion turbine, with and without inlet air cooling, and calculation of the parasitic power consumption of inlet air cooling system components. The latter includes compressor power for vapor compression chillers, steam and pump power for absorption chillers, pump power for the various fluid circulation systems, and fan power for the cooling tower.

The model is also capable of computing annual performance data with the ability to aggregate performance calculated for each operating hour of the year, based on hourly weather data. The performance analysis is based on user-specified information that is described later in the report.

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[2] The Typical Meteorological Year (TMY) is a "typical" year of weather prepared from a large data base of the SOLMET data and is available for 239 stations in the U.S. The data include one complete year of hourly values (8760 values) for each of the following parameters: direct (beam) solar radiation, total horizontal solar radiation, dry-bulb (temperature), wet-bulb (temperature), dew-point (humidity), wind speed and cloud cover. These data are provided in a single file in ASCII format. The SOLMET (SOLar and METeorological) hourly database was first prepared for 26 cities that had measured hourly solar radiation data. Then it was extended to 222 additional locations by using models to generate synthetic solar radiation measures for these locations.

3.3. Model Limitations

Although the model is a powerful analysis tool, it has the following limitations:

1) The inlet air temperature is fixed to three values (52 °F, 42 °F and 0 °F).

2) The storage media is limited to ice or water (the choice between ice and water is implicit, i.e., if the air is cooled to 42 °F, ice storage is assumed; and for 52 °F, water storage is assumed).

3) In general, cooling in multiple stages is not possible. Currently, the model cools the air in a single stage when the air is cooled to 42 °F or 52 °F, and in two stages when the air is cooled to 0 °F.

4) The component sizing basis is limited to meeting 100% of the peak design conditions.

4. Getting Started

4.1. Hardware and Software Requirements

The CTCOOL computer program was written for use on personal computers with an Intel 80486© or higher processor running Windows 95® or Windows NT 4.0® or higher operating system. It also requires a VGA or better graphics monitor with at least 800 x 600 resolution.

4.2. Installation Instructions

Installation consists of simply creating a unique directory, preferably named "CTCOOL," and then copying all the files from the distribution diskettes into the newly created directory. The distribution diskettes contain one executable file (ctcool.exe); the rest of the files are either typical meteorological year (TMY) weather files (with extension *.wth) for various cities in the United States or the example input files (with extensions *.ctc).

4.3. Starting CTCOOL

Once the executable file and the weather files are all copied into a user-created directory, the program can be executed by double clicking the "ctcool.exe" file.

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5. Input Screens

The inputs required to carry out the performance analysis using CTCOOL are entered using the various menus shown in Figure 5.1. Although the menu-driven format allows the user to move easily throughout the program, most of the menu items are only activated by a previous menu selection. Therefore, the first level menu items on the main input screen are arranged in the order each should be defined. For example, the user must complete the inputs under the menu item "Project" before entering the inputs under the other menu items. However, the inputs can be changed at any time before or after an analysis is completed.

In addition to other commands, the command to open an existing input file or to save an input file can be found under the "File" menu item. A brief description of each menu item is provided in the sections that follow.

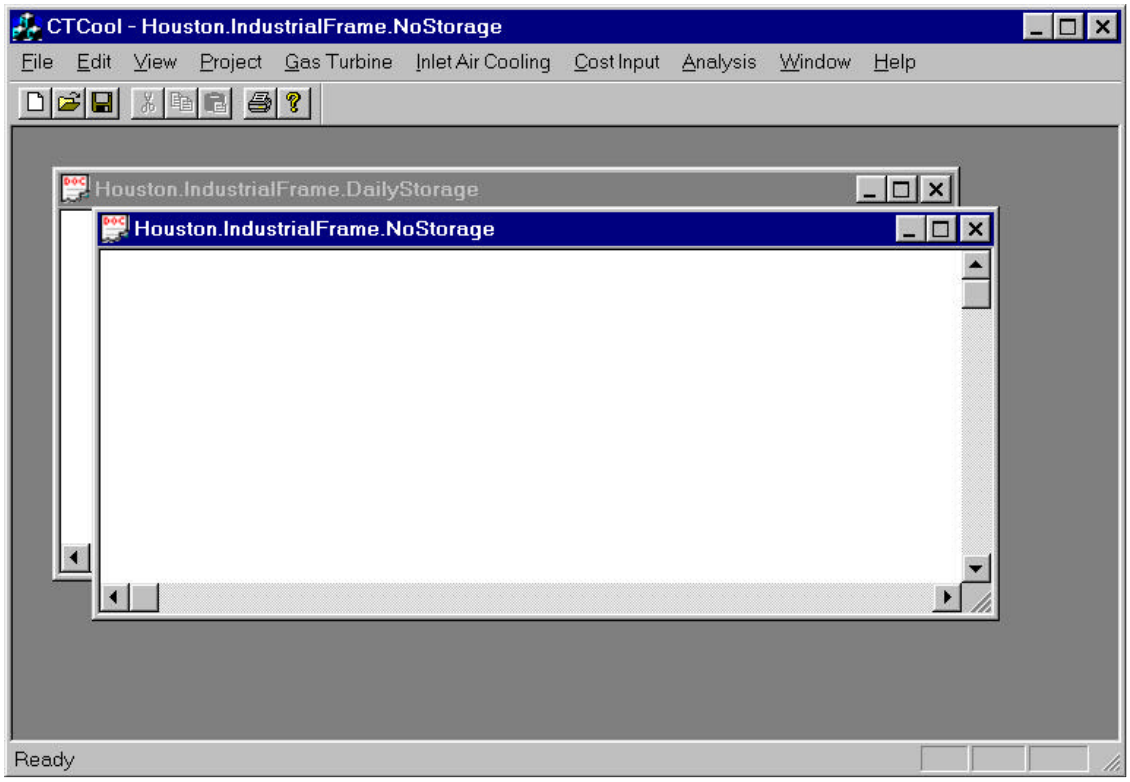

Figure 5.1 CTCOOL Main Screen

5.1. Project Information Dialog

The "Project Information" dialog (Figure 5.2) is the only item under the "Project" menu heading. This dialog allows file "labeling" with general project information and selection of the weather file. The general project information has no affect on the analysis; it only serves as an identifier. However, the selection of the weather file is critical to the analysis. Several weather files are provided with the software; these files have been derived from the TMY weather tapes. The dialog provides the user the ability to select a custom weather file by checking the "User Defined" button. After the "User Defined" button is checked, the "Get File Name" button becomes active and the user can provide the path and the

file name by double clicking on the button. The format of the weather file is described in Appendix B. It is the **user's responsibility** to make sure the data are consistent and are in the order specified in Appendix B. The changes made to the inputs will only be recorded if the user clicks on the "OK" button. By clicking on the "Cancel" button, the user can choose not to record the changes made. The functionality of the "OK" and "Cancel" buttons is identical on all input screens.

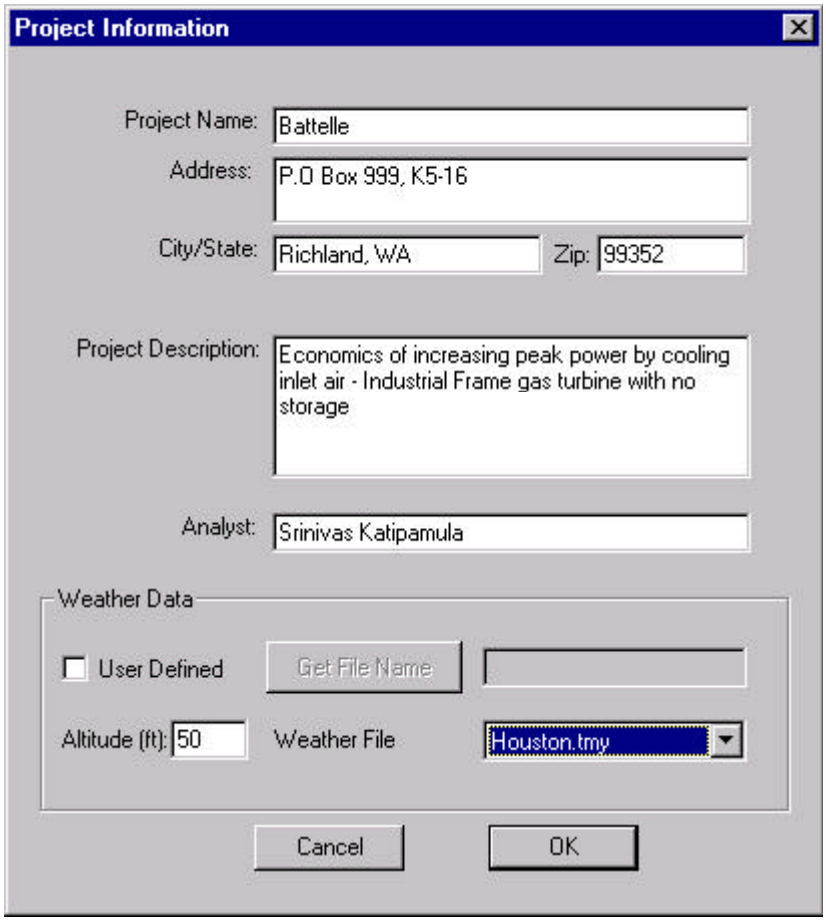

Figure 5.2 Project Information Dialog

5.2. ISO Ratings Input Dialog

After the project information is entered and a weather file is selected, the user must specify the type of gas turbine that is being considered for the analysis. The "ISO Ratings" input dialog (Figure 5.3) is part of the "Gas Turbine" menu item. CTCOOL supports three different types of gas turbine configurations, as shown in Figure 5.3. After the gas turbine type is selected, the heat rate, power output, and inlet air mass flow rate at ISO conditions (59 °F dry-bulb temperature, 60% relative humidity, and sea level elevation) for the selected turbine have to be entered. The last two entries (GT Power, and ST Power) in the ISO ratings input dialog are only needed for a combined-cycle industrial frame gas turbine; therefore, if that option is not checked, the input boxes corresponding to those entries are grayed out. All of the input dialogs have a "Default" button. By clicking on the "Default" button the default set of values will be filled in for every required input. These inputs are based on an evaluation of inlet air cooling technology conducted by PNNL and may not be valid for all situations. On this input dialog, in addition to the "Default ISO Ratings" button, there is also a button for "Default Performance Curves." If the user chooses to use the default ISO rating, the user may also choose to default the performance curves for

power, heat rate, and mass flow rate (and exhaust flow rate for combined-cycle). The input for these curves is described later in this section.

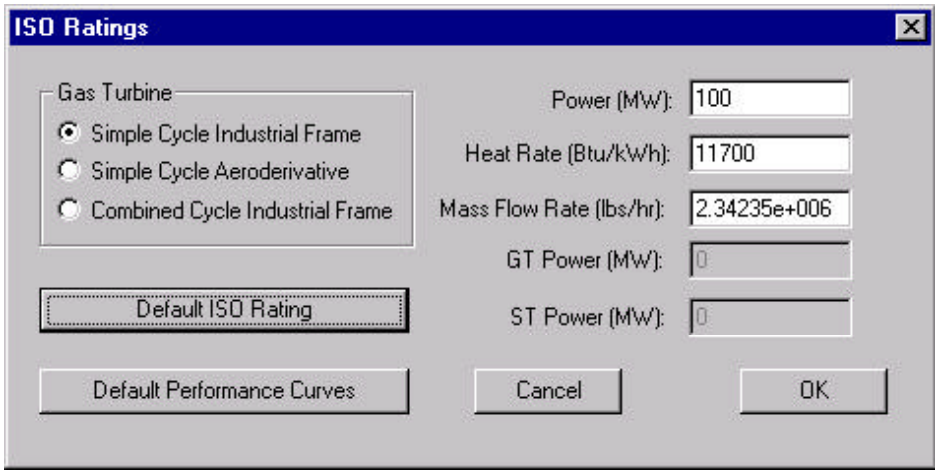

Figure 5.3 ISO Ratings Input Dialog

5.3. Heat Rate Performance Coefficients Input Dialog

Analysis of inlet air cooling systems requires knowledge of the power plant's performance (heat rate, power output, and mass flow rate) as a function of various operating conditions (altitude, dry-bulb temperature, humidity, turbine inlet pressure drop, and turbine exit pressure drop). The impact of operating conditions on power output and heat rate directly affect inlet air cooling economics, of course, and understanding the impact on mass flow rate is crucial to sizing cooling system components.

The model uses up to five adjustment factors (F1 for altitude, F2 for dry-bulb temperature, F3 for humidity, F4 for excess inlet pressure drop, and F5 for excess exhaust pressure drop) to estimate the heat rate for non-ISO conditions (see Figure 5.4). The adjustment factors are assumed to vary linearly with the change in the independent variable (altitude, dry-bulb temperature, absolute humidity, excess inlet pressure drop and excess exhaust pressure drop). Therefore, the slope and intercept of each of the five adjustments have to be entered. The combustion turbine manufacturer generally provides performance data for non-ISO conditions. For the dry-bulb temperature, the user has the flexibility to segment the linear relationship into one or more regions (up to three). For example, two regions are used in Figure 5.4 to define the impact of dry-bulb temperature on heat rate.

The "Heat Rate Performance Coefficients" input dialog is part of the "Gas Turbine" menu item. The default input values can be assigned by clicking on the "Default" button or by clicking "Default Performance Curves" from the "ISO Ratings" input dialog (Figure 5.3). There are different sets of default values depending on the choice of the gas turbine type. The inputs are used to estimate the adjustment factors for each hour. The adjustment factors are multiplied by the ISO performance data to calculate performance at non-ISO conditions.

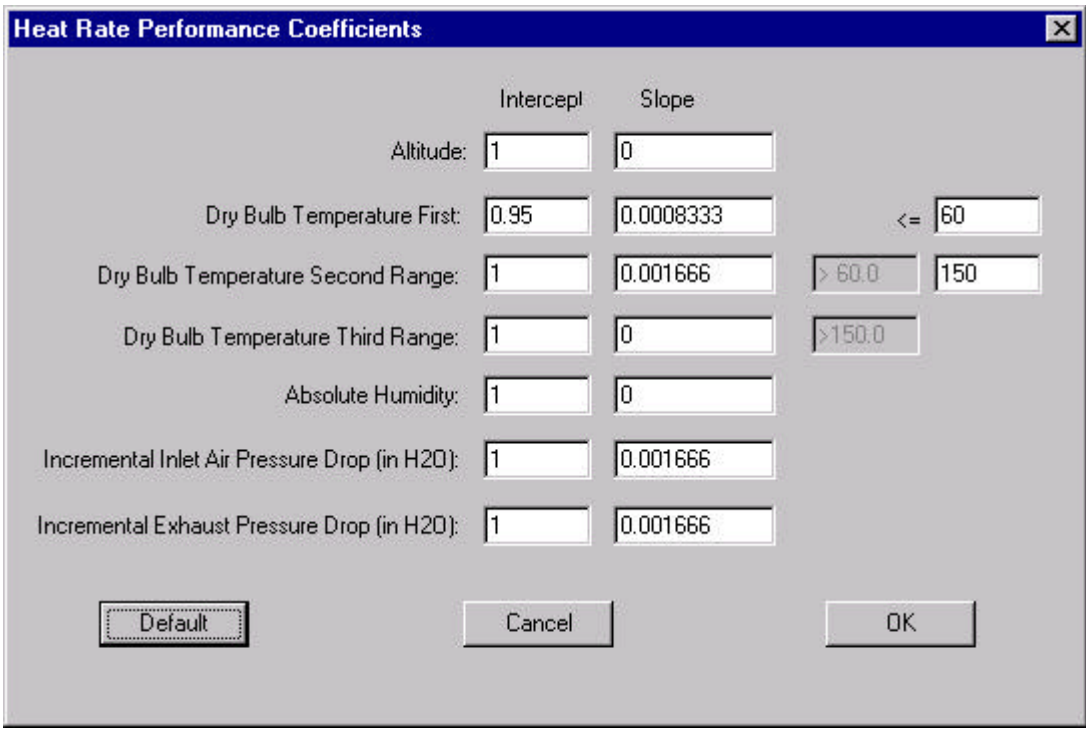

Figure 5.4 Heat Rate Performance Coefficients Input Dialog

5.4. Power Performance Coefficients Input Dialog

Like the approach just described for modeling the heat rate, the model uses up to five adjustment factors (F1 for altitude, F2 for dry-bulb temperature, F3 for humidity, F4 for excess inlet pressure drop, and F5 for excess exhaust pressure drop) to estimate the power at non-ISO conditions (see Figure 5.5). The adjustment factors are assumed to vary linearly with the change in the independent variable (altitude, dry-bulb temperature, absolute humidity, excess inlet pressure drop and excess exhaust pressure drop). Therefore, the slope and intercept of each of the five adjustments have to be entered. The combustion turbine manufacturer generally provides performance data for non-ISO conditions. For the dry-bulb temperature, the user has the flexibility to segment the linear relationship into one or more regions (up to three).

The "Power Performance Coefficients" input dialog is part of the "Gas Turbine" menu item. The default input values can be assigned by clicking on the "Default" button or by clicking "Default Performance Curves" from the "ISO Ratings" input dialog (Figure 5.3). There are different sets of default values depending on the choice of the gas turbine type. The inputs are used to estimate the adjustment factors for each hour. The adjustment factors are multiplied by the ISO performance data to calculate performance at non-ISO conditions.

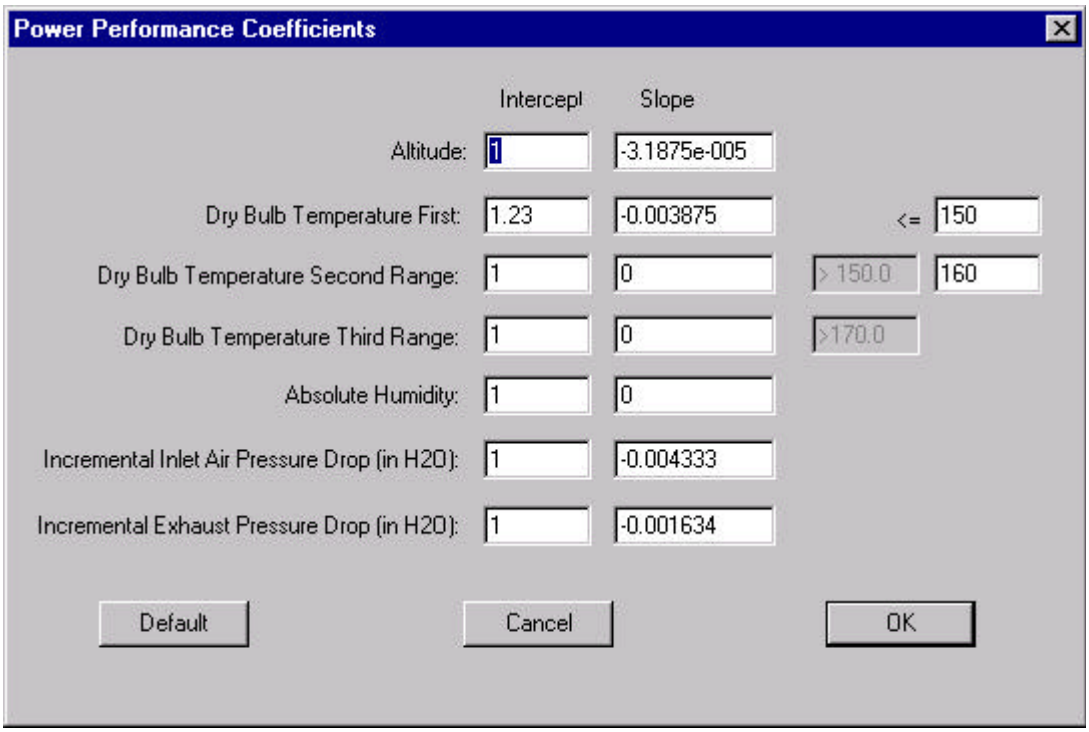

Figure 5.5 Power Performance Coefficients Input Dialog

5.5. Air Flow Rate Performance Coefficients Input Dialog

As for heat rate and power, the model uses up to five adjustment factors (F1 for altitude, F2 for dry-bulb temperature, F3 for humidity, F4 for excess inlet pressure drop, and F5 for excess exhaust pressure drop) to estimate the air flow rate at non-ISO conditions (see Figure 5.6). The adjustment factors are assumed to vary linearly with the change in the independent variable (altitude, dry-bulb temperature, absolute humidity, excess inlet pressure drop and excess exhaust pressure drop). Therefore, the slope and intercept of each of the five adjustments have to be entered. The combustion turbine manufacturer generally provides performance data for non-ISO conditions. For the dry-bulb temperature, the user has the flexibility to segment the linear relationship into one or more regions (up to three).

The "Air Flow Performance Coefficients" input dialog is part of the "Gas Turbine" menu item. The default input values can be assigned by clicking on the "Default" button or by clicking "Default Performance Curves" from the "ISO Ratings" input dialog (Figure 5.3). The inputs are used to estimate the adjustment factors for each hour. The adjustment factors are multiplied by the ISO performance data to calculate performance at non-ISO conditions.

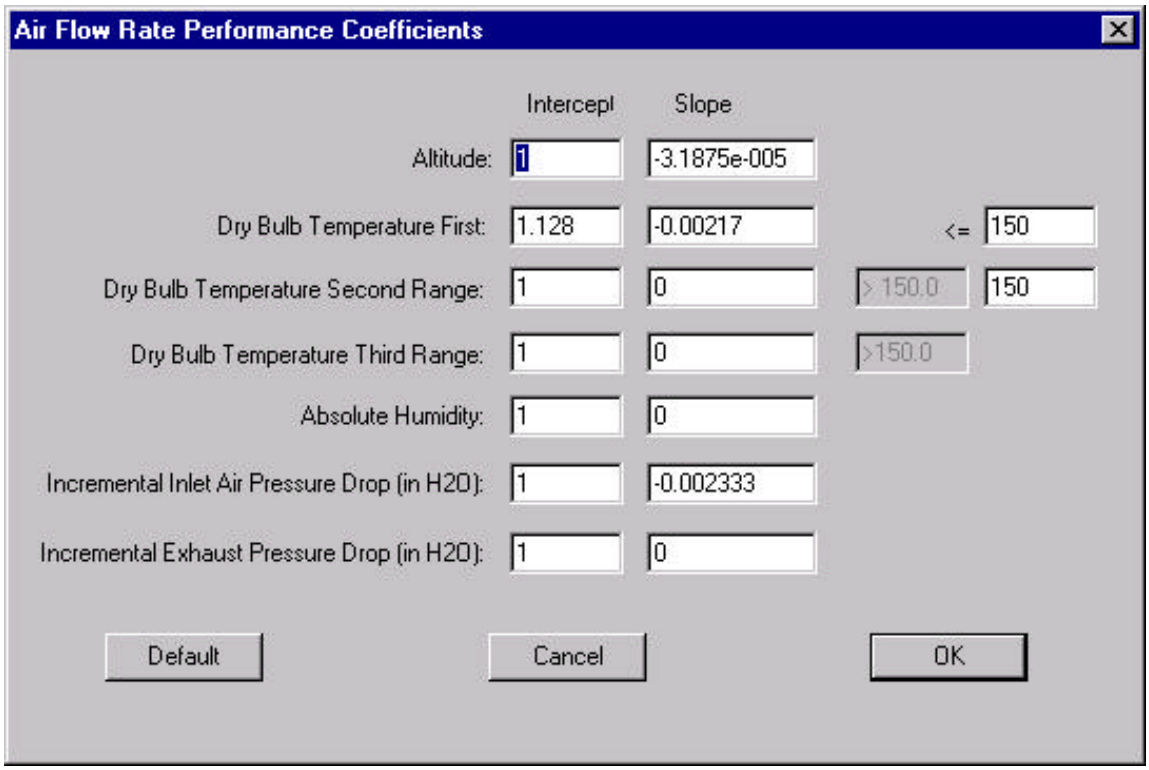

Figure 5.6 Air Flow Rate Performance Coefficients Input Dialog

5.6. Exhaust Flow Rate Performance Coefficients Input Dialog

The inputs for the exhaust flow rate performance coefficient are only needed if a combined-cycle industrial frame gas turbine is selected from the "ISO Ratings" input dialog (Figure 5.3). Again, the model uses up to five adjustment factors (F1 for altitude, F2 for dry-bulb temperature, F3 for humidity, F4 for excess inlet pressure drop, and F5 for excess exhaust pressure drop; see Figure 5.7) to estimate the exhaust flow rate at non-ISO conditions. The adjustment factors are assumed to vary linearly with the change in the independent variable (altitude, dry-bulb temperature, absolute humidity, excess inlet pressure drop and excess exhaust pressure drop). Therefore, the slope and intercept of each of the five adjustments have to be entered. The combined-cycle plant equipment manufacturers generally provide performance data for non-ISO conditions. For the dry-bulb temperature, the user has the flexibility to segment the linear relationship into one or more regions (up to three).

The "Exhaust Flow Performance Coefficients" input dialog is part of the "Gas Turbine" menu item. The default input values can be assigned by clicking on the "Default" button or by clicking "Default Performance Curves" from the "ISO Ratings" Input Dialog (Figure 5.3). The inputs are used to estimate the adjustment factors for each hour. The adjustment factors are multiplied by the ISO performance data to calculate performance at non-ISO conditions.

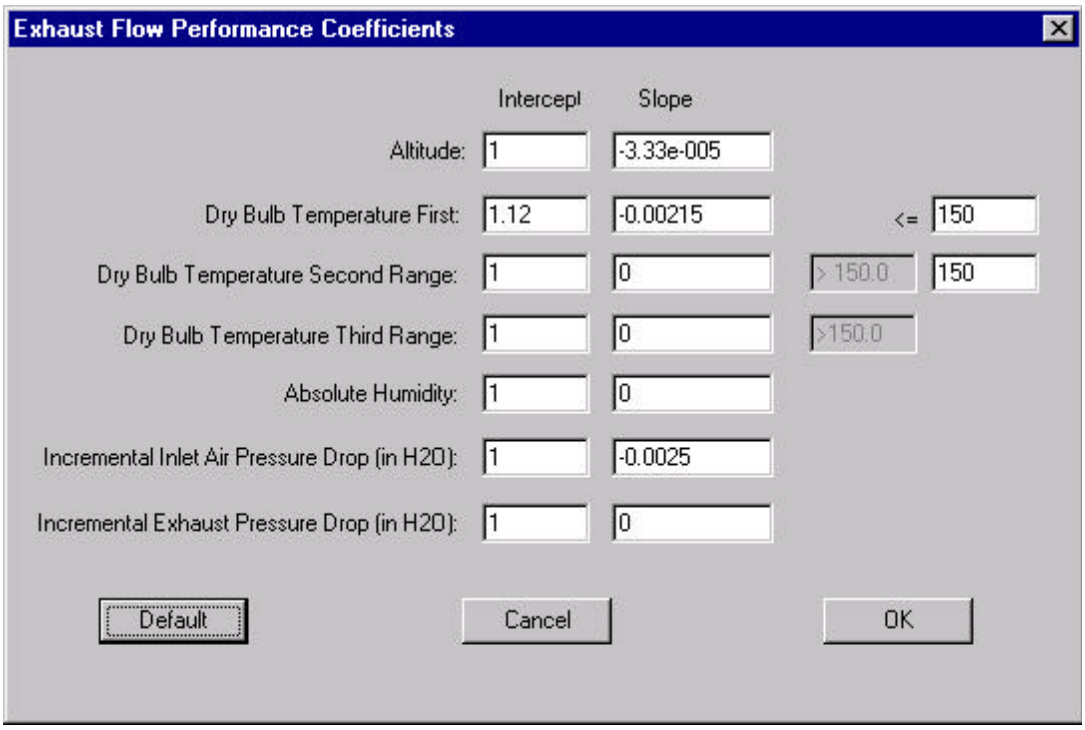

Figure 5.7 Exhaust Flow Performance Coefficients Input Dialog

5.7. Gas Turbine Operating Schedules Input Dialog

The operating schedule for the gas turbine can be input using the dialog box shown in Figure 5.8. The "Gas Turbine Operating Schedules" input dialog is part of the "Gas Turbine" menu item. The user has the flexibility to specify when the gas turbine is started and stopped by the day-of-week and the month-of-year. For the inputs provided in Figure 5.8, the gas turbine will operate between 1400 hours and 1800 hours (4 hours of operation) every day except Saturday and Sunday for the month of April. Similar inputs can be provided for the other months. To simplify inputs, additional buttons are provided at the bottom of the dialog. For example, the user can enter the schedule for 1 month and then copy the same schedule to the other 11 months just by clicking on the "Copy" button (any previous values associated with the other months will be overwritten with the values of the current month). Also, by simply clicking the "Clear" button, the schedule for any month can be cleared. Unlike the "Copy" button, the "Clear" button only clears the current selection (month).

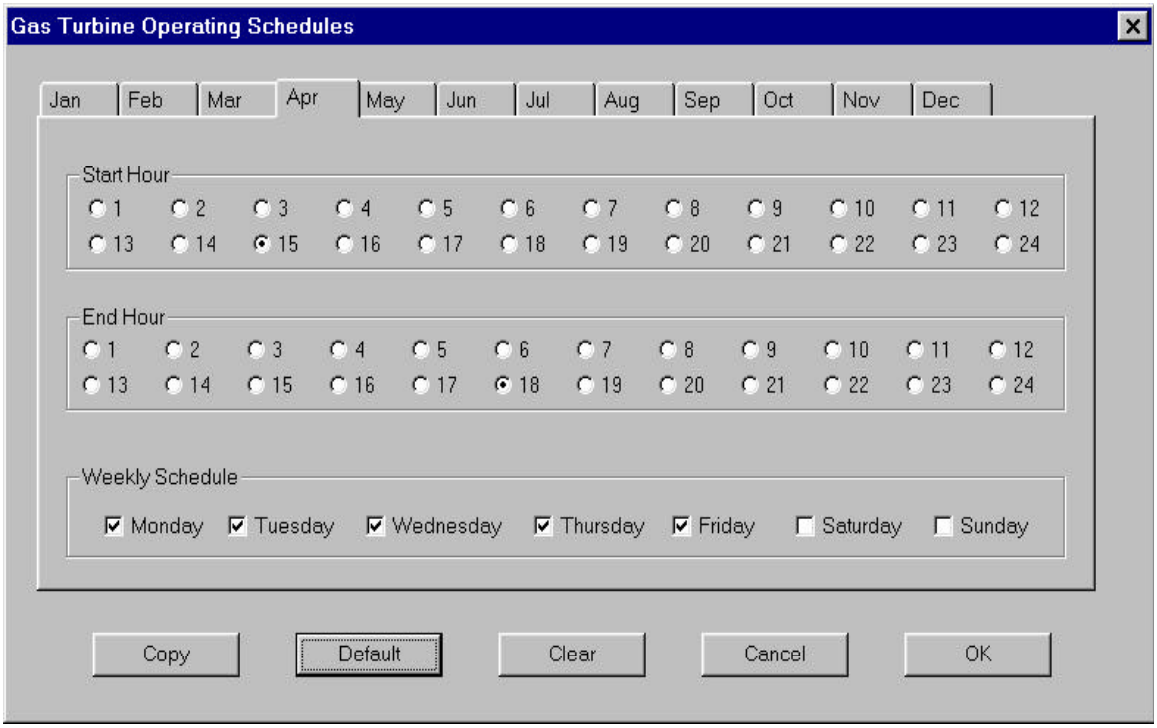

Figure 5.8 Gas Turbine Operating Schedules Input Dialog

5.8. Inlet Air Cooling Configuration Input Dialog

CTCOOL supports two different types of cooling options, as shown in Figure 5.9. The "Inlet Air Cooling Configuration" input dialog is the first item under the menu item "Inlet Air Cooling." By checking the "No Inlet Air Cooling" option, the annual performance of the gas turbine without cooling the inlet air can be analyzed. The inlet air temperature is a function of wet-bulb temperature for the evaporative cooling option, and for the refrigerative option, it is a user-defined input (described later in this section).

Figure 5.9 Inlet Air Cooling Configuration Input Dialog

5.9. Cooling Operating Schedule Input Dialog

The operating schedule for the cooling system can be input using the dialog shown in Figure 5.10. The "Cooling Operating Schedules" input dialog is part of the "Inlet Air Cooling" menu item and is only accessible if either of the inlet air cooling options is checked in the previous input dialog (Figure 5.9). The cooling schedule is independent of the gas turbine schedule. However, if the selected cooling period is greater than the gas turbine operation, the inlet air will only be cooled during gas turbine operation.

The functionality of this input dialog is similar to the explanation provided for the gas turbine schedule (see Section 5.7)

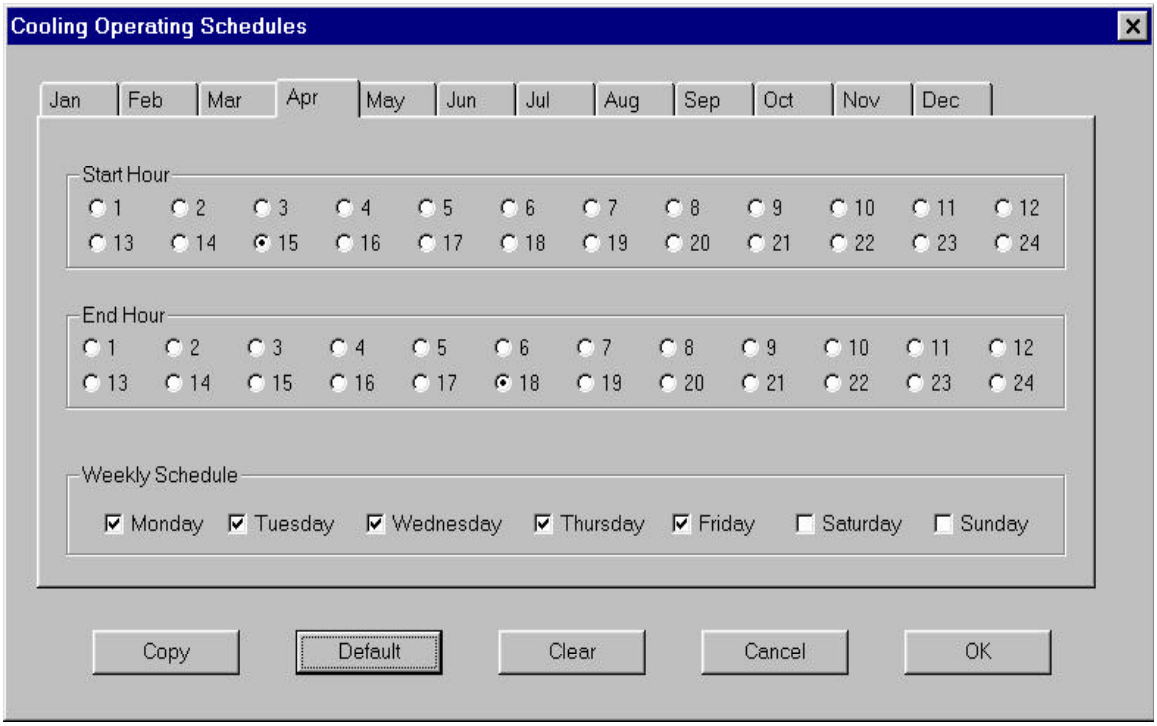

Figure 5.10 Cooling Operating Schedule Input Dialog

5.10. Evaporative Cooling Input Dialog

The input dialog shown in Figure 5.11 is part of the "Inlet Air Cooling" menu item. The "Evaporator Cooling Configuration" input dialog is only accessible if the evaporative cooling option is selected in the "Inlet Air Cooling Configuration" input dialog (Figure 5.9). Only four inputs are required to simulate the performance impacts with evaporative cooling. Although the water consumption for cooling air evaporatively is separately calculated, a user-specified input is used to calculate the additional water that is consumed to accommodate blowdown requirements. The evaporator effectiveness is used to calculate the temperature of the inlet air at the exit of the evaporator. The user may also specify the parasitic power requirement of pumps, fans, etc., as a percent of the ISO power. Adding an evaporative cooling device increases the inlet air pressure drop, resulting in decreased power output and increased heat rate, all else equal. The user-specified additional pressure drop associated with the presence of the evaporator in the inlet air stream (inches of water) is used to adjust the ISO performance.

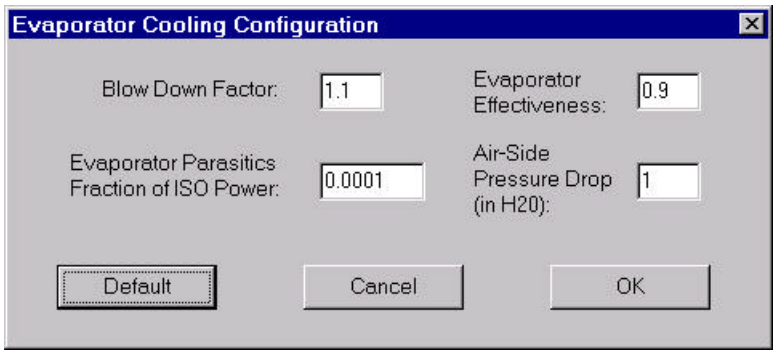

Figure 5.11 Evaporative Cooling Inputs Dialog

5.11. Refrigerative Cooling Input Dialog

The input dialog shown in Figure 5.12 is part of the "Inlet Air Cooling" menu item. The "Refrigerative Cooling Configuration" input dialog is only accessible if the refrigerative cooling option is selected in the "Inlet Air Cooling Configuration" input dialog (Figure 5.9). The user may select one of the three fixed inlet air temperatures: 0 °F, 42 °F or 52 °F. When inlet air is cooled to 0 °F, the second stage chiller selection and chiller configuration buttons are activated. For 0 °F inlet air cooling, the air is cooled in two stages, first to 42 °F and then from 42 °F to 0 °F. The model will not allow a lithium bromide chiller to be used with 0 °F or 42 °F inlet air temperatures. However, the user should use caution in selecting a chiller configuration that will meet the inlet air temperature requirement.

Adding a refrigerative cooling device increases the inlet air pressure drop, resulting in decreased power output and increased heat rate, all else equal. The user-specified additional pressure drop associated with the presence of the refrigerative cooling coils in the inlet air stream (inches of water) is used to adjust the ISO performance. There is also an additional exhaust side pressure drop when a heat recovery steam generator (HRSG) is added to a simple-cycle turbine to provide steam for a thermally-activated chiller. The pressure drop should be specified in terms of inches of water. If an HRSG is not used, the value should be set to zero. The generator cooling factor is used to allow for additional cooling capacity to cool the generator. Supplemental generator cooling will likely be required for its capacity to match the increased capacity of the turbine on a hot summer day.

This dialog also provides the user the choice of selecting the chiller type and chiller performance characteristics. The tool supports five different types of chillers: 1) single-stage lithium bromide, 2) double-stage lithium bromide, 3) aqua-ammonia, 4) vapor compression, and 5) complex-compound. For information on complex-compound chillers, refer to Brown, Katipamula, and Konynenbelt (1996). After the chiller type is selected, clicking the "Chiller Configuration" button will raise the chiller input characteristics dialog.

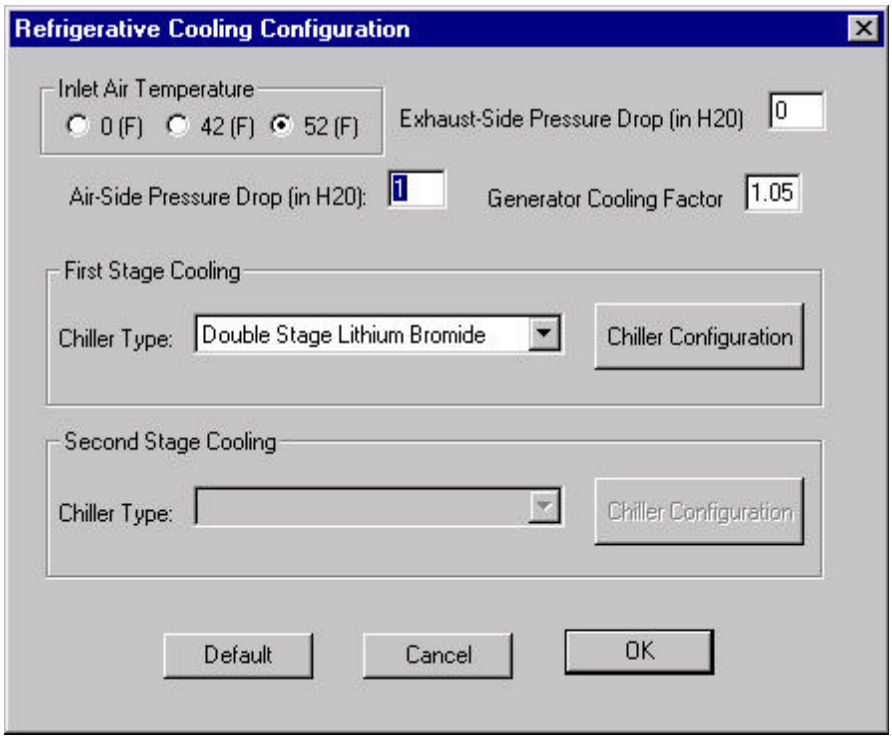

Figure 5.12 Refrigerative Cooling Configuration Dialog

5.12. Aqua-Ammonia Chiller Performance Data Input Dialog

The "Aqua-Ammonia Performance Data" input dialog is accessed by selecting an aqua-ammonia chiller for cooling and clicking the "Chiller Configuration" button from the "Refrigerative Cooling Configuration" input dialog.

Chiller performance is characterized by its coefficient-of-performance (COP), "relative capacity," heat rejection rate, and, for absorption-type chillers, the steam conditions and parasitic electric power consumption.

The COP is the amount of cooling obtained per unit of input energy, with the input and output measured in the same units. COP is assumed to vary as a function of wet-bulb temperature, as shown in Figure 5.13. The COP equations assume the condenser temperature is 15 °F higher than the wet-bulb temperature. In general, the wet-bulb temperature is limited to a minimum of 25 °F for modeling chiller COP, except for lithium bromide chillers, which have higher limits.

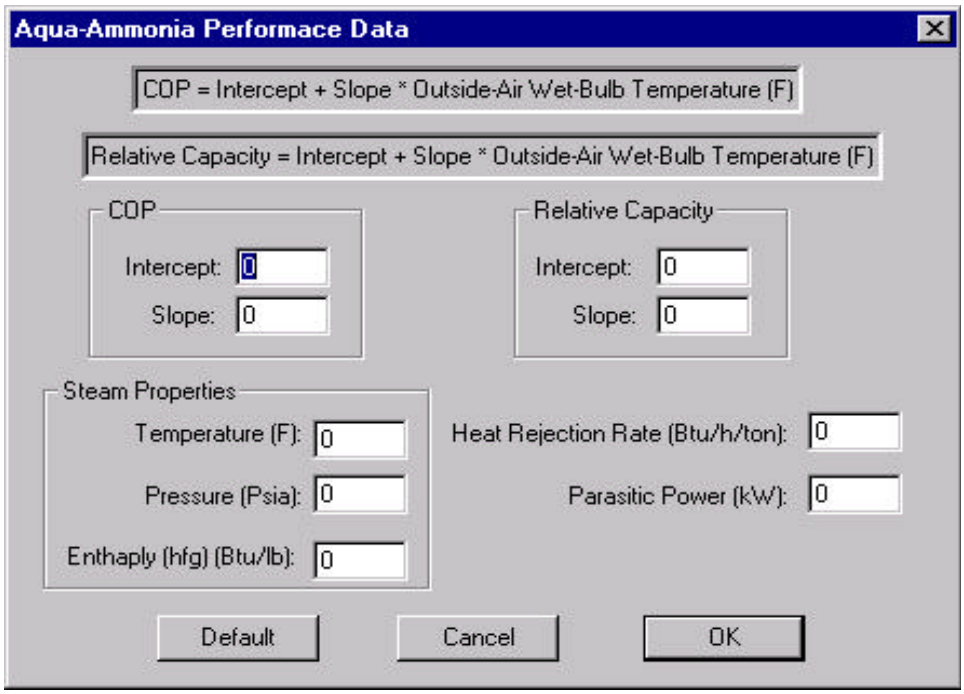

Figure 5.13 Aqua-Ammonia Chiller Performance Data Input Dialog

Like the COP, the cooling capacity of a chiller is a function of the condenser temperature, and is expressed here as a function of the wet-bulb temperature. The chiller cost equations presented later are based on chiller capacity at "rated" temperature conditions. Capacity is typically less at temperature conditions corresponding to the peak-cooling load. The "relative" capacity equation is applied to adjust the peak chiller capacity to the rated chiller capacity.

All energy absorbed by a chiller at the evaporator and input to the chiller in the form of steam or electricity must ultimately be rejected at a cooling tower. The heat rejection rate should be specified per ton of cooling. The parasitic power input generally applies to the thermally-activated chillers only.

5.13. Complex-Compound Chiller Performance Data Input Dialog

The "Complex-Compound Chiller Performance Data" input dialog (Figure 5.14) is accessed by selecting a complex-compound chiller for cooling and clicking the "Chiller Configuration" button from the "Refrigerative Cooling Configuration" dialog (Figure 5.12).

The input requirements are similar to that of the aqua-ammonia chiller, with a second entry for heat rejection. The "storage" version of the complex-compound chiller is actually an integrated chiller-storage device that is sized based on ton-hours rather than tons. The first heat rejection rate is used to size heat rejection equipment, while the second heat rejection rate (the one associated with the storage option) is used to calculate the annual cooling load. Both the COP and the relative capacity are assumed to vary linearly as a function of the wet-bulb temperature.

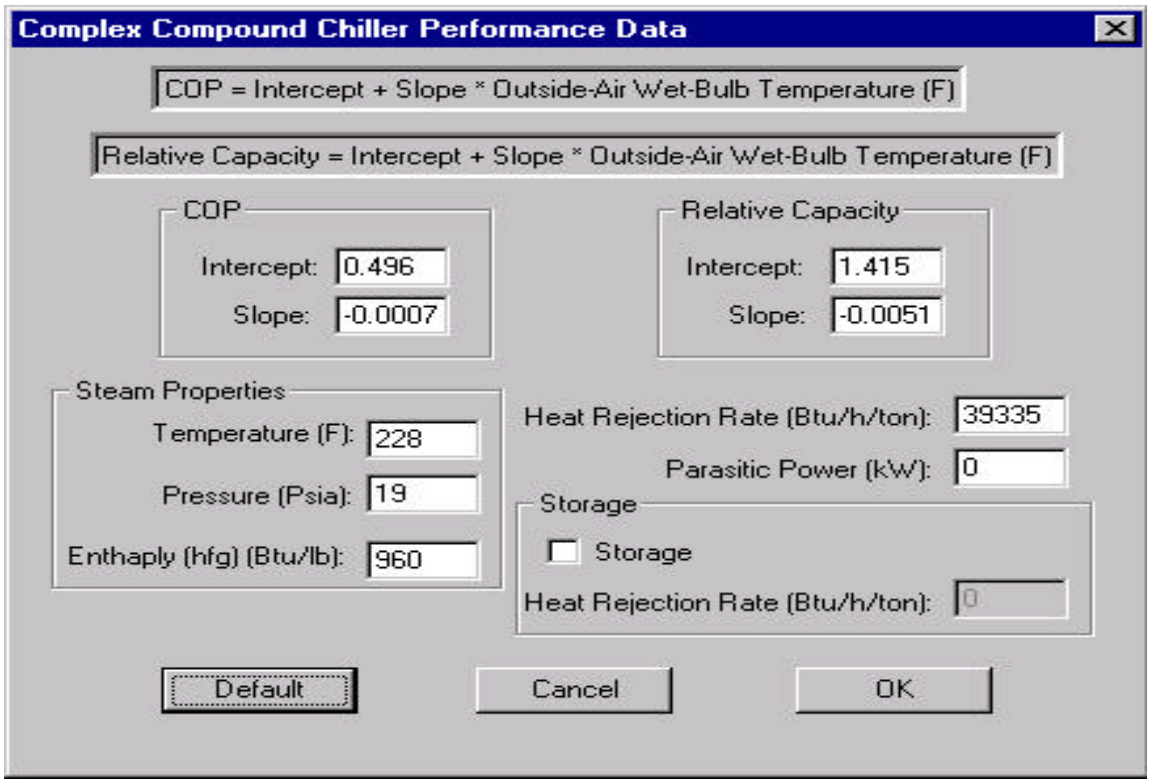

Figure 5.14 Complex-Compound Chiller Performance Data Input Dialog

5.14. Lithium Bromide Chiller Performance Data Input Dialog

The "Lithium Bromide Chiller Performance Data" input dialog (Figure 5.15) is accessed by selecting a single- or double-stage lithium bromide chiller for cooling and clicking the "Chiller Configuration" button from the "Refrigerative Cooling Configuration" dialog (Figure 5.12). The model will not allow the selection of lithium bromide chillers, if the inlet air is cooled below 52 °F.

The input requirements are similar to those for the other thermally-activated chillers, with an additional specification for limiting the wet-bulb temperature because the cooling water for lithium bromide chillers must generally be no lower than 75 °F. The relative capacity is assumed to vary linearly as a function of the difference between the wet-bulb temperature and the limiting wet-bulb temperature. The following equation formats are used to compute the COP:

COP = Intercept + Slope * (Wet-Bulb – Limiting Wet-Bulb)^Exponent

where "Exponent" is 1.5 for single-stage lithium bromide chillers and 1.2 for double-stage lithium bromide chillers.

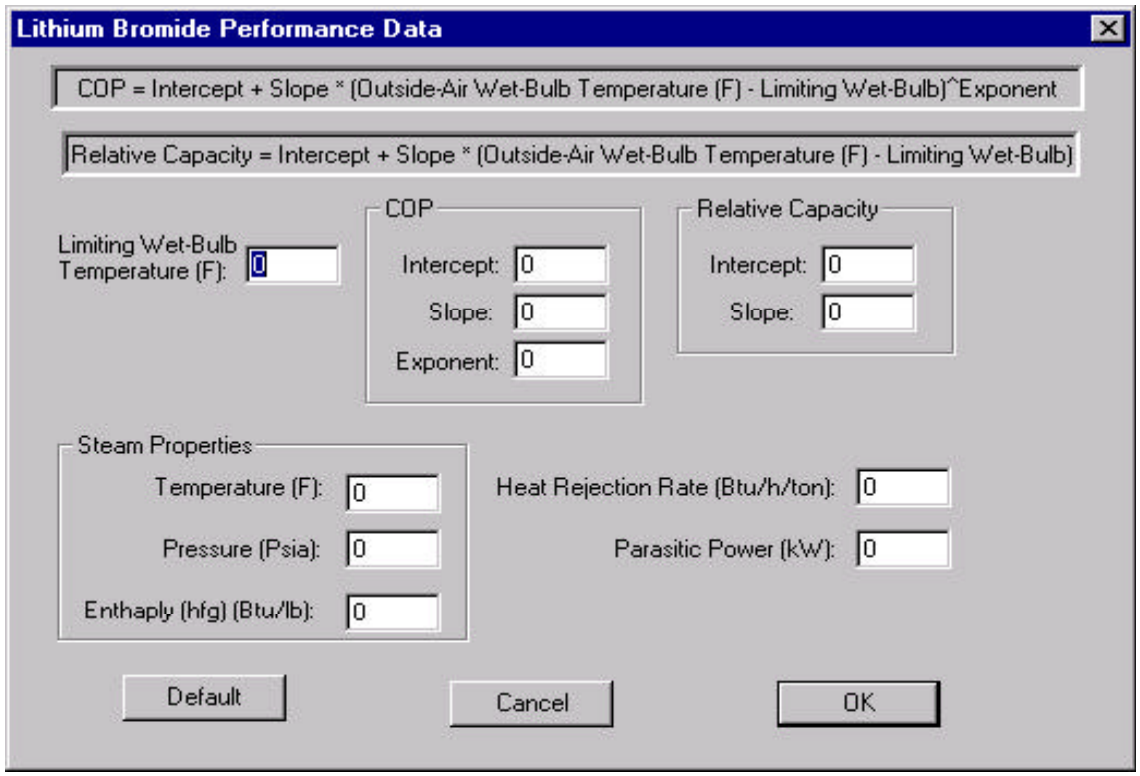

Figure 5.15 Lithium Bromide Performance Data Input Dialog

5.15. Vapor Compression Chiller Performance Data Input Dialog

The "Vapor Compression Performance Data" input dialog (Figure 5.16) is accessed by selecting a vapor compression chiller for cooling and clicking the "Chiller Configuration" button from the "Refrigerative Cooling Configuration" dialog (Figure 5.12). The relative capacity is assumed to vary linearly as a function of the wet-bulb temperature, while the COP is assumed to vary exponentially as a function of wet-bulb temperature.

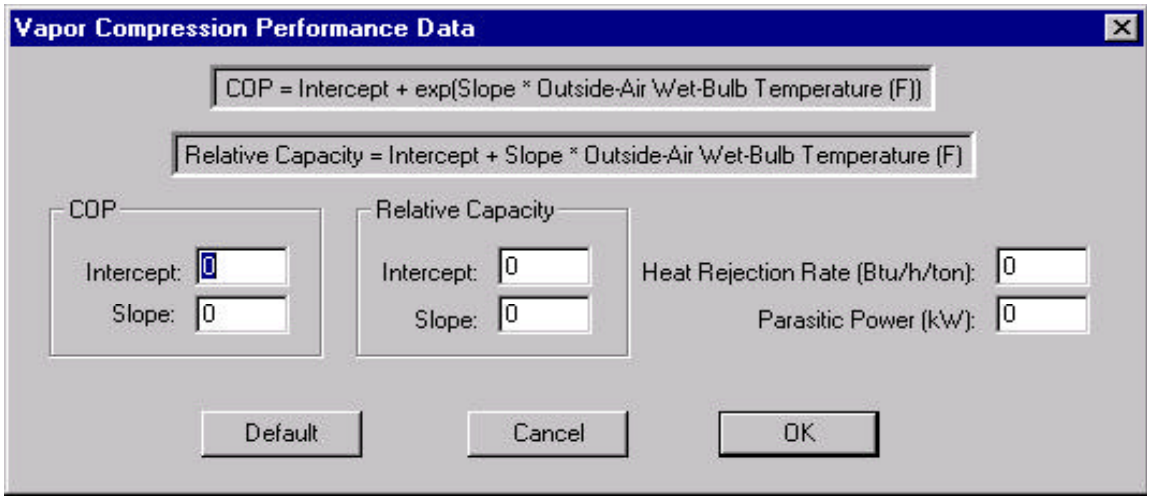

Figure 5.16 Vapor Compression Chiller Performance Data Input Dialog

5.16. Refrigerative Cooling Sizing Inputs Dialog

The "Refrigerative Cooling Sizing Inputs" dialog (Figure 5.17) is accessed from the "Inlet Air Cooling" menu item. This option is only available if the refrigerative cooling option is selected in the "Inlet Air Cooling Configuration" input dialog (Figure 5.9). The inputs related to second stage cooling can only be entered if the inlet air is cooled to 0 °F (Figure 5.9).

The CTCOOL analysis involves sizing various fluid circulation loops and systems depending on the type of chiller and storage selected. Loops that are sized include the ammonia loop connecting the chiller evaporator and other chiller components (except for lithium bromide chillers), the chilled water loop connecting lithium bromide chillers and storage with the inlet air cooling coils, the cooling water loop connecting the chiller with the cooling tower, the steam loop connecting the steam source with absorption-type chillers, and storage water loops connecting relatively warm storage water with evaporators in the ice generator or water chiller. The reader is referred to Brown, Katipamula, and Konynenbelt (1996) for further discussion of these inputs.

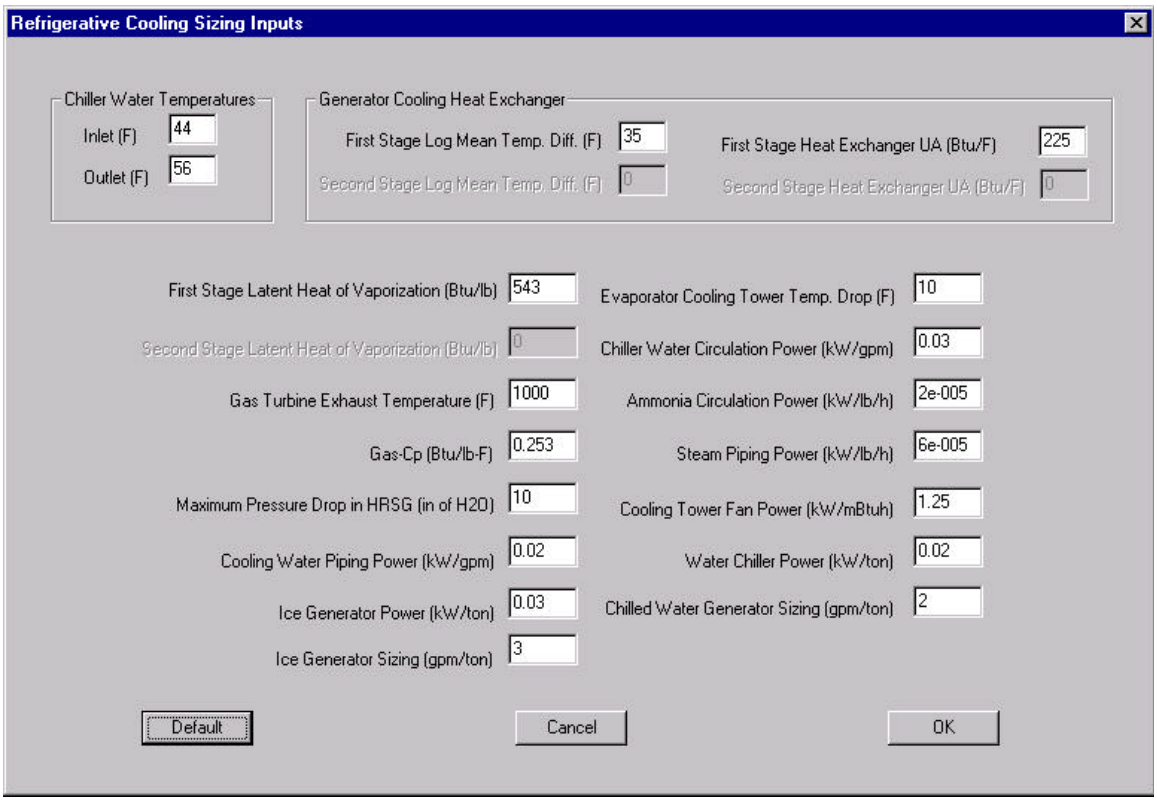

Figure 5.17 Refrigerative Cooling Sizing Inputs Dialog

5.17. Thermal Storage Configuration Input Dialog

The "Thermal Storage Configuration" input dialog (Figure 5.18) is accessed from the "Inlet Air Cooling" menu item. This option is only available if the refrigerative cooling option is selected in the "Inlet Air Cooling Configuration" input dialog (Figure 5.9). CTCOOL allows for daily or weekly storage with load-shifting or load-leveling options. In addition to selecting the type of storage and type of strategy, the storage and defrost efficiency have to be entered. The latter factor accounts for defrosting energy added to the storage tank that must be removed in subsequent ice-making cycles. The second stage efficiency inputs can only be entered if the inlet air is cooled to 0 °F (Figure 5.9).

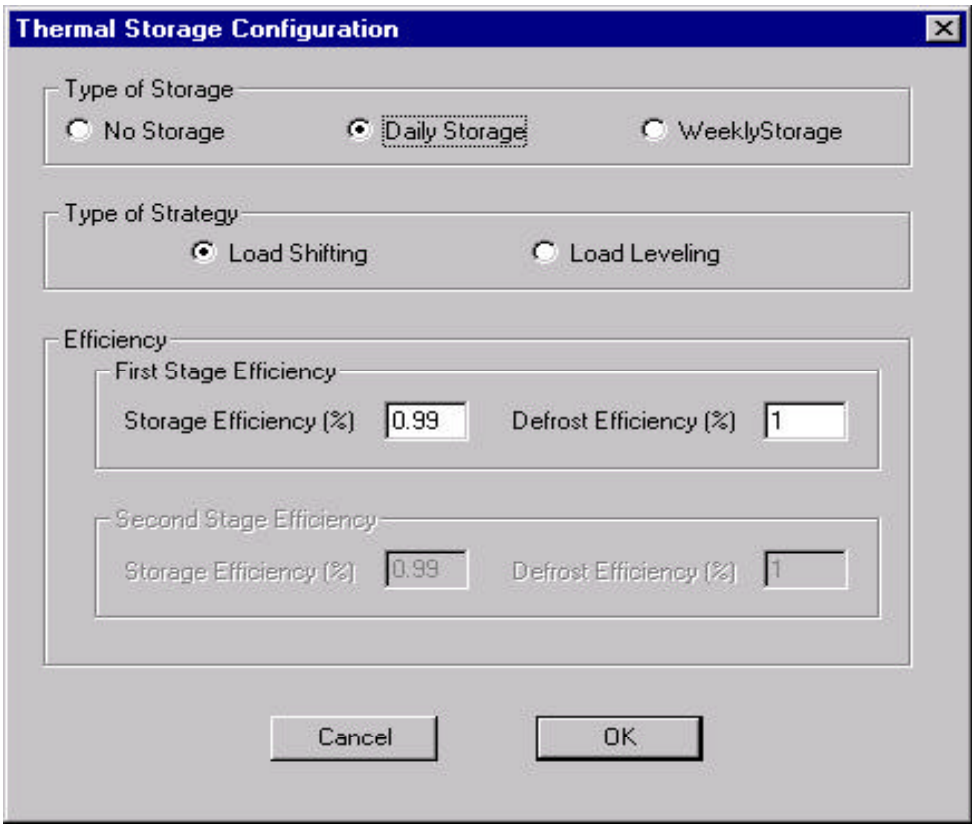

Figure 5.18 Thermal Storage Configuration Dialog

5.18. Evaporative Cooling Cost Inputs Dialog

The "Evaporative Cooling Cost Inputs" dialog (Figure 5.19) is accessed from the "Cost Input" menu item. This option is only available if the evaporative cooling option is selected in the "Inlet Air Cooling Configuration" input dialog (Figure 5.9). For evaporative cooling, only three inputs are required. The cost of the evaporative coil is estimated as a function of the ISO power. The operation and maintenance cost, "O & M Cost", is estimated as a percent of the total evaporator cost. Finally, there is the unit cost ($/gal) associated with the water consumed in the process.

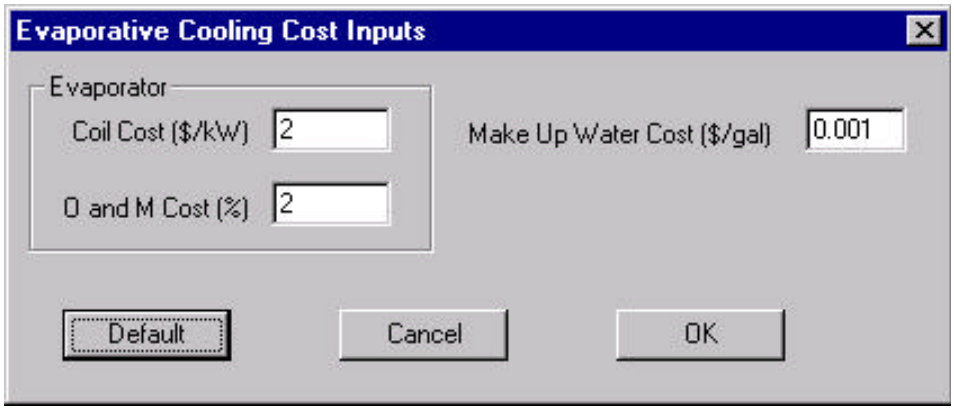

Figure 5.19 Evaporative Cooling Cost Inputs

5.19. Refrigerative Cooling Cost Input Dialog

The "Refrigerative Cooling Costs" input dialog (Figure 5.20) is accessed from the "Cost Input" menu item. This option is only available if the refrigerative cooling option is selected in the "Inlet Air Cooling Configuration" input dialog (Figure 5.9).

All default costs are in terms of mid-1994 dollars and are intended to include all costs associated with equipment and material purchase and installation. All components except the complex compound chiller are mature, commercially available products. Complex compound chiller costs were estimated based on a mature production scenario, i.e., the first unit would probably be more expensive.

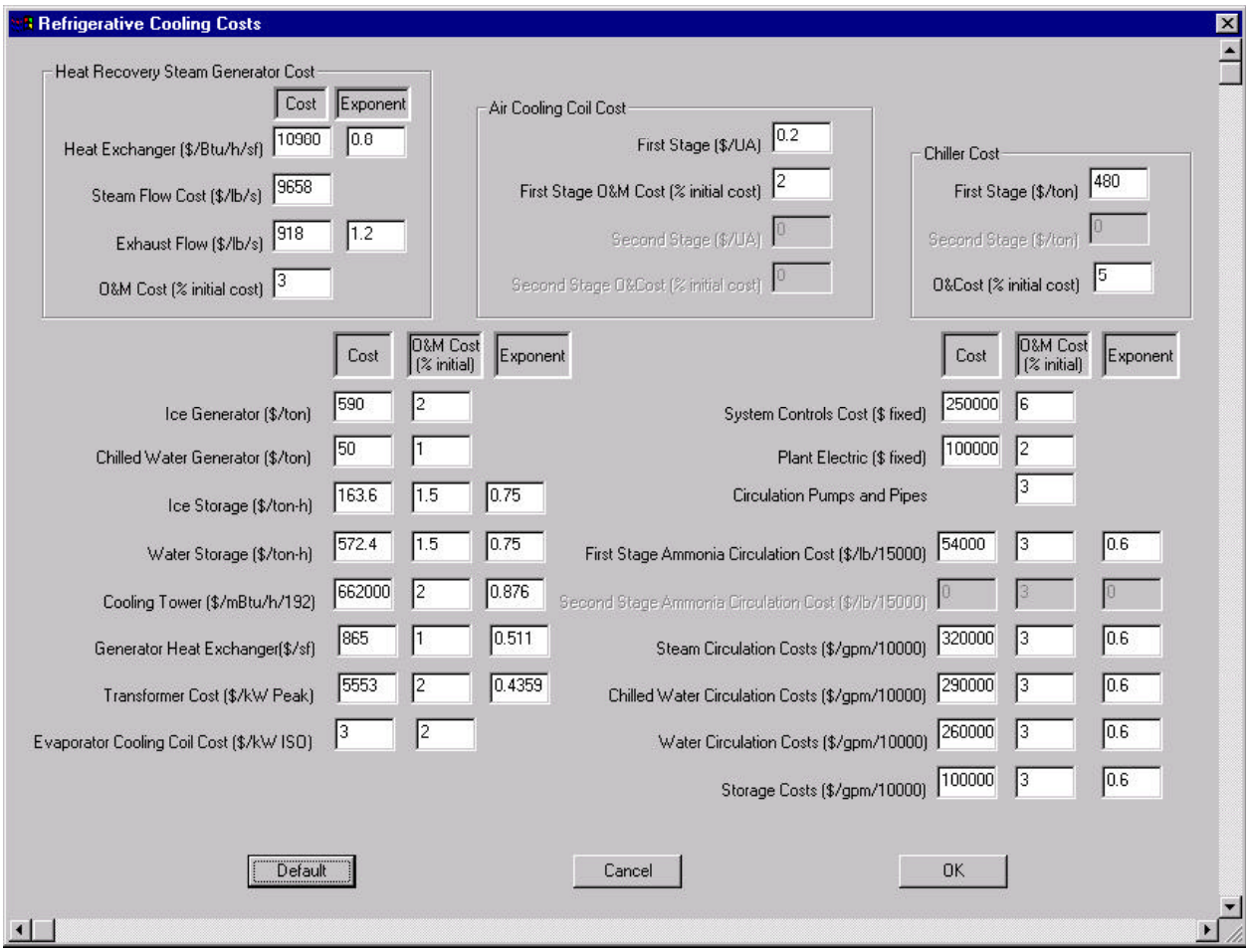

Figure 5.20 Refrigerative Cooling Cost Input Dialog

5.20. NPV Input Dialog

The "NPV Inputs" dialog (Figure 5.21) is accessed from the "Cost Input" menu item. This option is only available if the evaporative or refrigerative cooling option is selected in the "Inlet Air Cooling Configuration" input dialog (Figure 5.9). The inputs shown in Figure 5.21 are needed to perform an economic analysis for a given set of inlet air cooling system options.

CTCOOL performs an economic analysis by comparing the benefit of incremental power and energy production with the cost of owning and operating the inlet air cooling system. With costs and benefits denominated in dollars, economic feasibility is determined by calculating the net present value (NPV) of the investment. The preferred cooling systems are those with the highest NPVs for a given set of application conditions. Refrigerative and evaporative cooling systems are compared against reference plants that are uncooled. Refrigerative cooling systems may also be compared against reference plants that are already evaporatively cooled. In either case, the annual kWh output and fuel input for the reference plant must be entered in this input dialog along with the values of incremental kW and kWh production and the cost of fuel. The model calculates the incremental on-peak generating capacity, so the reference on-peak generating capacity does not need to be entered.

The model assumes that all incremental kWh production occurs during the on-peak period; i.e., the inlet air is only cooled during the on-peak period. Therefore, if the model is set up to cool inlet air around the clock, the valuation of incremental kWh production and NPV calculations will be incorrect. The value of off-peak kWh must be specified for cooling systems with storage, where part of the parasitic power consumption occurs during the off-peak period.

The multipliers are the present value factors for incremental electricity production, O&M costs, fuel costs, and initial construction costs. For example, incremental electricity production valued at $1 per year has a present value of $10.39 based on the multiplier (present value factor) shown in Figure 5.21. The multipliers must be determined externally to CTCOOL.

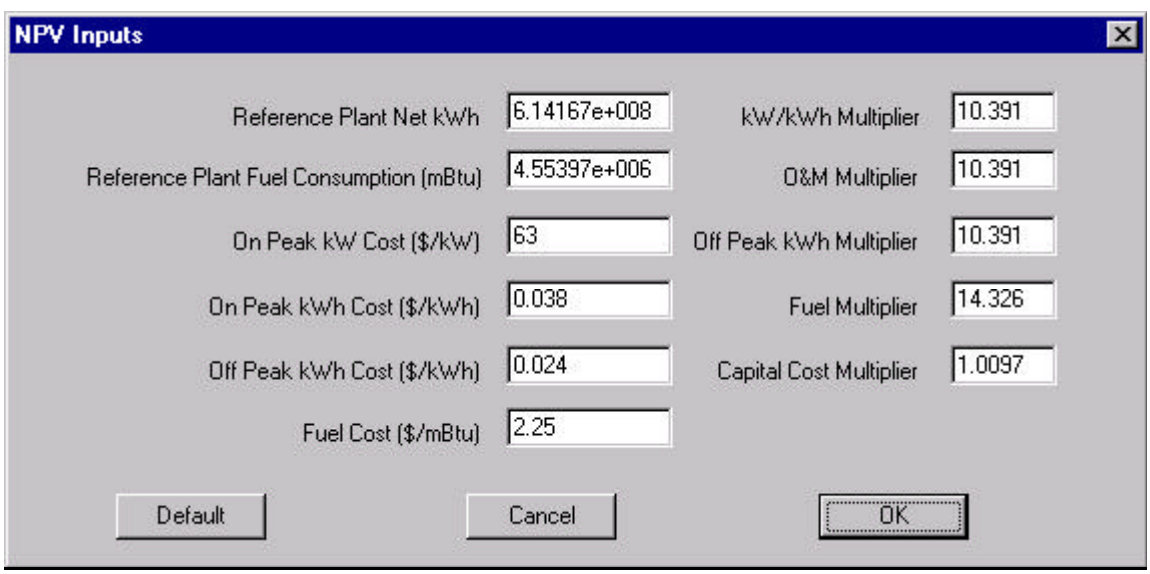

Figure 5.21 NPV Inputs Dialog

5.21. Input Summary Dialog

The "Inputs" dialog (Figure 5.22) is accessed from the "View" menu. This dialog prepares a summary of the file inputs. The tabulated inputs may also be printed to a file by clicking on the "Output to File" button.

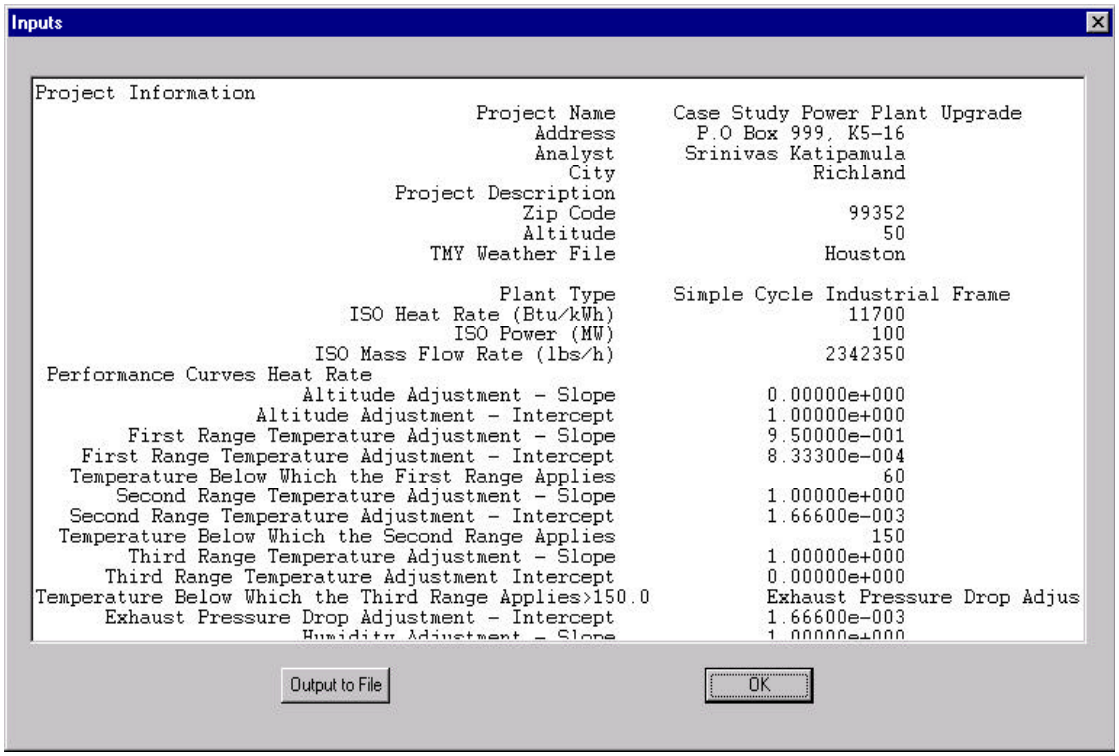

Figure 5.22 Echo of Inputs Dialog

5.22. Gas Turbine Performance Analysis Output Dialog

The "GT Analysis Results" dialog (Figure 5.23) is accessed from the "Analysis" menu. After all the inputs are selected/entered, the annual performance of the gas turbine may be simulated. First, the weather data must be read by clicking the button "Get Weather Data" in Figure 5.23. After the weather data are processed, the button "Run GT Analysis" may be clicked to simulate the gas turbine performance. This button will be disabled until the weather data file is processed. After the analysis is completed, the results are tabulated in the output box, as shown in Figure 5.23. The tabulated results may also be printed to a file by clicking on the "Output to File" button.

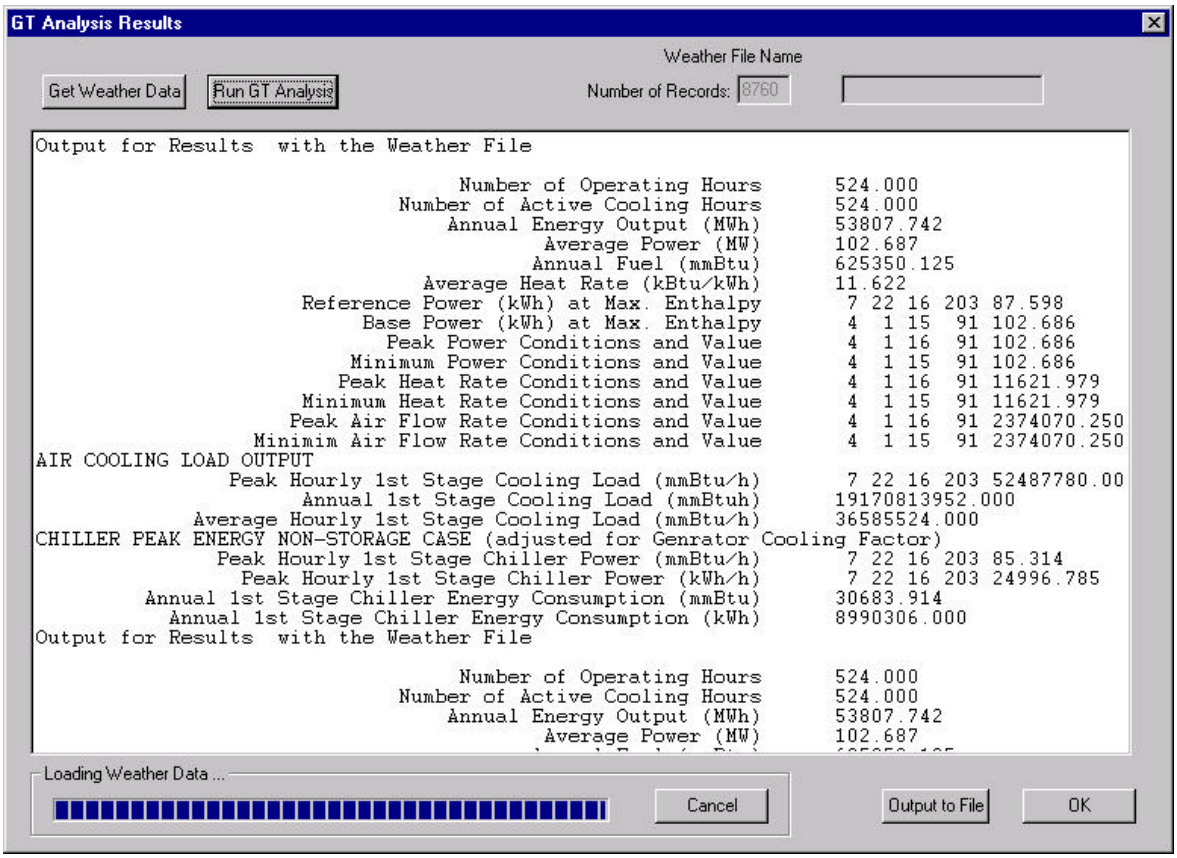

Figure 5.23 Gas Turbine Performance Analysis Output Dialog

5.23. Cooling Equipment Sizing Output Dialog

The "Cooling Equipment Sizing" results dialog (Figure 5.24) may be accessed from the "Analysis" menu only after the gas turbine annual performance is simulated. The tabulated results may also be printed to a file by clicking on the "Output to File" button.

5.24. Cooling Equipment Costs Output Dialog

The "Cooling Equipment Costs" output dialog (Figure 5.25) may be accessed from the "Analysis" menu only after simulating the gas turbine annual performance and sizing the cooling equipment. The tabulated results may also be printed to a file by clicking on the "Output to File" button.

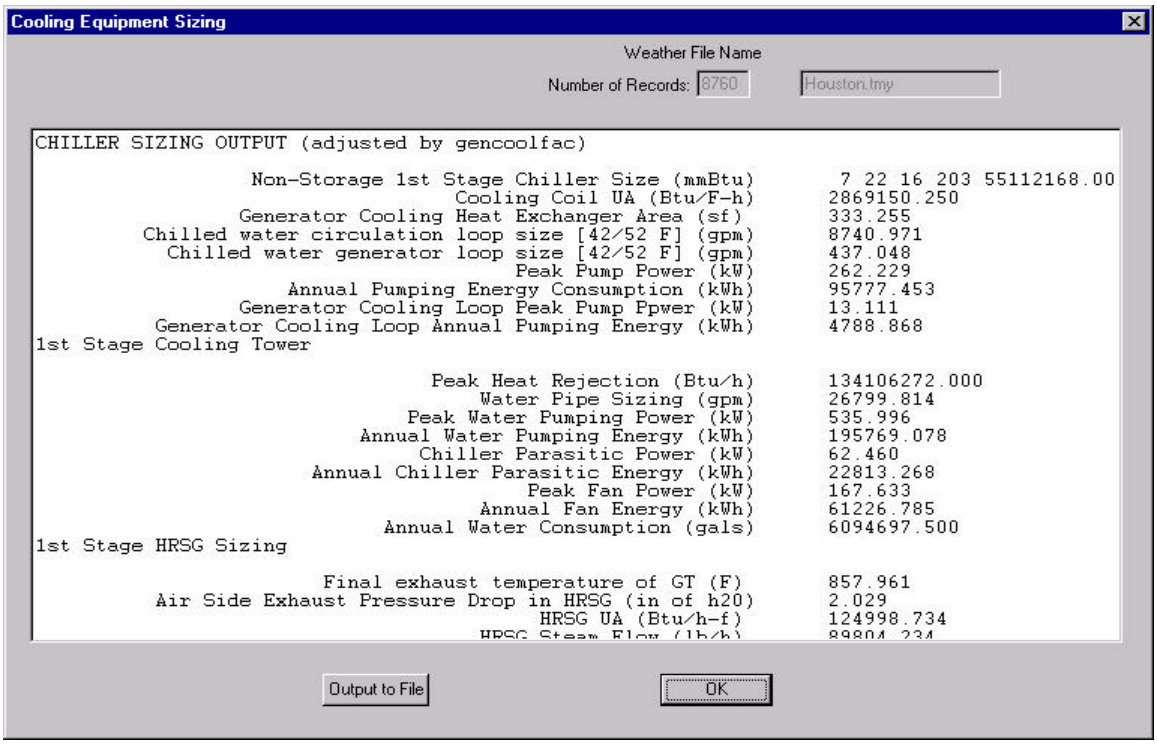

Figure 5.24 Cooling Equipment Sizing Calculations

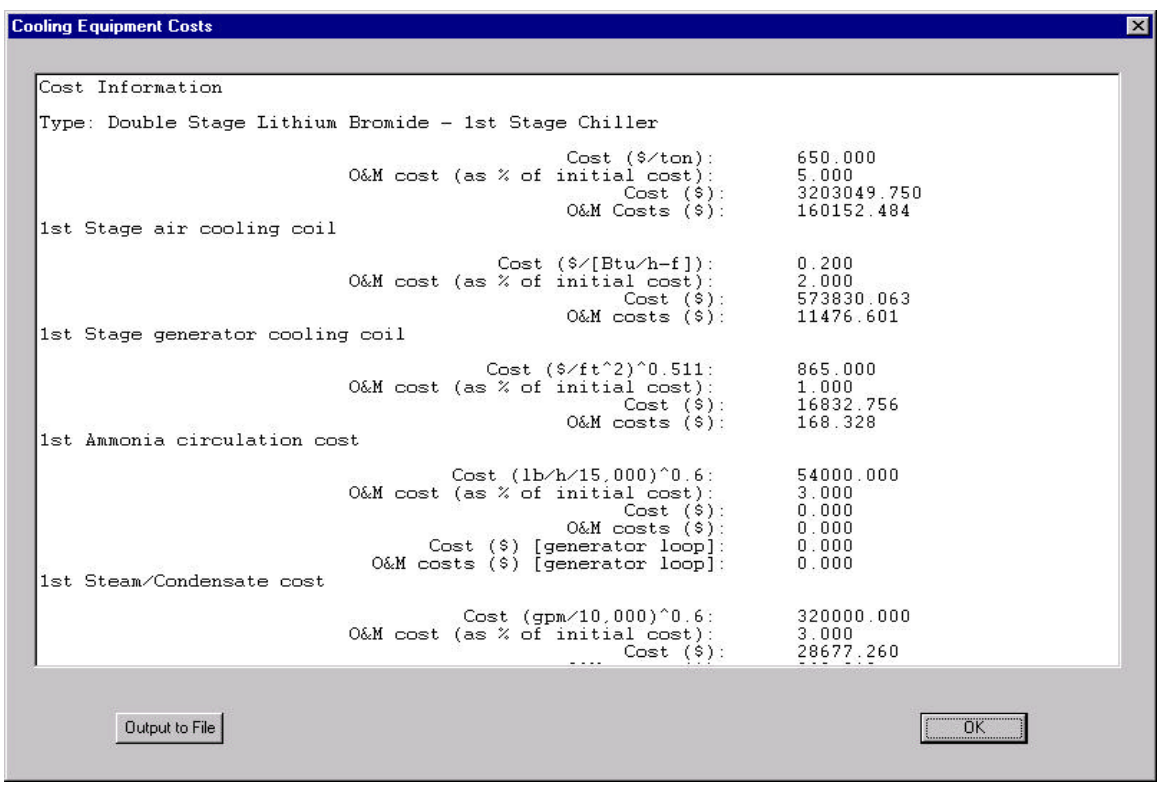

Figure 5.25 Cooling Equipment Cost Estimates

5.25. NPV Calculations Results Dialog

The "NPV Calculations" results dialog (Figure 5.26) may be accessed from the "Analysis" menu only after simulating the gas turbine annual performance, sizing the cooling equipment, and estimating the cooling equipment costs. The tabulated results may also be printed to a file by clicking on the "Output to File" button.

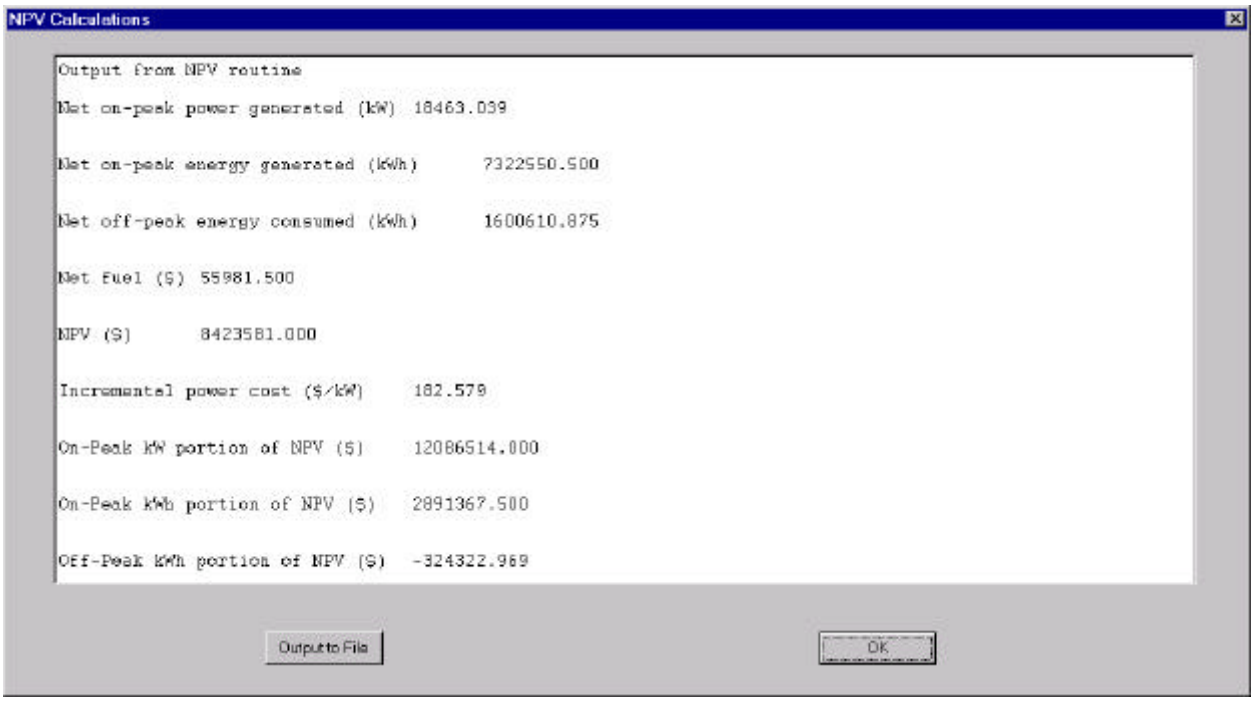

Figure 5.26 NPV Calculations Results

6. Sample Case

A sample case, including both the inputs and outputs from the CTCOOL software, is presented in this section.

The various inputs are shown in Figures 6.1 through 6.12. The gas turbine is assumed to operate on weekdays between 2 p.m. and 6 p.m. (4 hours) for the months of April through September. The inlet air cooling schedule is identical to that of the gas turbine operating schedule.

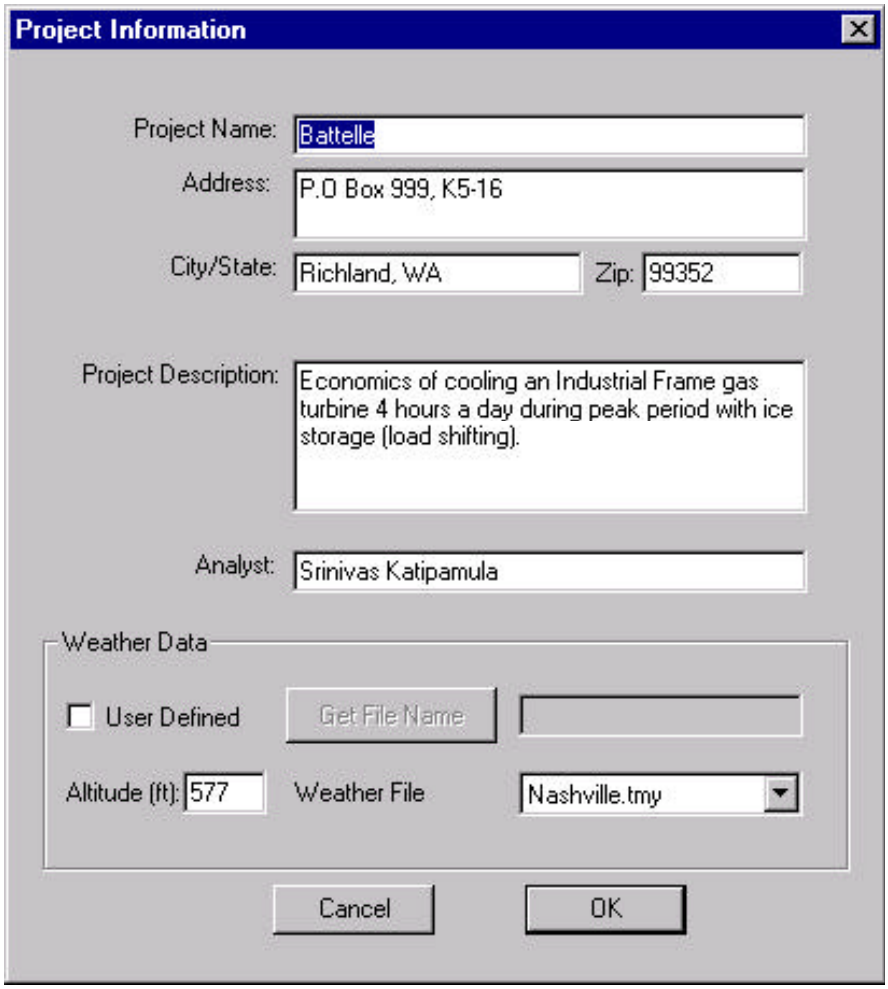

Figure 6.1 Project Information

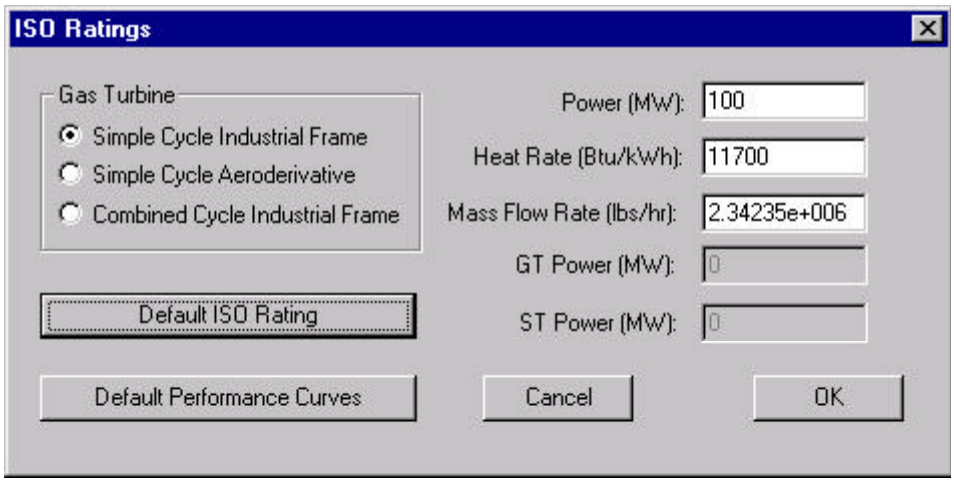

Figure 6.2 ISO Rating for the Industrial Frame Gas Turbine

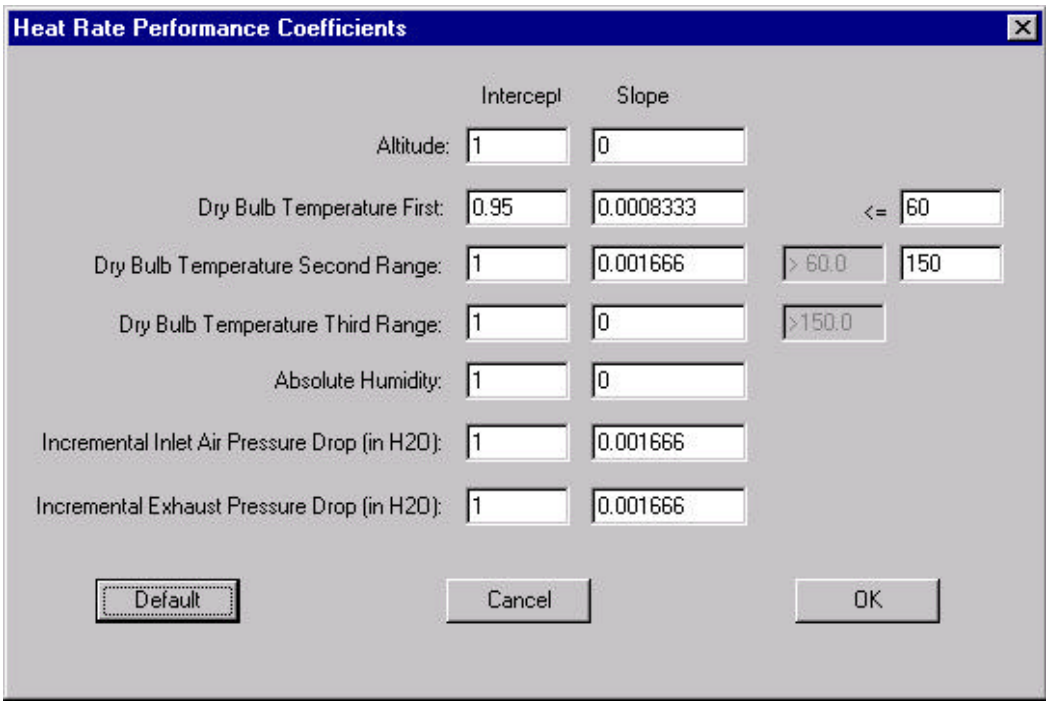

Figure 6.3 Heat Rate Performance Coefficients

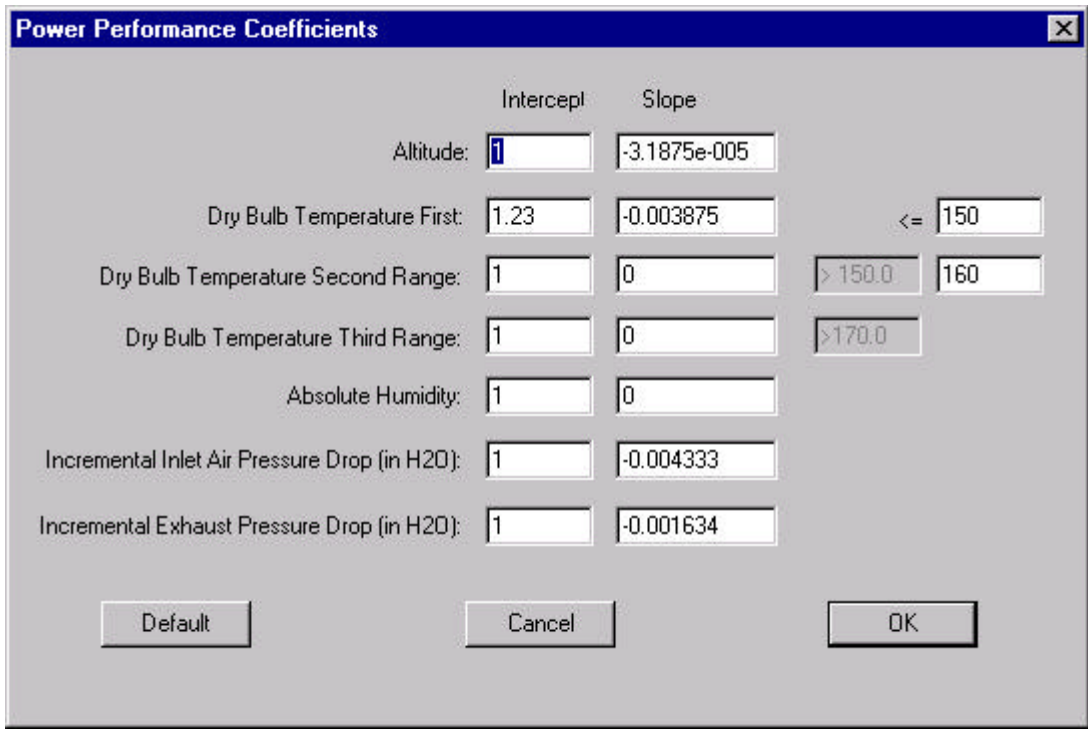

Figure 6.4 Power Performance Coefficients

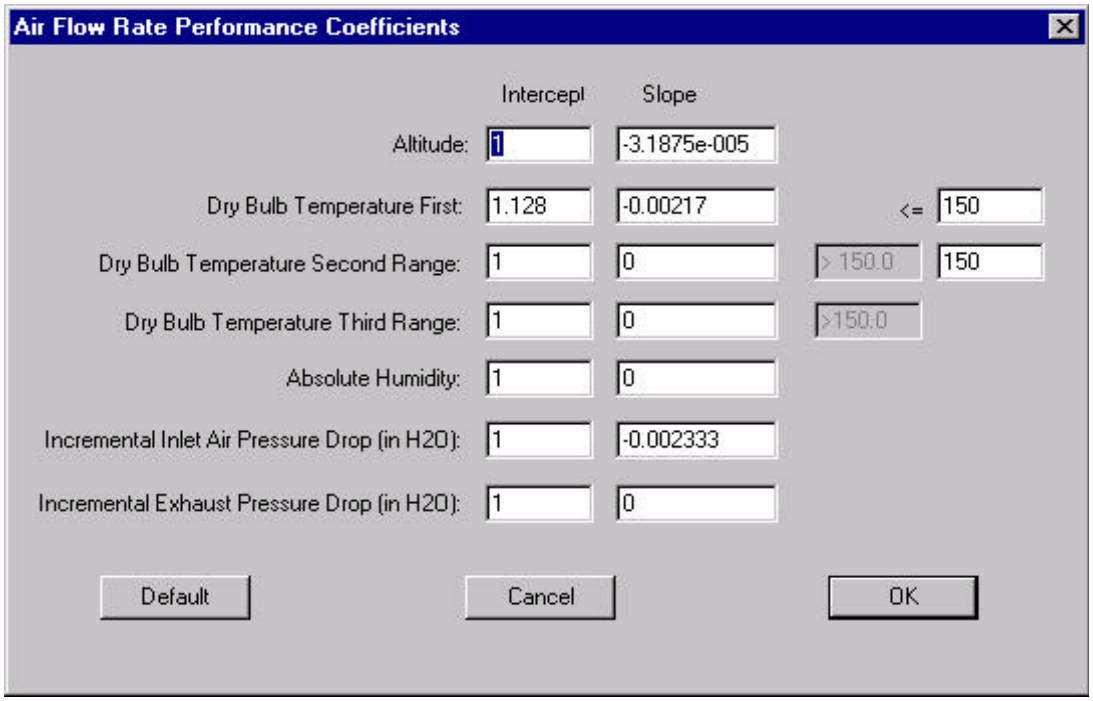

Figure 6.5 Air Flow Rate Performance Coefficients

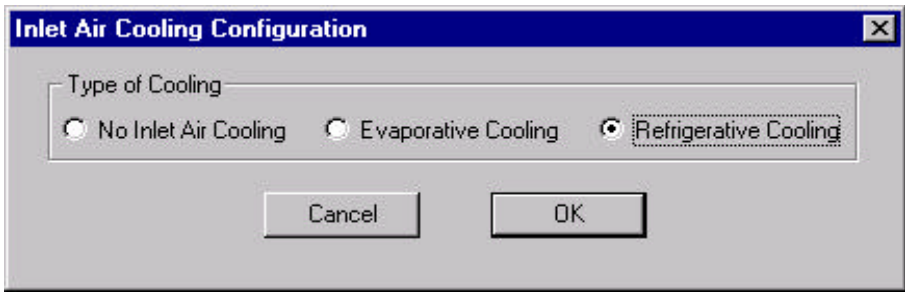

Figure 6.6 Inlet Air Cooling Configuration

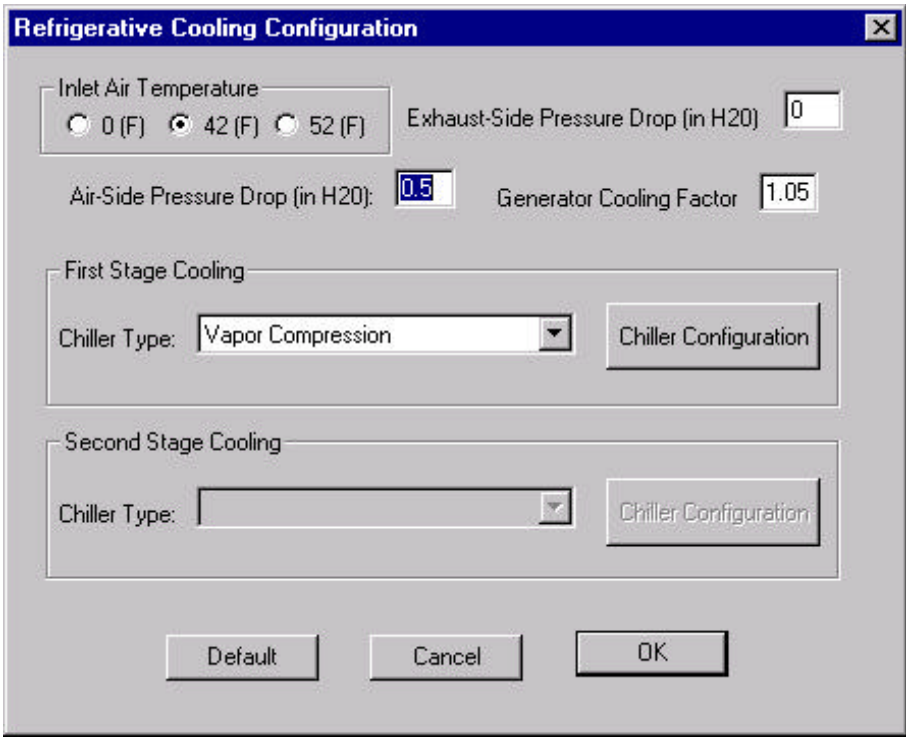

Figure 6.7 Refrigerative Cooling Configuration

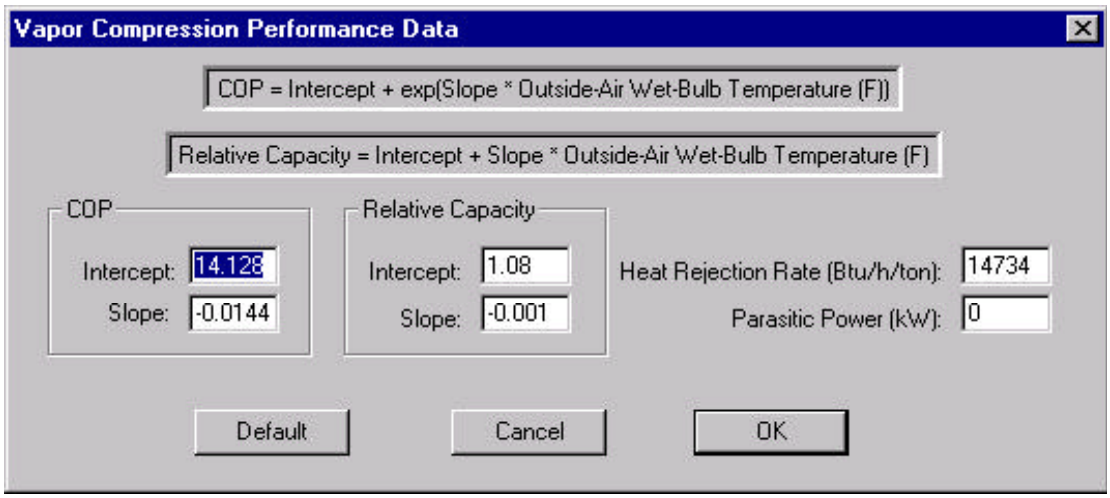

Figure 6.8 Vapor Compression Performance Data

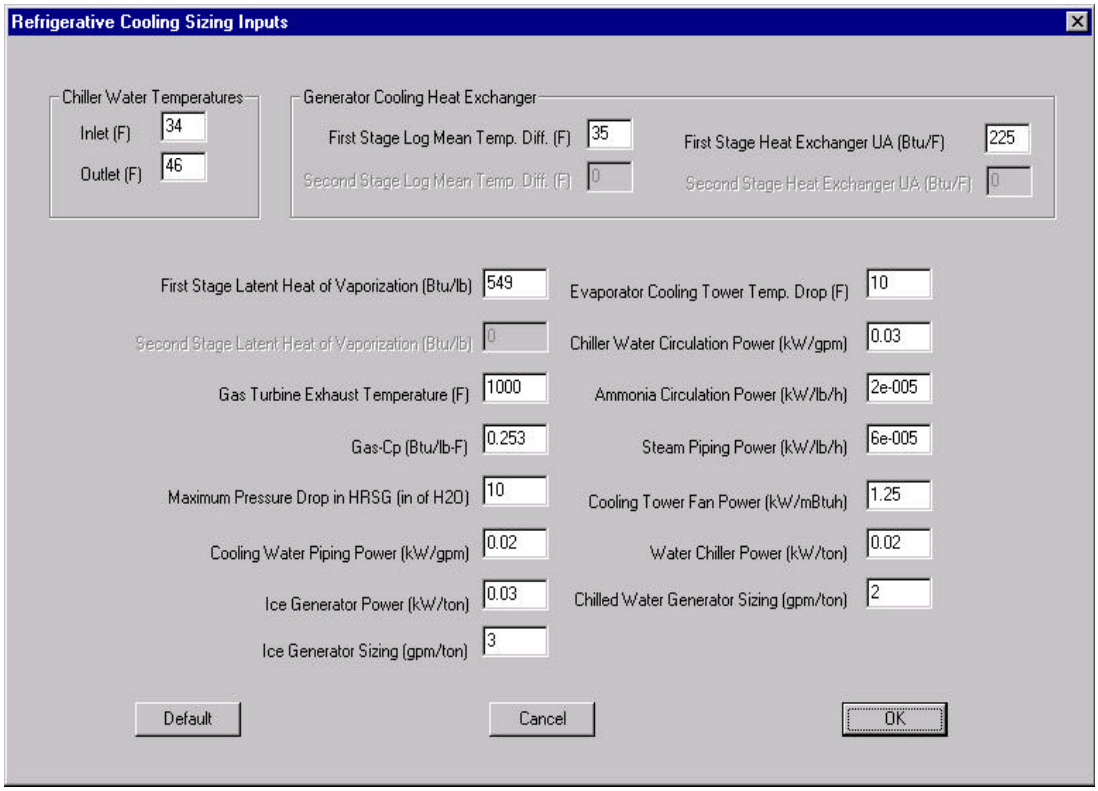

Figure 6.9 Refrigerative Cooling Sizing Inputs

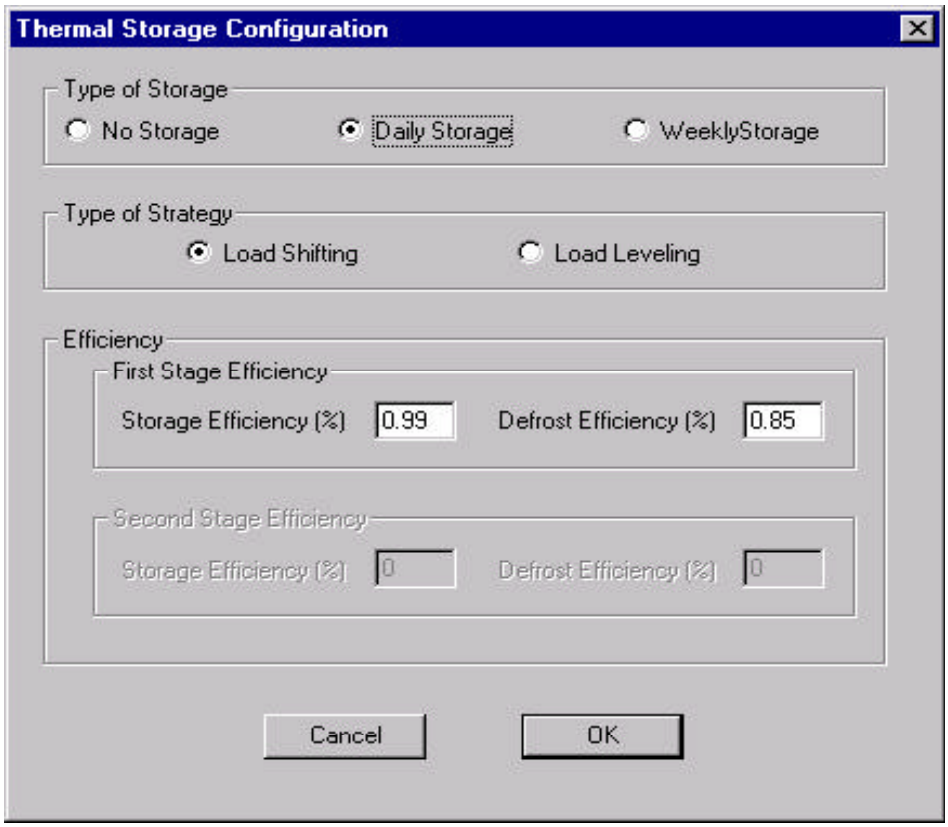

Figure 6.10 Thermal Storage Configuration

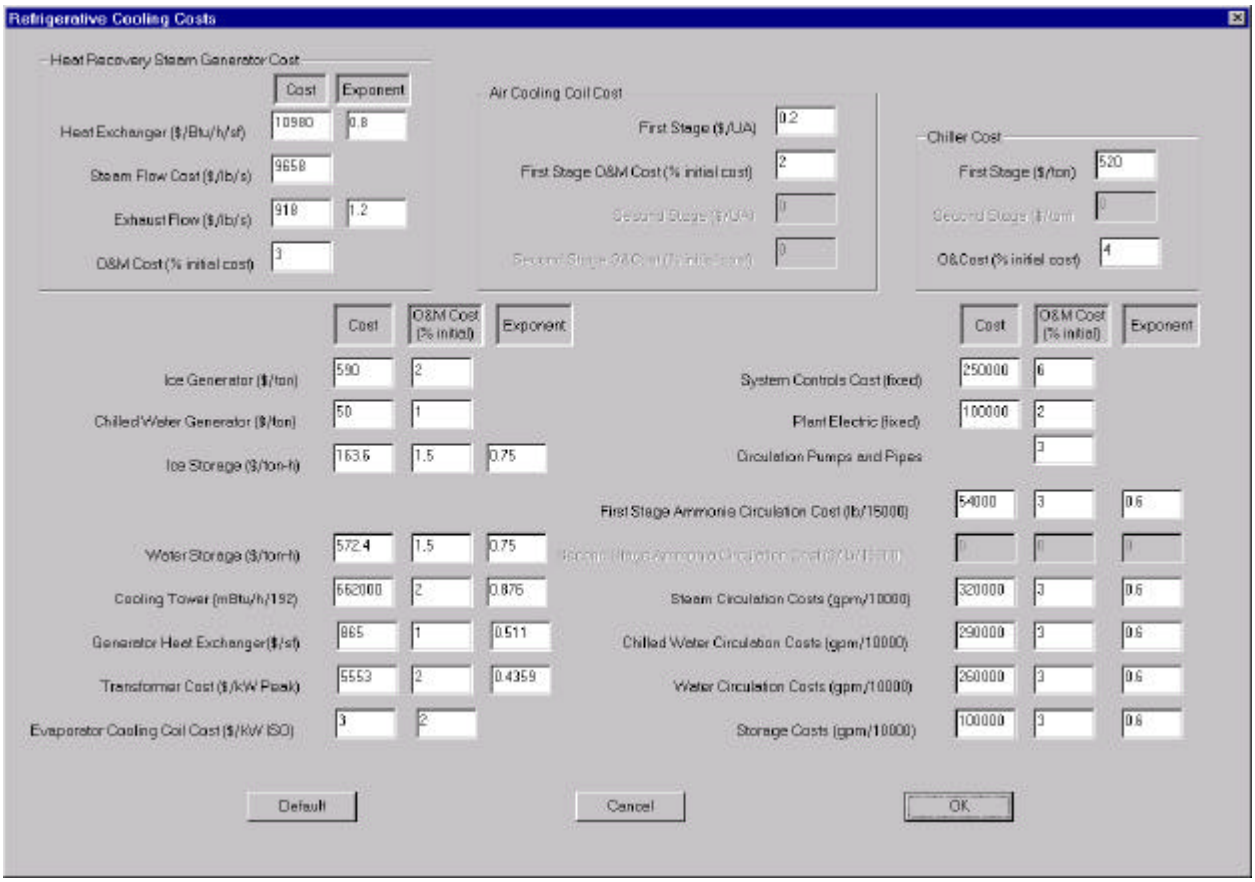

Figure 6.11 Refrigerative Cooling Costs

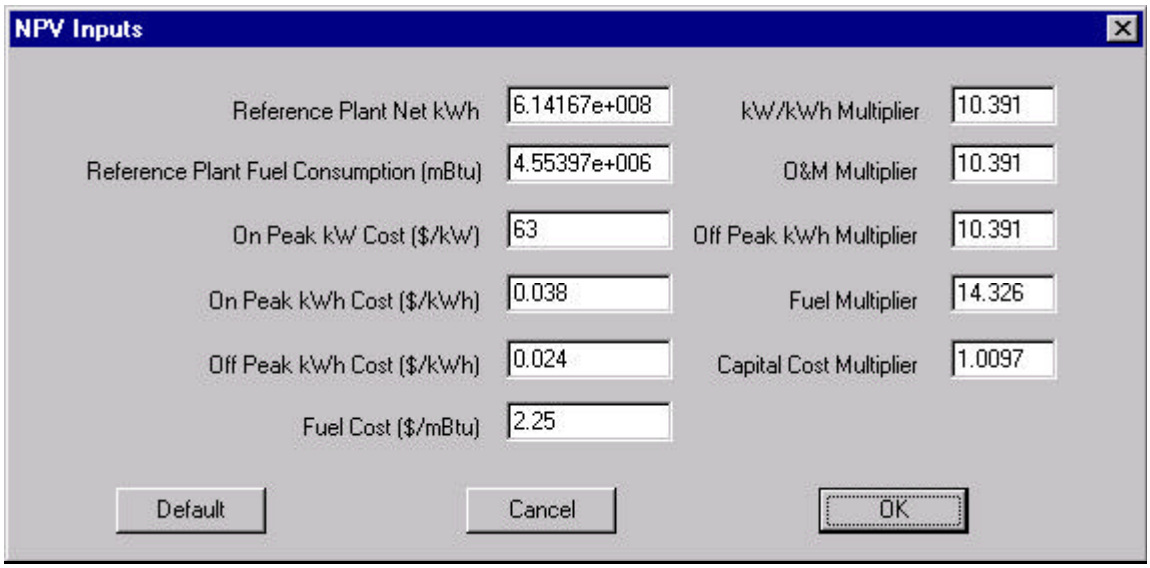

Figure 6.12 NPV Inputs

6.1. Annual, Peak and Minimum Values for Selected Performance Parameters

Annual gas turbine performance, along with the minimum and peak values for various performance parameters, is listed below. Also shown below are the annual and peak-cooling loads. For the peak and the minimum values, the first four figures represent the month, day of the month, hour of the day, and day of the year, respectively, while the last figure represents the actual value of the variable.

```
GAS TURBINE PERFORMANCE
Number of Operating Hours                          524
Number of Active Cooling Hours                     524
Annual Energy Output (MWh)                         54777.2
Average Power (MW)                                 104.537
Annual Fuel (mmBtu)                                631800.5
Average Heat Rate (kBtu/kWh)                       11.534
Reference Power (MW) at Max. Enthalpy              8 20 15 232 85.7
Base Power (MW) at Max. Enthalpy                   4  1 15  91 104.5
Peak Power Conditions and Value (MW)               4  8 18  98 105.3
Minimum Power Conditions and Value (MW)            4  1 15  91 104.5
Peak Heat Rate Conditions and Value (kBtu/kWh)     4  1 16  91 11.534
Minimum Heat Rate Conditions and Value (kBtu/kWh)  4  8 18  98 11.515
Peak Air Flow Rate Conditions and Value (lb/h)     4  8 18  98 2391207
Minimum Air Flow Rate Conditions and Value (lb/h) 4  1 15  91 2381240

AIR COOLING LOAD OUTPUT
Peak Hourly 1st Stage Cooling Load (mmBtu/h)       8 20 15 232 62.650
Annual 1st Stage Cooling Load (mmBtu)              20155.80
Average Hourly 1st Stage Cooling Load (mmBtu/h)    38.465
Peak Daily 1st Stage Cooling Load (mmBtu/day)      231.407
```

CHILLER PEAK ENERGY NON-STORAGE CASE (adjusted for Generator Cooling Factor)

Peak Hourly 1st Stage Chiller Power (mmBtu/h)	8 20 15 232 14.5
Peak Hourly 1st Stage Chiller Power (kWh/h)	8 20 15 232 4255
Annual 1st Stage Chiller Energy Consumption (mmBtu)	4112
Annual 1st Stage Chiller Energy Consumption (kWh)	1204913

6.2. Cooling Equipment Sizing Output

The sizing results for various components of the cooling plant are shown below.

Daily Load Shifting 1st Stage Chiller Size (mmBtu/h)	14.437
1st Stage Daily Load Shifting Storage Size (mmBtu)	245.432
Cooling Coil UA (Btu/F-h)	2883862.5
Generator Cooling Heat Exchanger Area (sf)	397.8
Chilled water circulation loop size [42/52 F] (gpm)	10433
Chilled water generator loop size [42/52 F] (gpm)	522
Peak Pump Power (kW)	313
Annual Pumping Energy Consumption (kWh)	100698
Generator Cooling Loop Peak Pump Power (kW)	15.7
Generator Cooling Loop Annual Pumping Energy (kWh)	5035
Ammonia loop (chiller-to-storage)	
Circulation loop size (lb/h)	26297
Peak pumping power (kW)	0.526
Annual pumping energy (kWh)	916
1st Stage ice generator sizing	
Circulation loop size (gpm)	3609
Peak pumping power (kW)	36
Annual pumping energy (kWh)	62874.6
1st Stage Cooling Tower	
Peak Heat Rejection (Btu/h)	17726422
Water Pipe Sizing (gpm)	3542.5
Peak Water Pumping Power (kW)	70.9
Annual Water Pumping Energy (kWh)	123420
Chiller Parasitic Power (kW)	0.000
Annual Chiller Parasitic Energy (kWh)	0.000
Peak Fan Power (kW)	22.2
Annual Fan Energy (kWh)	38599.8
Annual Water Consumption (gals)	3842335
Peak auxiliary power consumption (kW)	328.7
Off-peak auxiliary power consumption (kW)	1001.8
Peak auxiliary annual energy consumption (kWh)	105733
Off-peak auxiliary annual energy consumption (kWh)	1600611

6.3. Cooling Equipment Cost Estimates

The cost estimates for various cooling components are shown below.

```
Vapor Compression - 1st Stage Chiller
Cost ($/ton):                                          520
O&M cost (as % of initial cost):                       4
Cost ($):                                              624985
O&M Costs ($):                                         24999

1st Stage air cooling coil
Cost ($/[Btu/h-f]):                                    0.2
O&M cost (as % of initial cost):                       2.0
Cost ($):                                              576773
O&M costs ($):                                         11535

1st Stage generator cooling coil
Cost ($/ft^2)^0.511:                                   865
O&M cost (as % of initial cost):                       1
Cost ($):                                              18426
O&M costs ($):                                         184

1st Stage cooling tower circulation cost
Cost (gpm/10,000)^0.6:                                 260000
O&M cost (as % of initial cost):                       3
Cost ($):                                              139494
O&M costs ($):                                         4184

1st Stage cooling tower cost
Cost (mmBtu/h-192)^0.876:                              662000
O&M cost (as % of initial cost):                       2
Cost ($):                                              82126
O&M costs ($):                                         1643

1st Stage Makeup Water Cost ($1/1000 gals)
Makeup Water Costs                                     3842

1st Stage Ice Generator/Storage
Generator cost ($/ton):                                590
Generator O&M cost as (% of initial cost):             2
Ice storage cost ($/ton-hrs)^0.75:                     163.6
Ice storage O&M cost (as % of initial cost):           1.5
Generator cost ($):                                    709827
Generator O&M costs ($):                               14197
 Ice storage cost ($):                                 279798
Ice storage O&M costs ($):                             4197

1st Chilled water circulation cost
Cost (gpm/10,000)^0.6:                                 290000
O&M cost (as % of initial cost):                       3
Cost ($):                                              297475
O&M costs ($):                                         8924
Cost ($) [generator loop]:                             49298
O&M costs ($) [generator loop]:                        14794
```

1st stage storage water charging cost (ice loop)	
Cost (gpm/10,000)^0.6:	100000
O&M cost (as % of initial cost):	3
Cost ($):	54257
O&M costs ($):	1628
1st Ammonia circulation cost (chiller-to-storage)	
Cost (lb/h/15,000)^0.6:	54000
O&M cost (as % of initial cost):	3
Cost ($):	75628
O&M costs ($):	2269
System control cost	
Fixed Cost ($):	250000
O&M cost (as % of initial cost):	6
O&M costs ($):	15000
Plant electric cost	
Fixed Cost ($):	100000
O&M cost (as % of initial cost):	2
O&M costs ($):	2000
Peak transformer cost	
Cost ($ kW)^0.4359:	5553
O&M cost (as % of initial cost):	2
Cost ($):	112868
O&M costs ($):	2257
Total capital cost ($)	3370955
Total O&M cost ($)	98339

6.4. Net Present Value Outputs

The net present value (NPV) outputs are shown below.

Net on-peak power generated (kW)	18463
Net on-peak energy generated (kWh)	7322551
Net off-peak energy consumed (kWh)	1600611
Net fuel ($)	55982
NPV ($)	8423581
Incremental power cost ($/kW)	182.6
On-Peak kW portion of NPV	12086514
On-Peak kWh portion of NPV	2891368
Off-peak kWh portion of NPV ($)	-324323
O&M portion of NPV ($)	-1021845
Fuel portion of NPV ($)	-1804480
Capital portion of NPV ($)	-3403653

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7. Reference

Brown, D.R., S. Katipamula, and J.H. Konynenbelt. 1996. A Comparative Assessment of Alternative Combustion Turbine Inlet Air Cooling Systems. PNNL-10966. Pacific Northwest National Laboratory, Richland, Washington.

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Appendix A

Model Structure

Appendix A Model Structure

The two flow-charts show the logical flow of the model structure.

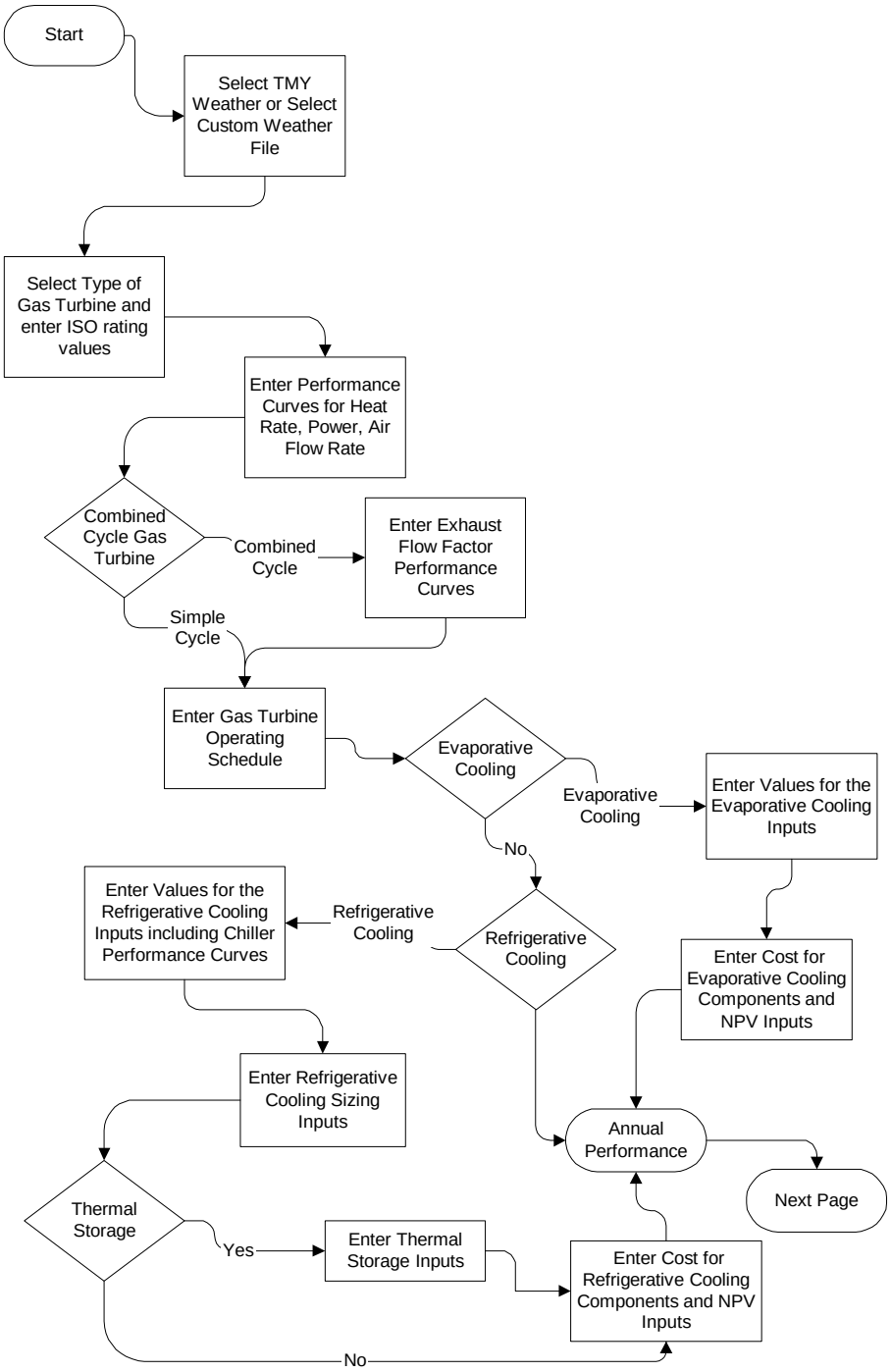

Figure A.1 Schematic of the CTCOOL Model Structure

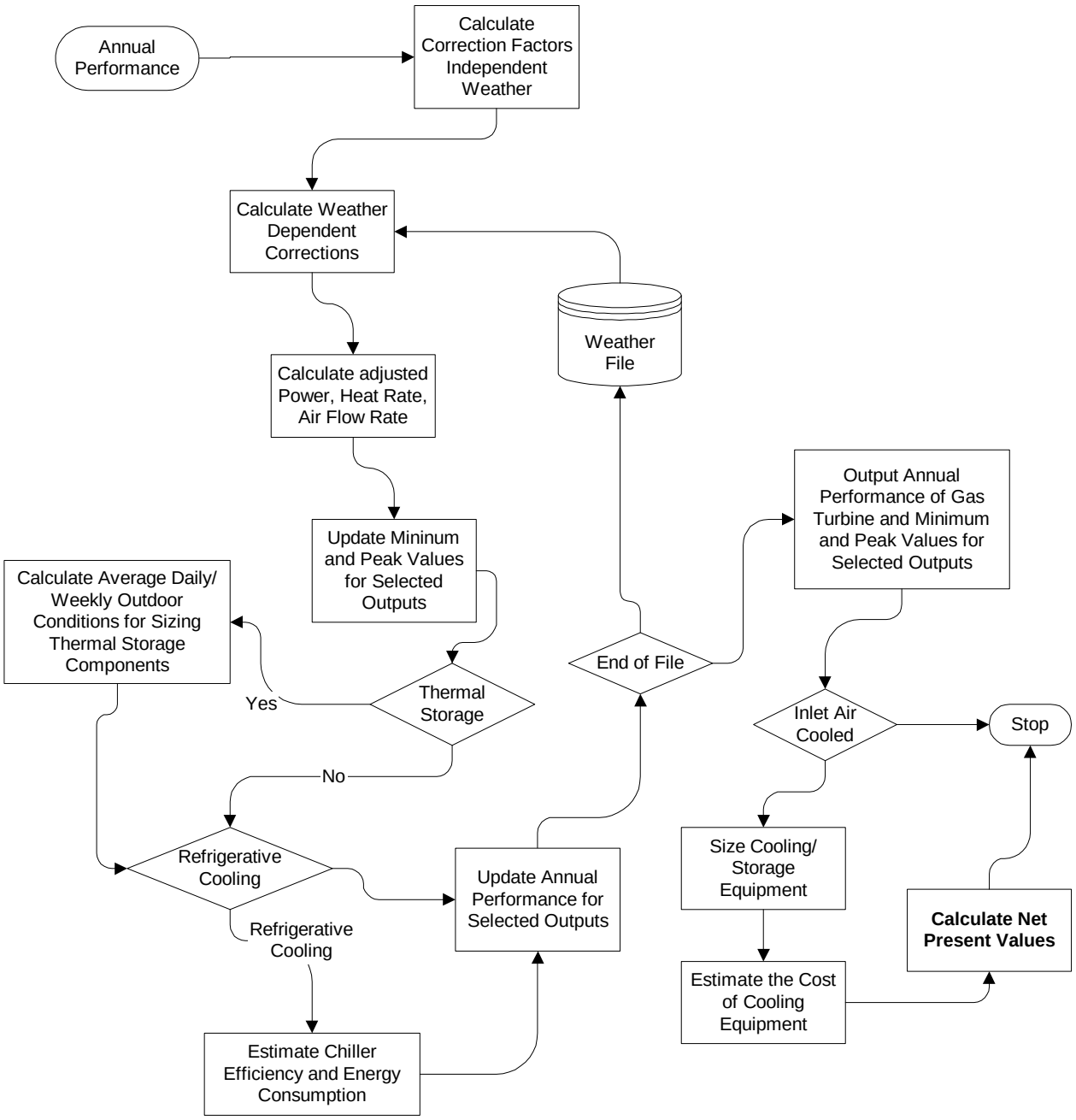

Figure A.2 Schematic of the CTCOOL Model Structure (cont.)

Appendix B

Custom Weather File Format

Appendix B Custom Weather File Format

Several weather files are provided with the software; these files have been derived from the TMY weather tapes. However, the user can use a custom weather file for the analysis. The format of the weather file is shown in Table B1, with sample data shown in Table B2. The file should be prepared in ASCII "space-delimited" format.

Table B1 - Format for the Custom Weather File

Column	Input
1	Month
2	Day of Month
3	Hour of Day
4	Day of Year
5	Day of Week (1=Sunday, 2=Monday, 3=Tuesday, 4=Wednesday, 5=Thursday, 6=Friday, 7=Saturday)
6	Dry-Bulb Temperature (^oF)
7	Wet-Bulb Temperature (^oF)
8	Dew-Point Temperature (^oF)

Table B2 – Sample Weather Data for Houston

Month	Day of Month	Hour of Day	Day of Year	Day of Week	Dry-Bulb (ºF)	Wet-Bulb (ºF)	Dew Point (ºF)
1	1	1	1	5	52	51	50.1
1	1	2	1	5	52	51	50.1
1	1	3	1	5	52	51	50.1
1	1	4	1	5	52	51	50.1
1	1	5	1	5	52	50	48.3
1	1	6	1	5	52	50	48.3
1	1	7	1	5	52	51	50.2
1	1	8	1	5	53	51	49.5
1	1	9	1	5	53	51	49.5
1	1	10	1	5	55	51	47.7
1	1	11	1	5	58	51	45
1	1	12	1	5	60	51	43.3
1	1	13	1	5	61	51	41.9
1	1	14	1	5	61	50	39.6
1	1	15	1	5	62	50	38.5
1	1	16	1	5	59	48	36.9
1	1	17	1	5	57	47	36.4
1	1	18	1	5	54	45	35.4
1	1	19	1	5	53	44	33.6
1	1	20	1	5	51	43	33.6
1	1	21	1	5	50	43	34.8
1	1	22	1	5	48	42	34.8
1	1	23	1	5	46	41	35.4
1	1	24	1	5	44	40	35.4

The project information dialog (Figure 5.2) provides the user the ability to select a custom weather file by checking the "User Defined" button. After the "User Defined" button is checked, the "Get File Name"

button becomes active and the user can provide the path and the file name of the custom weather file by double clicking on the button. It is the **user's responsibility** to make sure the data are consistent and are in the order specified (Table B1).

Appendix C

Component Sizing

Appendix C Component Sizing

The equations and assumptions used to size cooling system components are presented in this Appendix.

C.1 Chiller Sizing

For direct air cooling applications (without storage) the chiller is sized to meet the peak air cooling load (pacl) and the cooling load required to cool the generator:

$$1^{st} \text{ Stage Chiller Size (Btu/h)} = 1^{st} \text{ Stage pacl x gcf}$$

where "gcf," the generator cooling factor, is the amount of additional cooling needed for the generator (a value of 1.05 is the default assumption). When the air is cooled to 0^oF, two chillers are sized. The first chiller meets the peak cooling load required to cool air from the ambient condition to 42^oF (1st stage). The second chiller meets the peak cooling load to cool the air from 42^oF to 0^oF (2nd stage).

$$2^{nd} \text{ Stage Chiller Size (Btu/h)} = 2^{nd} \text{ Stage pacl x gcf}$$

When daily storage is used, the total load the chiller is required to provide is the sum of the peak daily air cooling load (pdcl), the daily generator cooling load, and the load required to offset the storage and defrost (ice storage only) losses. For load-shifting storage, the chiller is sized to generate the total load while operating only during the period when there is no inlet air cooling:

$$\text{Chiller Size, Daily Load-Shifting (Btu/h)} = (\text{pdcl x gcf x seff x deff})/\text{nop}$$

where "seff" and "deff" are storage and defrost efficiencies, respectively, and "nop" is the number of hours the chiller can operate when inlet air is not being cooled. For example, if the inlet air is cooled 4 hours per day with an electrically-driven chiller, "nop" is 20 hours. For thermally-activated chillers, "nop" is the difference between the number of plant operating hours and the number of inlet air cooling hours (because the thermal energy required to run the chiller is extracted from the power plant). For load-leveling storage, the chiller is sized to provide the total daily load while operating 24 hours a day (for electrically-driven chillers) or while the plant is operating (for thermally-activated chillers).

$$\text{Chiller Size, Daily Load-Leveling (Btu/h)} = (\text{pdcl x gcf x seff x deff})/\text{op}$$

where "op" is either 24 or the number of plant operating hours. When weekly storage is used, the total load the chiller is required to provide is the sum of the peak weekly air cooling load, the weekly generator cooling load and load required to offset the storage and defrost (ice storage only) losses. For load-shifting, the chiller is sized to generate the total load while operating only during the period when there is no inlet air cooling. The same equations used to size chillers for daily load-shifting and load-leveling are used for weekly load-shifting and load-leveling. The "nop" for weekly storage is either multiplied by 5 or 7, for thermally activated chillers, depending on whether the plant operates on weekends or not. For electrically-driven chillers with the plant operating on weekends, the daily "nop" is multiplied by 7; when the plant is not operating on weekends, the weekly "nop" is:

$$\text{weekly nop} = (\text{daily nop x 5}) + 48$$

C.2 Storage Sizing

The storage size for daily load-shifting is:

Daily Storage Size, Load-Shifting (Btu) = (pdcl x gcf)/seff

The storage size for daily load-leveling is:

Daily Storage Size, Load-Leveling (Btu) = (pdcl x gcf)/seff - (chiller size x noc x deff)

where "noc" is the number of inlet cooling hours. The storage size for weekly load-shifting is:

Weekly Storage Size, Load-Shifting (Btu) = (pwcl x gcf)/seff - (chiller size x nop x 4 x deff)

where "pwcl" is the peak weekly cooling load. The storage size for weekly load-leveling is:

Weekly Storage Size, Load-Leveling (Btu) = (pwcl x gcf)/seff - (chiller size x (noc + 24 x 4) x deff)

C.3 Inlet Air Cooling Coil Sizing

Both the 1st and 2nd stage inlet air cooling coil UAs are sized as follows:

Coil UA Size (Btu/h-^{o}F) = pacl/? T_{lm}

where "? T_{lm}" is the log mean temperature difference. The heat exchangers required to cool the generator are sized as follows:

Generator Cooling Coil Size (sf) = pacl x (gcf - 1) /(? T_{lm} x U)

The values used for "? T_{lm}" and "U" are shown in Table C.1.

Table C1 - ?T_{lm} and U Values Used for Sizing Generator Cooling Coil

Mode	? T_{lm}	U
Inlet air cooling to 42°F or 52°F	35	225
Inlet air cooling to 0°F		
1st Stage	10	225
2nd Stage	60	225

C.4 Chilled Water Circulation System Sizing

The chilled water (CW) circulation loop is sized as follows:

CW Circulation System Size (gpm) = pacl / (500.4 x (T_o - T_i))

For LiBr chillers or water storage cases, T_o and T_i are 56°F and 44°F, respectively; and for ice storage, T_o and T_i are 46°F and 34°F, respectively. The circulation system needed to cool the generator is calculated similarly:

$$\text{CW Circulation System Size Generator Loop (gpm)} = \text{pacl} \times (\text{gcf} - 1) / (500.4 \times (T_o - T_i))$$

The peak pumping pump required to pump water is calculated as follows:

$$\text{Peak Pumping Power (kW)} = \text{CW Circulation System Size} \times \text{kW_gpm}$$

where "kW_gpm" is the power (kW) required to pump 1 gallon of CW per minute. The same equation is used for calculating the generator peak pumping power. Annual pumping energy for both the main CW loop and the generator cooling loop is calculated as follows:

$$\text{Annual CW Pumping Energy (kWh)} = \text{Peak Pumping Power} \times \text{acl/pacl}$$

where "acl" is the annual cooling load.

C.5 Ammonia Circulation System Sizing

The ammonia circulation loop is sized as follows:

$$\text{Ammonia Circulation System Sizing (lb/h)} = \text{pacl} / \text{Ammonia hfg}$$

where "Ammonia hfg" is the latent heat of vaporization for ammonia. For ammonia chillers operating at an evaporator temperature of 32°F, the hfg is 543 Btu/lb, and for chillers operating at 20°F and -20°F evaporator temperatures, hfg values are 549 Btu/lb and 583 Btu/lb, respectively. Typically, the first 1st stage chiller operates at 32°F or 20°F evaporator temperature, while the 2nd stage chiller operates at -20°F evaporator temperature. When ice or water storage is used, an additional ammonia loop between the chiller and storage tank is sized similarly:

$$\text{Ammonia Chiller-to-Storage Sizing (lb/h)} = \text{Storage Size} / \text{Ammonia hfg}$$

The circulation loop needed to cool the generator is calculated as follows:

$$\text{Ammonia Circulation Size for Generator Loop (lb/h)} = \text{pacl} \times (\text{gcf} - 1) / \text{Ammonia hfg}$$

The peak pumping pump required to pump ammonia is calculated as follows:

$$\text{Peak Pumping Power (kW)} = \text{Ammonia Circulation System Size} \times \text{kW_lbh}$$

where "kW_lbh" is the power (kW) required to pump 1 pound of ammonia per hour. The same equation is used for calculating the generator peak pumping power. Annual pumping energy for the main ammonia loop, chiller-to-storage loop, and the generator loop is calculated as follows:

$$\text{Annual Ammonia Pumping Energy (kWh)} = \text{Peak Pumping Power} \times \text{acl/pacl}$$

C.6 Ice Generator Sizing

The size of the ice generator is same as the chiller. The circulation loop for the ice generator is sized as follows:

Ice Generator Circulation Loop Sizing (gpm) = Chiller Size x ice_gpmpt / 12000

where "ice_gpmpt" is the circulation loop flow rate in gallons per minute per ton of chilling capacity. The peak pumping power is calculated as follows:

Peak Pumping Power for Ice Generator (kW) = Chiller Size x ikW_ton / 12000

where "ikW_ton" is the kW per ton of chilling capacity. Annual pumping power is calculated as follows:

Annual Pumping Energy (kWh) = Peak Pumping Power x acl x gcf / (seff x Chiller Size)

C.7 Chilled Water Generator

The size of the chilled water generator is same as the chiller. The circulation loop for the chilled water generator is sized as follows:

CW Generator Circulation Loop Sizing (gpm) = Chiller Size / (12000 x h2o_gpmpt)

where "h2o_gpmpt" is the circulation loop flow rate in gallons per minute per ton of chilling capacity. The peak pumping power is calculated as follows:

Peak Pumping Power for CW Generator (kW) = Chiller Size / (12000 x wkW_ton)

where "wkW_ton" is the kW per ton of chiller capacity. Annual pumping power is calculated as follows:

Annual Pumping Power (kWh) = Peak Pumping Power x acl x gcf / (seff x Chiller Size)

C.8 Cooling Tower Sizing

The peak heat rejection rate at the cooling tower for both 1st and 2nd stage is calculated as follows:

Peak Heat Rejection Rate (Btu/h) = Chiller Size x Chiller Heat Rejection Rate / 12000

where the chiller heat rejection rate varies by chiller technology. The water pipe sizing for the cooling tower is:

Cooling Water Sizing (gpm) = Peak Heat Rejection Rate / (500.4 x ? T_e)

where ? T_e is assumed to be 10°F. The peak pumping power is calculated as follows:

Peak Pumping Power (kW) = Cooling Water Sizing x ckW_gpm

where "ckW_gpm" is 0.02. Annual pumping energy is calculated as follows:

Annual Pumping Energy (kWh) = Peak Pumping Power x acl x gcf / (seff x Chiller Size)

The peak fan power for the cooling tower is calculated as follows:

Peak Fan Power (kW) = Peak Heat Rejection Rate x kW_mmBtu/1E6

where "kW_mmBtu" is 1.25. The annual fan energy consumption is calculated as follows:

Annual Fan Energy (kWh) = (Peak Fan Power x acl x gcf)/(seff x Chiller Size)

C.9 Chiller Parasitic Energy

The LiBr chiller has additional parasitic consumption, which is calculated as follows:

Annual Parasitic (kWh) = (Chiller Size / 12000 x kW_ton) x acl x gcf / Chiller Size

where "kW_ton" is 0.0136 kW/ton for a single stage LiBr chiller and 0.028 kW/ton for a double-stage LiBr chiller.

C.10 Heat Recovery Steam Generators (HRSG)

The heat recovery steam generator for thermally-activated chillers with a simple-cycle power plant is sized as follows:

HRSG UA (Btu/h-^{o}F) = Peak Chiller Energy Consumption/? T_{lm}

where "? T_{lm}" is the log mean temperature difference:

? T_{lm} = ((gt_fex - feed_t) - (gt_ex - feed_t))/(log((gt_fex - feed_t)/(gt_ex - feed_t))

where "gt_ex" is the temperature of the exhaust gas leaving the gas turbine (1000 ^{o}F) and "gt_fex" is the temperature of the exhaust gas leaving the HRSG. The exhaust gas leaving the HRSG is calculated as follows:

gt_fex (^{o}F) = gt_ex - Peak Chiller Energy Consumption/(mass x C_p Gas)

where "mass" is the mass flow rate of the exhaust gas and a value of 0.25 Btu/ ^{o}F-lb is assumed for the exhaust gas C_p. The required steam flow is calculated as follows:

Steam Flow (lb/h) = Peak Chiller Energy Consumption/(Steam hfg)

A maximum pressure drop equivalent to 10 inches of water is assumed to occur in the HRSG, if all the energy from the exhaust gas is extracted to generate steam (i.e., if gt_fex = 300°F). However, only a fraction of the exhaust flow energy is required in most cases to meet the peak energy requirements of the thermally-activated chiller:

Fraction of Energy Required = (gt_ex - gt_fex)/(gt_ex - 300)

The required exhaust airflow to meet the peak chiller energy consumption is calculated as follows:

Exhaust Flow Rate (lb/h) = Chiller Energy Consumption/(Cp Gas x (gt_ex - gt_fex))

C.11 Power Reduction Due to Steam Extraction

Power is lost when a thermally-activated chiller is used with a combined-cycle power plant because the steam extracted from the combined-cycle to run the chiller would otherwise be used to generated electricity. The foregone electricity production is calculated as follows:

Power Reduction (kW) = (-0.0079 + 0.0172 x $\log_e$ (Steam Pressure, psia))

The amount of steam that has to be extracted is calculated as follows:

Steam Flow (lb/h) = Peak Chiller Energy Consumption/(Steam hfg)

Appendix D

Performance Analysis

Appendix D Performance Analysis

The performance analysis includes calculation of the power output and fuel consumption of the combustion turbine, with and without inlet air cooling, plus the parasitic power consumption of inlet air cooling system components. The latter includes compressor power for vapor compression chillers, steam and pump power for absorption chillers, pump power for the various fluid circulation systems, and fan power for the cooling tower.

Annual performance data are calculated by aggregating performance calculated for each operating hour of the year, based on hourly weather data. Performance assumptions were developed based on a combination of information collected from equipment vendors and published literature. Specific assumptions are presented below for the power plants and cooling system components.

D.1 Power Plants

Three power plant types were evaluated in this study: 1) simple-cycle industrial turbine, 2) simple-cycle aeroderivative turbine, and 3) combined-cycle (based on an industrial-type turbine). Analysis of inlet air cooling systems requires knowledge of the power plant's (combustion turbine <u>and</u> steam turbine, for combined-cycle plants) performance (power output, heat rate, and mass flow rate) as a function of various operating conditions (altitude, dry-bulb temperature, humidity, turbine inlet pressure drop, and turbine exit pressure drop). The impact of operating conditions on power output and heat rate directly affect inlet air cooling economics, of course, while understanding the impact on mass flow rate is crucial to sizing cooling system components.

Power output, heat rate, and inlet air mass flow rate at ISO conditions (59 °F dry-bulb temperature, 60% relative humidity, and sea level elevation), and the correction factors for calculating these performance characteristics at non-ISO conditions are presented below for each power plant type. Up to five adjustment factors (F1 for altitude, F2 for dry-bulb temperature, F3 for humidity, F4 for excess inlet pressure drop, and F5 for excess exhaust pressure drop) are multiplied by the ISO performance to calculate performance at non-ISO conditions.

<u>Simple-Cycle Industrial Frame Combustion Turbine</u>

ISO Power Output = 100 MW
ISO Heat Rate = 11,700 Btu/kWh based on HHV
ISO Inlet Air Mass Flow Rate = 2,342,346 lb/h.

Power Adjustment Factors:

F1 = 1.0 - 3.1875 E-5 * h; h = altitude, feet
F2 = 1.23 - 3.875 E-3 * T; T = inlet air dry-bulb temperature, °F
F3 = 1.0
F4 = 1.0 - 4.333 E-3 * P_i; P_i = excess inlet air pressure loss, inches H_2O
F5 = 1.0 - 1.634 E-3 * P_e; P_e = excess exhaust air pressure loss, inches H_2O.

Heat Rate Adjustment Factors:

F1 = 1.0
F2 = 0.95 + 8.333 E-4 * T for T <= 60 °F; F2 = 1.0 + 1.666 E-3 * (T - 60) for T > 60 °F
F3 = 1.0
F4 = 1.0 + 1.666 E-3 * P_i
F5 = 1.0 + 1.666 E-3 * P_e.

Inlet Air Mass Flow Rate Adjustment Factors:

F1 = 1.0 - 3.1875 E-5 * h
F2 = 1.128 - 2.17 E-3 * T
F3 = 1.0
F4 = 1.0 - 2.333 E-3 * P_i
F5 = 1.0.

<u>Simple-Cycle Aeroderivative Combustion Turbine</u>

ISO Power Output = 45 MW
ISO Heat Rate = 10,000 Btu/kWh based on HHV
ISO Inlet Air Mass Flow Rate = 1,093,482 lb/h.

Power Adjustment Factors:

F1 = 1.0 - 3.3125 E-5 * h
F2 = 1.125 for T <= 32 °F;F2 = 1.125 - 5.06 E-3 * (T - 32) for T > 32 °F
F3 = 1.0
F4 = 1.0 - 3.6 E-3 * P_i
F5 = 1.0 - 1.4 E-3 * P_e.

Heat Rate Adjustment Factors:

F1 = 1.0
F2 = 0.94 + 1.00 E-3 * T for T <= 60 °F; F2 = 1.0 + 1.666 E-3 * (T - 60) for T > 60 °F
F3 = 1.0
F4 = 1.0 + 1.2 E-3 * P_i
F5 = 1.0 + 1.1875 E-3 * P_e.

Inlet Air Mass Flow Rate Adjustment Factors:

F1 = 1.0 - 3.3125 E-5 * h
F2 = 1.15 - 1.875 E-3 * T for T <= 32 °F; F2 = 1.09 - 3.21 E-3 * (T - 32) for 32 °F < T < 60 °F
F2 = 1.0 - 4.67 E-3 * (T - 60) for T >= 60 °F
F3 = 1.0
F4 = 1.0 - 2.55 E-3 * P_i
F5 = 1.0 - 3.125 E-3 * P_e.

Combined-Cycle

ISO Power Output = 225 MW
Combustion Turbine Portion = 155 MW
Steam Turbine Portion = 70 MW
ISO Heat Rate = 7300 Btu/kWh based on HHV
ISO Inlet Air Mass Flow Rate = 3,365,475 lb/h.

Power Adjustment Factors for Combustion Turbine Portion:

F1 = 1.0 - 3.33 E-5 * h
F2 = 1.22 - 3.7 E-3 * T
F3 = 1.0
F4 = 1.0 - 3.2 E-3 * P_i
F5 = 1.0 - 1.2 E-3 * P_e.

Power Adjustment Factors for Steam Turbine Portion:

The power adjustment factors for the steam turbine require a more complicated format. In general, steam turbine power is presumed to be proportional to steam production from the HRSG, with an adjustment for the impact of ambient temperature on condenser temperature. The general form for steam turbine power adjustment is as follows:

P/P_{ISO} = exhaust flow factor * exhaust temperature factor * condenser temperature factor.

Each of these three factors is defined below.

Exhaust Flow Factor = F1*F2*F3*F4*F5, where

F1 = 1.0 - 3.33 E-5 * h
F2 = 1.12 - 2.15 E-3 * T
F3 = 1.0
F4 = 1.0 - 2.5 E-3 * P_i
F5 = 1.0.

Exhaust Temperature Factor = $(T_{exhaust} - 550)/542$, where

$T_{exhaust} = 1054 + 0.517\ T + P_i + 0.8\ P_e$ for T <= 60 °F or
$T_{exhaust} = 1085 + 0.850\ (T - 60) + P_i + 0.8\ P_e$ for T > 60 °F.

Condenser Temperature Factor =

1 - 1.9 E-4 * $(T_{wb} - 54)$ for T_{wb} <= 54 °F or
1 - 9.0 E-4 * $(T_{wb} - 54)$ for T_{wb} > 54 °F, and

T_{wb} = ambient wet-bulb temperature, °F.

Heat Rate Adjustment Factors for Combustion Turbine Portion:

F1 = 1.0
F2 = 0.94 + 1.00 E-3 * T for T <= 60 °F; F2 = 1.0 + 1.5 E-3 * (T - 60) for T > 60 °F
F3 = 1.0
F4 = 1.0 + 1.7 E-3 * P_i
F5 = 1.0 + 1.45 E-3 * P_e.

Heat Rate Adjustment Factors for Combined-Cycle:

Overall Heat Rate = CT Heat Rate * CT Power / CC Power
7300 = CT Heat Rate * 155/225; Reference CT Heat Rate = 10,597 Btu/kWh
Adjusted CT Heat Rate = 10,597 * F1 * F2 * F3 * F4 * F5
Adjusted CC Power = Adj. CT Power + Adj. ST Power
Adjusted CC Heat Rate = Adj. CT HR * Adj. CT Power / Adj. CC Power.

Inlet Air Mass Flow Rate Adjustment Factors:

F1 = 1.0 - 3.33 E-5 h
F2 = 1.12 - 2.15 E-3 T
F3 = 1.0
F4 = 1.0 - 2.5 E-3 P_i
F5 = 1.0.

D.2 Chillers

The cost and performance of the chiller are key to the cost effectiveness of inlet air cooling systems. Chiller performance is characterized by its coefficient-of-performance (COP), "relative capacity," heat rejection rate, and, for absorption-type chillers, the steam conditions and additional electric power consumption, if any. Performance assumptions for complex-compound, vapor-compression, aqua-ammonia, and lithium bromide chillers are presented below.

The COP, of course, is the amount of cooling obtained per unit of input energy, with the input and output measured in the same units. The COP varies considerably between chiller types and for the same chiller type, depending on the evaporating and condensing temperature, with the difference between these two temperatures defined as the "lift." In general, COP increases as the lift is reduced. Absorption-type chillers also come in single- and double-stage versions. Double-stage chillers use waste heat from the higher temperature stage to drive the lower temperature stage. A higher COP results, but the double-stage unit is more expensive. COP equations are presented here as a function of wet-bulb temperature for explicit chiller types, number of stages, and evaporator temperatures. The COP equations assume the condenser temperature is 15 °F higher than the wet-bulb temperature. In general, the wet-bulb temperature is limited to a minimum of 25 °F for COP modeling purposes, with more restrictive assumptions listed below for lithium bromide chillers.

Like the COP, the cooling capacity of a chiller is a function of the condenser temperature, and is expressed here as a function of the wet-bulb temperature. The chiller cost equations presented in Section 8 are based on chiller capacity at "rated" temperature conditions. Capacity is typically less at temperature conditions corresponding to the peak cooling load. Thus, the "relative" capacity equation must be applied to adjust the peak chiller capacity to the rated chiller capacity.

All energy absorbed by a chiller at the evaporator and input to the chiller in the form of steam or electricity must ultimately be rejected at a cooling tower. Thus, heat rejection requirements are directly related to chiller COP and vary considerably, especially when absorption systems are compared to vapor compression systems.

Complex-Compound Chillers:

CC chiller 1 (CC-1); single stage, 20 °F evaporator:

COP = 0.496 - 7.0 E-4 * Twb
relative capacity = 1.652 - 0.0079 * Twb
steam temperature = 228 °F; pressure = 19 psia; h_{fg} = 960 Btu/lb
heat rejection rate = 39,335 Btu/hr per ton
parasitic power = 0.

CC chiller 2 (CC-2); single stage, 32 °F evaporator:

COP = 0.496 - 7.0 E-4 * Twb
relative capacity = 1.415 - 0.0051 * Twb
steam temperature = 228 °F; pressure = 19 psia; h_{fg} = 960 Btu/lb
heat rejection rate = 39,211 Btu/hr per ton
parasitic power = 0.

CC chiller 3 (CC-3); double stage, 20 °F evaporator:

COP = 0.867 - 2.5 E-3 * Twb
relative capacity = 1.566 - 0.0069 * Twb
steam temperature = 351 °F; pressure = 137 psia; h_{fg} = 870 Btu/lb
heat rejection rate = 29,991 Btu/hr per ton
parasitic power = 0.

CC chiller 4 (CC-4); double stage, 32 °F evaporator:

COP = 0.850 - 1.85 E-3 * Twb
relative capacity = 1.334 - 0.0041 * Twb
steam temperature = 351 °F; pressure = 137 psia; h_{fg} = 870 Btu/lb
heat rejection rate = 29,094 Btu/hr per ton
parasitic power = 0.

CC chiller 5 (CC-5); single stage, -20 °F evaporator:

COP = 0.42 - 6.75 E-4 * Twb
relative capacity = 1.231 - 0.003 * Twb
steam temperature = 293 °F; pressure = 60 psia; h_{fg} = 915 Btu/lb

heat rejection rate = 44,787 Btu/hr per ton
parasitic power = 0.

CC chiller 6 (CC-6); single-stage, storage type, -20 °F evaporator:

COP = 0.52
relative capacity = 1
steam temperature = 213 °F; pressure = 15 psia; h_{fg} = 969 Btu/lb
heat rejection rate = 21,624 Btu/hr per ton
for sizing heat rejection equipment
heat rejection rate = 35,076 Btu/hr per ton
for calculating annual heat rejection load
parasitic power = 0.

The storage version of the complex-compound chiller is actually an integrated chiller/storage device that is sized based on ton-hours rather than tons.

Ammonia Vapor Compression Chillers:

VC chiller 1 (VC-1); 20 °F evaporator:

COP = 14.1277 exp(-0.0144 * Twb)
relative capacity = 1.08 - 0.001 * Twb
heat rejection rate = 14,734 Btu/hr per ton
parasitic power = 0.

VC chiller 2 (VC-2); 32 °F evaporator:

COP = 20.1417 exp(-0.0171 * Twb)
relative capacity = 1.265 - 0.0033 * Twb
heat rejection rate = 14,334 Btu/hr per ton
parasitic power = 0.

VC chiller 3 (VC-3); -20 °F evaporator:

COP = 5.863 exp(-0.0123 * Twb)
relative capacity = 1.008 - 0.001 * Twb
heat rejection rate = 17,444 Btu/hr per ton
parasitic power = 0.

Aqua-Ammonia Chillers

AA chiller 1 (AA-1); 20 °F evaporator:

COP = 0.820 - 0.00374 * Twb
relative capacity = 1.5625 - 0.0075 * Twb
steam temperature = 252 °F; pressure = 31 psia; h_{fg} = 944 Btu/lb
heat rejection rate = 34,263 Btu/hr per ton
parasitic power = 0.

AA chiller 2 (AA-2); 32 °F evaporator:

COP = 0.857 - 0.00372 * Twb
relative capacity = 1.5625 - 0.0075 * Twb
steam temperature = 240 °F; pressure = 25 psia; h_{fg} = 952 Btu/lb
heat rejection rate = 32,754 Btu/hr per ton
parasitic power = 0.

AA chiller 3 (AA-3); -20 °F evaporator:

COP = 0.8672 - 0.00635 * Twb
relative capacity = 1.5625 - 0.0075 * Twb
steam temperature = 315 °F; pressure = 83 psia; h_{fg} = 899 Btu/lb
heat rejection rate = 42,691 Btu/hr per ton
parasitic power = 0.

Lithium Bromide Chillers

LiBr chiller 1 (LiBr-1); single-stage:

COP = 0.671 - 9.7 E-4 * $(Twb - 65)^{1.5}$
relative capacity = 1.22 - 0.024 * (Twb - 65)
steam temperature = 244 °F; pressure = 27 psia; h_{fg} = 950 Btu/lb
heat rejection rate = 29,200 Btu/hr per ton
parasitic power = 0.0136 kW/ton.

The minimum cooling water temperature allowable for single-stage LiBr chillers is 75 °F. Therefore, Twb is limited to a minimum of 65 °F.

LIBr chiller 2 (LiBr-2); double-stage, 32 °F evaporator:

COP = 1.237 - 4.258 E-3 * $(Twb - 68)^{1.2}$
relative capacity = 1.074 -0.0255 * (Twb - 68)
steam temperature = 347 °F; pressure = 130 psia; h_{fg} = 873 Btu/lb
heat rejection rate = 21,800 Btu/hr per ton
parasitic power = 0.028 kW/ton.

The minimum cooling water temperature allowable for double-stage LiBr chillers is 78°F. Therefore, Twb is limited to a minimum of 68 °F.

D.3 Storage

Storage is a significant component of a CTIAC system, but can be effectively modeled with a few simple inputs. Thermal losses through the walls of the storage tank are typically no more than 1%, but other loss mechanisms have a much greater impact. Water storage relies on the creation of a thermocline to separate the hot and cold portions of the water. The difference between hot and cold temperatures, hence hot and cold bouyancies, is not much in this application of water/thermal storage. Therefore, losses across the thermocline are relatively high. The combined water storage efficiency was assumed to be 85%. Ice storage is really a mixture of water and ice. During discharge, warm return water is sprayed on top of the ice to facilitate melting. Here, mixing is not only beneficial, but required. During the charging cycle, however, periodic defrosting of the ice generator is required to release the ice from the evaporator

surfaces. The defrost energy increases the refrigeration load and chiller capacity, but, unlike the water storage losses or the 1% loss through the tank walls, it does not affect the capacity of the storage tank. The defrost efficiency factor was presumed to be 85% (i.e., the chiller load and refrigeration capacity are increased by a factor of 1/0.85 to account for the defrost energy, which must be removed). The defrost efficiency is separate from the ice storage efficiency, which was presumed to be 99%.

D.4 Fluid Circulation

CTIAC systems include several fluid circulation loops, with the specific number and type depending on the type of chiller and storage. Included here are the ammonia loop connecting the chiller evaporator and other chiller components (except for LiBr chillers), the chilled water loop connecting LiBr chillers and storage with the inlet air cooling coils, the cooling water loop connecting the chiller with the cooling tower, the steam loop connecting the steam source with absorption-type chillers, and storage water loops connecting relatively warm storage water with evaporators in the ice generator or water chiller. The following pumping power rules-of-thumb were established from a collection of published data describing CTIAC systems.

Ammonia Loop Power: 0.005 kW/ton or 2.0 E-5 kW/lb per hr ammonia

Chilled Water Loop Power: 0.03 kW/gpm water

Cooling Water Loop Power: 0.02 kW/gpm water

Steam Loop Power: 0.03 kW/gpm or 6.0 E-5 kW/lb per hr steam

Storage Charging Water Loop Power

Ice Generation: 0.03 kW/ton
Chilled Water Generation: 0.02 kW/ton

D.5 Evaporative Cooling

Evaporative cooling towers are used by every chiller to reject heat. In addition, evaporative inlet air cooling is an alternative to refrigerative inlet air cooling. Water consumption for both types of evaporators is calculated based on the thermodynamic requirement plus a 10% allowance for blowdown and carryover. The inlet air evaporator consumes 0.01 kW/gpm of water circulating through the cooler or 0.3 kW/gpm evaporating. The cooling tower fan consumes 1.25 kW/MMBtu per hour of heat rejected.

D.6 Auxiliary System Cooling

Cooling the inlet air of the turbine will increase its shaft power output, but care must be taken not to exceed the operating capabilities of other power plant components, specifically the generator, transformer, and lube oil system. Depending on the capabilities of these components, supplemental cooling may be required. This study assumed that supplemental cooling equal to 5% of the inlet air cooling load would be required. Chillers and storage components were oversized by this factor, and a separate cooling loop and heat exchanger was provided to cool the auxiliary equipment.

D.7 Turbine Inlet and Exit Excess Pressure Drop

Adding an inlet air cooling device increases the compressor pressure drop, resulting in decreased power output and increased heat rate. Although these losses are relatively small compared to the improvement in these same factors achieved from cooling the air, they are significant enough to warrant inclusion in the analysis. A similar effect occurs if a heat recovery steam generator is added to a simple-cycle turbine to provide steam for an absorption-type chiller. The increase in exhaust pressure drop also reduces power output and increases heat rate, but by a relatively small amount. The specific assumptions used for excess inlet and exhaust pressure drop are shown below.

Inlet Air Cooling Coil Air-Side Pressure Drop:

Evaporative Cooling: 1.0 in. of water
Refrigerative Cooling to 42 °F: 0.5 in. of water
Refrigerative Cooling to 0 °F: 1.3 in. of water

HRSG Air-Side Pressure Drop:

Maximum pressure drop for full heat recovery = 10 in. water
Actual pressure drop = 10 in. * $(1000 - T_r)/(1000-300)$
T_r = calculated stack reject temperature, °F

D.8 Combined -Cycle Steam Extraction Power Output Reduction

Absorption-type chillers require a source of low pressure steam. For simple-cycle power plants, an HRSG and related steam/condensate piping are added to provide the steam. In combined-cycle plants, steam is presumed to be extracted from the lower pressure stages of the steam turbine. While no additional fuel is required in either case (except for the effect of increasing the exhaust pressure when adding an HRSG), extracting steam from combined-cycle plant's steam turbine does incur a loss of power production that would otherwise have been produced. The loss increases as the extraction pressure increases and was described by the following relationship:

kW/lb per hr steam = $-0.0079 + 0.0172 \ln (P)$
P = steam pressure, psia.

Appendix E

Cost Analysis

Appendix E Cost Analysis

Initial capital cost and annual operating and maintenance cost estimating equations were developed for each cooling system component. Cost data acquired from equipment vendors, published literature describing inlet air cooling systems, and cost estimating reference books were the basis for developing the equations. The equations are applied to equipment sizing results to produce system cost estimates peculiar to each case evaluated.

All costs were estimated in mid-1994 dollars and are intended to include all costs associated with equipment and material purchase and installation. All components except for the complex-compound chiller are mature, commercially available products. Complex-compound chiller costs were estimated based on a mature production scenario, i.e., the first unit would probably be more expensive.

Note that no equations for estimating annual operating labor costs are presented. No incremental operating labor is presumed to be required based on advice received from personnel at an operating inlet air cooling system.

E.1 Chiller Capital Costs

A striking observation for chillers is the extremely high cost of the aqua-ammonia systems compared to the other options. Otherwise, all chillers increase in cost as the evaporator temperature is lowered, and double-stage units cost more than single-stage units. Absorption chiller costs are less sensitive to evaporator temperature than vapor compression chillers, resulting in a significant cost advantage for complex-compound chillers at the lower evaporating temperature required for cooling inlet air to 0 °F.

LiBr Absorption Chillers:

44 °F water, single stage = $480/ton
44 °F water, double stage = $650/ton.

Complex-Compound Absorption Chillers:

-20 °F evaporator temperature, single stage = $490/ton
+20 °F evaporator temperature, single stage = $390/ton
+20 °F evaporator temperature, double stage = $550/ton
+32 °F evaporator temperature, single stage = $380/ton
+32 °F evaporator temperature, double stage = $530/ton
-20 °F evaporator temperature, single-stage storage = $310/ton-hr.

Note: The storage version of the complex-compound absorption chiller integrates the chiller and storage into a single component. The cost for both is based on the required storage size.

Aqua/Ammonia Absorption Chillers:

-20 °F evaporator temperature = $3000/ton
+20 °F evaporator temperature = $2800/ton
+32 °F evaporator temperature = $2700/ton.

Ammonia Vapor Compression Chillers:

-20 °F evaporator temperature = $780/ton

+20 °F evaporator temperature = $520/ton
+32 °F evaporator temperature = $480/ton.

E.2 Storage Capital Costs

Except for the LiBr units, the chiller costs presented above do not include the cost of the evaporator. Without storage, refrigerant (ammonia) evaporation for the other units occurs directly in the inlet air cooling coils. Otherwise, refrigerant is evaporated in an ice or chilled water generator, depending on the storage type selected. Note that chilled water can be generated in a relatively simple and inexpensive, shell-and-tube type heat exchanger while ice generation requires more complex and costly ice-harvesting equipment. Water and ice storage tanks are essentially the same, but ice storage (actually an ice/water mixture) is less costly per ton-hr because of its higher chill energy density.

Ice Generator = $590/ton

Chilled Water Generator = $50/ton

Ice Storage = $163.6 $(\text{ton-hr})^{0.75}$.

Water Storage = $572.4 $(\text{ton-hr})^{0.75}$

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E.3 Inlet Air Coil Capital Costs

The cooling coils include the coil, its enclosure, and foundation. For refrigerative cooling, the same cost is assumed for either water or ammonia fluid circulating through the coils. The higher cost for the 0 °F air cooling coils is based on having three 50% capacity banks (with two cooling and one defrosting at any point in time) and the extra cost of dampers required to isolate each of the banks for defrosting.

Evaporative Cooling Coils = $3/kW of ISO turbine capacity

Refrigerative Cooling Coils:
For 42 °F or 52 °F air = $0.20/UA; UA in Btu/hr-°F
For 0 °F air = $0.33/UA; UA in Btu/hr-°F.

E.4 Fluid Circulation Capital Costs

Individual estimates for pumps and piping were aggregated to develop the water circulation cost equations shown below. Unit cost differences are directly related to the length of the piping run and pumping head. An insulation premium was also added for the steam/condensate pipe. Ammonia circulation costs are much lower than water circulation because ammonia passes through a phase change, so its mass flow rate is much lower for the same thermal duty. Note that ammonia circulation costs are doubled for 0 °F inlet air temperature applications to allow for hot refrigerant bypass for defrosting.

Water Circulation:
steam/condensate = $320,000 * $(\text{gpm}/10{,}000)^{0.6}$
chilled water = $290,000 * $(\text{gpm}/10{,}000)^{0.6}$

cooling water = $260,000 * (gpm/10,000)$^{0.6}$
storage charging water = $100,000 * (gpm/10,000)$^{0.6}$.

Ammonia Circulation:
For 42 °F or 52 °F air = $54,000 (lb/hr/15,000)$^{0.6}$
For 0 °F air = $108,000 (lb/hr/15,000)$^{0.6}$.

E.5 Miscellaneous Equipment Capital Costs

Capital cost estimating equations for other cooling system components are listed below. Heat recovery steam generators are only used for simple-cycle power plants with absorption-type chillers.

Cooling Tower = $662,000 * (MMBtu/hr/192)$^{0.876}$

System Controls = $250,000 (fixed for all systems)

Plant Electrical = $100,000 (fixed for all systems)

Transformer = $5,553 * (peak auxiliary kW)$^{0.4359}$

Heat Recovery Steam Generator:
$10,980 * (UA)$^{0.8}$ +
$9,658 * (SF) +
$918 * (EF)$^{1.2}$ * (1000 - TR)/300

UA = heat transfer surface conductance, Btu/s-°F
SF = steam flow, lb/s
EF = turbine exhaust flow, lb/s
TR = exhaust gas reject temperature after HRSG, °F

Generator Cooling Loop Heat Exchanger = $865 (HTA)$^{0.511}$

HTA = heat exchanger heat transfer area, square feet.

E.6 Annual Maintenance Costs

Annual maintenance costs were estimated as a percentage of initial capital cost for each inlet air cooling component. In addition, make-up water, for water consumed at either an evaporative inlet air cooler or a cooling tower, was estimated to cost $1/1000 gallons. Maintenance cost estimating percentages are listed below.

Component	Percentage
LiBr chiller	5.0
NH_3/H_2O chiller	4.0
Vapor compression chiller	4.0
Complex compound chiller	3.0
Ice generator	2.0
Chilled water generator	1.0
Ice storage	1.5
Water storage	1.5
Direct contact evaporative air cooler	2.0
Inlet air coil	
42 °F and 52 °F air	2.0
0 °F air	3.0
Circulation (pumps and pipes)	3.0
Cooling tower	2.0
Controls	6.0
Plant electrical	2.0
Transformer	2.0
HRSG	3.0
Generator cooling heat exchanger	1.0

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