ANNUAL ENERGY CONSUMPTION OF RECIPROCATING REFRIGERATION SYSTEMS FOR HUMIDITY CONTROL

BY

THOMAS J. MEITL

B.S., Kansas State University

A MASTER'S THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY

Manhattan, Kansas

1985

APPROVED BY:

TH Ball

Major Professor

TABLE OF CONTENTS

LIST OF FIGURES

FIGURE

LIST OF FIGURES (cont.)

LIST OF TABLES

TABLE

NOMENCLATURE

SYMBOL	DEFINITION
$a_1 - a_9$	Modeling constants
AREA	Cooling coil face area (ft^2)
$b_1 - b_9$	Modeling constants
$C_1 - C_{12}$	Modeling constants
CAP	Subsystem capacity (kW)
CF	Cooling coil airflow rate correction factor
COP	Coefficient of Performance
C_p	Specific Heat (BTU/lb-^oF)
$d_1 - d_9$	Modeling constants
dT	Temperature difference (^oF)
DX	Direct Expansion
$F_1 - F_5$	Modeling constants
FFL	Fraction of Full Load,power
FLP	Subsystem Full Load Power (kW)
fpm	Air velocity at the coil face (fpm)
GPM	Water flow rate (gpm)
h	Enthalpy (BTU/lb)
$\dot{m}$	Mass flow rate (lb/hour)
P	Compressor power consumption (kW)
PLR	Partial Load Ratio,capacity
Q	Amount of heat transferred (BTU/hour)
T	Temperature (^oF)
W	Moisture content of air (lb water/lb air)

NOMENCLATURE (cont.)

SUBSCRIPTS	DEFINITION
c	Condenser
ci	Inlet condenser water
co	Outlet condenser water
cond	Outlet condenser water, Leverenz-Bergan Model
cw	Chilled water
c1	Input condenser variable, Subsystem Model
,D	Conditions at design (base)
db	Dry-bulb
e	Evaporator
e1	Input evaporator variable, Subsystem Model
m	Medium in contact with the evaporator or condenser (air or water)
wb	Wet-bulb

The English Engineering System of units (Btu, Fahrenheit, etc.) was used for this study with the exception of the Leverenz-Bergan and Subsystem models of Chapter 3 which were originally developed in SI units (kW, Celsuis, etc.).

ACKNOWLEDGEMENTS

I would like to thank my major advisor, Dr. Herbert D. Ball, for his help through the course of this research. Also, thanks are extended to Dr. Robert L. Gorton and Dr. Joseph F. Merklin for serving on my committee. Thanks is also extended to Mary Lough for careful and efficient typing of this thesis.

CHAPTER I

INTRODUCTION

The moisture content of air has a major influence on many industrial operations conducted today. Because of this, it is often necessary to limit the moisture content of air on a year-round basis.

Dehumidification, the removal of water from air, is essential in the handling of a variety of products. With the storage of materials and products, maintaining a low air-moisture content level retards the formation of corrosion and may eliminate the need for surface-protecting materials. In the food industry, a reduced humidity level slows the growth of molds and increases product storage life. In manufacturing, the use of dehumidified air during critical processes often results in a higher quality final product. Also, worker productivity is related to personal comfort, which is influenced by the moisture content of the air.

The actual dehumidification process depends on the temperature and pressure relationships which exist between the air, its moisture content, and surrounding objects. Air, in general, is composed of a mixture of different gases with each exerting an individual partial pressure which, when summed, equals the total overall pressure of the mixture. Moisture contained in air is in the form of a vapor and therefore also maintains a partial pressure as part of the overall mixture pressure. This partial pressure cannot exceed the saturation pressure of water for the temperature of the mixture. Because saturation pressure varies directly with temperature, it is possible to limit the moisture content of air by lowering the temperature of the mixture, forcing some of the water to condense out. Also, differences in vapor-pressure will drive moisture from an area of high vapor-pressure to an area of low vapor-pressure. Therefore, dehumidification may also be accomplished through the use of sorbent materials, which utilize a low vapor-pressure to extract excess moisture from the air.

Commercial dehumidification of air in most instances is accomplished by the use of one or both of the following types of systems: desiccant systems and refrigeration systems. Both methods have achieved widespread popularity in industry and the selection of the method to be utilized often depends on the industrial operation to be performed. Desiccant systems, which utilize sorbent materials, are best suited for operations which require high air temperature-low moisture content levels. Refrigeration systems, which dehumidify by cooling the air, are often utilized in situations were air of high moisture content must be processed and relatively low final air temperatures are required. Combinations of the two systems are frequently used to handle air processing situations which fall between these two extremes.

Both of the methods presented above are capable of handling large quantities of air for dehumidification. The selection of the actual system to be utilized is often based on a required exit air condition, initial cost, and the energy requirements of the system at design conditions. This procedure, however, may not select the most energy efficient system available because, on an annual basis, the majority of the system operational time is spent at conditions far less severe than those of design. As a result, a method for predicting system energy requirements on an annual basis should be included in the selection process. To accomplish this, ASHRAE* commissioned Research Project 298 RP "Analysis of Open Sorption and Refrigeration Humidity Control Systems".

The objective of ASHRAE Research Project 298 RP was to develop an analytical method for estimating the annual energy consumption and cost-effectiveness of the different dehumidification systems available, including refrigeration, open-sorption (desiccant) dehumidification, and combinations of both refrigeration and open-sorption dehumidification. From this, the optimum dehumidification system for the desired application may be determined.

1.1 OBJECTIVE

The objective of this study was to develop mathematical models for predicting the energy requirements of refrigeration systems when subjected to variations in both dehumidification load and ambient air conditions which would influence system operation on a year-round basis. The study focused primary on reciprocating vapor-compression refrigeration systems** and was performed in support of ASHRAE Research Project 298 RP, "Analysis of Open-Sorption and Refrigeration Humidity Control Systems".

---

*American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc.

**Treatment of open-sorption dehumidification systems may be found in Reference [1].

CHAPTER II

REFRIGERATION

Dehumidification of air by refrigeration requires the removal of sensible/latent energy from the air stream being processed. For this process, the air must be exposed to a low temperature medium for a significant period of time, allowing the transfer of enough energy to take place between the air stream and the medium for the air to be cooled to a predetermined level. At this level, the air is capable of supporting less moisture than before and any excess moisture is forced to condense out of the mixture to insure that the saturation pressure of water is not exceeded. A typical refrigeration process for the cooling/dehumidification of air is illustrated on a psychrometric chart in Figure 2.1.

Refrigeration systems may be subdivided into two general categories: Absorption and Vapor-Compression. Both methods utilize a low pressure-temperature fluid to absorb heat from the desired medium (air) and a high pressure-temperature fluid to later reject the heat to another medium. The difference between the two systems is in the type of fluids utilized and the way in which the high and low pressure sides of the systems are generated.

2.1 ABSORPTION REFRIGERATION

In the absorption system (Figure 2.2), refrigerant vapor at a low pressure is absorbed in a solution (the absorbent) and the resulting solution is pumped to a high pressure container (the generator). At the

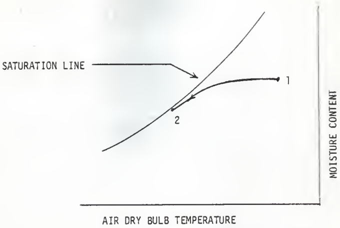


Figure 2.1 Psychrometric chart for cooling/dehumidification by refrigeration.

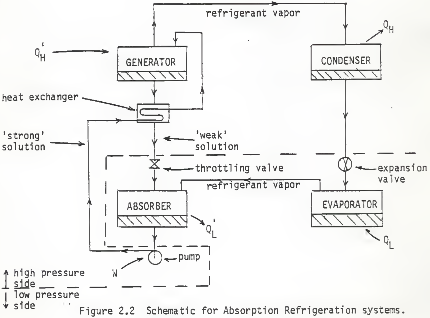


Figure 2.2 Schematic for Absorption Refrigeration systems.

generator, heat is added to the solution, forcing the absorbent to reject the refrigerant. The liquid absorbent is then returned to the absorber while the refrigerant vapor is sent to the condenser. At the condenser, the refrigerant rejects energy to the surroundings and condenses. The liquid refrigerant then flows from the condenser, through a pressure reduction valve, and into the evaporator. Here, heat is absorbed from the desired medium, allowing the refrigerant to revaporize. Finally, the refrigerant returns to the absorber to repeat the process.

For the above mentioned system to operate efficiently, a pair of substances must be utilized which have such a strong attraction for one another that on contact one will immediately absorb the other. Two such refrigerant-absorption pairs have achieved widespread commercial use: Ammonia-Water and Water-Lithium Bromide. Also, efficient system operation requires that a relatively high temperature source must be available at the generator while a method of external cooling is utilized at both the absorber and the condenser.

Capacity control for an absorption refrigeration system may be accomplished by varying the energy source input at the generator or by restricting the flow rate of the solution between the absorber and the generator. By either method the amount of refrigerant boiled off in the generator is reduced, resulting in a stronger solution in the absorber. This solution absorbs refrigerant at a slower rate than before and an increase in the pressure-temperature level of the evaporator results. Finally, the higher temperature level of the evaporator slows the rate of energy transfer from the desired medium to the level needed to match the desired load.

2.2 Vapor-Compression Refrigeration

The primary function of all refrigeration systems is to transfer heat from a relatively low temperature medium to a medium at a higher temperature. To accomplish this, the vapor-compression refrigeration system operates on a cycle based on the energy absorbed or rejected by a single refrigerant as it shifts between its liquid and vapor phases. By controlling the pressures (and therefore temperatures) at which the phase changes occur, it becomes possible to transfer the energy between the two mediums, with the requirement that some additional energy in the form of work also be supplied to the system.

A schematic of a typical vapor-compression system is presented in Figure 2.3. This system is composed of four primary components: compressor, condenser, expansion value, and evaporator, arranged to provide a high pressure-temperature side (condenser) where energy is rejected from the system and a low pressure-temperature side (evaporator) where energy is absorbed. The remaining components (compressor and expansion valve) are utilized to provide for the transition of the refrigerant between the two sides of the system and also to regulate the flow rate of the refrigerant between the other components. A description of the operation cycle of a vapor-compression refrigeration system follows.

2.2a Vapor-Compression Refrigeration Cycle

In the vapor-compression cycle, refrigerant vapor initially at a low pressure-temperature level is compressed to a higher level and flows to the condenser. At the condenser, the high temperature level of the refrigerant allows energy to be transferred out, resulting in the condensation of the refrigerant at the elevated pressure. From here, the liquid refrigerant passes through a pressure reduction (expansion) valve and flows into the

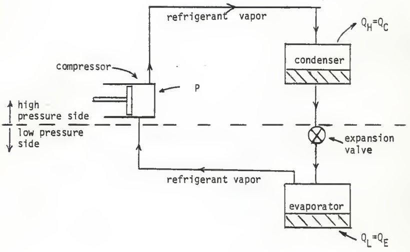


Figure 2.3 Schematic for Vapor-Compression Refrigeration systems.

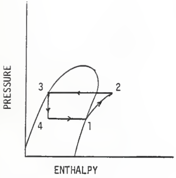


Figure 2.4 a

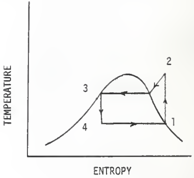


Figure 2.4 b

STEPS	PROCESS
1 - 2	Compression of the refrigerant vapor.
2 - 3	Condensation of the refrigerant to a liquid.
3 - 4	Expansion of the refrigerant to a lower pressure.
4 - 1	Evaporation of the refrigerant to a vapor.

Figure 2.4 Ideal Vapor-Compression Refrigeration Cycle.

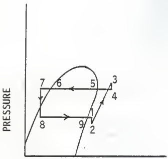


Figure 2.5 a

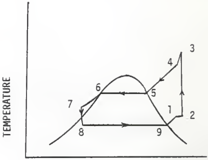


Figure 2.5 b

STEPS	PROCESS
1 - 2	Pressure drop-Temperature rise in suction line.
2 - 3	Compression of the refrigerant vapor.
3 - 4	Pressure drop-Temperature drop in discharge line.
4 - 5	Desuperheating of the refrigerant vapor.
5 - 6	Condensation of the refrigerant.
6 - 7	Subcooling of the refrigerant liquid.
7 - 8	Presssure drop of expansion valve,piping.
8 - 9	Evaporation of the refrigerant.
9 - 1	Superheating of the refrigerant vapor.

Figure 2.5 Actual Vapor-Compression Refrigeration Cycle.

evaporator. The relatively low temperature level of the refrigerant caused by the pressure reduction allows the refrigerant to absorb heat from the desired medium, resulting in its revaporization. Finally, the refrigerant vapor flows back to the compressor to repeat the cycle again.

The above mentioned cycle is illustrated in its idealized form on Pressure-Enthalpy (Figure 2.4a) and Temperature-Entropy diagrams (Figure 2.4b).

The actual refrigeration cycle deviates from the ideal cycle primarily because of additional heat transfer to or from the surroundings and pressure drops associated with fluid flow. Also, vapor entering the compressor will likely be in a superheated condition to eliminate the possibility of damaging the machine by attempting to compress a liquid. Refrigerant leaving the condenser is usually subcooled below its saturation point to prevent the flashing of some of the refrigerant to its vapor phase before reaching the expansion valve, which would severely limit the flow capacity of the valve due to the high specific volume of vapor relative to liquid. Thus, the actual refrigeration cycle would probably approach those illustrated in Figures 2.5a and 2.5b.

2.2b Vapor-Compression Refrigeration/Dehumidification Systems

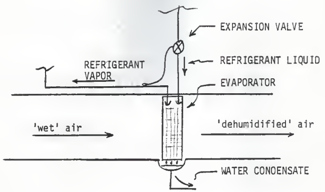
Commercial vapor-compression refrigeration systems utilized for the dehumidification of air generally fall into one of two categories: DX-coil systems, and chilled-water systems. Chilled-water systems are generally favored over DX-coil systems for large, complex multi-coil operations while DX-coil systems are used extensively for the smaller, single coil operations. For either type of system, the basic schematic illustrated in Figure 2.3 is appropriate.

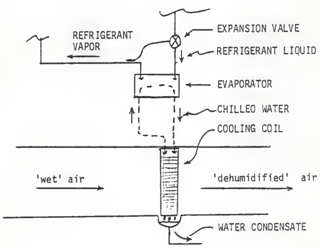
DX-Coil Systems

With DX-coil systems, the air stream to be processed is cooled directly by contact with the evaporator (Figure 2.6), which is maintained at a temperature lower than the final desired temperature of the air. Thus, the energy removed from the air stream is utilized directly to evaporate the refrigerant.

Chilled-Water Systems

With chilled-water systems, the low temperature of the evaporator is utilized to cool a supply of water. This water is then circulated to another coil(s) to absorb the energy necessary for the cooling/dehumidification of the air stream (Figure 2.7). Thus, the energy removed from the air stream is first utilized to heat the water supply which is then used to cause the vaporization of the refrigerant in the evaporator.


EXPANSION VALVE
REFRIGERANT VAPOR
REFRIGERANT LIQUID
EVAPORATOR
'wet' air
'dehumidified' air
WATER CONDENSATE


EXPANSION VALVE
REFRIGERANT VAPOR
REFRIGERANT LIQUID
EVAPORATOR
CHILLED WATER
COOLING COIL
'wet' air
'dehumidified' air
WATER CONDENSATE

CHAPTER III

VAPOR-COMPRESSION SYSTEM MODELING - FULL LOAD OPERATION

In determining the energy requirements of a refrigeration system, the type of model utilized is dependent on how the system loading is varied. If the loading is dynamic in nature, then the equations which describe the interaction of the various components will generally include differential equations. If, however, the loading is under relatively steady-state conditions, then the equations describing system interaction will be more algebraic in nature.

The objective of this study is to develop a model which may be utilized to predict the overall energy requirements of reciprocating vapor-compression refrigeration/dehumidification systems when subjected to variable loading conditions. In general, the operational dynamic time constants of a dehumidification system are quite short compared to those of typical variable dehumidification loads. Thus, the assumption of steady-state loading is valid provided the variable load is broken down into sections of time spent at each loading level for the model to consider individually. This was the approach utilized for the modeling of the refrigeration/dehumidification systems in this study.

3.1 System Modeling - Full Load Operation

As discussed earlier, the function of the refrigeration system is to transfer energy from a relatively low temperature medium to another, usually higher temperature medium. The basic system to be modeled, as illustrated in Figure 3.1, is composed of the following components:

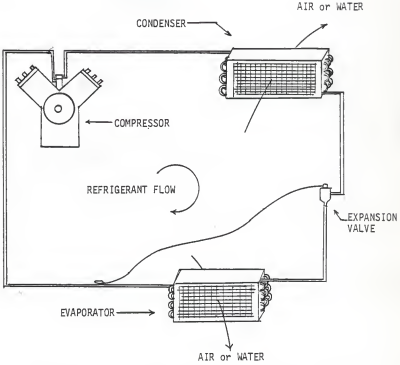


Figure 3.1 Basic Vapor-Compression Refrigeration system.

reciprocating compressor, condenser, evaporator, and expansion valve. A refrigerant, such as R-22, is circulated between the various components, providing the means by which heat transfer may occur.

With steady-state system loading, two different modeling approaches may be utilized to predict overall performance. The first, as presented in ASHRAE's "Procedures for Simulating the Performance of Components and Systems for Energy Calculations" [2], involves the use of energy balance equations for each system component. These equations are coupled together by common variables, such as refrigerant temperatures and mass flow rates. This coupling, in turn, allows the equations to be manipulated (e.g., by use of Successive Substitution or Newton-Raphson iteration) to determine system performance for a given set of conditions, as shown by Stoecker [3], [4]. The use of this approach in system modeling provides excellent agreement with manufacturers equipment catalog data (usually within $\pm$1-1/2%) but requires the recalculation of the constants in the energy balance equations for each change in equipment size (e.g. constants for a 15 ton system will not be appropriate for a 30 ton system).

The second approach, as developed by Hittle [5], Leverenz and Bergan [6], involves the use of an input/output model to directly relate performance to the input conditions for a subsystem (condensing unit, chiller unit, etc.) of the overall system. The equations used in the model are based on the overall performance data of the subsystem, eliminating the need to solve the energy balance equations for each of the subsystems individual components. Thus, the number of variables in the overall model is reduced, simplifying the iteration process. Also, the generalized nature of the model allows for the use of the same equation constants for a wide range of equipment sizes while still providing performance predictions within $\pm$ 3-1/2% of manufacturers catalog data. However, some of the components of a system, such as Direct-Expansion evaporator coils, can not be categorized in a generalized subsystem model because of the wide variation in model size which may be placed in a particular system by the designer. Therefore, an overall system model based on generalized subsystems input/output models may still, depending upon the system, also require one or more of its components to be modeled individually. For this reason, both component and the subsystem modeling procedures will be presented in the following sections.

A summary of the equations for the component, subsystem, and Leverenz-Bergan models is presented in Appendix A.

3.2 Component Modeling – Full Load Operation

Reciprocating Compressor

The primary function of the compressor is to increase the pressure and temperature of the refrigerant gas from the level at which it vaporized to a level which would give the gas the ability to condense back to a liquid (Figure 2.4 — Step 2-3). To accomplish this, a reciprocating compressor utilizes one or more pistons moving in cylinders equipped with suction (inlet) and discharge (outlet) valves which allows the refrigerant gas to be pumped to the required pressure and temperature levels. The amount of refrigerant gas which the compressor handles under these varying operating conditions plays a fundamental role in both the overall load capacity of the system and the energy input requirements of the system.

The primary concerns in modeling a compressor involve the describing of the mass flow rate which the compressor can supply and also the power input for the compressor in providing that flow rate. The compressor mass flow rate is a function of the specific volumes of the gas at both the suction and discharge sides of the compressor. Also, as the pressure difference between the two sides increases, the mass flow rate decreases. This is due to the residual gas left in the cylinder after discharge which must first be expanded back to the suction pressure level before more gas can be drawn into the cylinder.

To make the modeling process more convenient, several simplifications are available. Instead of describing compressor operation in terms of specific volumes and pressures, the operational parameters are often described in terms of saturation temperature (for which a unique saturation pressure and specific volume would exist for the saturated refrigerant vapor). Also, the compressor mass flow rate may be combined with the enthalpy rise of the refrigerant in the evaporator into a term called the compressor capacity. Thus, compressor operation may be described in terms of compressor capacity (Q_e) and power requirements (P), using saturation temperature as the variables (T_e,T_c).

$$Q_e = f(T_e,T_c)$$

$$P = f(T_e,T_c)$$

where Q_e = Compressor Capacity

P = Compressor Power

T_e = Saturated Suction (evaporator) Temperature

T_c = Saturated Discharge (condenser) Temperature

The following equations are typical of those used in refrigerant compressor modeling [2].

$$Q_e = a_1 + a_2T_e + a_3T_e^2 + a_4T_c + a_5T_c^2 + a_6T_eT_c + a_7T_e^2T_c + a_8T_eT_c^2 + a_9T_e^2T_c^2 \quad (3.1)$$

$$P = b_1 + b_2T_e + b_3T_e^2 + b_4T_c + b_5T_c^2 + b_6T_eT_c + b_7T_e^2T_c + b_8T_eT_c^2 + b_9T_e^2T_c^2 \quad (3.2)$$

Condenser

The primary function of the condenser is to transfer energy from the refrigerant gas to another medium, thereby allowing the refrigerant to undergo a phase change from a vapor to a liquid state. The energy rejected includes the compressor capacity (which is also the energy absorbed to the evaporator) and the mechanical work of compression, as given below.

$$Q_c = Q_e + P \quad (3.3)$$

where Q_c = heat rejected at the condenser

Q_e = compressor capacity

P = compressor power

The heat rejection in the condenser, as illustrated in Figure 2.5 (points 4 through 7), approaches that of a constant pressure process. Also, the assumption of a single temperature for the refrigerant through most of the condenser may be justified by the following two reasons: 1) the majority of the energy transfer occurs because of the phase change of the refrigerant, and 2) the increased temperature difference due to the initial desuperheating of the refrigerant is offset by lower heat transfer coefficients [4]. Because the overall heat transfer coefficients do not change appreciably, a constant heat-exchanger effectiveness may be assumed for the condenser. Thus, the heat transfer at the condenser (Q_c) may be described in terms of the condensing temperature (T_c) and the conditions of the medium (air or water) to which the energy is being transferred, as shown below.

$$Q_c = f(T_c, T_m, \dot{m}_m)$$

where Q_c = Condenser Load

T_c = Condensing Temperatures

T_m = Entering medium temperature

$\dot{m}_m$ = Medium mass flow rate

In general form, the rate of heat transfer at the condenser may be modeled by the following equation,

$$Q_c = F_1 + F_2 (T_c - T_m) \tag{3.4}$$

where F_1 and F_2 are constants denoting the physical capacity of the condenser for a preset air or water mass flow rate.

Another condenser characteristic to be included in the model is the amount of refrigerant subcooling which the condenser is capable of supplying. Each degree of refrigerant subcooling increases compressor capacity by approximately 1/2% by reducing the percentage of the liquid flashed during expansion, thereby providing refrigerant to the evaporator in a condition somewhat closer to that of a saturated liquid [7] (Figure 2.4 — points 3 to 4 and Figure 2.5 — points 6 through 8). The percentage increase in compressor capacity is generally presented in manufacturers' catalog data as a function of the temperature difference between the refrigerant and the entering air or water temperature, and is represented by the equation below.

$$\text{subcooling} = F_3 + F_4 (T_c - T_m) + F_5(T_c - T_m)^2 + \ldots \tag{3.5}$$

where subcooling = % increase in compressor capacity due to subcooling

F_3, F_4, F_5 = Constants

For some applications, it may also be necessary to know the temperature of the medium (air or water) leaving the condenser. This may be found from the energy balance equation for the medium, as given below.

$$Q_c = \dot{m}_m C_p(T_{m,out} - T_{m,in}) \tag{3.6}$$

where C_p = specific heat

$T_{m,out}$ = leaving medium temperature

$T_{m,in}$ = entering medium temperature

Evaporators

The primary function of the evaporator is to transfer energy from the desired medium (e.g. air or water) into the refrigerant, thereby allowing the refrigerant to undergo a phase change from a liquid to a vapor state.

As with the condenser, the energy transfer in the evaporator approaches that of a constant refrigerant pressure-temperature process. However, unlike the condenser, the assumption of a constant heat-exchanger effectiveness is not valid due to variations in the overall heat transfer coefficient (which are caused by changes in the boiling heat transfer coefficient with variations in evaporator load) [4]. Instead, a higher order polynomial must be used for the evaporator as compared to the condenser. Thus, heat transfer at the evaporator may be described in terms of saturated evaporating temperature (T_e) and the conditions of the medium (air or water) from which the energy is being transferred, as shown below.

$$Q_e = f(T_e, \dot{m}_m, h_m)$$

where Q_e = Evaporator load

T_e = Saturated Evaporating Temperature

$\dot{m}_m$ = Medium mass flow rate

h_m = Entering medium enthatlpy

For modeling purposes, the evaporators are divided into two categories: DX-coil and Chilled water.

DX-coil. Direct Expansion Coils, as discussed in Chapter 2, utilize direct contact between the air stream and the evaporator to provide the cooling/dehumidification desired for the air. Because the energy transfer from the air stream involves both latent and sensible energies, the heat transfer rate should be described in terms of entering air enthalpy and not dry-bulb temperature. For modeling convenience, entering air wet-bulb temperature is generally substituted for the entering air enthalpy in the energy balance equations. Thus, DX-coil evaporator operation may be described by the equations [2]

$$Q_e = CF*AREA*[C_1 + C_2T_e + C_3T_e^2 + C_4T_{wb} + C_5T_{wb}^2 + C_6T_eT_{wb} + C_7T_e^2T_{wb} \\ C_8T_eT_{wb}^2 + C_9T_e^2T_{wb}^2] \tag{3.7}$$

$$CF = C_{10} + C_{11}\ fpm + C_{12}\ fpm^2 \tag{3.8}$$

where Q_e = Evaporator load

T_e = Saturated evaporating temperature

T_{wb} = Entering air wet-bulb temperature

AREA = Face area of coil

CF = Correction factor for air flow rate

fpm = Air velocity to coil

C_1-C_{12} = Constants

Chilled-water. Water chilling evaporators, as discussed in Chapter 2, are used to cool a water supply which is circulated to other coils to absorb the cooling/dehumidification load. Chilled water evaporator performance may be described by the equations [2].

$$Q_e = d_1 + d_2(GPM) + d_3(GPM)^2 + d_4(dT) + d_5(dT)^2 + d_6(GPM)(dT) \\ + d_7(GPM)^2(dT) + d_8(GPM)(dT)^2 + d_9(GPM)^2(dT)^2 \tag{3.9}$$

dT = (inlet water temperature)-(saturated evaporating temperature) (3.10)

where Q_e = Evaporator load

GPM = Water flow rate

$d_1 \sim d_9$ = constants

Expansion Valve

The function of the expansion valve is to throttle the flow of refrigerant to the evaporator and decrease the pressure and temperature of the liquid refrigerant to a level at which it will have the ability to vaporize easily. This is illustrated in Figure 2.5 as process 7-8. For this throttling process, very little heat transfer occurs at the expansion device compared to the rest of the system. Therefore, an equation describing the process is not necessary, provided that the valve was properly sized to provide the evaporator with a good supply of liquid refrigerant for a wide range of operating conditions [4].

Miscellaneous

Refrigerant Piping Pressure Drops. With the flow of refrigerant in pipes between the primary components of a refrigeration system, it is inevitable that some pressure drop will occur in the refrigerant. This process is illustrated in Figure 2.5 as process 1-2 and 3-4. The effect of these pressure drops is for the compressor to operate with a suction pressure lower than that at the evaporator exit and a discharge pressure higher than that at the condenser inlet. Because the compressor is modeled in terms of saturation temperatures (and therefore saturation pressures), each pressure drop can be expressed in terms of an Equivalent Temperature Change [8]. Thus, the compressor suction temperature would be the evaporator temperature minus the equivalent temperature change of the suction line and the discharge temperature would be the condenser temperature plus the equivalent temperature change for the discharge line.

The actual magnitude of the pressure drop depends on the diameter and length of the piping and the flow rate of the refrigerant. Therefore, the effect of piping pressure drops on system performance may vary widely from one application to the next for a particular system capacity size. Generally, systems with components spaced closely together would not appear to suffer appreciably from piping pressure drops and probably would not require the use of equivalent temperatures in modeling. However, applications which require long refrigerant lines between system components must account for pressure drops when predicting system performance under various operating conditions.

3.3. Subsystem Modeling - Full Load Operation

Many of the refrigeration components on the market (e.g. compressors, condensers, and evaporators) are available on an individual basis or as prepackaged units which combine one or more of the units together. This packaging of components into a subsystem unit allows the manufacturer to optimize the various components around each compressor. For this case, it is possible to generalize the overall performance of such units by a set of algebraic equations without the need to model each of the individual refrigeration components, as performed by Leverenz and Bergan [6] for a water chiller. Because of the versatility of the Leverenz-Bergan Model, it was used as a basis for the subsystem modeling in this study.

A brief summary of the Leveren-Bergan Model is presented next, followed by the subsystem model.

Leverenz-Bergan Chiller Model

The performance of a water chiller system depends strongly on the pressure-temperature conditions of the refrigerant at the compressor.

These conditions, in turn, are closely tied to the water temperatures at both the evaporator and the condenser. Because of this dependency, system capacity and power requirements may be described in terms of leaving evaporator water temperature (T_{cw}) and entering condensing water temperature (T_{cond}) instead of refrigerant evaporating (T_e) and condensing (T_c) temperatures. Also, this dependency eliminates the need to directly account for variations in water flow rates for the model (the secondary support systems which determine T_{cw} and T_{cond}, such as cooling coil and cooling tower models, are, however, dependent on the water flow rates; as will be discussed in Chapter 5).

The model utilizes a set of quadratic equations to directly relate the full load capacity and power requirements of the system to the two water temperatures (T_{cw} and T_{cond}). These equations were developed utilizing normalized variables based on a typical operational condition of the system, T_{cw} = 7.8 °C (46 °F) and T_{cond} = 29.4 °C (85 °F), and the capacity and power consumed at these temperatures. Using these conditions, Leverenz and Bergan showed that the model is capable of predicting the full load capacity and power requirements of a chiller system within ± 3% of manufacturers catalog data for the normal range of condenser and evaporator water temperatures typically encountered in actual operation. For the model, only the tabulated capacity and power requirements of the chiller at the above mentioned typical operational condition is required as input to indicate system size. The rest of the modeling constants are appropriate for all chillers regardless of size.

The chiller model was validated by Leverenz and Bergan against performance data of a 20-ton* reciprocating chiller operating in an

---

* A ton of refrigeration is equal to 12,000 BTU per hour of operation.

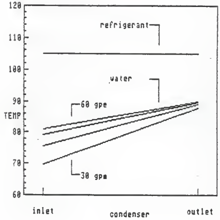
experimental facility designated to simulate a conventional four-zone HVAC system. The difference between the model and measured chiller performance was less than 5.4% for all operational points. However, it should be noted (for reasons to be discussed later) that chiller was operated only at relatively high water flow rates.

Subsystem Model

Four basic subsystem component groupings were examined:

1. Compressor
2. Compressor-water cooled condenser (water cooled condensing unit)
3. Compressor-chilled water evaporator (compressor-chiller unit)
4. Compressor-chilled water evaporator-water cooled condenser (chiller unit).

Manufacturers catalog data for a subsystem unit is generally presented in terms of refrigerant temperatures and/or water temperatures (entering or leaving), with the water flow rates varied to achieve a 10 ^{o}F water temperature rise (or drop) for each operating condition. For this situation, the use of either entering or leaving water temperatures in the subsystem model is appropriate in reproducing catalog data, as performed by Leverenz and Bergan. However, as can be seen in Figure 3.2, the inlet water temperature varies significantly with a change in water flow rate while the outlet water temperature remains relatively stable. Because the subsystem model (and the Leverenz-Bergan model) do not take into account variations in water flow rates, leaving water temperatures must always be utilized instead of entering water temperatures, else substantial errors in performance predictions may result if a water flow rate is used which does not provide a 10 ^{o}F water temperature swing at full load conditions.



Plot of condenser refrigerant and water temperatures at the inlet and outlet of a 15-ton water cooled condensing unit @ evap.temp = 40 F , cond.temp = 105 F

Figure 3.2 : Variation of water temperature with flow rates for a water cooled condensing unit.

For each subsystem, a typical operational condition was selected for use as the basis for the normalized variables in the model. Constants were then derived (using a least squares fit) for each subsystem using the catalog data* for a 15 or 20 ton unit (Table 3.A). Utilizing these constants, the performance of each subsystem was checked against catalog data for the original unit size, a 30-ton unit, and a 80-ton unit (Appendix B). This comparison showed all of the performance predictions (e.g. the capacity and power requirements) for each subsystem to be within $\pm$ 3-1/2% of the catalog valves, regardless of system size (Appendix B). It should be noted, however, that because of the variations which exist in component design between the different manufacturers, some subsystem units may not fall within the above mentioned modeling accuracy of 3-1/2%. For this situation, a new set of modeling constants may be required for the unit(s) in question. For most subsystem units, however, the parameters presented in Table 3.A are appropriate.

Summary

The use of a subsystem model for predicting overall refrigeration performance reduces the number of components requiring modeling, simplifying the iteration process. Also, the generalized nature of the subsystem model minimizes the amount of input required from catalog data to describe subsystem size and operational characteristics, while still providing reasonable full load performance estimates. The use of the subsystem model is the subject of Chapter 5.

---

*Catalog data based on 16°F of subcooling for those subsystems without condensers, actual subcooling for those with condensers. 15°F of superheat for all subsystems.

SUBSYSTEM	Tel,D	Tcl,D	TEMPERATURE RANGE USED IN PARAMETER SELECTION Tel	TEMPERATURE RANGE USED IN PARAMETER SELECTION Tcl	Tratio	A1	A2	A3	B1	B2	B3
compressor unit	4.44 C (40) F	40.55 C (105) F	-1.11~10 C (30~50) F	29.4~51.67 C (85~125) F	2.9	.9973	-.03694	.0004335	-.4141	1.7717	-.3694
water-cooled condensing unit	4.44 C (40) F	35 C (95) F	-1.11~10 C (30~50) F	29.4~40.56 C (85~105) F	2.9	.9982	-.03952	.004931	-.2244	1.5438	-.3219
compressor-chiller unit	7.78 C (46) F	40.55 C (105) F	4.44~10 C (40~50) F	29.4~51.67 C (85~125) F	2.9	.9982	-.02990	.0001711	-.2484	1.6600	-.4105
chiller unit	7.78 C (46) F	35 C (95) F	4.44~10 C (40~50) F	29.4~40.56 C (85~105) F	2.9	.9978	-.02943	.0001472	-.1119	1.5815	-.4702

TABLE 3. A SUBSYSTEM MODELING PARAMETERS

CHAPTER IV

VAPOR COMPRESSION SYSTEM MODELING - PARTIAL LOAD OPERATION

Typically, refrigeration systems are selected for an installation based on the ability of the system to handle a particular operational (design) load. This load, however, generally occurs, on an annual basis, as only a small percentage of the system operational time. For the majority of the time, the system encounters a load condition which is far less severe than the design load. Thus, the manner in which a system handles these off-design conditions has a strong influence on the system annual energy consumption and must be included in the overall system model.

The natural response for a refrigeration system to a drop in evaporator load is to decrease the pressure and temperature of the refrigerant in the evaporator. These lower conditions, in turn, reduce the compessor capacity and power requirements until the new load is matched. However, the lower temperature of the evaporator results in over-cooling of the air (or water) and problems such as coil frost formation (or chilled water freeze up) may occur. Therefore, a method to control the capacity of the system is usually incorporated into initial system design.

4.1 Refrigeration System Capacity Control

Capacity control for a refrigeration system may be achieved by modifying the performance characteristics of one or more of the components of the system (e.g. compressor, condenser, and/or evaporator) to match the varying load. Because the compressor is the major power consuming component of a system, control methods which reduce its capacity and therefore the power requirements are by far the most widely utilized in industry. Condenser and evaporator control methods, on the other hand, are not as widely utilized due to their limited abilities to reduce system power requirements with a reduced load*. For this reason, only compressor capacity control methods were considered in this study as the primary means of system capacity control.

Compressor Capacity Control

Compressor capacity control may be obtained by using one of the following methods: cycling, hot gas bypass, back-pressure regulator, and cylinder unloading. All of these methods, in effect, reduce the amount of refrigerant vapor supplied by the compressor to the condenser. This reduced supply, in turn, results in less liquid refrigerant at the evaporator and a lower system capacity. A brief explanation of each of the methods follows.

With cycling, the compressor is started and stopped as needed to provide the necessary refrigerant flow to match the evaporator load. Because the compressor is not operating continuously, substantial power consumption savings are possible. However, frequent cycling of a system shortens the life of the compressor, motor, and starter system. For this reason, cycling is generally used only with small durable compressors and also with systems when frequent cycling of a particular compressor is not likely to occur.

---

* Proper expansion valve operation requires a minimum pressure differential across the valve. To insure this, condensing temperature is generally not allowed to fall below a preset level. This is usually accomplished by condenser fan cycling or dampers and also by not allowing the entering condenser water to fall below a minimum temperature.

With hot gas bypass, refrigerant from the compressor bypasses the condenser through a pressure reduction valve which feeds the evaporator (or the compressor suction line). This method generally provides for precise capacity control. However, the refrigerant which the compressor must process remains close to that of full loading, resulting in near full load power consumption regardless of the actual evaporator load.

With a back-pressure regulator, the flow of refrigerant between the evaporator and the compressor is throttled, resulting in the compressor operating at a lower suction pressure-temperature level than the evaporator. The lower suction pressure-temperature level causes a reduction in the mass flow rate to the compressor and, therefore, a reduced power requirement. The lower suction level, however, does result in a lower coefficient of performance (COP) for the system.

The most widely utilized method of compressor capacity control is the unloading of one or more cylinders in a multi-cylinder compressor by holding the suction valve of the cylinder(s) open throughout the entire motion of the piston(s). With the unloading of a cylinder, a step reduction in compressor refrigerant mass flow rate occurs, resulting in lower compressor capacity and power requirements. While some losses occur because of friction and the pumping of refrigerant into and out of the unused cylinders, the resulting COP of a cylinder unloading system is still greater than that of a back-pressure regulated system, as shown by Garland [9]. For the above reasons, cylinder unloading was chosen as the primary means of system capacity control for this study.

4.2 Cylinder Unloading - Component Model

As discussed in section 4.1, a step reduction occurs in both capacity and power requirements with the unloading of each cylinder. With system partial load operation, it is unlikely for the load to perfectly match the step reduction in capacity available with the unloading of a cylinder. Instead, the compressor would be required to fluctuate between an unloading step of under capacity and one of over capacity to enable it to match the total overall load. For this situation, Hittle [5] proposed the use of a quadratic equation fit to the unloading step data to estimate compressor performance (equation 4.1). Also, a linear equation may be used to represent the condition of compressor cycling between the last unloading position and the off-position (equation 4.2). A plot of these equations is presented in Figure 4.1.

$$FFL = C_1 + C_2(PLR) + C_3(PLR)^2 \qquad (4.1)$$

$$FFL = C_4(PLR) \qquad (4.2)$$

where C_1 = 0.1456

C_2 = .9555

C_3 = -0.1048

C_4 = Constant dependent on number of cylinders allowed to unload

PLR = Ratio of actual capacity to the full load capacity (also the ratio of number of cylinders loaded to the total number of cylinders

FFL = Ratio of actual power to the full load power

Because of the normalized nature of the variables, the C_1 through C_3 constants should be appropriate for all compressors while the C_4 constant is dependent only on the PLR and FFL which occurs at the last cylinder unloading step before shut down (example below).

Example

For a 4 cylinder compressor which can unload down to 1 cylinder,

$$PLR = 1/4$$

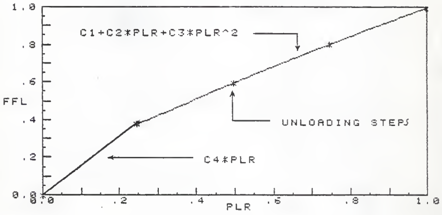


Figure 4.1 Plot of Fraction of Full Load (FFL) versus Partial Load Ratio (PLR) for a four-cylinder compressor.

and from Equation 4.1,

$$FFL = 0.378$$

therefore $C_4 = \quad = 1.512$ for this compressor.

Equation Summary

$$PLR = (Q_{e_{\text{partial load}}})/(Q_{e_{\text{full load}}})$$

If PLR greater than that of the last unloading step

Then $\quad FFL = C_1 + C_2(PLR) + C_3(PLR)^2$

Else $\quad FFL = C_4(PLR)$

where $FFL = (P_{\text{partial load}})/(P_{\text{full load}})$

4.3 Cylinder Unloading - Subsystem Model

The subsystems modeled in the previous chapter are all composed of refrigerant components grouped around a compressor. As such, the capacity and power requirements of each subsystem will decrease with the unloading of each compressor cylinder. For this situation, the equations outlined in section 4.2 (Equations 4.1 and 4.2) are applicable for each subsystem. However, the lower compressor capacity resulting from the unloading of a cylinder requires a reduction in the temperature difference between the refrigerant and the water at the evaporator (or condenser). For a given leaving water temperature, this requires a higher evaporator (or lower condenser) refrigerant temperature. This, in turn, reduces the losses in compressor efficiency for the subsystem which normally occurs with cylinder unloading. To account for this in the subsystem model*, a new set of constants for equation 4.1 were developed for each type of subsystem.

*Because the compressor in the component and the compressor subsystem models are described only in terms of refrigerant temperatures, the constants developed by Hittle are adequate for these two situations.

These constants (Table 4.D) and the associated development procedures are presented on the following pages.

4.3a Subsystem Cylinder Unloading Constants

In developing a set of constants for each type of subsystem, a method of determining net subsystem efficiency at partial loading was required. To do this, each subsystem* was described using both the subsystem and the component models. Then, the full load output of both models and the partial load output of the component model (using Hittle's constants for Equation 4.1) were calculated for a variety of positions through the normal operational temperature range of the subsystem. From this, the PLR's and FFL's required by each subsystem model to generate the partial load output of the component models was found. These data were then used to generate the constants required by each subsystem for use in the fraction of full load equation (Equation 4.1).

Component Model

The subsystems to be modeled are:

1) compressor-water cooled condenser (water-cooled condensing unit)
2) compressor - chilled water evaporator (compressor-chiller unit)
3) compressor-chilled water evaporator-water cooled condenser (chiller unit)

The subsystems modeled were Trane 15-ton units (RWUB C15M, CCAB C15M, and CGWB C15M) which, like most available, use the same compressor (and also condenser and evaporator where applicable) for all three units. Because of this, the problem of breaking such subsystems down into individual components was greatly simplified. The performance data of each component

*15-ton subsystem units from Trane Air Conditioning were utilized in this process.

COMPRESSOR

```
Qe=subcool*(A1+A2*Te+A3*Te^2+A4*Tc+A5*Tc^2+A6*Te*Tc+A7*Te^2*Tc+A8*Te*Tc^2+A9*Te^2*Tc^2)
P=B1+B2*Te+B3*Te^2+B4*Tc+B5*Tc^2+B6*Te*Tc+B7*Te^2*Tc+B8*Te*Tc^2+B9*Te^2*Tc^2
```

```
A1=  17.8676431628            B1= -13.61523303
A2= -8.68496697674E-2         B2=  1.321157156
A3=  7.79868275349E-3         B3= -.02127362189
A4= -.157051968372            B4=  .455646584
A5=  4.15010446512E-4         B5= -.002043495701
A6=  5.4130744186E-3          B6= -.0249864538
A7= -9.55775851163E-5         B7=  .000393825407
A8= -2.76754412093E-5         B8=  1.281939044E-4
A9=  3.79140383256E-7         B9= -1.834600845E-6
```

WATER-COOLED CONDENSER

```
Qc=Qe+P
Qc=Z*(DT)
DT=Tc-Tw_in
Z=C1+C2*(GPM)+C3*(GPM^2)
subcool=D1+D2*(DT)+D3*(DT^2)
```

```
C1=.09348214359                D1=.98716936174
C2=2.25321428658E-2            D2=.00461114887
C3=-1.4833333318E-4            D3=-3.2717021235E-5
```

CHILLED-WATER EVAPORATOR

```
DT=Tw_in-Te
Qe=F1+F2*GPM+F3*GPM^2+F4*DT+F5*DT^2+F6*GPM*DT+F7*GPM^2*DT+F8*GPM*DT^2+F9*GPM^2*DT^2
```

```
F1=44.0665489
F2=-2.07091539
F3=.0241712729
F4=-4.22431777
F5=.11196747
F6=.231325698
F7=-.00256336483
F8=-.0053368574
F9=.00006085379
```

TABLE 4.A 15-ton refrigeration system component equations and constants.

...out..		....in...		.....refrig.....			.......................full load							
Tel	Tcl	Tel	Tcl	Te	Tc	subcl	Qe_cat	Qe_pred	%	P_cat	P_pred	%	GPMc	GPMe
40.0	85.0	50.0	75.0	31.3	98.5	16.5	15.10	15.08	-.12	14.40	14.41	.06	46.03	36.20
40.0	90.0	50.0	80.0	31.6	103.3	16.4	14.70	14.68	-.15	14.90	14.89	-.09	45.39	35.23
40.0	95.0	50.0	85.0	31.8	108.1	16.3	14.20	14.23	.21	15.30	15.38	.51	44.65	34.15
40.0	100.0	50.0	90.0	32.1	113.0	16.2	13.80	13.80	.01	15.80	15.89	.60	43.97	33.12
40.0	105.0	50.0	95.0	32.4	117.8	16.1	13.40	13.36	-.27	16.40	16.43	.16	43.28	32.07
42.0	85.0	52.0	75.0	32.9	98.8	16.8	15.60	15.61	.09	14.60	14.58	-.14	47.43	37.47
42.0	90.0	52.0	80.0	33.2	103.7	16.7	15.20	15.19	-.07	15.10	15.08	-.10	46.75	36.45
42.0	95.0	52.0	84.9	33.4	108.4	16.5	14.70	14.74	.26	15.60	15.59	-.07	46.01	35.37
42.0	100.0	52.0	90.0	33.8	113.3	16.4	14.30	14.30	-.02	16.10	16.13	.17	45.32	34.31
42.0	105.0	52.0	95.0	34.1	118.1	16.3	13.90	13.85	-.38	16.60	16.68	.49	44.62	33.23
44.0	85.0	54.0	75.0	34.5	99.2	17.0	16.10	16.15	.30	14.80	14.76	-.27	48.83	38.75
44.0	90.0	54.0	80.0	34.8	104.1	16.9	15.70	15.71	.07	15.30	15.29	-.10	48.14	37.71
44.0	95.0	54.0	85.0	35.2	108.9	16.8	15.30	15.27	-.22	15.80	15.83	.17	47.44	36.64
44.0	100.0	54.0	90.0	35.4	113.7	16.7	14.80	14.79	-.04	16.30	16.37	.42	46.68	35.51
44.0	105.0	54.0	95.0	35.8	118.5	16.5	14.40	14.36	-.29	16.90	16.95.	.32	46.04	34.46
46.0	85.0	56.0	75.0	36.2	99.6	17.3	16.60	16.69	.56	15.00	14.94	-.39	50.26	40.06
46.0	90.0	56.0	80.0	36.5	104.4	17.1	16.20	16.25	.33	15.50	15.48	-.13	49.57	39.01
46.0	95.0	56.0	85.0	36.8	109.2	17.0	15.80	15.80	-.03	16.00	16.04	.27	48.86	37.91
46.0	100.0	56.0	90.0	37.1	114.1	16.9	15.30	15.33	.18	16.60	16.62	.14	48.13	36.79
46.0	105.0	56.0	95.0	37.4	118.8	16.8	14.90	14.86	-.25	17.10	17.21	.64	47.41	35.67
48.0	85.0	58.0	75.0	37.8	100.0	17.5	17.20	17.25	.29	15.10	15.13	.17	51.72	41.40
48.0	90.0	58.0	80.0	38.1	104.8	17.4	16.80	16.79	-.07	15.70	15.69	-.04	51.00	40.29
48.0	95.0	58.0	85.0	38.4	109.6	17.3	16.30	16.33	.16	16.20	16.27	.43	50.29	39.18
48.0	100.0	58.0	90.0	38.8	114.5	17.1	15.80	15.85	.29	16.80	16.87	.41	49.54	38.03
48.0	105.0	58.0	95.0	39.1	119.2	17.0	15.40	15.37	-.22.	17.40	17.47.	.43	48.81	36.88
50.0	85.0	60.0	75.0	39.4	100.4	17.8	17.70	17.79	.50	15.30	15.30	.03	53.14	42.69
50.0	90.0	60.0	80.0	39.7	105.2	17.6	17.30	17.32	.13	15.90	15.88	-.10	52.41	41.58
50.0	95.0	60.0	85.0	40.0	110.0	17.5	16.80	16.84	.23	16.40	16.49	.54	51.63	40.41
50.0	100.0	60.0	90.0	40.4	114.8	17.4	16.40	16.37	-.16	17.00	17.12	.69	50.98	39.30
50.0	105.0	60.0	95.0	40.7	119.6	17.3	15.90	15.88	-.12	17.60	17.74	.80	50.23	38.11

WHERE

Tel = EVAPORATOR WATER TEMPERATURES
Tcl = CONDENSER WATER TEMPERATURES
Te = EVAPORATOR REFRIGERANT TEMP.
Tc = CONDENSER REFRIGERANT TEMP.
GPMe = EVAPORATOR WATER FLOW RATE
GPMc = CONDENSER WATER FLOW RATE
subcl = DEGREES OF SUBCOOLING

Qe_cat = CATALOG SYSTEM CAPACITY
Qe_pred= PREDICTED SYSTEM CAPACITY
P_cat = CATALOG SYSTEM POWER
P_pred = PREDICTED SYSTEM POWER

TABLE 4.8 Comparision of predicted performance versus catalog data for the 15-ton chiller unit component model.

was directly available for the compressor (CVWB unit) and the condenser (RWVB unit), while the data for the evaporator was found by comparing the compressor (CUWB unit) and the compressor-chilled water evaporator (CCAB unit) data to find all of the necessary variables (Q_e, GPM, T_e, $T_{water\ in}$). Using these data as a base, a least-squares fit was performed for each equation. The resulting equations (Table 4.A), predict performance values well within 1-1/2% of catalog values. As a farther check, the equations were used to predict the performance of the chiller unit through its entire operational temperature range (Table 4.B). Examination of these results showed all of the predicted capacity values (column 9) to be within 0.56% (column 10) of the catalog capacity values (column 8). Also, the predicted power consumption (column 12) was within 0.80% (column 13) of the catalog power consumption (column 11) for all of the operational temperatures. Thus, these equations should provide reasonable estimates of performance for all three subsystems.

Subsystem Model

The Trane 15-ton subsystem units were originally used to derive the constants for the subsystem models. Thus, the constants represented in Table 3.A are appropriate for the subsystems being examined in this chapter. The base capacity and power is presented below for each subsystem.

	T_{el}, D	T_{cl}, D	CAP, D	FLP, D
water-cooled condensing unit	4.44 °C (40 °F)	35 °C (95 °F)	59.1 kW (16.8 tons)	16.5 kW
compressor-chiller unit	7.78 °C (46 °F)	40.56 °C (105 °F)	56.6 kW (16.1 tons)	15.6 kW
chiller unit	7.78 °C (46 °F)	35 °C (95 °F)	55.5 kW (15.8 tons)	16 kW

Table 4.C

15-Ton Subsystem Base Capacity and Power Requirements

Model Comparison

As a basis for comparison, the operational temperature range of each subsystem was assumed to be the same as those utilized in the initial calculation of the subsystem modeling constants. These temperature ranges were broken down into the following intervals:

Water Cooled Condenser Unit		Compressor-Chiller Unit		Chiller Unit	
T_e	T_{cl}	T_{el}	T_c	T_{el}	T_{cl}
30 F	85 F	40 F	85 F	40 F	85 F
40 F	95 F	45 F	95 F	45 F	90 F
50 F	105 F	50 F	105 F	50 F	95 F
	115 F		115 F		100 F
	125 F		125 F		105 F

For each interval, partial load performance was calculated utilizing the component model at 100, 80, 60, 40, and 20 percent of cylinder loading. Also, the full load capacity and power predictions of the subsystem models were found. From these data, the PLR's and FFL's required by the subsystem models to provide the same partial load performance values as the component models were calculated using the following equations:

$$PLR_{subsystem} = \text{(component model partial load capacity)/} \\ \text{(subsystem model full load capacity)}$$

$$FFL_{subsystem} = \text{(component model partial load power)/} \\ \text{(subsystem model full load power)}$$

New constants for equation 4.1 were derived from these data for each of the subsystems and are presented in Table 4.D. Also, plots of the PLR and FFL data points are illustrated in Figures 4.2 through 4.8.

	Compressor	Water Cooled Condensing Unit	Compressor-Chiller Unit	Chiller Unit
C_1 =	0.1456	0.129	0.1545	0.1334
C_2 =	0.9555	0.8849	0.8323	0.7439
C_3 =	-0.1048	-0.01302	0.01672	0.1209

Table 4.D

Subsystem Model Cylinder Unloading Constants (Equation 4.1)

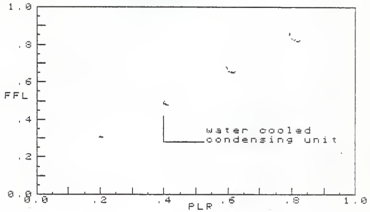


Figure 4.2 Plot of Fraction of Full Load (FFL) versus Partial Load Ratio (PLR) for the Water Cooled Condensing Unit of Section 4.3.

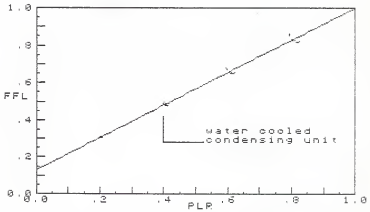


Figure 4.3 Least Squares Fit of FFL versus PLR data for the Water Cooled Condensing Unit of Section 4.3.

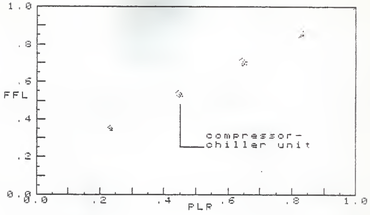


Figure 4.4 Plot of Fraction of Full Load (FFL) versus Partial Load Ratio (PLR) for the Compressor-Chiller Unit of Section 4.3.

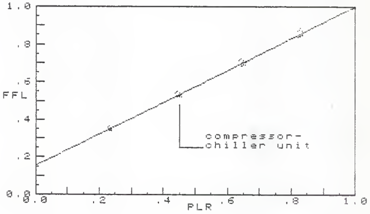


Figure 4.5 Least Squares Fit of FFL versus PLR data for the Compressor-Chiller Unit of Section 4.3.

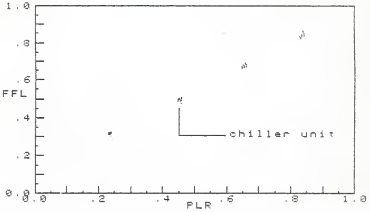


Figure 4.6 Plot of Fraction of Full Load (FFL) versus Partial Load Ratio (PLR) for the Chiller Unit of Section 4.3.

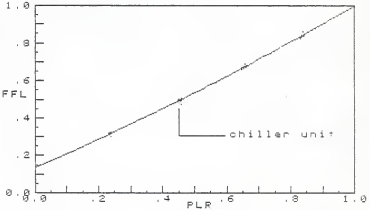


Figure 4.7 Least Squares Fit of FFL versus PLR data for the Chiller Unit of Section 4.3.

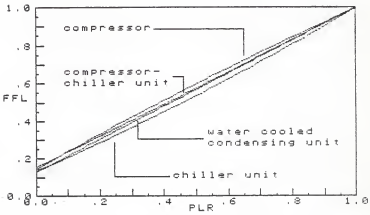


Figure 4.8 Plot of Partial Load Performance (FFL vs. PLR) for a Compressor, Compressor-Chiller Unit, Water Cooled Condensing Unit, and a Chiller Unit.

CHAPTER V

APPLICATIONS

5.1 Secondary Support Models

The performance of a refrigeration system is strongly influenced by the operational characteristics of its support systems (cooling towers, cooling coil controls, etc.). Because of this, models of these support systems must be included in the overall model used to predict refrigeration system performance.

5.1A Cooling Tower Model

When recirculated water is used in liquid-cooled condensers, it is generally cooled by being routed through a cooling tower. At the cooling tower, some of the water is allowed to evaporate into the surrounding air. The energy required for evaporation is drawn from the unevaporated water, resulting in a drop in the temperature of the remaining liquid water. The cooled water is then returned to the condenser to absorb more energy from the refrigerant undergoing condensation.

The model for counterflow cooling tower performance developed by Webb and Villacres [10] provides excellent correlation with manufacturers data and was used as the basis for the cooling tower model in this study.

The Webb-Villacres model is capable of handling three operational variations:

1. Fixed air and water flow rates with leaving water temperature allowed to vary with load.

2. Fixed water flow rate and leaving water temperature with a variable air flow rate.

3. Fixed air flow rate and leaving water temperature with a variable water flow rate.

Because most cooling towers are not designed for a significant variation in water flow rate [10], only the first two cooling tower operational cases were included in the overall model. Also, some cooling towers provide a fixed leaving water temperature by fan cycling. To approximate the operational time of these fans in the overall model, the ratio of the required temperature drop to the temperature drop of first operation case was used (Equation 5.1).

$$\%\ \text{fan Operational Time} = [(T_{co} - T_{ci,min})/(T_{co} - T_{ci})]*100 \qquad (5.1)$$

where T_{co} = leaving condenser water temp. (entering cooling tower)

$T_{ci,min}$ = minimum entering condenser water temp. (leaving cooling tower)

T_{ci} = leaving cooling tower water temp. of operational case #1

Pitfall of Webb-Villacres Model - The iteration technique utilized in the Webb-Villacres model was the bisection method (where high and low estimates of a variable are used to determine its actual value). However, because of the nature of the NTU Equation in the model, the use of too low of an initial estimate of the variable may result in the cooling tower model providing an incorrect leaving water temperature or air flow rate (Figure 5.1). To prevent this from occurring, the variable should first be evaluated at the high estimate (T_{ci} equal to T_{co} or tower air flow equal to design air flow) and then have the estimate reduced in small increments

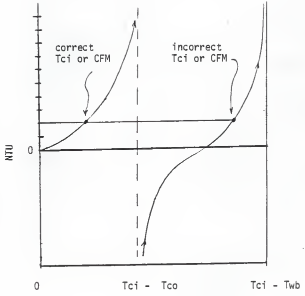


Figure 5.1 Typical plot of NTU versus cooling tower temperature range for a low ambient wet-bulb. (normal operation should ALWAYS be to the left of the discontinuity) .

until a low estimate is available. At this point, the bisection method may be used to find the final value of the variable. This approach, while requiring more iteration steps, was found to always provide the correct solution provided small enough increment steps were used.

5.1B. Cooling Coil Model

Dehumidification by refrigeration requires the removal of energy from the air-water vapor mixture, resulting in a reduction in air temperature and the condensation of the undesired water vapor from the mixture. Cooling coils for this purpose tend to be of multiple-row construction which, in dehumidifying, provide leaving air conditions at or very near that of saturation (Figure 5.2) [11]. These coils, because of the relatively large amount of heat transfer surface available, are able to operate at higher refrigerant and/or chilled water temperatures than coils of less depth for dehumidifying air to a given moisture content level [12]. Thus, coil frost formaton and/or chilled water icing are less likely to occur with 'deep' coils. Also, the higher refrigerant temperature improves compressor efficiency and reduces power consumption. For these reasons, the cooling coil used in this model was assumed to be of a 'deep' multiple-row construction.

As was stated above, 'deep' multi-row cooling coils provide dehumidified air at or very near that of saturaton. To simplify coil modeling, the dehumidified air leaving the coil was assumed to be at saturation. In doing so, the need for equations describing the variation of the leaving air conditions with coil loading was eliminated. It should be noted, however, that this assumption will provide for a slight over estimation of the actual cooling coil load required to provide a given leaving air-moisture content level (Figure 5.3).

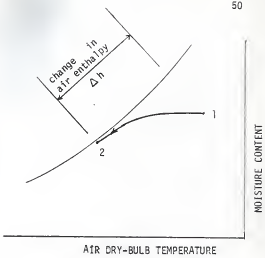


Figure 5.2 Psychrometric chart for dehumidification by a deep, multiple-row cooling coil.

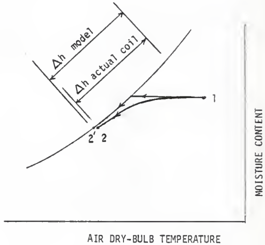


Figure 5.3 Psychrometric chart for dehumidification for the cooling coil model.

Cooling Load

The load to be absorbed at the cooling coil is given by

$$Q_e = \dot{m}_{air}(h_1 - h_2) \tag{5.2}$$

where h_1 = entering air enthalpy

h_2 = leaving air enthalpy

Q_e = coil load

$\dot{m}_{air}$ = mass flow rate of the air

and

$$h = 0.24(T_{db}) + W(1061 + 0.444(T_{db})) \tag{5.3}$$

where h = enthalpy of the air

T_{db} = dry bulb temperature

W = moisture content of air

Air enthalpy, as can be seen in Equation 5.3, is a function of both dry-bulb temperature and the moisture content of the air. However, coil conditions are frequently known in terms of dry-bulb and wet-bulb temperatures. For this situation, air enthalpy may be estimated from the wet-bulb temperature utilizing the subroutines developed by Webb and Villacres [9]. Thus, air enthalpy may be estimated by either of the above methods for most of the circumstances which a cooling coil would normally encounter. The calculation procedure used to determine coil load (Q_e) is outlined below.

Calculation Procedure

1) Calculated entering air enthalpy by either

   a) Eqn. 5.3 (T_{db}, W)

   b) Wet-bulb (T_{db}, T_{wb})

2) If only entering dry-bulb and wet-bulb specified, calculate entering moisture content from Equation 5.3

3) For dehumidification, leaving wet-bulb is approximately equal to leaving dry-bulb, calculate enthalpy based on this wet-bulb
4) From Eqn. 5.3, calculate saturated moisture content of Step #3
5) If saturated moisture content is less than inlet moisture content, then leaving air enthalpy is enthalpy of Step #3
   If saturated moisture content is greater than inlet moisture content, then use inlet moisture content, outlet dry bulb, and Equation 5.3 to calculate leaving air enthalpy
6) Calculate coil load (Q_e) using Equation 5.2

5.2 Generalized Model-Refrigeration/Dehumidification System

For the purposes of this study, four basic component variations of a refrigeration/dehumidification system were examined (Figures 5.4 through 5.7):

1. Compressor unit - DX-coil - air cooled condenser system
2. Water cooled condensing unit - DX-coil - cooling tower system
3. Compressor chiller unit - chilled water coil - air cooled condenser system
4. Chiller unit - chilled water coil - cooling tower system

These systems were all incorporated into a generalized computer program. This program, which is listed in Appendix C, is described below.

5.2A System Controls

DX-coil systems. As outlined in Chapter 3, DX-coil evaporator load is a function of the refrigerant temperature and of the inlet air conditions (and therefore outlet air conditions). For a DX-coil to provide a given cooling/dehumidification load (cooling coil model), the coil must operate with a specific refrigerant temperature (as defined by Eqns. 3.7, 3.8).

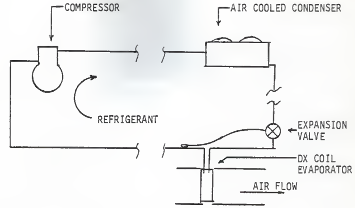


Figure 5.4 Compressor Unit--DX Coil--Air Cooled Condenser system.

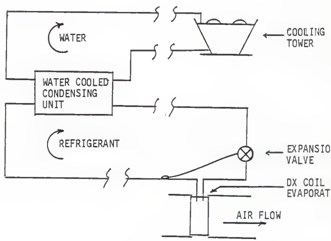


Figure 5.5 Water Cooled Condensing Unit--DX coil--Cooling Tower system.

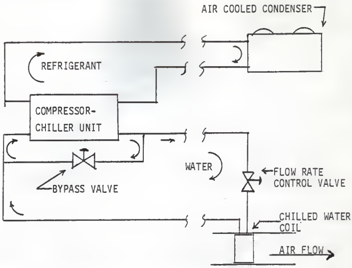


Figure 5.6 Compressor Chiller Unit--Chilled Water Coil--Air Cooled Condenser system.

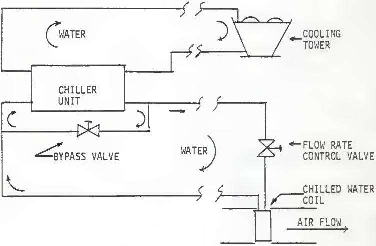


Figure 5.7 Chiller Unit--Chilled Water Coil--Cooling Tower system.

This temperature, in turn, varies with coil loading conditions (e.g. maintaining of fixed evaporator temperature would result in the over cooling/dehumidifying of the air stream for a reduction in the inlet air wet-bulb temperature or air flow rate). For a DX-coil system, the variable evaporator temperature and system capacity required for the desired air outlet conditions may be provided by regulating the cylinder unloading process (such as by a thermostat in the outlet air duct). This was the procedure used in this model.

Chilled-water systems. The schematic used for modeling chilled-water systems is illustrated in Figure 5.8. As with the DX-coil systems, coil temperature of a chilled-water system must be allowed to vary with inlet air conditions to provide the desired levels of cooling/dehumidifying. To accomplish this, a thermostat in the air stream is generally used to control a valve at the coil to regulate the chilled-water flow rate through the coil. Thus, cooling coil chilled-water flow rate will vary directly with the coil load. Most systems, however, maintain a constant chilled-water flow rate at the evaporator. To allow for this, a chilled water-cooling coil bypass circuit is generally included in these systems; with either a bypass circuit at each coil or a single bypass circuit located near the evaporator. The use of these circuits eliminates the need for modeling individual chilled-water coil performance (e.g. coil water flow rates for each loading). Instead, the load absorbed at the coil (from the cooling coil model) and an energy balance at the evaporator (Equation 5.4) may be used to predict evaporator performance.

$$Q_e = \dot{m}_m C_p (T_{cw,r} - T_{cw,s}) \qquad (5.4)$$

where Q_e = evaporator load (cooling coil model)

$\dot{m}$ = water flow rate through evaporator

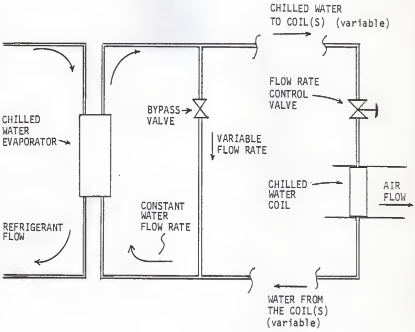


Figure 5.8 Chilled water flow route schematic.

$T_{cw,r}$ = return mixed water temperature

$T_{cw,s}$ = chilled water supply temperature

C_p = specified heat of water

For these systems, cylinder unloading is typically used to match system capacity (and therefore evaporator capacity) to the desired load by providing either a preset supply or return mixed chilled water temperature. Thus, as coil load increases (or decreases), the number of compressor cylinders loaded must increase (or decrease) and the water temperature drop across the evaporator must change (as given by Equation 5.4) for the system to maintain the preset water temperature. For a system with return water temperature control, the response to an increase coil load is to provide colder chilled water. For a system which controls supply water temperature, the return water temperature will increase with coil loading.

Air-cooled condenser systems. As briefly mentioned in Chapter 4, to maintain proper expansion valve operation, the condensing temperature is generally not allowed to fall below a preset level. To accomplish this, the air flow rate across the condenser is generally reduced at low ambient temperatures by either the use of dampers or by cycling the fans. For this study, a minimum condensing temperature was included in the model. A method of estimating fan cycling time, however, is not currently part of the model.

Water-cooled condenser system. As mentioned in the previous section, cooling towers do not operate efficiently at water flow rates other than those of design. Because of this, the water flow rate of the condenser and the cooling tower were assumed to be the same at all operating conditions (e.g. no bypass circuits). However, proper expansion valve operation requires that entering condensing water temperature does not fall below a preset level (as would be the case for low ambient wet-bulb temperatures). For this situation, the leaving cooling tower water temperature is maintained at the minimum entering condensing water temperature by either cooling tower dampers or fan cycling — as outlined in section 5.1.

5.2B. Use of the Generalized Model

The flow chart for the computer program listed in Appendix C is illustrated in Figure 5.9. The use of this model may be divided into four steps:

1. Determining system equipment constants
2. Calculating T_{e1} of subsystem model
3. Iterating for T_{c1} of subsystem model
4. Performance output

System equipment constants. Constants for 15-ton systems of all four of the component variations handled by this model are included in the constants subroutine of the program. The procedures for finding the constants for the appropriate system is as follows:

a) subsystem unit — input catalog data for capacity and power at the base temperatures as outlined in Chapter 3.
   — input appropriate subsystem constants and cycling PLR as outlined in Chapters 3 and 4.
b) DX-coil — input constants for equation fitting of Equations 3.7, 3.8 (constants given for 6-row Aerofin coils)
   — input face area of coil.
c) Chilled-water coil — select either return or supply water temperature control
   — input water temperature chosen for the above
   — input chilled-water flow rate

d) Air-cooled condensers — input constants for equation fit of Equations 3.4, 3.5 (constants given for Equation 3.5 for all Trane air-cooled condensers).

— input minimum condensing temperature

e) Cooling towers — input Webb-Villacres cooling tower constants (program of Appendix G).

— input cooling tower design air and water flow rates

— input minimum condensing water temperature

Calculating T_{el}. From the known inlet air conditions at the coil and the desired outlet conditions, the cooling coil load may be calculated (Q_e — cooling coil model). For a DX-coil, equation 3.7 and 3.8 may be solved directly utilizing the load and inlet conditions to find T_e. For a chilled-water system, equation 5.4 may be solved directly for the leaving chilled water temperature (T_{cw}). The appropriate temperature is then used as the T_{el} variable in the subsystem model.

Iterating T_{cl}. Unlike the T_{el} variable, the T_{cl} must be found through an iteration process. First, an estimate of the T_{cl} variable is made. Utilizing this estimate of T_{cl} and the known T_{el}, the full load capacity and power of the system is calculated (subsystem model). Using these values and the cooling coil load, the partial load power is calculated (from Equations 4.1, 4.2). The partial load power and cooling coil load is summed to find the condenser load. The condenser load is then used to find a new condensing temperature (T_c) for an air-cooled condenser (Eqns. 3.4, 3.5) or a new leaving condensing water temperature (T_{co}) for water-cooled condenser (cooling tower model and Eqn. 3.6). These values are then used as the new estimates of T_{cl}; with the process being repeated until the convergence criteria is met.

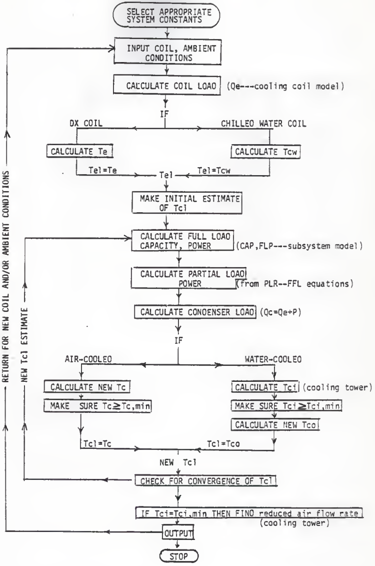


Figure 5.9 Generalized model flowchart

Performance Output. At this stage, the partial load performance for the cooling tower - if desired - may be calculated (reduced cfm, fan cycling). The performance data for the system at this steady-state condition may then be multiplied by the number of hours of occurrence of this condition for inclusion in the total annual performance estimate*. The process is then repeated starting with Step #2 (calculating T_{e1}) until all of the likely operational coil and ambient conditions have been accounted for.

5.3 Typical Annual Performance

The annual energy consumption of a refrigeration system is dependent on equipment size, location (for ambient conditions), and the variable annual cooling/dehumidification load. As such, energy requirements of a system may vary widely from one application to another and should therefore be calculated on an individual basis.

5.3A Single Coil Systems

Load. To illustrate the use of the generalized model for annual energy consumption estimates, the performance of each of the systems included in the model was calculated for the following two load conditions:

1) Use of the system for pre-cooling ambient air to a present level.
2) Use of the system for cooling/dehumidifying recirculated air to a preset level.

---

*It should be noted, however, that this includes only compressor energy requirements. The annual energy requirements of the fans and pumps, because they operate generally at constant flow rate, may be approximated by multiplying operational time by the design power consumption of each of these items.

For both loads single bin* ambient data for New Orleans, LA was utilized. Also, both loads set 55 F as the temperature to which the air was to be processed.

System. The 15-ton refrigeration systems utilized in the development of the modeling constants for this study were also used for these performance predictions. The equipment modeled is as follows:

1. Trane CUAB C15M compressor - 6 row Aerofin DX coil - Trane CAUA 200 air cooled condenser.
2. Trane CUAW C15M water cooled condensing unit - 6 row Aerofin DX coil - Baltimore Aircoil VXT-15 cooling tower.
3. Trane CCWB C15M compressor chiller unit - Trane CAVA 200 air cooled condenser.
4. Trane CGWB C15M chiller unit - Baltimore Aircoil VXT-15 cooling tower.

The minimum condensing temperature and entering condensing water temperature were set at 85 F and 75 F (which, from manufacturers catalogs, appear to be typical values), respectively. Also, the chilled water was maintained at either a supply temperature of 45 F or a return temperature of 55 F, (depending on the system). The water flow rates for evaporator (34.5 gpm) and the condenser (44.8 gpm) were selected to provide a 10 F water temperature swing at the largest loading condition. Also, the face area for the DX coil (5 sq. ft.) was chosen to provide a coil face velocity of 400 fpm at this condition. All of these conditions and associated

---

*Single bin data is typically presented in terms of dry-bulb temperature (usually grouped into 5 degree intervals), the number of hours of occurrence of each temperature interval, and the mean coincident wet-bulb temperature of each interval.

equipment modeling constants are available in the constants subroutine of the computer program listed in Appendix C.

Performance. The performance calculations for the systems were presented in Appendix D. A summary of the annual compressor power consumption estimates for the pre-cooling load is presented below.

System	Annual Power
Compressor-DX coil-Cooling Tower	48300 (kW-hrs)
Compressor-DX Coil-Air Cooled Condenser	48690
Chiller Unit - Cooling Tower (55 F return)	49480
Compressor Chiller Unit-Air Cooled Condenser (55 F return)	50090
Chiller Unit - Cooling Tower (45 F supply)	50390
Compressor Chiller Unit-Air Cooled Condenser (45 F supply)	51790

As expected, the annual compressor power requirements for DX coil systems are lower than for chilled water systems (due to higher evaporating temperatures for each loading condition). Also, systems which control chilled water return temperature are more energy efficient than systems which maintain a fixed supply temperature (again because of higher evaporator temperatures). Also, systems utilizing cooling towers operate at lower annual power consumption levels than systems with air cooled condensers due to lower condensing temperatures (this, however, may vary from location to location depending on ambient dry-bulb and wet-bulb conditions). The second set of performance calculations followed similar trends as the first set. It should be noted, however, that the coil load conditions for this calculation run were arbitrarily chosen to illustrate use of the model. As such, the annual power consumption values calculated may or may not be typical of refrigeration systems used for cooling/dehumidifying recirculating air at this location. These performance predictions including a listing of temperatures, air and water flow rates, and load and power values for each temperature bin may be found in Appendix D.

5.3B Multiple Coil (Chilled Water) Systems

As was discussed in Chapter 2, water chiller systems are used primarily for multiple coil applications. Because the chilled water system modeled earlier utilized an energy balance at the evaporator to predict performance, the generalized model can easily be adapted to handle as many chilled water cooling coils as are desired. For this situation, the evaporator load to be used in equation 5.4 is simply the sum of all of the individual cooling coil loads (from the cooling coil model). From this, system performance can be calculated in the same manner as before.

To illustrate the use of the model for multiple coil applications, the generalized computer program of Appendix C was modified to handle specifically a two coil-chiller unit-cooling tower system (Appendix E). This program was then utilized to make performance predictions for the refrigeration system while providing pre-cooling of ambient make up air and after-cooling of air from a HONEYCOMBE dehumidifier; with the conditions to and from each coil being provided to this program from a model prediction for that specific brand of dehumidifier [1] (eliminating the need to include the dehumidifier in the program). The schematic for the system is presented in Figure 5.10.

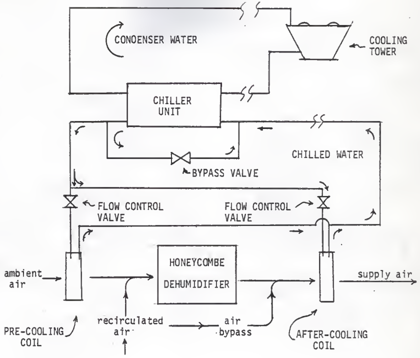


FIGURE 5.10 Schematic for a Two Coil-Chiller unit-Cooling Tower system.

Load. Performance calculations were generated for two different data sets; with the first data set based on single bin ambient data and the second set based on double bin ambient data* (both for New Orleans, LA).

System. Based on the largest total cooling load, a 20-ton Trane chiller unit and a 20-ton Baltimore AIRCOIL cooling tower were used in the computer program. An evaporator water flow rate of 42.9 gpm and condenser water flow rate of 55.4 gpm were selected. Also, chilled water supply temperature was held at 45 F and the minimum entering condensing water temperature at 75 F.

Performance. The performance calculations (along with coil-air conditions) are presented in Appendix F. A summary of system annual performance values for each data sets is presented below.

Annual	Single Bin	Double Bin
Pre-cool coil load (MMBTU)	123.63	133.66
After-cool coil load (MMBTU)	838.88	858.72
Total coil load (MMBTU)	962.51	992.38
Compressor power (KW-HR)	76801	79689
Hours of operation	8417	8537

Examination of this table shows that all of the annual performance values for the single bin are consistently lower than the double bin performance values. The single bin annual power consumption, however, was within 4% of the double bin value. Thus, the use of single bin ambient data (which requires less processing time due to fewer data points than with double bin data) may provide adequate power consumption estimates for most refrigeration system applications.

---

*Double bin data is typically presented in terms of dry-bulb temperature (usually grouped into 5 degree intervals) and the hours of occurrence of each wet-bulb temperature (usually grouped into 2 degree intervals) for each dry-bulb temperature interval.

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

6.1 Summary and Conclusions

The main objective of this study was to develop a generalized model for predicting the annual energy requirements of reciprocating vapor-compression refrigeration/dehumidification systems. The annual performance of a system is directly dependent on location and the variable load to which the system is to be subjected. The model developed is capable of handling the following systems:

1. Compressor - DX-coil - air cooled condenser systems.
2. Water cooled condensing unit - DX-coil - cooling tower systems.
3. Compressor chilled unit - chilled water coil(s) - air cooled condenser systems.
4. Chiller unit - chilled water coils(s) - cooling tower systems.

For each of these systems, cylinder unloading was used for partial load system capacity control. The model agrees with catalog data ($\pm$3-1/2%) for a wide range of equipment variations and sizes. Also, the use of subsystem units in the model greatly reduces the amount of data required for individual system performance modeling.

6.2 Recommendations

It is recommended that:

1. A procedure be developed for including system annual energy requirements and costs into the equipment selection procedure.

2. The generalized model be extended to other refrigeration systems utilizing screw and centrifugal compressors.
3. The model be verified experimentally to determine, if any, the shortcomings of the model in making annual energy consumption estimates for actual operation systems.

REFERENCES

[1] "Analysis of Open-Sorption and Refrigeration Humidity Control Systems", Research Project 298 RP, Final Report to ASHRAE, June 1985.

[2] W.F. Stoecker, editor, "Procedures for Simulating the Performance of Components and Systems for Energy Calculations, ASHRAE, 1976.

[3] W.F. Stoecker, Design of Thermal Systems", McGraw-Hill, 1980.

[4] W.F. Stoecker, "Refrigeration and Air Conditioning", McGraw-Hill, 1982.

[5] D.C. Hittle, "An Algorithm for Modeling Direct-Expansion Air-Cooled Condensing Unit", ASHRAE Transactions, Volume 88, part 2, 1982.

[6] D.J. Leverenz, N.E. Bergan, "Development and Validation of a Reciprocating Chiller Model for Hour Energy Analysis Programs", ASHRAE Transactions, Volume 89, part 1A, 1983.

[7] Carrier Air Conditioning Company, "Handbook of Air Conditioning System Design", McGraw-Hill, 1965.

[8] E.G. Pita, "Refrigeration Principles and Systems - An Energy Approach", John Wiley & Sons, 1984.

[9] M.W. Garland, "Compressor Capacity Control for Air-Conditioning System Partial Load Operation", ASHRAE Transactions, Volume 86, part 1, 1980.

[10] R.L. Webb, A. Villacres, "Algorithms for Perfomance Simulation of Cooling Towers, Evaporative Condensers, and Fluid Coolers", ASHRAE Transactions, Volume 90, part 2B, 1984.

[11] ASHRAE Handbook, 1979 Equipment, ASHRAE, New York, 1979.

[12] M.A. Ramsey, "Tested Solutions to: Design Problems in Air Conditioning and Refrigeration", International Press, Inc., 1966.

APPENDIX A

REFRIGERATION SYSTEM MODELING EQUATION SUMMARY

Equation Summary - Component Model - Full Load Operation

Compressor

$$Q_e = a_1 + a_2T_e + a_3T_e^{\,2} + a_4T_c + a_5T_c^{\,2} + a_6T_eT_c + a_7T_e^{\,2}T_c + a_8T_eT_c^{\,2} + a_9T_e^{\,2}T_c^{\,2}$$

$$P = b_1 + b_2T_e + b_3T_e^{\,2} + b_4T_c + b_5T_c^{\,2} + b_6T_eT_c + b_7T_e^{\,2}T_c + b_8T_eT_c^{\,2} + b_9T_e^{\,2}T_c^{\,2}$$

Condenser

$$Q_c = Q_e + P$$

$$Q_c = F_1 + F_2(T_c - T_m)$$

$$\text{subcooling} = F_3 + F_4(T_c - T_m) + F_5(T_c - T_m)^2$$

air-cooled --- T_m = ambient dry bulb temperature

water-cooled --- T_m = inlet water temperature

Evaporator

DX-Coil

$$Q_e = (CF)(AREA)[C_1 + C_2T_e + C_3T_e^{\,2} + C_4T_{wb} + C_5T_{wb}^{\,2} + C_6T_eT_{wb} + C_7T_e^{\,2}T_{wb} \\ C_8T_eT_{wb}^{\,2} + C_9T_e^{\,2}T_{wb}^{\,2}]$$

$$CF = C_{10} + C_{11}\ \text{fpm} + C_{12}\ \text{fpm}^2$$

Chilled-water

$$Q_e = d_1 + d_2(\mathrm{GPM}) + d_3(\mathrm{GPM})^2 + d_4(\mathrm{dT}) + d_5(\mathrm{dT})^2 + d_6(\mathrm{GPM})(\mathrm{dT})$$

$$+ d_7(\mathrm{GPM})^2(\mathrm{dT}) + d_8(\mathrm{GPM})(\mathrm{dT})^2 + d_9(\mathrm{GPM})^2(\mathrm{dT})^2$$

dT = (inlet water temperature)-(saturated evaporating temperature)

where $\left.\begin{array}{l} a_1 \sim a_9 \\ b_1 \sim b_9 \\ F_1 \sim F_5 \\ C_1 \sim C_{12} \\ d_1 \sim d_9 \end{array}\right\}$ = constants to be found for each system

(generally by a least-squares fit of each equation to manufacturers catalog data)

Equation Summary - Leverenz-Bergan Chiller Model - Full Load Operation

$$T_{eq} = (T_{cond} - T_{cond,D})/T_{ratio} - (T_{cw} - T_{cw,D})$$

$$ADCR = a_1 + a_2(T_{eq}) + a_3(T_{eq})^2$$

$$CAP = (ADCR)(CAP,_D)$$

$$COPC = (T_{cond} + 273.2)/(T_{cond} - T_{cw})$$

$$COPCD = (T_{cond,D}) + 273.2)/T_{cond,D} - T_{cw,D})$$

$$COPCN = (COPC)/(COPCD)$$

$$COPAN = b_1 + b_2(COPCN) + b_3(COPCN)^2$$

$$COPAD = (CAP,_D)/(FLP,_D)$$

$$COPA = (COPAN)(COPAD)$$

$$FLP = (CAP)/(COPA)$$

where

all input and output in Kilowatts and degrees Celsius

input -

T_{cw} = leaving chilled water temperature

T_{cond} = entering condensing water temperature

T_{ratio}, $T_{cw,D}$, $T_{cond,D}$, $a_1 \sim a_3$, $b_1 \sim b_3$ = constants for all chiller systems

$CAP,_D$ = System capacity at $T_{cw,D}$ and $T_{cond,D}$ temperatures

$FLP,_D$ = System power at $T_{cw,D}$ and $T_{cond,D}$ temperatures

output -

CAP = predicted system capacity

FLP = predicted system power

(for definitions of T_{eq}, ADCR, ---- See subsystem model - equation summary)

Equation Summary - subsystem model - Full Load

$$T_{eq} = (T_{c1} - T_{c1,D})/T_{ratio} - (T_{e1} - T_{e1,D})$$

$$ADCR = A_1 + A_2(T_{eq}) + A_3(T_{eq})^2$$

$$CAP = (ADCR)(CAP_D)$$

$$COPC = (T_{c1} + 273.2)/(T_{c1} - T_{e1})$$

$$COPCD = (T_{c1,D} + 273.2)/(T_{c1,D} - T_{e1,D})$$

$$COPCN = COPC/COPCD$$

$$COPAN = B_1 + B_2(COPCN) + B_3(COPCN)^2$$

$$COPAD = (CAP,_D)/(FLP,_D)$$

$$COPA = (COPAN)(COPAD)$$

$$FLP = CAP/COPA$$

where

	compressor	water-cooled condensing unit	compressor-chiller unit	chiller unit
T_{e1} =	evap. temp	evap. temp	leaving evap. water temp.	leaving evap. water temp.
$T_{e1,D}$ =	base evap. temp	base evap. temp	base leaving evap. water temp	base leaving evap. water temp
T_{c1} =	cond. temp	leaving cond. water temp.	cond. temp.	leaving cond. water temp.
$T_{c1,D}$ =	base cond. temp	base leaving cond. water temp.	base cond. temp	base leaving cond. water temp

where

all input and output in Kilowatts and degrees Celsius

input - T_{e1} and T_{c1} temperatures

T_{ratio}, $T_{e1,D}$, $T_{c1,D}$, $A_1 \sim A_3$, $B_1 \sim B_3$ = constants for appropriate system (Table 3.A)

$CAP,_D$ = System capacity at $T_{e1,D}$ and $T_{c1,D}$ temperatures

$FLP,_D$ = System power at $T_{e1,D}$ and $T_{c1,D}$ temperatures output - CAP = predicted system capacity

FLP = predicted system power

<u>where</u>

T_{eq} = Equivalent operational temperature

T_{ratio} = ratio of change in condenser temperature (refrig. or water) required for a given change in evaporator temperature (refrig. or water) which maintains a given capacity.

ADCR = Actual versus Design (base) Capacity Ratio

COPC = Coefficient of Performance, Carnot

COPCD = Coefficient of Performance, Carnot at Design (base) temperatures

COPCN = Normalized Carnot Coefficient of Performance

COPA = Coefficient of Performance, Actual

COPAD = Coefficient of Performance, Actual at Design (base) temperatures

COPAN = Normalized Actual Coefficient of Performance

APPENDIX B

COMPARISION OF SUSBSYSTEM MODELING ACCURACY

COMPRESSOR CHARACTERISTICS

BASE	EVAPORATOR TEMP.	(deg.F)	40.0
BASE	CONDENSER TEMP.	(deg.F)	105.0
BASE	COMPRESSOR LOAD	(tons)	17.3
BASE	COMPRESSOR POWER	(kws)	15.9

Tel ..(deg.F)..	Tcl	Qe_pred ....(tons)....	Qe_cat	dif. %	P_pred ..(kws)...	P_cat	dif. %
30.0	85.0	16.2	16.0	1.1	13.2	13.1	.8
35.0	85.0	17.9	17.8	.8	13.4	13.4	.3
40.0	85.0	19.8	19.7	.6	13.6	13.6	.1
45.0	85.0	21.8	21.7	.5	13.8	13.8	-.2
50.0	85.0	23.9	23.9	.1	14.0	13.9	.4
30.0	105.0	13.9	14.0	-.5	15.2	14.9	1.8
35.0	105.0	15.5	15.6	-.4	15.6	15.4	1.6
40.0	105.0	17.3	17.3	-.3	16.0	15.9	.9
45.0	105.0	19.1	19.2	-.6	16.4	16.4	-.2
50.0	105.0	21.0	21.1	-.3	16.6	16.9	-1.7
30.0	125.0	11.9	11.9	.1	16.7	16.8	-.6
35.0	125.0	13.4	13.3	.4	17.4	17.6	-1.0
40.0	125.0	14.9	14.9	.1	18.1	18.3	-1.1
45.0	125.0	16.6	16.6	-.1	18.7	19.0	-1.5
50.0	125.0	18.4	18.3	.4	19.2	19.7	-2.4

where

Tel = saturated evaporating temperature
Tcl = saturated condensing temperature
Qe_pred = capacity predicted by model
Qe_cat = manufacturers catalog data
P_pred = power requirements predicted by model
P_cat = power requirements from catalog

constants

Tratio = 2.9
A1 = .9973
A2 =-.03694
A3 = .0004335
B1 = -.4141
B2 = 1.7717
B3 = -.3694

COMPRESSOR CHARACTERISTICS

BASE	EVAPORATOR TEMP.	(deg.F)	40.0
BASE	CONDENSER TEMP.	(deg.F)	105.0
BASE	COMPRESSOR LOAD	(tons)	34.1
BASE	COMPRESSOR POWER	(kws)	29.8

Tel	Tcl	Qe_pred	Qe_cat	dif.	P_pred	P_cat	dif.
..(deg.F)..		....(tons)....		%	..(kws)...		%
30.0	85.0	31.9	32.2	-1.0	24.8	24.0	3.1
35.0	85.0	35.4	35.6	-.7	25.2	24.6	2.4
40.0	85.0	39.1	39.2	-.4	25.5	25.2	1.3
45.0	85.0	43.0	43.0	-.0	25.8	25.6	.8
50.0	85.0	47.1	47.0	.3	26.2	26.1	.3
30.0	105.0	27.5	27.9	-1.6	28.4	28.1	1.2
35.0	105.0	30.6	31.0	-1.2	29.3	29.0	1.1
40.0	105.0	34.0	34.1	-.3	30.1	29.8	.9
45.0	105.0	37.6	37.5	.3	30.7	30.5	.6
50.0	105.0	41.5	41.0	1.1	31.1	31.1	.1
30.0	125.0	23.5	23.7	-.9	31.3	31.6	-.9
35.0	125.0	26.3	26.3	.1	32.7	32.9	-.7
40.0	125.0	29.4	29.2	.7	33.9	34.1	-.5
45.0	125.0	32.7	32.1	1.9	35.1	35.2	-.4
50.0	125.0	36.2	35.3	2.6	36.0	36.3	-.7

COMPRESSOR CHARACTERISTICS

BASE	EVAPORATOR TEMP.	(deg.F)	40.0
BASE	CONDENSER TEMP.	(deg.F)	105.0
BASE	COMPRESSOR LOAD	(tons)	81.1
BASE	COMPRESSOR POWER	(kws)	64.7

Tel	Tcl	Qe_pred	Qe_cat	dif.	P_pred	P_cat	dif.
..(deg.F)..		....(tons)....		%	..(kws)...		%
30.0	85.0	75.8	74.9	1.2	53.7	52.5	2.4
35.0	85.0	84.1	83.3	.9	54.7	53.5	2.2
40.0	85.0	92.9	92.1	.8	55.4	54.4	1.8
45.0	85.0	102.2	101.5	.7	56.0	55.0	1.9
50.0	85.0	112.1	111.3	.7	56.8	55.4	2.5
30.0	105.0	65.3	65.3	.0	61.8	61.2	.9
35.0	105.0	72.8	73.0	-.2	63.7	63.1	.9
40.0	105.0	80.9	81.1	-.3	65.3	64.7	.9
45.0	105.0	89.5	89.6	-.1	66.6	66.0	.9
50.0	105.0	98.6	98.7	-.1	67.6	67.1	.7
30.0	125.0	55.9	55.0	1.6	68.0	68.5	-.8
35.0	125.0	62.6	62.3	.5	70.9	71.6	-.9
40.0	125.0	69.9	69.6	.5	73.7	74.5	-1.1
45.0	125.0	77.8	77.4	.5	76.1	77.3	-1.5
50.0	125.0	86.2	85.6	.6	78.3	79.9	-2.1

COMPRESSOR-CONDENSER CHARACTERISTICS

BASE	EVAPORATOR TEMP.	(deg.F)	= 40.0
BASE	CONDENSER-WATER TEMP.	(deg.F)	= 95.0
BASE	COMPRESSOR LOAD	(tons)	= 23.7
BASE	COMPRESSOR POWER	(kws)	= 17.3

Tei	Tci	Qe_pred	Qe_cat	dif.	P_pred	P_cat	dif.
..(deg.F)..		....(tons)....		%	..(kws)...		%
30.0	85.0	20.4	20.5	-.5	15.2	14.9	1.9
35.0	85.0	22.9	23.0	-.6	15.6	15.4	1.5
40.0	85.0	25.5	25.6	-.4	16.0	15.8	1.1
45.0	85.0	28.3	28.4	-.3	16.2	16.2	.2
50.0	85.0	31.3	31.4	-.3	16.4	16.4	.2
30.0	95.0	18.8	18.9	-.5	16.1	16.0	.8
35.0	95.0	21.1	21.2	-.3	16.8	16.7	.4
40.0	95.0	23.7	23.7	-.2	17.3	17.3	.1
45.0	95.0	26.3	26.3	.2	17.7	17.8	-.3
50.0	95.0	29.2	29.2	.1	18.1	18.2	-.8
30.0	105.0	17.3	17.2	.7	16.9	16.9	-.1
35.0	105.0	19.5	19.4	.6	17.7	17.8	-.5
40.0	105.0	21.9	21.8	.5	18.5	18.6	-.8
45.0	105.0	24.5	24.3	.7	19.1	19.3	-1.1
50.0	105.0	27.2	27.0	.8	19.6	19.9	-1.5

where

Tei = saturated evaperating temperature
Tci = leaving condensing water temperature
Qe_pred = capacity predicted by model
Qe_cat = manufacturers catalog data
P_pred = power requirements predicted by model
P_cat = power requirements from catalog

constants

Tratio = 2.9
A1 = .9982
A2 =-.03952
A3 = .004931
B1 = -.2244
B2 = 1.5438
B3 = -.3219

COMPRESSOR-CONDENSER CHARACTERISTICS

BASE	EVAPORATOR TEMP.	(deg.F)	= 40.0
BASE	CONDENSER WATER TEMP.	(deg.F)	= 95.0
BASE	COMPRESSOR LOAD	(tons)	= 38.1
BASE	COMPRESSOR POWER	(kws)	= 28.4

Tei	Tci	Qe_pred	Qe_cat	dif.	P_pred	P_cat	dif.
..(deg.F)..		....(tons)....		%	..(kws)...		%
30.0	85.0	32.8	33.2	-1.2	24.9	24.5	1.7
35.0	85.0	36.7	36.9	-.4	25.7	25.2	1.8
40.0	85.0	41.0	40.9	.2	26.2	25.8	1.7
45.0	85.0	45.5	45.2	.7	26.6	26.3	1.3
50.0	85.0	50.3	49.8	1.1	27.0	26.6	1.4
30.0	95.0	30.2	30.8	-1.8	26.5	26.6	-.5
35.0	95.0	34.0	34.3	-.9	27.5	27.5	.1
40.0	95.0	38.0	38.1	-.2	28.4	28.4	.1
45.0	95.0	42.4	42.2	.4	29.1	29.1	.1
50.0	95.0	47.0	46.5	1.0	29.6	29.8	-.6
30.0	105.0	27.8	28.5	-2.3	27.7	28.5	-2.8
35.0	105.0	31.4	31.9	-1.6	29.1	29.7	-2.1
40.0	105.0	35.2	35.4	-.5	30.3	30.8	-1.7
45.0	105.0	39.3	39.3	.1	31.3	31.9	-1.8
50.0	105.0	43.8	43.3	1.1	32.2	32.8	-1.9

COMPRESSOR-CONDENSER CHARACTERISTICS

BASE	EVAPORATOR TEMP.	(deg.F)	= 40.0
BASE	CONDENSER WATER TEMP.	(deg.F)	= 95.0
BASE	COMPRESSOR LOAD	(tons)	= 88.4
BASE	COMPRESSOR POWER	(kws)	= 66.6

Tei	Tci	Qe_pred	Qe_cat	dif.	P_pred	P_cat	dif.
..(deg.F)..		....(tons)....		%	..(kws)...		%
30.0	85.0	76.1	77.7	-2.1	58.4	57.9	.9
35.0	85.0	85.3	86.3	-1.2	60.2	59.4	1.3
40.0	85.0	95.1	95.4	-.3	61.5	60.8	1.2
45.0	85.0	105.6	105.1	.5	62.5	61.8	1.1
50.0	85.0	116.8	115.4	1.2	63.2	62.6	1.0
30.0	95.0	70.2	71.9	-2.4	62.1	62.5	-.6
35.0	95.0	78.9	79.9	-1.3	64.6	64.7	-.2
40.0	95.0	88.2	88.4	-.2	66.6	66.6	.1
45.0	95.0	98.3	97.6	.7	68.3	68.2	.1
50.0	95.0	109.0	107.3	1.6	69.5	69.5	-.0
30.0	105.0	64.6	66.2	-2.5	65.0	66.7	-2.6
35.0	105.0	72.8	73.7	-1.2	68.2	69.5	-1.9
40.0	105.0	81.7	81.8	-.1	71.0	72.0	-1.3
45.0	105.0	91.3	90.3	1.1	73.5	74.2	-1.0
50.0	105.0	101.5	99.5	2.0	75.5	76.1	-.8

COMPRESSOR CHILLER UNIT CHARACTERISTICS

BASE	CHILLED WATER TEMP.	(deg.F)	= 46.0
BASE	CONDENSER TEMP.	(deg.F)	=105.0
BASE	COMPRESSOR LOAD	(tons)	= 16.1
BASE	COMPRESSOR POWER	(kws)	= 15.6

Tel	Tcl	Qe_pred	Qe_cat	dif.	P_pred	P_cat	dif.
..(deg.F)..		....(tons)....		%	..(kws)...		%
40.0	85.0	16.3	16.2	.7	13.2	13.2	.2
44.0	85.0	17.4	17.3	.6	13.3	13.3	.2
46.0	85.0	18.0	17.9	.3	13.4	13.4	-.1
50.0	85.0	19.1	19.1	-.1	13.5	13.5	.3
40.0	95.0	15.4	15.3	.6	14.3	14.1	1.2
44.0	95.0	16.5	16.5	-.2	14.4	14.3	.7
46.0	95.0	17.0	17.0	.0	14.4	14.4	.3
50.0	95.0	18.1	18.2	-.5	14.5	14.7	-1.2
40.0	105.0	14.5	14.5	-.0	15.3	15.0	1.8
44.0	105.0	15.5	15.6	-.4	15.5	15.4	.5
46.0	105.0	16.1	16.1	-.2	15.6	15.6	-.3
50.0	105.0	17.2	17.2	-.3	15.7	15.9	-1.4
40.0	115.0	13.6	13.6	.1	16.1	16.1	.3
44.0	115.0	14.6	14.7	-.4	16.5	16.5	-.3
46.0	115.0	15.2	15.2	-.3	16.6	16.7	-.7
50.0	115.0	16.2	16.3	-.5	16.8	17.1	-1.7

where

Tel = leaving chilled water temperature
Tcl = saturated condensing temperature
Qe_pred = capacity predicted by model
Qe_cat = manufacturers catalog data
P_pred = power requirements predicted by model
P_cat = power requirements from catalog

constants

Tratio = 2.9
A1 = .9982
A2 =-.02990
A3 = .0001711
B1 = -.2484
B2 = 1.6600
B3 = -.4105

COMPRESSOR CHILLER UNIT CHARACTERISTICS

BASE	CHILLED WATER TEMP.	(deg.F)	= 46.0
BASE	CONDENSER TEMP.	(deg.F)	=105.0
BASE	COMPRESSOR LOAD	(tons)	= 32.0
BASE	COMPRESSOR POWER	(kws)	= 29.3

Tel	Tcl	Qe_pred	Qe_cat	dif.	P_pred	P_cat	dif.
..(deg.F)..		....(tons)....		%	..(kws)...		%
40.0	85.0	32.4	32.5	-.2	24.9	24.1	3.1
44.0	85.0	34.6	34.7	-.3	25.0	24.5	2.2
46.0	85.0	35.7	35.8	-.3	25.1	24.6	2.2
50.0	85.0	37.9	38.1	-.4	25.4	25.0	1.8
40.0	95.0	30.6	30.7	-.3	26.8	26.3	1.9
44.0	95.0	32.7	32.8	-.3	27.0	26.8	.9
46.0	95.0	33.8	33.9	-.3	27.1	27.0	.5
50.0	95.0	36.0	36.1	-.3	27.3	27.4	-.5
40.0	105.0	28.8	28.9	-.3	28.7	28.4	1.0
44.0	105.0	30.9	30.9	-.0	29.1	29.0	.2
46.0	105.0	31.9	32.0	-.2	29.2	29.3	-.3
50.0	105.0	34.1	34.1	-.0	29.5	29.8	-1.2
40.0	115.0	27.1	27.1	-.1	30.3	30.5	-.6
44.0	115.0	29.1	29.0	.3	30.9	31.2	-.9
46.0	115.0	30.1	30.0	.4	31.2	31.5	-1.1
50.0	115.0	32.2	32.0	.7	31.6	32.1	-1.6

COMPRESSOR CHILLER UNIT CHARACTERISTICS

BASE	CHILLED WATER TEMP.	(deg.F)	= 46.0
BASE	CONDENSER TEMP.	(deg.F)	=105.0
BASE	COMPRESSOR LOAD	(tons)	= 78.0
BASE	COMPRESSOR POWER	(kws)	= 64.1

Tel	Tcl	Qe_pred	Qe_cat	dif.	P_pred	P_cat	dif.
..(deg.F)..		....(tons)....		%	..(kws)...		%
40.0	85.0	79.0	78.4	.8	54.4	53.0	2.6
44.0	85.0	84.3	84.0	.4	54.8	53.6	2.2
46.0	85.0	87.0	86.9	.1	55.0	53.9	2.0
50.0	85.0	92.5	92.7	-.3	55.7	54.4	2.3
40.0	95.0	74.6	74.3	.4	58.7	57.8	1.5
44.0	95.0	79.7	79.7	.1	59.2	58.7	.8
46.0	95.0	82.4	82.5	-.1	59.3	59.0	.6
50.0	95.0	87.7	88.2	-.5	59.7	59.7	-.1
40.0	105.0	70.2	70.1	.2	62.7	62.4	.5
44.0	105.0	75.3	75.3	-.0	63.6	63.6	-.0
46.0	105.0	77.9	78.0	-.2	63.9	64.1	-.3
50.0	105.0	83.1	83.5	-.5	64.4	65.1	-1.0
40.0	115.0	66.0	65.7	.4	66.3	66.8	-.7
44.0	115.0	70.9	70.8	.2	67.6	68.4	-1.1
46.0	115.0	73.4	73.4	.1	68.2	69.1	-1.3
50.0	115.0	78.6	78.7	-.2	69.1	70.5	-2.0

CHILLER CHARACTERISTICS

BASE	CHILLED WATER TEMP.	(deg.F)	= 46.0
BASE	CONDENSER WATER TEMP.	(deg.F)	= 95.0
BASE	COMPRESSOR LOAD	(tons)	= 15.8
BASE	COMPRESSOR POWER	(kws)	= 16.0

Tel	Tcl	Qe_pred	Qe_cat	dif.	P_pred	P_cat	dif.
..(deg.F)..		....(tons)....		%	..(kws)...		%
40.0	85.0	15.1	15.1	.1	14.7	14.4	2.0
44.0	85.0	16.1	16.1	.3	14.9	14.8	.7
46.0	85.0	16.7	16.6	.4	15.0	15.0	.1
50.0	85.0	17.7	17.7	.2	15.3	15.3	-.0
40.0	95.0	14.2	14.2	.3	15.6	15.3	2.0
44.0	95.0	15.3	15.3	-.3	15.9	15.8	.4
46.0	95.0	15.8	15.8	-.2	16.0	16.0	-.2
50.0	95.0	16.8	16.8	.1	16.2	16.4	-1.4
40.0	105.0	13.4	13.4	-.1	16.4	16.4	.2
44.0	105.0	14.4	14.4	-.1	16.8	16.9	-.6
46.0	105.0	14.9	14.9	-.1	17.0	17.1	-.9
50.0	105.0	15.9	15.9	.1	17.2	17.6	-2.2

where

Tel = leaving evaporator water temperature
Tcl = leaving condenser water temperature
Qe_pred = capacity predicted by model
Qe_cat = manufacturers catalog data
P_pred = power requirements predicted by model
P_cat = power requirements from catalog

constants

Tratio = 2.9
A1 = .9978
A2 =-.02943
A3 = .0001472
B1 = -.1119
B2 = 1.5815
B3 = -.4702

CHILLER CHARACTERISTICS

BASE CHILLED WATER TEMP. (deg.F) = 46.0
BASE CONDENSER WATER TEMP. (deg.F) = 95.0
BASE COMPRESSOR LOAD (tons) = 31.2
BASE COMPRESSOR POWER (kws) = 30.1

Tel	Tcl	Qe_pred	Qe_cat	dif.	P_pred	P_cat	dif.
..(deg.F)..		....(tons)....		%	..(kws)...		%
40.0	85.0	29.8	30.1	-.9	27.6	27.1	2.0
44.0	85.0	31.9	32.1	-.7	28.0	27.8	.9
46.0	85.0	32.9	33.1	-.6	28.3	28.1	.6
50.0	85.0	35.0	35.1	-.3	28.8	28.7	.3
40.0	95.0	28.1	28.4	-1.0	29.4	29.0	1.3
44.0	95.0	30.1	30.3	-.6	29.8	29.8	.2
46.0	95.0	31.1	31.2	-.2	30.1	30.1	-.2
50.0	95.0	33.2	33.2	-.0	30.4	30.8	-1.2
40.0	105.0	26.4	26.6	-.6	30.9	30.9	.1
44.0	105.0	28.4	28.4	-.0	31.6	31.7	-.3
46.0	105.0	29.4	29.4	-.0	31.9	32.1	-.7
50.0	105.0	31.4	31.2	.7	32.4	32.9	-1.6

CHILLER CHARACTERISTICS

BASE CHILLED WATER TEMP. (deg.F) = 46.0
BASE CONDENSER WATER TEMP. (deg.F) = 95.0
BASE COMPRESSOR LOAD (tons) = 75.9
BASE COMPRESSOR POWER (kws) = 65.0

Tel	Tcl	Qe_pred	Qe_cat	dif.	P_pred	P_cat	dif.
..(deg.F)..		....(tons)....		%	..(kws)...		%
40.0	85.0	72.6	72.4	.3	59.7	58.6	1.9
44.0	85.0	77.5	77.5	.0	60.6	59.8	1.3
46.0	85.0	80.1	80.2	-.2	61.0	60.4	1.0
50.0	85.0	85.2	85.6	-.5	62.1	61.4	1.2
40.0	95.0	68.4	68.4	.0	63.4	62.8	1.0
44.0	95.0	73.3	73.4	-.2	64.4	64.3	.2
46.0	95.0	75.7	75.9	-.2	64.9	65.0	-.2
50.0	95.0	80.8	81.1	-.4	65.7	66.4	-1.1
40.0	105.0	64.3	64.3	.0	66.8	66.9	-.2
44.0	105.0	69.1	69.0	.1	68.2	68.7	-.7
46.0	105.0	71.5	71.5	-.0	68.9	69.6	-1.1
50.0	105.0	76.4	76.5	-.1	69.9	71.4	-2.0

APPENDIX C

GENERALIZEO RECIPROCATING REFRIGERATION/OEHUMIOIFICATION SYSTEM COMPUTER PROGRAM

```
10 REM         GENERALIZED PROGRAM FOR REFRIGERATION SYSTEM PERFORMANCE PREDICTIONS
20 REM         FOR THE FOLLOWING SYSTEMS:
30 REM                   COMPRESSOR--DX COIL--REMOTE AIR COOLED CONDENSER
40 REM                   WATER COOLED CONDENSING UNIT--DX COIL--COOLING TOWER
50 REM                   COMPRESSOR CHILLER UNIT--CHILLED WATER COIL--REMOTE AIR COOLED CONDENSER
60 REM                   CHILLER UNIT--CHILLED WATER COIL--COOLING TOWER
70 REM        WITH CYLINDER UNLOADING MATCHED TO HOLD DESIRED DX-COIL LEAVING AIR TEMP             -OR-
80 REM        WITH CYLINDER UNLOADING MATCHED TO HOLD DESIRED EVAPORATOR LEAVING OR ENTERING WATER TEMP
90 REM        CONSTANT WATER FLOW RATE AT EVAPORATORS,CONDENSERS  (WATER BYPASS--IF NECESSARY--AT COIL)
100 REM
110 REM              THE CONSTANTS DESCRIBING EQUIPMENT SIZE ARE ALREADY INCLUDED IN THE CONSTANT SUBROUTI
NE
120 REM              THE COIL CONDITIONS ARE PROVIDED TO THE PROGRAM IN THE COIL_CONDITIONS SUBROUTINE
130 !
140 !
150 !
160 DIM Qsens(200),Qlat(200),Qe(200),P(200),hrs(200),TDB(200),TWB(200)
170 DIM Tdb(200),wb_wi(200),CFMe(200),Ts(200),UNITS$(200)
180 !
190 !
200 !
210 !     INDICATE THE TYPE OF SYSTEM TO BE UTILIZED BY SETTING SYSTEM EQUAL TO THE APPROPRIATE VALUE
220 !      1 =          COMPRESSOR UNIT-------DX COIL------REMOTE AIR-COOLED CONDENSER
230 !      2 =  WATER COOLED CONDENSING UNIT--DX COIL--------COOLING TOWER
240 !      3 =  COMPRESSOR CHILLER UNIT-----WATER COIL-----REMOTE AIR-COOLED CONDENSER
250 !      4 =            CHILLER UNIT-----WATER COIL-------COOLING TOWER
260 !
270 !
280 SYSTEM=4 !                          SELECTED FROM THE ABOVE VALUES
290 !
300 !
310 GOSUB CONSTANTS  !                                  input equipment constants
320 GOSUB COIL_CONDITIONS  !                            input all coil loading conditions
330 GOSUB HEADING  !                                    system heading
340 GOSUB HEADING2  !                                   data  heading
350 !
360 FOR x=1 TO X !                                      loop for each system loading
370 !
380  GOSUB COIL_DATA  !                                 individual coil data
390  GOSUB COIL  !                                      find cooling load
400  IF evaporator$="air" THEN GOSUB DX_EVAP  !         finds Tel=Te
410  IF evaporator$="water" THEN GOSUB WATER_EVAP  !    finds Tel=Tcw
420 Tcl=95 !                                            initial estimate of Tcl
430 !
440 INTERATION:
450 GOSUB UNIT_MODEL  !                                 finds full load capacity,power
460 GOSUB PLR  !                                        finds partial load power
470 Qc=Qe+P/3.516 !                                     calculation of condenser load
480 IF condenser$="air" THEN GOSUB AIR_COND  !          find new Tcl (air-cooled condenser)
490 IF condenser$="water" THEN GOSUB WATER_COND  !      find new Tcl (cooling tower)
500 IF ABS (Tc2-Tcl)<.1 THEN GOTO CHECK  !              check for convergence
510 Tcl=Tc2
520 GOTO INTERATION  !                                  continue interation of Tcl
530 !
540 CHECK:
550 IF condenser$="air" THEN GOTO 570
560 IF Tc_in=Tc_min THEN GOSUB FORCED_Tci  !            find reduced cooling tower cfm--if necessary
570 GOSUB OUTPUT  !                                     system steady state performance output
580 !
590 NEXT x !                                            next load condition for processing
600 GOSUB OUTPUT2  !                                    annual load output
610 !
620 END
630 !
640 !
650 COIL:
660 IF UNITS$="WB" THEN GOTO wet_bulb
670 IF UNITS$="G" THEN GOTO grains
690 wet_bulb: !                               inlet enthalpy from wet-bulb
700 Twb=wb_wi
710 Temp=Twb
720 GOSUB HVTA
730 Hin=H
740 Wi=(Hin-.24*Tdb)/(1061+.444*Tdb) !        inlet air moisture content
```

```
750 GOTO dehumid_ck
770 grains: !                                     inlet enthalpy from moisture content
780 Wi=wb_wi/7000
790 GOSUB WETbulb
800 Hin=.24*Tdb+Wi*(1061+.444*Tdb)
810 GOTO dehumid_ck
830 dehumid_ck: !                                 checking for dehumidification
840 Temp=Ts
850 GOSUB HVTA
860 IF WS>Wi THEN GOTO no_dehumid  ELSE GOTO dehumid
880 no_dehumid:
890 WS=Wi
900 Hout=.24*Ts+Wi*(1061+.444*Ts) !   finding enthalpy for no dehumidification
910 GOTO loading
930 dehumid:
940 Hout=H !                          enthalpy for dehumidification
950 GOTO loading
970 loading:
980 IF Tdb<Ts THEN Hout=Hin
990 Qe=CFMe*(Hin-Hout)*4.5 !                      total load in Btuh
1000 Qlat=(Wi-WS)*(1061+.444*Tdb)*CFMe*4.5 !       sensible load in Btuh
1010 Qsens=Qe-Qlat !                              latent   load in Btuh
1020 Qsens=Qsens/12000 !                          load in Tons
1030 Qlat=Qlat/12000 !                            load in Tons
1040 Qe=Qe/12000 !                                load in Tons
1050 RETURN
1060 !
1070 !
1080 UNIT_MODEL:
1090 Tc1=(Tc1-32)*5/9 !                                   convert deg.F to deg.C
1100 Te1=(Te1-32)*5/9 !                                   convert deg.F to deg.C
1110 Teq=(Tc1-Tc1_DES)/Tratio-(Te1-Te1_DES)
1120 ADCR=A1+A2*Teq+A3*Teq^2
1130 CAP=ADCR*CAP_DES !                                   capacity in kws
1140 COPC=(Tc1+273.2)/(Tc1-Te1)
1150 COPCO=(Tc1_DES+273.2)/(Tc1_DES-Te1_DES)
1160 COPCN=COPC/COPCO
1170 COPAN=B1+B2*COPCN+B3*COPCN^2
1180 COPAO=CAP_DES/FLP_DES
1190 COPA=COPAN*COPAO
1200 FLP=CAP/COPA !                                     power in kws
1210 Tc1=Tc1*9/5+32 !                                   convert deg.C to deg.F
1220 Te1=Te1*9/5+32 !                                   convert deg.C to deg.F
1230 CAP=CAP/3.516 !                                    convert kws to tons (CAP)
1240 RETURN
1250 !
1260 !
1270 PLR:
1280 IF condenser$="water" THEN GOTO 1310
1290 CAP=CAP/subcl*subcool !                            correcting for subcooling of air cooled condenser
1300 !                                                  (already included in water cooled condenser systems)
1310 PLR=Qe/CAP
1320 IF PLR>CYCLE THEN GOTO 1350
1330 FFL=PLR*C4 !                                       cycling operation
1340 GOTO 1360
1350 FFL=C1+C2*PLR+C3*PLR^2 !                           continuous operation
1360 P=FFL*FLP
1370 RETURN
1380 !
1390 !
1400 DX_EVAP: !                                        find evap. temp. (Newton-Ralphson)
1410 Te=40
1420 fpm=CFMe/AREA
1430 CF=e1+e2*fpm+e3*fpm^2
1440 EV=Qe-AREA*CF*(E1+E2*Twb+E3*Twb^2+E4*Te+E5*Te*Twb+E6*Te*Twb^2+E7*Te^2+E8*Twb*Te^2+E9*Twb^2*Te^2)
1450    DERIV=-(AREA*CF*(E4+E5*Twb+E6*Twb^2+2*E7*Te+2*E8*Twb*Te+2*E9*Twb^2*Te))
1460    Te=Te-EV/DERIV
1470  IF ABS (EV/DERIV)>.001 THEN GOTO 1440
1480 Te1=Te
1490 RETURN
1500 !
1510 !
1520 WATER_EVAP: !                                     finds chilled water temp.
1530 IF wtr_cntrl$="return" THEN GOTO 1560
1540 Tcw=Tcntrl_temp !                                 supply water temp control
```

```
1550 GOTO 1570
1560 Tcw=Tcntrl_temp-Qe*24/GPMe !                        return water temp control
1570 Tel=Tcw !                                                 chilled water temp
1580 RETURN
1590 !
1600 !
1610 AIR_COND: !                                         finds condensing temp. and subcooling
1620 DTc=(Qc-F1)/F2
1630 subcool=D1+D2*DTc+D3*DTc^2
1640 Tc=TDB+DTc
1650 IF Tc<Tc_min THEN Tc=Tc_min !                        holding min cond. temp
1660 Tc2=Tc
1670 RETURN
1680 !
1690 !
1700 WATER_COND: !                                        finds condensing water temp.
1710 Tco=Tc1
1720 GOSUB FREE_Tci
1730 IF Tci<Tc_min THEN Tc_in=Tc_min ELSE Tc_in=Tci ! holding min. water temp
1740 Tc_out=Tc_in+Qc*24/GPMc !                            new leaving water temp
1750 Tc2=Tc_out
1760 RETURN
1770 !
1780 !
1790 COOLING_TWR:
1800 !
1810 FREE_Tci: !                                         floating leaving water temp
1820 no_answer=0
1830 CFMt=CFM_twr
1840 R=0
1850 INTER$="initial"
1860 R1g=GPMc*8.33/(CFM_twr*.0712)
1870 NTUA=C*(R1g^(-(1+XN)))
1880 initial:
1890 R=R+2
1900 GOTO 1930
1910 final:
1920 R=(R1+Rs)/2
1930 A=Tco-R-TWB
1940 GOSUB COUNT
1950 NTU=CT/R1g
1960  IF ABS (NTUA-NTU)/NTUA<.001 THEN GOTO 2020
1970     IF NTUA>NTU THEN Rs=R
1980      IF NTUA<NTU THEN R1=R
1990      IF NTUA<NTU THEN INTER$="final"
2000        IF INTER$="final" THEN GOTO final
2010  GOTO initial
2020 Tci=Tco-R
2030 RETURN
2040 !
2050 FORCED_Tci: !                                    fixed leaving water temp
2060 R=Tco-Tc_in
2070 A=Tc_in-TWB
2080 CFMt=CFM_twr
2090 CFMh=CFM_twr
2100 INTER$="initially"
2110 GOTO 2190
2120 initially:
2130 IF CFMt/CFM_twr>.3 THEN CFMt=CFMt-.1*CFM_twr ELSE CFNt=CFNt-.01*CFN_twr
2140 IF CFNt=0 THEN no_answer=1
2150 IF CFNt=0 THEN GOTO 2290
2160 GOTO 2190
2170 finally:
2180 CFMt=(CFMh+CFM1)/2
2190 R1g=GPNc*8.33/(.0712*CFMt)
2200 NTUA=C*(R1g^(-(1+XN)))
2210 GOSUB COUNT
2220 NTU=CT/R1g
2230  IF ABS ((NTUA-NTU)/NTUA)<.001 THEN GOTO 2290
2240     IF NTUA>NTU THEN CFMh=CFMt
2250     IF NTUA<NTU THEN CFM1=CFMt
2260     IF NTUA<NTU THEN INTER$="finally"
2270       IF INTER$="finally" THEN GOTO finally
2280  GOTO initially
2290 RETURN
```

```
2300 !
2310 !
2320 COUNT: !                                   cooling tower subroutine
2330 CT=0
2340 Y(1)=.1
2350 Y(2)=.3
2360 Y(3)=.2
2370 Y(4)=.3
2380 Temp=TWB
2390 GOSUB HVTA
2400 HA=H
2410 TW=TWB+A
2420 FOR I=1 TO 4
2430   TW=TW+Y(I)*R
2440   HA=HA+RIg*Y(I)*R
2450   Temp=TW
2460   GOSUB HVTA
2470   HT=H
2480   DH=HT-HA
2490   CT=CT+1/DH
2500 NEXT I
2510 CT=CT*R/4
2520 RETURN
2530 !
2540 !
2550 HVTA: !                                       enthalpy subroutine
2560 PB=101325
2570 GOSUB PSAT
2580 WS=.622*(PS/(PB-PS))
2590 H=.24*Temp+WS*(1061+.444*Temp)
2600 RETURN
2610 !
2620 !
2630 WETbulb: !                                    wet-bulb subroutine
2640 Temp_H=Tdb
2650 Temp_L=0
2660 h=.24*Tdb+Wi*(1061+.444*Tdb)
2670 PB=101325
2680 Temp=(Temp_H+Temp_L)/2
2690 GOSUB PSAT
2700 Twb=Temp
2710 WS=.622*(PS/(PB-PS))
2720 H=.24*Twb+WS*(1061+.444*Twb)
2730 IF ABS (H-h)<.005 THEN GOTO 2770
2740  IF H>h THEN Temp_H=Temp
2750  IF H<h THEN Temp_L=Temp
2760 GOTO 2680
2770 RETURN
2780 !
2790 !
2800 PSAT: !                                 saturated pressure subroutine
2810 A(1,1)=6.41533947
2820 A(1,2)=6.41542599
2830 A(1,3)=6.44302266
2840 A(1,4)=6.53947838
2850 A(2,1)=.0821033478
2860 A(2,2)=.0724023398
2870 A(2,3)=.0698803796
2880 A(2,4)=.0656725584
2890 A(3,1)=-.000340068991
2900 A(3,2)=-.000264361973
2910 A(3,3)=-2.0606744496E-4
2920 A(3,4)=-.000160854949
2930  TS=(Temp-32)*5/9
2940  IF TS>22.0322 THEN GOTO 2950 ELSE GOTO 2990
2950   IF TS<46.0304 THEN J=3 ELSE GOTO 2970
2960   GOTO 3050
2970    IF TS<72.0476 THEN J=4 ELSE GOTO 3030
2980    GOTO 3050
2990  IF TS>.00895 THEN J=2 ELSE GOTO 3010
3000   GOTO 3050
3010    IF TS>-19.94 THEN J=1 ELSE GOTO 3030
3020    GOTO 3050
3030 PRINT "           TEMPERATURE IS OUTSIDE OF OUR RANGE"
3040 PAUSE
```

```
3050 PS=EXP (A(1,J)+TS*(A(2,J)+TS*A(3,J)))
3060 RETURN
3070 !
3080 !
3090 HEADING:
3100  PRINT CHR$ (12)
3110 PRINT "                                                                                    SYSTEM"
3120 PRINT "                        --------------------------------------------------------------------------------
--------------------------"
3130 IF SYSTEM=1 THEN PRINT USING "49X,75A" ; "COMPRESSOR-------DX COIL------REMOTE AIR-COOLED CONDENSER"
3140 IF SYSTEM=2 THEN PRINT USING "34X,75A" ; " WATER COOLED CONDENSING UNIT--DX COIL--------COOLING TOWER
"
3150 IF SYSTEM=3 THEN PRINT USING "34X,75A" ; " COMPRESSOR CHILLER UNIT-----WATER COIL-----REMOTE AIR-COOL
ED CONDENSER"
3160 IF SYSTEM=4 THEN PRINT USING "47X,75A" ; "CHILLER UNIT-----WATER COIL-------COOLING TOWER"
3170 IF evaporator$="water" THEN PRINT USING "44X,14A,6A,15A,DD.D,6A" ; "chilled water ",wtr_cntrl$," temp
. held at  ",Tcntrl_temp," deg.F"
3180 PRINT
3190 PRINT
3200 RETURN
3210 !
3220 !
3230 OUTPUT:
3240 !
3250  D$=" *** "
3260  F$="  ***** "
3270  G$="   *****"
3280 IF condenser$="water" THEN GOTO 3310
3290 PRINT USING "3D,X,5(4D.D),6D,3(5D.D),3(4D.D),2X,3(6A),8A,2(8A),X,5A" ; x,TDB,TWB,Tdb,Twb,Ts,CFMe,Tei,
Tci,PLR*100,Qe,P,Qc,D$,D$,D$,F$,F$,F$,D$
3300 GOTO 3380
3310 P_C1=(Tco-Tc_in)/(Tco-Tci)*100
3320 P_C2=CFMt/CFM_twr*100
3330 IF no_answer=0 THEN GOTO 3360
3340 PRINT USING "3D,X,5(4D.D),6D,3(5D.D),3(4D.D),3(4D.D),5D.D,X,3(9A)" ; x,TDB,TWB,Tdb,Twb,Ts,CFMe,Tei,Tc
i,PLR*100,Qe,P,Qc,Tco,Tci,Tc_in,P_C1,G$,G$,F$
3350 GOTO 3380
3360 PRINT USING "3D,X,5(4D.D),6D,3(5D.D),6(4D.D),5D.D,X,2(8D),6D.D" ; x,TDB,TWB,Tdb,Twb,Ts,CFMe,Tei,Tci,P
LR*100,Qe,P,Qc,Tco,Tci,Tc_in,P_C1,CFM_twr,CFMt,P_C2
3370 !
3380 IF x=1 THEN LOAD_annual=0 !                     annual variables
3390 IF x=1 THEN Qsens_annual=0
3400 IF x=1 THEN Qlat_annual=0
3410 IF x=1 THEN hrs_annual=0
3420 IF x=1 THEN P_annual=0
3430 Qsens(x)=Qsens*12
3440 Qlat(x)=Qlat*12
3450 Qe(x)=Qe*12
3460 P(x)=P
3470 Qsens_annual=Qsens_annual+Qsens(x)*hrs(x)
3480 Qlat_annual=Qlat_annual+Qlat(x)*hrs(x)
3490 hrs_annual=hrs_annual+hrs(x)
3500 LOAD_annual=LOAD_annual+Qe(x)*hrs(x)
3510 P_annual=P_annual+P*hrs(x)
3520 RETURN
3530 !
3540 OUTPUT2:
3550 PRINT
3560 PRINT
3570 PRINT "                                    STEADY STATE                                          ANNU
AL"
3580 PRINT "                          ----------------------------------------     --------------------
-------------------"
3590 PRINT "              CASE  Qsensible  Qlatent  Qtotal   Ql/Qt   POWER     hours       load/yr
     power/yr   "
3600 PRINT "              #     ..........(Mbtuh) .......   %    (kws)     (hrs)    (MBTU)
%    (kw-hrs)       %"
3610 PRINT "              ----- --------- -------- -------  ----- ------    ------  -------- ---
--  --------    -----"
3620 FOR I=1 TO X
3630 P_C4=Qlat(I)/Qe(I)*100
3640 Q5=Qe(I)*hrs(I)
3650 P_C5=Q5/LOAD_annual*100
3660 P6=P(I)*hrs(I)
3670 P_C6=P6/P_annual*100
```

```
3680 PRINT USING "19X,4D,9D.D,7D.D,7D.D,6D.D,6D.D,10D,11D,5D.D,11D,7D.D" ; I,Qsens(I),QIat(I),Qe(I),P_C4,P
(I),hrs(I),Q5,P_C5,P6,P_C6
3690 NEXT I
3700 !
3710 PRINT USING "3/"
3720 PRINT USING "30X,34A,5D.DD,18A" ; "Annual latent   cooling load   = ";QIat_annual/1000,"   MMBtu"
3730 PRINT USING "30X,34A,5D.DD,18A" ; "Annual sensible cooling load   = ";Qsens_annual/1000,"   MMBtu"
3740 PRINT USING "30X,34A,5D.DD,18A" ; "Annual total    cooling load   = ";LDAD_annual/1000,"   MMBtu"
3750 PRINT
3760 PRINT USING "30X,34A,8D,18A" ; "Annual hours of operation      = ";hrs_annual,"   Hours"
3770 PRINT USING "30X,34A,8D,19A" ; "Annual Power Consumption       = ";P_annual," Kw-hrs"
3780 RETURN
3790 !
3800 !
3810 HEADING2:
3820 PRINT "                                                          SYSTEM  STEADY  STATE  OPERATION "
3830 PRINT "          ---------------------------------------------------------------------------------------
------------------------------"
3840 PRINT "     .ambient.   ..........coil........     ...........system............   ................
...cooling tower.................."
3850 PRINT "CASE   TDB   TWB   Tdb   Twb   Ts    CFM    TeI   TcI    PLR    Qe    P    Qc    Tco    Tci  T
c_in    %   CFM_twr   CFMt    %"
3860 PRINT " #   ...........deg.F.............   cfm    ..deg.F..    %   tons  kws  tons   ...deg.F....
..    (Qtwr)   ...cfm...   (CFMtwr)"
3870 PRINT "---  ----- ----- ----- ----- ----- ------  ----  ----- ------ ----- ----  -----  ----  ----- -
----  ------ ------- ------- -------"
3880 RETURN
3890 !
3900 !
3910 CONSTANTS:
3920 ON SYSTEM GOTO UNIT1 ,UNIT2 ,UNIT3 ,UNIT4
3930 !
3940 UNIT1: !                              (compressor subsytem) system
3950 condenser$="air" !                   constants good for all unit1 systems--lines 3950 to 4080
3960 evaporator$="air"
3970 Tratio=2.9
3980 A1=.9973
3990 A2=-.03694
4000 A3=.0004335
4010 B1=-.4141
4020 B2=1.7717
4030 B3=-.3694
4040 C1=.1456
4050 C2=.9555
4060 C3=-.1048
4070 Tc1_DES=(105-32)*5/9 !               temp at which CAP_DES,FLP_DES selected
4080 Te1_DES=(40-32)*5/9 !                temp at which CAP_DES,FLP_DES selected
4090 !
4100 !             UNIT_CONSTANTS
4110 CAP_DES=17.3*3.516 !        compressor capacity in kws ( from mach. data )
4120 FLP_DES=15.9 !              compressor power    in kws ( from mach. data )
4130 CYCLE=.25 !                                    PLR at which system cycles
4140 C4=(C1+C2*CYCLE+C3*CYCLE^2)/CYCLE !            system cycling constant
4150 !
4160 !             DX-COIL_CONSTANTS
4170 GOSUB DX_CONSTANTS  !               loading  DX_EVAP constants
4180 AREA=5 !                            DX-EVAP face area selected by designer(sq.ft.)
4190 !
4200 !           CONDENSER_CONSTANTS
4210 subcl=1.075 !                       unit base on 16 deg.F subcooling
4220 subcool=subcl !                     used to find actual subcooling
4230 D1=.9774 !                          constants for subcooling
4240 D2=.0044685 !
4250 D3=-.00002978 !
4260 F2=.94203 !                         load/dt  of air-cooled condenser (tons/deg.F)
4270 F1=-3.4047 !                        offset from zero of above curve
4280 Tc_min=85 !                         minimum condensing temperature (deg.F)
4290 RETURN
4300 !
4310 !
4320 UNIT2: !                            (water cooled condensing subsystem)system
4330 condenser$="water" !                constants good for all unit2 systems--lines 4330 to 4460
4340 evaporator$="air"
4350 Tratio=2.9
4360 A1=.9982
```

```
4370 A2=-.03952
4380 A3=.004931
4390 B1=-.2244
4400 B2=1.5438
4410 B3=-.3219
4420 C1=.129
4430 C2=.8849
4440 C3=-.01302
4450 Tcl_DES=(95-32)*5/9 !                   temp at which CAP_DES,FLP_DES selected
4460 Tel_DES=(40-32)*5/9 !                   temp at which CAP_DES,FLP_DES selected
4470 !
4480 !              UNIT CONSTANTS
4490 CAP_DES=16.8*3.516 !        compressor capacity in kws ( from mach. data)
4500 FLP_DES=16.6 !              compressor power    in kws ( from mach. data)
4510 CYCLE=.25 !                                    PLR at which system cycles
4520 C4=(C1+C2*CYCLE+C3*CYCLE^2)/CYCLE !            system cycling constant
4530 !
4540 !              DX-EVAP CONSTANTS
4550 GOSUB DX_CONSTANTS  !       loading DX-EVAP constants
4560 AREA=5 !                    DX-EVAP face area selected by designer(sq.ft.)
4570 !
4580 !              COOL_TWR_CONSTANTS
4590 C=1.23146 !                 Webb-Villacres cooling tower constant
4600 XN=.9193 !                  Webb-Villacres cooling tower constant
4610 CFM_twr=4800 !              air flow rate of cooling tower ( cfm )
4620 GPMC=44.9 !                 water flow rate of tower,condenser ( gpm )
4630 Tc_min=75 !                 minimum entering condensing water temp(deg.F)
4640 RETURN
4650 !
4660 !
4670 UNIT3: !                    (compressor chiller subsystem)system
4680 condenser$="air" !          constants good for all unit3 systems--lines 4680 to 4810
4690 evaporator$="water"
4700 Tratio=2.9
4710 A1=.9982
4720 A2=-.0299
4730 A3=.0001711
4740 B1=-.2484
4750 B2=1.66
4760 B3=-.4105
4770 C1=.1545
4780 C2=.8323
4790 C3=.01672
4800 Tcl_DES=(105-32)*5/9 !             temp at which CAP_DES,FLP_DES selected
4810 Tel_DES=(46-32)*5/9 !              temp at which CAP_DES,FLP_DES selected
4820 !
4830 !              UNIT CONSTANTS
4840 CAP_DES=16.1*3.516 !     comp-chiller capacity in kws ( from mach. data )
4850 FLP_DES=15.6 !           comp-chiller power    in kws ( from mach. data )
4860 CYCLE=.25 !                                    PLR at which system cycles
4870 C4=(C1+C2*CYCLE+C3*CYCLE^2)/CYCLE !            system cycle constant
4880 !
4890 !              CONDENSER_CONSTANTS
4900 subcl=1.075 !                     unit based of 16 deg.F subcooling
4910 subcool=subcl !                   used to find actual subcooling
4920 D1=.9774 !                        constants for subcooling(all TRANE condensers)
4930 D2=.0044685 !                     "
4940 D3=-.00002978 !                   "               "
4950 F2=.94203 !                       load/dt of air-cooled condenser (tons/deg.F)
4960 F1=-3.4047 !                      offset from zero of above curve
4970 Tc_min=85 !                       minimum condensing temperature (deg.F)
4980 !
4990 !              COIL_CONSTANTS
5000 wtr_cntrl$="supply" !        chilled water temp control--return or supply
5010 Tcntrl_temp=45 !             temp at which above water is held at (deg.F)
5020 GPMe=34.5 !                  chilled water flow rate   (gpm)
5030 RETURN
5040 !
5050 !
5060 UNIT4: !                         (chiller substem) system
5070 condenser$="water" !         constants good for all unit4 systems--lines 5070 to 5200
5080 evaporator$="water"
5090 Tratio=2.9
5100 A1=.9978
5110 A2=-.02943
```

```
5120 A3=.0001472
5130 B1=-.1119
5140 B2=1.5815
5150 B3=-.4702
5160 C1=.1334
5170 C2=.7439
5180 C3=.1209
5190 Tc1_DES=(95-32)*5/9 !                 temp at which CAP_DES,FLP_DES selected
5200 Te1_DES=(46-32)*5/9 !                 temp at which CAP_DES,FLP_DES selected
5210 !
5220 !               UNIT_CONSTANTS
5230 CAP_DES=15.8*3.516 !      chiller  capacity  in kws ( from mach. data )
5240 FLP_DES=16 !              chiller  power     in kws ( from mach. data )
5250 CYCLE=.25 !                                  PLR at which system cycles
5260 C4=(C1+C2*CYCLE+C3*CYCLE^2)/CYCLE !          system cycling constant
5270 !
5280 !               COIL_CONSTANTS
5290 wtr_cntrl$="supply" !     chilled water temp control--return or supply
5300 Tcntrl_temp=45 !          temp at which above water is held at (deg.F)
5310 GPMe=34.5 !               chilled water flow rate (gpm)
5320 !
5330 !               COOL_TWR_CONSTANTS
5340 C=1.23146 !               Webb-Villacres cooling tower constant
5350 XN=.9193 !                Webb-Villacres cooling tower constant
5360 CFM_twr=4800 !            air flow rate of cooling tower ( cfm )
5370 GPMc=44.8 !               water flow rate of tower,condenser ( gpm )
5380 Tc_min=75 !               minimum entering condensing water temp (deg.F)
5390 RETURN
5400 !
5410 !
5420 DX_CONSTANTS: !           constants for all 6-row AEROFIN 'wet' DX-coil evaporators
5430 E1=6.08552
5440 E2=-.0866875
5450 E3=.000813752
5460 E4=-.242335
5470 E5=.0021156
5480 E6=.00001460892
5490 E7=.0018496
5500 E8=-.00001131545
5510 E9=-.000000323206
5520 e1=.0911429
5530 e2=.002245425
5540 e3=-.000000855422
5550 RETURN
5560 !
5570 !
5580 COIL_CONDITIONS:
5590 !
5600 !           ambient                              coil conditions
5610 ! EX                                     inlet                      outlet
5620 ! --      -----------------      ---------------------------      -------
5630 !         dry-bulb ,wet-bulb,    dry-bulb,wet-bulb DR grains,cfm , dry-bulb
5640 !  DATA     95    ,   81    ,      95    ,  75,WB,           2500,   55
5650 !  DATA     95    ,   81    ,      95    ,             120,G, 2500,   55
5660 !
5670 X=1
5680  READ hrs(X),TDB(X),TWB(X),Tdb(X),wb_wi(X),UNITS$(X),CFMe(X),Ts(X)
5690  IF TDB(X)=0 THEN GOTO 5720
5700  X=X+1
5710  GOTO 5680
5720  X=X-1
5730 RETURN
5740 !
5750  DATA 7,97,79,97,79,WB,2000,55
5760  DATA 142,92,78,92,78,WB,2000,55
5770  DATA 574,87,77,87,77,WB,2000,55
5780  DATA 1008,82,75,85,76,WB,2000,55
5790  DATA 1526,77,73,80,74,WB,2000,55
5800  DATA 1325,72,69,76,72,WB,1850,55
5810  DATA 925,67,63,74,70,WB,1611,55
5820  DATA 779,62,58,70,68,WB,1500,55
5830  DATA 701,57,53,68,64,WB,1500,55
5840 DATA 0,0,0,0,0,WB,0,0
5850 !
5860 COIL_DATA:
```

```
5870 TD8=TD8(x)
5880 TW8=TW8(x)
5890 Tdb=Tdb(x)
5900 wb wi=wb wi(x)
5910 UNITS$=UNITS$(x)
5920 CFMe=CFMe(x)
5930 Ts=Ts(x)
5940 RETURN
```

APPENDIX D

TYPICAL ANNUAL PERFORMANCE OUTPUT

(single coil systems)

SYSTEM

COMPRESSOR-------DX COIL------REMOTE AIR-COOLED CONDENSER

SYSTEM STEADY STATE OPERATION

CASE	.ambient.		..........coil........				..........system..........						.....cooling tower.....							
#	TDB	TWB	Tdb	Twb	Ts	CFM	Tel	Tcl	PLR	Qe	P	Qc	Tco	Tci	Tc_in	%	CFM_twr	CFMt	%
	..........deg.F..........					cfm	..deg.F..		%	tons	kws	tons	...deg.F......			(Qtwr)	...cfm...		(CFMtwr)
1	97.0	79.0	97.0	79.0	55.0	2000	37.8	121.2	99.8	14.5	17.4	19.5	***	***	***	*****	*****	*****	***
2	92.0	78.0	92.0	78.0	55.0	2000	38.8	114.8	88.6	13.7	15.5	18.1	***	***	***	*****	*****	*****	***
3	87.0	77.0	87.0	77.0	55.0	2000	39.7	108.5	78.7	13.0	13.7	16.9	***	***	***	*****	*****	*****	***
4	82.0	75.0	82.0	75.0	55.0	2000	41.4	101.2	64.5	11.5	11.3	14.7	***	***	***	*****	*****	*****	***
5	77.0	73.0	77.0	73.0	55.0	2000	43.0	94.1	52.6	10.1	9.2	12.7	***	***	***	*****	*****	*****	***
6	72.0	69.0	72.0	69.0	55.0	2000	45.9	85.5	35.3	7.5	6.5	9.4	***	***	***	*****	*****	*****	***
7	67.0	63.0	67.0	63.0	55.0	2000	49.9	85.0	17.8	4.0	3.7	5.1	***	***	***	*****	*****	*****	***
8	62.0	58.0	62.0	58.0	55.0	2000	53.4	85.0	6.0	1.4	1.3	1.8	***	***	***	*****	*****	*****	***

CASE	STEADY STATE					ANNUAL				
#	Qsensible	Qlatent	Qtotal	Ql/Qt	POWER	hours	load/yr		power/yr	
	..........(Mbtuh)			%	(kws)	(hrs)	(MBTU)	%	(kw-hrs)	%
1	92.3	81.8	174.1	47.0	17.4	7	1219	.2	122	.3
2	81.3	83.5	164.7	50.7	15.5	142	23393	3.8	2195	4.5
3	70.3	85.3	155.6	54.8	13.7	574	89327	14.5	7843	16.1
4	59.3	78.7	138.0	57.0	11.3	1008	139147	22.6	11355	23.3
5	48.3	73.0	121.3	60.2	9.2	1526	185079	30.1	14064	28.9
6	37.3	52.6	90.0	58.5	6.5	1325	119232	19.4	8638	17.7
7	26.4	21.7	48.1	45.2	3.7	925	44471	7.2	3465	7.1
8	15.4	1.7	17.1	10.0	1.3	779	13312	2.2	1005	2.1

Annual latent cooling load = 343.26 MMBtu
Annual sensible cooling load = 271.92 MMBtu
Annual total cooling load = 615.18 MMBtu

Annual hours of operation = 6286 Hours
Annual Power Consumption = 48687 Kw-hrs

(ambient makeup air data run)

SYSTEM

WATER COOLED CONDENSING UNIT--DX COIL--------COOLING TOWER

SYSTEM STEADY STATE OPERATION

CASE #	.ambient. TDB deg.F	TWB deg.F	...coil... Tdb deg.F	Twb deg.F	Ts deg.F	CFM cfm	...system... Tel deg.F	Tcl deg.F	PLR %	Qe tons	P kws	Qc tons	...cooling tower... Tco deg.F	Tci deg.F	Tc_in deg.F	% (Qtwr)	CFM_twr cfm	CFMt cfm	% (CFMtwr)
1	97.0	79.0	97.0	79.0	55.0	2000	37.8	95.7	90.5	14.5	15.3	18.9	95.7	85.7	85.7	100.0	4800	4800	100.0
2	92.0	78.0	92.0	78.0	55.0	2000	38.8	94.3	83.5	13.7	14.1	17.7	94.3	84.7	84.7	100.0	4800	4800	100.0
3	87.0	77.0	87.0	77.0	55.0	2000	39.7	92.6	76.4	13.0	13.0	16.7	92.6	83.6	83.6	100.0	4800	4800	100.0
4	82.0	75.0	82.0	75.0	55.0	2000	41.4	89.3	62.9	11.5	11.0	14.6	89.3	81.4	81.4	100.0	4800	4800	100.0
5	77.0	73.0	77.0	73.0	55.0	2000	43.0	86.0	50.7	10.1	9.4	12.8	86.0	79.1	79.1	100.0	4800	4800	100.0
6	72.0	69.0	72.0	69.0	55.0	2000	45.9	80.1	31.3	7.5	6.8	9.4	80.1	74.7	75.0	94.8	4800	4463	93.0
7	67.0	63.0	67.0	63.0	55.0	2000	49.9	77.7	13.8	4.0	3.6	5.0	77.7	71.1	75.0	41.0	4800	1629	33.9
8	62.0	58.0	62.0	58.0	55.0	2000	53.4	75.9	4.1	1.4	1.2	1.8	75.9	68.4	75.0	12.5	4800	455	9.5

CASE #	STEADY STATE Qsensible (Mbtuh)	Qlatent (Mbtuh)	Qtotal	Ql/Qt %	POWER (kws)	ANNUAL hours (hrs)	load/yr (MBTU)	%	power/yr (kw-hrs)	%
1	92.3	81.8	174.1	47.0	15.3	7	1219	.2	107	.2
2	81.3	83.5	164.7	50.7	14.1	142	23393	3.8	2004	4.1
3	70.3	85.3	155.6	54.8	13.0	574	89327	14.5	7465	15.5
4	59.3	78.7	138.0	57.0	11.0	1008	139147	22.6	11130	23.0
5	48.3	73.0	121.3	60.2	9.4	1526	185079	30.1	14303	29.6
6	37.3	52.6	90.0	58.5	6.8	1325	119232	19.4	9052	18.7
7	26.4	21.7	48.1	45.2	3.6	925	44471	7.2	3297	6.8
8	15.4	1.7	17.1	10.0	1.2	779	13312	2.2	942	2.0

Annual latent cooling load	=	343.26	MMBtu
Annual sensible cooling load	=	271.92	MMBtu
Annual total cooling load	=	615.18	MMBtu
Annual hours of operation	=	6286	Hours
Annual Power Consumption	=	48300	Kw-hrs

(ambient makeup air data run)

SYSTEM

COMPRESSOR CHILLER UNIT-----WATER COIL-----REMOTE AIR-COOLED CONDENSER
chilled water supply temp. held at 45.0 deg.F

SYSTEM STEADY STATE OPERATION

CASE	.ambient.		..........coil........				............system.............						....................cooling tower..................						
	TOB	TWB	Tdb	Twb	Ts	CFM	Tei	Tci	PLR	Qe	P	Qc	Tco	Tci	Tc_in	%	CFM_twr	CFMt	%
#	...........deg.F............					cfm	..deg.F..		%	tons	kws	tons	...deg.F......			(Qtwr)	...cfm...		(CFMtwr)
1	97.0	79.0	97.0	79.0	55.0	2000	45.0	121.2	101.8	14.5	17.4	19.5	###	###	###	#####	#####	#####	###
2	92.0	78.0	92.0	78.0	55.0	2000	45.0	114.9	93.0	13.7	15.6	18.2	###	###	###	#####	#####	#####	###
3	87.0	77.0	87.0	77.0	55.0	2000	45.0	108.5	85.0	13.0	13.9	16.9	###	###	###	#####	#####	#####	###
4	82.0	75.0	82.0	75.0	55.0	2000	45.0	101.3	72.8	11.5	11.6	14.8	###	###	###	#####	#####	#####	###
5	77.0	73.0	77.0	73.0	55.0	2000	45.0	94.3	61.9	10.1	9.7	12.9	###	###	###	#####	#####	#####	###
6	72.0	69.0	72.0	69.0	55.0	2000	45.0	85.7	44.3	7.5	7.1	9.5	###	###	###	#####	#####	#####	###
7	67.0	63.0	67.0	63.0	55.0	2000	45.0	85.0	24.0	4.0	4.7	5.3	###	###	###	#####	#####	#####	###
8	62.0	58.0	62.0	58.0	55.0	2000	45.0	85.0	8.6	1.4	1.7	1.9	###	###	###	#####	#####	#####	###

CASE	STEADY STATE					ANNUAL				
	Qsensible	Qlatent	Qtotal	Ql/Qt	POWER	hours	load/yr		power/yr	
#	..........(Mbtuh)		....	%	(kws)	(hrs)	(MBTU)	%	(kw-hrs)	%
1	92.3	81.8	174.1	47.0	17.4	7	1219	.2	122	.2
2	81.3	83.5	164.7	50.7	15.6	142	23393	3.8	2211	4.3
3	70.3	85.3	155.6	54.8	13.9	574	89327	14.5	7968	15.4
4	59.3	78.7	138.0	57.0	11.6	1008	139147	22.6	11721	22.6
5	48.3	73.0	121.3	60.2	9.7	1526	185079	30.1	14786	28.6
6	37.3	52.6	90.0	58.5	7.1	1325	119232	19.4	9368	18.1
7	26.4	21.7	48.1	45.2	4.7	925	44471	7.2	4305	8.3
8	15.4	1.7	17.1	10.0	1.7	779	13312	2.2	1308	2.5

Annual latent cooling load = 343.26 MMBtu
Annual sensible cooling load = 271.92 MMBtu
Annual total cooling load = 615.18 MMBtu

Annual hours of operation = 6286 Hours
Annual Power Consumption = 51789 Kw-hrs

(ambient makeup air data run)

SYSTEM

COMPRESSOR CHILLER UNIT-----WATER COIL-----REMOTE AIR-COOLED CONDENSER
chilled water return temp. held at 55.0 deg.F

SYSTEM STEADY STATE OPERATION

CASE	.ambient.		..........coil.......				...........system.............						...................cooling tower..................						
	TOB	TWB	Tdb	Twb	Ts	CFM	Tel	Tcl	PLR	Qe	P	Qc	Tco	Tci	Tc_in	%	CFM_twr	CFMt	%
#	..........deg.F...........					cfm	..deg.F..		%	tons	kws	tons	...deg.F......			(Qtwr)	...cfm...		(CFMtwr)
1	97.0	79.0	97.0	79.0	55.0	2000	44.9	121.2	101.9	14.5	17.4	19.5	###	###	###	#####	#####	#####	###
2	92.0	78.0	92.0	78.0	55.0	2000	45.5	114.8	92.3	13.7	15.5	18.1	###	###	###	#####	#####	#####	###
3	87.0	77.0	87.0	77.0	55.0	2000	46.0	108.5	83.6	13.0	13.7	16.9	###	###	###	#####	#####	#####	###
4	82.0	75.0	82.0	75.0	55.0	2000	47.0	101.2	70.4	11.5	11.4	14.7	###	###	###	#####	#####	#####	###
5	77.0	73.0	77.0	73.0	55.0	2000	48.0	94.2	59.0	10.1	9.4	12.8	###	###	###	#####	#####	#####	###
6	72.0	69.0	72.0	69.0	55.0	2000	49.8	85.6	41.2	7.5	6.8	9.4	###	###	###	#####	#####	#####	###
7	67.0	63.0	67.0	63.0	55.0	2000	52.2	85.0	21.5	4.0	4.3	5.2	###	###	###	#####	#####	#####	###
8	62.0	58.0	62.0	58.0	55.0	2000	54.0	85.0	7.5	1.4	1.5	1.9	###	###	###	#####	#####	#####	###

CASE	STEADY STATE					ANNUAL				
	Qsensible	Qlatent	Qtotal	Ql/Qt	POWER	hours	load/yr		power/yr	
#	..........(Mbtuh)		....	%	(kws)	(hrs)	(MBTU)	%	(kw-hrs)	%
1	92.3	81.8	174.1	47.0	17.4	7	1219	.2	122	.2
2	81.3	83.5	164.7	50.7	15.5	142	23393	3.8	2200	4.4
3	70.3	85.3	155.6	54.8	13.7	574	89327	14.5	7878	15.7
4	59.3	78.7	138.0	57.0	11.4	1008	139147	22.6	11452	22.9
5	48.3	73.0	121.3	60.2	9.4	1526	185079	30.1	14300	28.5
6	37.3	52.6	90.0	58.5	6.8	1325	119232	19.4	8995	18.0
7	26.4	21.7	48.1	45.2	4.3	925	44471	7.2	3957	7.9
8	15.4	1.7	17.1	10.0	1.5	779	13312	2.2	1185	2.4

Annual latent cooling load = 343.26 MMBtu
Annual sensible cooling load = 271.92 MMBtu
Annual total cooling load = 615.18 MMBtu

Annual hours of operation = 6286 Hours
Annual Power Consumption = 50089 Kw-hrs

(ambient makeup air data run)

SYSTEM

CHILLER UNIT-----WATER COIL-------COOLING TOWER
chilled water supply temp. held at 45.0 deg.F

SYSTEM STEADY STATE OPERATION

CASE #	ambient TDB (deg.F)	ambient TWB (deg.F)	coil Tdb (deg.F)	coil Twb (deg.F)	coil Ts (deg.F)	coil CFM cfm	system Tei (deg.F)	system Tci (deg.F)	system PLR %	system Qe tons	system P kws	system Qc tons	cooling tower Tco (deg.F)	cooling tower Tci (deg.F)	cooling tower Tc_in (deg.F)	cooling tower % (Qtwr)	cooling tower CFM_twr cfm	cooling tower CFMt cfm	cooling tower % (CFMtwr)
1	97.0	79.0	97.0	79.0	55.0	2000	45.0	95.6	93.9	14.5	15.0	18.8	95.6	85.7	85.7	100.0	4800	4800	100.0
2	92.0	78.0	92.0	78.0	55.0	2000	45.0	94.2	88.1	13.7	14.0	17.7	94.2	84.7	84.7	100.0	4800	4800	100.0
3	87.0	77.0	87.0	77.0	55.0	2000	45.0	92.6	82.5	13.0	13.0	16.7	92.6	83.6	83.6	100.0	4800	4800	100.0
4	82.0	75.0	82.0	75.0	55.0	2000	45.0	89.4	71.9	11.5	11.2	14.7	89.4	81.4	81.4	100.0	4800	4800	100.0
5	77.0	73.0	77.0	73.0	55.0	2000	45.0	86.1	62.0	10.1	9.7	12.9	86.1	79.2	79.2	100.0	4800	4800	100.0
6	72.0	69.0	72.0	69.0	55.0	2000	45.0	80.1	44.5	7.5	7.1	9.5	80.1	74.7	75.0	94.3	4800	4425	92.2
7	67.0	63.0	67.0	63.0	55.0	2000	45.0	77.8	23.5	4.0	4.5	5.3	77.8	71.2	75.0	42.5	4800	1695	35.3
8	62.0	58.0	62.0	58.0	55.0	2000	45.0	76.0	8.3	1.4	1.6	1.9	76.0	68.4	75.0	13.2	4800	480	10.0

CASE #	STEADY STATE Qsensible (Mbtuh)	STEADY STATE Qlatent (Mbtuh)	STEADY STATE Qtotal (Mbtuh)	STEADY STATE Ql/Qt %	STEADY STATE POWER (kws)	ANNUAL hours (hrs)	ANNUAL load/yr (MBTU)	ANNUAL load/yr %	ANNUAL power/yr (kw-hrs)	ANNUAL power/yr %
1	92.3	81.8	174.1	47.0	15.0	7	1219	.2	105	.2
2	81.3	83.5	164.7	50.7	14.0	142	23393	3.8	1987	3.9
3	70.3	85.3	155.6	54.8	13.0	574	89327	14.5	7468	14.8
4	59.3	78.7	138.0	57.0	11.2	1008	139147	22.6	11314	22.5
5	48.3	73.0	121.3	60.2	9.7	1526	185079	30.1	14735	29.2
6	37.3	52.6	90.0	58.5	7.1	1325	119232	19.4	9447	18.7
7	26.4	21.7	48.1	45.2	4.5	925	44471	7.2	4118	8.2
8	15.4	1.7	17.1	10.0	1.6	779	13312	2.2	1218	2.4

Annual latent cooling load = 343.26 MMBtu
Annual sensible cooling load = 271.92 MMBtu
Annual total cooling load = 615.18 MMBtu

Annual hours of operation = 6286 Hours
Annual Power Consumption = 50392 Kw-hrs

(ambient makeup air data run)

SYSTEM

CHILLER UNIT-----WATER COIL-------COOLING TOWER
chilled water return temp. held at 55.0 deg.F

SYSTEM STEADY STATE OPERATION

CASE	.ambient.		..........coil.......				..........system.............						..................cooling tower..................						
#	TDB	TWB	Tdb	Twb	Ts	CFM	Tel	Tcl	PLR	Qe	P	Qc	Tco	Tci	Tc_in	%	CFM_twr	CFMt	%
	..........deg.F...........					cfm	..deg.F..		%	tons	kws	tons	...deg.F......			(Qlwr)	...cfm...		(CFMtwr)
1	97.0	79.0	97.0	79.0	55.0	2000	44.9	95.7	94.0	14.5	15.0	18.8	95.7	85.7	85.7	100.0	4800	4800	100.0
2	92.0	78.0	92.0	78.0	55.0	2000	45.5	94.2	87.5	13.7	13.9	17.7	94.2	84.7	84.7	100.0	4800	4800	100.0
3	87.0	77.0	87.0	77.0	55.0	2000	46.0	92.6	81.2	13.0	12.9	16.6	92.6	83.6	83.6	100.0	4800	4800	100.0
4	82.0	75.0	82.0	75.0	55.0	2000	47.0	89.3	69.6	11.5	11.0	14.6	89.3	81.4	81.4	100.0	4800	4800	100.0
5	77.0	73.0	77.0	73.0	55.0	2000	48.0	86.1	59.1	10.1	9.4	12.8	86.1	79.1	79.1	100.0	4800	4800	100.0
6	72.0	69.0	72.0	69.0	55.0	2000	49.8	80.2	41.4	7.5	7.0	9.5	80.2	74.7	75.0	94.8	4800	4459	92.9
7	67.0	63.0	67.0	63.0	55.0	2000	52.2	77.8	21.1	4.0	4.4	5.3	77.8	71.1	75.0	42.1	4800	1676	34.9
8	62.0	58.0	62.0	58.0	55.0	2000	54.0	76.0	7.2	1.4	1.7	1.9	76.0	68.4	75.0	13.2	4800	482	10.0

CASE	STEADY STATE					ANNUAL				
#	Qsensible	Qlatent	Qtotal	Ql/Qt	POWER	hours	load/yr		power/yr	
	.........(Mbtuh).......			%	(kws)	(hrs)	(MBTU)	%	(kw-hrs)	%
1	92.3	81.8	174.1	47.0	15.0	7	1219	.2	105	.2
2	81.3	83.5	164.7	50.7	13.9	142	23393	3.8	1975	4.0
3	70.3	85.3	155.6	54.8	12.9	574	89327	14.5	7377	14.9
4	59.3	78.7	138.0	57.0	11.0	1008	139147	22.6	11058	22.3
5	48.3	73.0	121.3	60.2	9.4	1526	185079	30.1	14304	28.9
6	37.3	52.6	90.0	58.5	7.0	1325	119232	19.4	9267	18.7
7	26.4	21.7	48.1	45.2	4.4	925	44471	7.2	4077	8.2
8	15.4	1.7	17.1	10.0	1.7	779	13312	2.2	1318	2.7

Annual latent cooling load = 343.26 MMBtu
Annual sensible cooling load = 271.92 MMBtu
Annual total cooling load = 615.18 MMBtu

Annual hours of operation = 6286 Hours
Annual Power Consumption = 49481 Kw-hrs

(ambient makeup air data run)

SYSTEM

COMPRESSOR-------DX COIL------REMOTE AIR-COOLED CONDENSER

SYSTEM STEADY STATE OPERATION

CASE	.ambient.		..........coil........				...........system.............						.....................cooling tower..................							
#	TDB	TWB	Tdb	Twb	Ts	CFM	Tei	Tci	PLR	Qe	P	Qc	Tco	Tci	Tc_in	%	CFM_twr	CFMt	%
	..........deg.F...........					cfm	..deg.F..		%	tons	kws	tons	...deg.F......			(Qtwr)	...cfm...		(CFMtwr)
1	97.0	79.0	97.0	79.0	55.0	2000	37.8	121.2	99.8	14.5	17.4	19.5	###	###	###	#####	#####	#####	###
2	92.0	78.0	92.0	78.0	55.0	2000	38.8	114.8	88.6	13.7	15.5	18.1	###	###	###	#####	#####	#####	###
3	87.0	77.0	87.0	77.0	55.0	2000	39.7	108.5	78.7	13.0	13.7	16.9	###	###	###	#####	#####	#####	###
4	82.0	75.0	85.0	76.0	55.0	2000	40.6	102.2	70.0	12.2	12.0	15.6	###	###	###	#####	#####	#####	###
5	77.0	73.0	80.0	74.0	55.0	2000	42.2	95.0	56.9	10.8	9.8	13.6	###	###	###	#####	#####	#####	###
6	72.0	69.0	76.0	72.0	55.0	1850	44.4	87.1	42.5	8.7	7.5	10.9	###	###	###	#####	#####	#####	###
7	67.0	63.0	74.0	70.0	55.0	1611	46.6	85.0	30.4	6.5	5.9	8.2	###	###	###	#####	#####	#####	###
8	62.0	58.0	70.0	68.0	55.0	1500	48.2	85.0	23.5	5.2	4.9	6.6	###	###	###	#####	#####	#####	###
9	57.0	53.0	68.0	64.0	55.0	1500	50.4	85.0	15.1	3.4	3.2	4.3	###	###	###	#####	#####	#####	###

CASE	STEADY STATE					ANNUAL				
#	Qsensible	Qlatent	Qtotal	Ql/Qt	POWER	hours	load/yr		power/yr	
	....	(Mbtuh)	....	%	(kws)	(hrs)	(MBTU)	%	(kw-hrs)	%
1	92.3	81.8	174.1	47.0	17.4	7	1219	.2	122	.2
2	81.3	83.5	164.7	50.7	15.5	142	23393	3.1	2195	3.7
3	70.3	85.3	155.6	54.8	13.7	574	89327	11.9	7843	13.4
4	65.9	80.8	146.7	55.1	12.0	1008	147900	19.8	12116	20.7
5	54.9	74.6	129.6	57.6	9.8	1526	197714	26.4	14949	25.5
6	42.7	62.0	104.7	59.2	7.5	1325	138738	18.5	9898	16.9
7	33.6	44.9	78.6	57.2	5.9	925	72669	9.7	5455	9.3
8	24.7	37.2	62.0	60.1	4.9	779	48265	6.5	3836	6.5
9	21.4	19.6	41.0	47.8	3.2	701	28750	3.8	2232	3.8

Annual latent cooling load = 423.30 MMBtu
Annual sensible cooling load = 324.68 MMBtu
Annual total cooling load = 747.97 MMBtu

Annual hours of operation = 6987 Hours
Annual Power Consumption = 58646 Kw-hrs

(recirculated air data run)

SYSTEM

WATER COOLED CONDENSING UNIT--DX COIL--------COOLING TOWER

SYSTEM STEADY STATE OPERATION

CASE #	TOB (ambient, deg.F)	TWB (ambient, deg.F)	Tdb (coil, deg.F)	Twb (coil, deg.F)	Ts (coil, deg.F)	CFM cfm (coil)	Tel (system, deg.F)	Tcl (system, deg.F)	PLR %	Qe tons	P kws	Qc tons	Tco (cooling tower, deg.F)	Tci (deg.F)	Tc_in (deg.F)	% (Qtwr)	CFM_twr (cfm)	CFMt (cfm)	% (CFMtwr)
1	97.0	79.0	97.0	79.0	55.0	2000	37.8	95.7	90.5	14.5	15.3	18.9	95.7	85.7	85.7	100.0	4800	4800	100.0
2	92.0	78.0	92.0	78.0	55.0	2000	38.8	94.3	83.5	13.7	14.1	17.7	94.3	84.7	84.7	100.0	4800	4800	100.0
3	87.0	77.0	87.0	77.0	55.0	2000	39.7	92.6	76.4	13.0	13.0	16.7	92.6	83.6	83.6	100.0	4800	4800	100.0
4	82.0	75.0	85.0	76.0	55.0	2000	40.6	90.1	69.0	12.2	11.8	15.6	90.1	81.7	81.7	100.0	4800	4800	100.0
5	77.0	73.0	80.0	74.0	55.0	2000	42.2	86.8	56.0	10.8	10.1	13.7	86.8	79.5	79.5	100.0	4800	4800	100.0
6	72.0	69.0	76.0	72.0	55.0	1850	44.4	81.1	39.1	8.7	7.8	10.9	81.1	75.2	75.2	100.0	4800	4800	100.0
7	67.0	63.0	74.0	70.0	55.0	1611	46.6	79.5	26.3	6.5	6.2	8.3	79.5	72.0	75.0	59.6	4800	2496	52.0
8	62.0	58.0	70.0	68.0	55.0	1500	48.2	78.5	19.3	5.2	4.8	6.5	78.5	69.7	75.0	39.7	4800	1571	32.7
9	57.0	53.0	68.0	64.0	55.0	1500	50.4	77.3	11.5	3.4	3.0	4.3	77.3	67.4	75.0	23.1	4800	875	18.2

CASE #	STEADY STATE Qsensible (Mbtuh)	Qlatent (Mbtuh)	Qtotal (Mbtuh)	Ql/Qt %	POWER (kws)	ANNUAL hours (hrs)	load/yr (MBTU)	load/yr %	power/yr (kw-hrs)	power/yr %
1	92.3	81.8	174.1	47.0	15.3	7	1219	.2	107	.2
2	81.3	83.5	164.7	50.7	14.1	142	23393	3.1	2004	3.4
3	70.3	85.3	155.6	54.8	13.0	574	89327	11.9	7465	12.7
4	65.9	80.8	146.7	55.1	11.8	1008	147900	19.8	11942	20.4
5	54.9	74.6	129.6	57.6	10.1	1526	197714	26.4	15344	26.1
6	42.7	62.0	104.7	59.2	7.8	1325	138738	18.5	10283	17.5
7	33.6	44.9	78.6	57.2	6.2	925	72669	9.7	5716	9.7
8	24.7	37.2	62.0	60.1	4.8	779	48265	6.5	3714	6.3
9	21.4	19.6	41.0	47.8	3.0	701	28750	3.8	2107	3.6

Annual latent cooling load = 423.30 MMBtu
Annual sensible cooling load = 324.68 MMBtu
Annual total cooling load = 747.97 MMBtu

Annual hours of operation = 6987 Hours
Annual Power Consumption = 58682 Kw-hrs

(recirculated air data run)

SYSTEM

COMPRESSOR CHILLER UNIT-----WATER COIL-----REMOTE AIR-COOLED CONDENSER
chilled water supply temp. held at 45.0 deg.F

SYSTEM STEADY STATE OPERATION

CASE #	.ambient. TDB	TWB	.coil. Tdb	Twb	Ts	CFM cfm	.system. Tel	Tcl	PLR %	Qe tons	P kws	Qc tons	.cooling tower. Tco	Tci	Tc_in	% (Qtwr)	CFM_twr	CFMt	% (CFMtwr)
	...deg.F...						..deg.F..						...deg.F...				...cfm...		
1	97.0	79.0	97.0	79.0	55.0	2000	45.0	121.2	101.8	14.5	17.4	19.5	###	###	###	#####	#####	#####	###
2	92.0	78.0	92.0	78.0	55.0	2000	45.0	114.9	93.0	13.7	15.6	18.2	###	###	###	#####	#####	#####	###
3	87.0	77.0	87.0	77.0	55.0	2000	45.0	108.5	85.0	13.0	13.9	16.9	###	###	###	#####	#####	#####	###
4	82.0	75.0	85.0	76.0	55.0	2000	45.0	102.3	77.6	12.2	12.3	15.7	###	###	###	#####	#####	#####	###
5	77.0	73.0	80.0	74.0	55.0	2000	45.0	95.2	66.2	10.8	10.3	13.7	###	###	###	#####	#####	#####	###
6	72.0	69.0	76.0	72.0	55.0	1850	45.0	87.3	51.7	8.7	8.0	11.0	###	###	###	#####	#####	#####	###
7	67.0	63.0	74.0	70.0	55.0	1611	45.0	85.0	38.7	6.5	6.4	8.4	###	###	###	#####	#####	#####	###
8	62.0	58.0	70.0	68.0	55.0	1500	45.0	85.0	30.7	5.2	5.5	6.7	###	###	###	#####	#####	#####	###
9	57.0	53.0	68.0	64.0	55.0	1500	45.0	85.0	20.5	3.4	4.0	4.6	###	###	###	#####	#####	#####	###

CASE #	STEADY STATE Qsensible	Qlatent	Qtotal	Ql/Qt %	POWER (kws)	ANNUAL hours (hrs)	load/yr (MBTU)	%	power/yr (kw-hrs)	%
	..........(Mbtuh)		....							
1	92.3	81.8	174.1	47.0	17.4	7	1219	.2	122	.2
2	81.3	83.5	164.7	50.7	15.6	142	23393	3.1	2211	3.6
3	70.3	85.3	155.6	54.8	13.9	574	89327	11.9	7968	12.8
4	65.9	80.8	146.7	55.1	12.3	1008	147900	19.8	12432	20.0
5	54.9	74.6	129.6	57.6	10.3	1526	197714	26.4	15707	25.3
6	42.7	62.0	104.7	59.2	8.0	1325	138738	18.5	10611	17.1
7	33.6	44.9	78.6	57.2	6.4	925	72669	9.7	5916	9.5
8	24.7	37.2	62.0	60.1	5.5	779	48265	6.5	4282	6.9
9	21.4	19.6	41.0	47.8	4.0	701	28750	3.8	2792	4.5

Annual latent cooling load = 423.30 MMBtu
Annual sensible cooling load = 324.68 MMBtu
Annual total cooling load = 747.97 MMBtu

Annual hours of operation = 6987 Hours
Annual Power Consumption = 62043 Kw-hrs

(recirculated air data run)

SYSTEM

COMPRESSOR CNILLER UNIT-----NATER COIL-----REMOTE AIR-COOLED CONDENSER
chilled water return temp. held at 55.0 deg.F

SYSTEM STEADY STATE OPERATION

CASE	.ambient.		..........coil........				...........system............						..................cooling tower..................						
	TOB	TWB	Tdb	Twb	Ts	CFM	Tei	Tcl	PLR	Qe	P	Qc	Tco	Tci	Tc_in	%	CFM_twr	CFMt	%
#	..........deg.F...........					cfm	..deg.F..		%	tons	kws	tons	...deg.F.....			(Qtwr)	...cfm...		(CFMtwr)
1	97.0	79.0	97.0	79.0	55.0	2000	44.9	121.2	101.9	14.5	17.4	19.5	###	###	###	#####	#####	#####	###
2	92.0	78.0	92.0	78.0	55.0	2000	45.5	114.8	92.3	13.7	15.5	18.1	###	###	###	#####	#####	#####	###
3	87.0	77.0	87.0	77.0	55.0	2000	46.0	108.5	83.6	13.0	13.7	16.9	###	###	###	#####	#####	#####	###
4	82.0	75.0	85.0	76.0	55.0	2000	46.5	102.2	75.6	12.2	12.1	15.7	###	###	###	#####	#####	#####	###
5	77.0	73.0	80.0	74.0	55.0	2000	47.5	95.0	63.1	10.8	9.9	13.6	###	###	###	#####	#####	#####	###
6	72.0	69.0	76.0	72.0	55.0	1850	48.9	87.2	48.7	8.7	7.7	10.9	###	###	###	#####	#####	#####	###
7	67.0	63.0	74.0	70.0	55.0	1611	50.4	85.0	35.6	6.5	6.1	8.3	###	###	###	#####	#####	#####	###
8	62.0	58.0	70.0	68.0	55.0	1500	51.4	85.0	27.9	5.2	5.3	6.7	###	###	###	#####	#####	#####	###
9	57.0	53.0	68.0	64.0	55.0	1500	52.6	85.0	18.3	3.4	3.6	4.5	###	###	###	#####	#####	#####	###

CASE	STEADY STATE					ANNUAL				
	Qsensible	Qlatent	Qtotal	Ql/Qt	POWER	hours	load/yr		power/yr	
#	.........(Mbtuh)		......	%	(kws)	(hrs)	(MBTU)	%	(kw-hrs)	%
1	92.3	81.8	174.1	47.0	17.4	7	1219	.2	122	.2
2	81.3	83.5	164.7	50.7	15.5	142	23393	3.1	2200	3.7
3	70.3	85.3	155.6	54.8	13.7	574	89327	11.9	7878	13.1
4	65.9	80.8	146.7	55.1	12.1	1008	147900	19.8	12215	20.3
5	54.9	74.6	129.6	57.6	9.9	1526	197714	26.4	15167	25.2
6	42.7	62.0	104.7	59.2	7.7	1325	138738	18.5	10213	17.0
7	33.6	44.9	78.6	57.2	6.1	925	72669	9.7	5684	9.4
8	24.7	37.2	62.0	60.1	5.3	779	48265	6.5	4115	6.8
9	21.4	19.6	41.0	47.8	3.6	701	28750	3.8	2557	4.3

Annual latent cooling load	=	423.30	MMBtu
Annual sensible cooling load	=	324.68	MMBtu
Annual total cooling load	=	747.97	MMBtu
Annual hours of operation	=	6987	Hours
Annual Power Consumption	=	60152	Kw-hrs

(recirculated air data run)

SYSTEM

CHILLER UNIT-----WATER COIL-------COOLING TOWER
chilled water supply temp. held at 45.0 deg.F

SYSTEM STEADY STATE OPERATION

CASE	.ambient.		..........coil.......				...........system............						.....................cooling tower..................							
	TDB	TWB	Tdb	Twb	Ts	CFM	Tel	Tcl	PLR	Qe	P	Qc	Tco	Tci	Tc_in	%	CFM_twr	CFMt	%
#	..........deg.F...........					cfm	..deg.F..		%	tons	kws	tons	...deg.F......			(Qtwr)	...cfm...		(CFMtwr)
1	97.0	79.0	97.0	79.0	55.0	2000	45.0	95.6	93.9	14.5	15.0	18.8	95.6	85.7	85.7	100.0	4800	4800	100.0
2	92.0	78.0	92.0	78.0	55.0	2000	45.0	94.2	88.1	13.7	14.0	17.7	94.2	84.7	84.7	100.0	4800	4800	100.0
3	87.0	77.0	87.0	77.0	55.0	2000	45.0	92.6	82.5	13.0	13.0	16.7	92.6	83.6	83.6	100.0	4800	4800	100.0
4	82.0	75.0	85.0	76.0	55.0	2000	45.0	90.2	76.7	12.2	12.0	15.6	90.2	81.7	81.7	100.0	4800	4800	100.0
5	77.0	73.0	80.0	74.0	55.0	2000	45.0	86.9	66.5	10.8	10.3	13.7	86.9	79.5	79.5	100.0	4800	4800	100.0
6	72.0	69.0	76.0	72.0	55.0	1850	45.0	81.2	52.1	8.7	8.1	11.0	81.2	75.2	75.2	100.0	4800	4800	100.0
7	67.0	63.0	74.0	70.0	55.0	1611	45.0	79.5	38.7	6.5	6.4	8.4	79.5	72.0	75.0	59.9	4800	2505	52.2
8	62.0	58.0	70.0	68.0	55.0	1500	45.0	78.6	30.4	5.2	5.4	6.7	78.6	69.7	75.0	40.4	4800	1603	33.4
9	57.0	53.0	68.0	64.0	55.0	1500	45.0	77.4	20.0	3.4	3.8	4.5	77.4	67.4	75.0	24.2	4800	917	19.1

CASE	STEADY STATE					ANNUAL				
	Qsensible	Qlatent	Qtotal	Ql/Qt	POWER	hours	load/yr		power/yr	
#	..........(Mbtuh)			%	(kws)	(hrs)	(MBTU)	%	(kw-hrs)	%
1	92.3	81.8	174.1	47.0	15.0	7	1219	.2	105	.2
2	81.3	83.5	164.7	50.7	14.0	142	23393	3.1	1987	3.3
3	70.3	85.3	155.6	54.8	13.0	574	89327	11.9	7468	12.3
4	65.9	80.8	146.7	55.1	12.0	1008	147900	19.8	12071	19.8
5	54.9	74.6	129.6	57.6	10.3	1526	197714	26.4	15745	25.9
6	42.7	62.0	104.7	59.2	8.1	1325	138738	18.5	10760	17.7
7	33.6	44.9	78.6	57.2	6.4	925	72669	9.7	5923	9.7
8	24.7	37.2	62.0	60.1	5.4	779	48265	6.5	4193	6.9
9	21.4	19.6	41.0	47.8	3.8	701	28750	3.8	2654	4.4

```
Annual latent   cooling load  =   423.30  MMBtu
Annual sensible cooling load  =   324.68  MMBtu
Annual total    cooling load  =   747.97  MMBtu

Annual hours of operation     =     6987  Hours
Annual Power Consumption      =    60906  Kw-hrs
```

(recirculated air data run)

SYSTEM

CHILLER UNIT-----WATER COIL-------COOLING TOWER
chilled water return temp. held at 55.0 deg.F

SYSTEM STEADY STATE OPERATION

CASE #	ambient TDB deg.F	ambient TWB deg.F	coil Tdb deg.F	coil Twb deg.F	coil Ts deg.F	coil CFM cfm	system Tei deg.F	system Tci deg.F	system PLR %	system Qe tons	system P kws	system Qc tons	cooling tower Tco deg.F	cooling tower Tci deg.F	cooling tower Tc_in deg.F	cooling tower % (Qtwr)	cooling tower CFM_twr cfm	cooling tower CFMt cfm	cooling tower % (CFMtwr)
1	97.0	79.0	97.0	79.0	55.0	2000	44.9	95.7	94.0	14.5	15.0	18.8	95.7	85.7	85.7	100.0	4800	4800	100.0
2	92.0	78.0	92.0	78.0	55.0	2000	45.5	94.2	87.5	13.7	13.9	17.7	94.2	84.7	84.7	100.0	4800	4800	100.0
3	87.0	77.0	87.0	77.0	55.0	2000	46.0	92.6	81.2	13.0	12.9	16.6	92.6	83.6	83.6	100.0	4800	4800	100.0
4	82.0	75.0	85.0	76.0	55.0	2000	46.5	90.1	74.9	12.2	11.8	15.6	90.1	81.7	81.7	100.0	4800	4800	100.0
5	77.0	73.0	80.0	74.0	55.0	2000	47.5	86.8	63.9	10.8	10.0	13.7	86.8	79.5	79.5	100.0	4800	4800	100.0
6	72.0	69.0	76.0	72.0	55.0	1850	48.9	81.1	49.0	8.7	7.9	11.0	81.1	75.2	75.2	100.0	4800	4800	100.0
7	67.0	63.0	74.0	70.0	55.0	1611	50.4	79.5	35.6	6.5	6.3	8.3	79.5	72.0	75.0	59.6	4800	2490	51.9
8	62.0	58.0	70.0	68.0	55.0	1500	51.4	78.6	27.6	5.2	5.4	6.7	78.6	69.7	75.0	40.4	4800	1599	33.3
9	57.0	53.0	68.0	64.0	55.0	1500	52.6	77.4	17.8	3.4	3.8	4.5	77.4	67.4	75.0	24.2	4800	916	19.1

CASE #	STEADY STATE Qsensible (Mbtuh)	STEADY STATE Qlatent (Mbtuh)	STEADY STATE Qtotal (Mbtuh)	STEADY STATE Ql/Qt %	STEADY STATE POWER (kws)	ANNUAL hours (hrs)	ANNUAL load/yr (MBTU)	ANNUAL load/yr %	ANNUAL power/yr (kw-hrs)	ANNUAL power/yr %
1	92.3	81.8	174.1	47.0	15.0	7	1219	.2	105	.2
2	81.3	83.5	164.7	50.7	13.9	142	23393	3.1	1975	3.3
3	70.3	85.3	155.6	54.8	12.9	574	89327	11.9	7377	12.3
4	65.9	80.8	146.7	55.1	11.8	1008	147900	19.8	11858	19.8
5	54.9	74.6	129.6	57.6	10.0	1526	197714	26.4	15334	25.6
6	42.7	62.0	104.7	59.2	7.9	1325	138738	18.5	10495	17.5
7	33.6	44.9	78.6	57.2	6.3	925	72669	9.7	5851	9.8
8	24.7	37.2	62.0	60.1	5.4	779	48265	6.5	4224	7.1
9	21.4	19.6	41.0	47.8	3.8	701	28750	3.8	2661	4.4

Annual latent cooling load = 423.30 MMBtu
Annual sensible cooling load = 324.68 MMBtu
Annual total cooling load = 747.97 MMBtu

Annual hours of operation = 6987 Hours
Annual Power Consumption = 59880 Kw-hrs

(recirculated air data run)

APPENOIX E

RECIPROCATING CHILLER--TWO COIL--
COOLING TOWER SYSTEM COMPUTER PROGRAM

```
10 REM        THIS PROGRAN IS FOR A   chiller-cooling tower-twin chilled water coil-system
20 REM        THE CONSTANTS DESCRIBING EQUIPMENT SIZE ARE ALREADY INCLUDED IN THE CONSTANT SUBROUTINE
22 REM        CYLINDER UNLOADING UTILIZED TO CONTROL CHILLED WATER TENPERATURE
23 REM        CONSTANT WATER FLOW RATES ASSUNED AT THE EVAPORATOR & CONDENSER---CHILLED WATER BYPASS
30 REM        THE COIL CONDITIONS ARE AVAILABLE TO THE PROGRAN THROUGH THE COIL_CONDITION
40 !
50 !
60 !
80 DIN Qsens(200),Qlat(200),Qe(200),P(200),hrs(200),TDB(200),TWB(200)
90 DIN Tdb_1(200),wb_wi_1(200),CFNe_1(200),Ts_1(200),UNITS_1$(200)
100 DIN Tdb_2(200),wb_wi_2(200),CFNe_2(200),Ts_2(200),UNITS_2$(200)
110 DIN COIL_1(200),COIL_2(200)
120 !
150 !
160 GOSUB CONSTANTS  !                                 input equipment constants
170 GOSUB COIL_CONDITIONS  !                           input all coil loading conditions
180 GOSUB HEADING  !                                   system heading
190 GOSUB HEADING2  !                                  data   heading
200 !
210 FOR x=1 TO X !                                     loop for each system loading
220 !
230  GOSUB COIL  !                                     finds cooling load
240  GOSUB WATER_EVAP  !                               finds Tel=Tcw
250 Tc1=95 !                                           initial estimate of Tcl
260 !
270 INTERATION:
280 GOSUB UNIT_MODEL  !                    finds full load capacity,power
290 GOSUB PLR  !                           finds partial load power
300 Qc=Qe+P/3.516 !                        calculation of condenser load
310 GOSUB WATER_COND  !                    find new Tc1 (cooling tower)
320 IF ABS (Tc2-Tc1)<.1 THEN GOTO CHECK  ! check for convergence
330 Tc1=Tc2
340 GOTO INTERATION  !                     continue interation of Tc1
350 !
360 CHECK:
370 IF Tc_in=Tc_min THEN GOSUB FORCED_Tci  !   find reduced cooling tower cfm---if necessary
380 GOSUB OUTPUT  !                        system steady state performance output
390 !
400 NEXT x !                               next load condition for processing
410 GOSUB OUTPUT3  !                       coil condition output
420 GOSUB OUTPUT2  !                       annual load output
430 END
440 !
450 !
460 COIL:
470 COIL_1=0 !                             initializing variables
480 COIL_2=0
490 Qe=0
500 Qlat=0
510 Qsens=0
520 !
530 FOR DATA_coil=1 TO 2 !                 two coil system
540 GOSUB COIL_DATA  !                     data for each coil
550 IF CFNe=0 THEN GOTO 950 !              for coil bypass
560 !
570 IF UNITS$="WB" THEN GOTO wet_bulb
580 IF UNITS$="G" THEN GOTO grains
600 wet_bulb: !                            inlet enthalpy from wet-bulb
610 Twb=wb_wi
620 Temp=Twb
630 GOSUB HVTA
640 Hin=H
650 Wi=(Hin-.24*Tdb)/(1061+.444*Tdb) !     inlet air moisture content
660 GOTO dehumid_ck
680 grains:
690 Wi=wb_wi/7000
700 Twb=Temp
710 Hin=.24*Tdb+Wi*(1061+.444*Tdb) !       inlet air enthalpy from moisture content
720 GOTO dehumid_ck
740 dehumid_ck: !                          checking for dehumidification
750 Temp=Ts
760 GOSUB HVTA
770 IF WS>Wi THEN GOTO no_dehumid  ELSE GOTO dehumid
790 no_dehumid:
```

```
800 WS=Wi
810 Hout=.24*Ts+Wi*(1061+.444*Ts) !   finding enthalpy for no dehueidification
820 GOTO loading
840 dehueid:
850 Hout=H !                          enthalpy for dehumidification
860 GOTO loading
880 loading:
890 IF Tdb<Ts THEN Hout=Hin
900 Qe=Qe+CFMe*(Hin-Hout)*4.5 !                   total    load in Btuh
910 Qlat=Qlat+(Wi-WS)*(1061+.444*Tdb)*CFMe*4.5 !   sensible load in Btuh
920 Qsens=Qe-Qlat !                               latent   load in Btuh
930 IF DATA_coil=1 THEN COIL_1=Qe !               #1 coil load
940 IF DATA_coil=2 THEN COIL_2=Qe-COIL_1 !        #2 coil load
950 NEXT DATA_coil
960 !
970 Qsens=Qsens/12000 !                             load in Tons
980 Qlat=Qlat/12000 !                               load in Tons
990 Qe=Qe/12000 !                                   load in Tons
1000 COIL_1=COIL_1/12000 !                          load in Tons
1010 COIL_2=COIL_2/12000 !                          load in Tons
1020 RETURN
1030 !
1040 !
1050 UNIT_MODEL:
1060 Tc1=(Tc1-32)*5/9 !                             convert deg.F to deg.C
1070 Te1=(Te1-32)*5/9 !                             convert deg.F to deg.C
1080 Teq=(Tc1-Tc1_DES)/Tratio-(Te1-Te1_DES)
1090 ADCR=A1+A2*Teq+A3*Teq^2
1100 CAP=ADCR*CAP_DES !                             capacity in kws
1110 COPC=(Tc1+273.2)/(Tc1-Te1)
1120 COPCD=(Tc1_DES+273.2)/(Tc1_DES-Te1_DES)
1130 COPCN=COPC/COPCD
1140 COPAN=B1+B2*COPCN+B3*COPCN^2
1150 COPAD=CAP_DES/FLP_DES
1160 COPA=COPAN*COPAD
1170 FLP=CAP/COPA !                               power in kws
1180 Tc1=Tc1*9/5+32 !                             convert deg.C to deg.F
1190 Te1=Te1*9/5+32 !                             convert deg.C to deg.F
1200 CAP=CAP/3.516 !                              convert kws to tons (CAP)
1210 RETURN
1220 !
1230 !
1240 PLR:
1250 PLR=Qe/CAP
1260 IF PLR>CYCLE THEN GOTO 1290
1270 FFL=PLR*C4 !                                 cycling operation
1280 GOTO 1300
1290 FFL=C1+C2*PLR+C3*PLR^2 !                     continuous operation
1300 P=FFL*FLP
1310 RETURN
1320 !
1330 !
1340 WATER_EVAP:
1350 IF wtr_cntrl$="return" THEN GOTO 1380
1360 Tcw=Tcntrl_teep !                           supply water teep control
1370 GOTO 1390
1380 Tcw=Tcntrl_teep-Qe*24/GPMe !                return water teep control
1390 Te1=Tcw !                                          chilled water teep
1400 RETURN
1410 !
1420 !
1430 WATER_COND:
1440 Tco=Tc1
1450 GOSUB FREE_Tci  !                            cooling tower for Tci
1460 IF Tci<Tc_ein THEN Tc_in=Tc_ein ELSE Tc_in=Tci !  holding min. water teep
1470 Tc_out=Tc_in+Qc*24/GPMc !                    new leaving water teep
1480 Tc2=Tc_out
1490 RETURN
1500 !
1510 !
1520 COOLING_TWR: !                       SLIGHTLY MODIFIED WEBB-VILLACRES COOLING TOWER
1530 !
1540 FREE_Tci: !                          floating leaving water teep.
1550 no_answer=0
1560 CFMt=CFM_twr
```

```
1570 R=0
1580 INTER$="initial"
1590 RIg=GPMc*8.33/(CFM_twr*.0712)
1600 NTUA=C*(RIg^(-(I+XN)))
1610 initial:
1620 R=R+2
1630 GOTO 1660
1640 final:
1650 R=(RI+Rs)/2
1660 A=Tco-R-TWB
1670 GOSUB COUNT
1680 NTU=CT/RIg
1690  IF ABS (NTUA-NTU)/NTUA<.001 THEN GOTO 1750
1700      IF NTUA>NTU THEN Rs=R
1710       IF NTUA<NTU THEN RI=R
1720       IF NTUA<NTU THEN INTER$="final"
1730          IF INTER$="final" THEN GOTO final
1740  GOTB initial
1750 Tci=Tco-R
1760 RETURN
1770 !
1780 FORCED_Tci: !                                  fixed leaving water temp
1790 R=Tco-Tc_in
1800 A=Tc_in-TWB
1810 CFMt=CFM_twr
1820 CFMh=CFM_twr
1830 INTER$="initially"
1840 GOTO 1920
1850 initially:
1860 IF CFMt/CFM_twr>.3 THEN CFMt=CFMt-.1*CFM_twr ELSE CFMt=CFMt-.01*CFM_twr
1870 IF CFMt=0 THEN no_answer=1
1880 IF CFMt=0 THEN GOTO 2020
1890 GOTO 1920
1900 finally:
1910 CFMt=(CFMh+CFMl)/2
1920 RIg=GPMc*8.33/(.0712*CFMt)
1930 NTUA=C*(RIg^(-(1+XN)))
1940 GOSUB COUNT
1950 NTU=CT/RIg
1960  IF ABS ((NTUA-NTU)/NTUA)<.001 THEN GOTO 2020
1970     IF NTUA>NTU THEN CFMh=CFMt
1980     IF NTUA<NTU THEN CFMl=CFMt
1990     IF NTUA<NTU THEN INTER$="finally"
2000        IF INTER$="finally" THEN GOTO finally
2010  GOTO initially
2020 RETURN
2030 !
2040 !
2050 COUNT: !                                       main cooling tower subroutine
2060 CT=0
2070 Y(1)=.1
2080 Y(2)=.3
2090 Y(3)=.2
2100 Y(4)=.3
2110 Temp=TWB
2120 GOSUB HVTA
2130 HA=H
2140 TW=TWB+A
2150 FOR I=1 TO 4
2160   TW=TW+Y(I)*R
2170   HA=HA+RIg*Y(1)*R
2180   Temp=TW
2190   GOSUB HVTA
2200   HT=H
2210   DH=HT-HA
2220   CT=CT+1/DH
2230 NEXT I
2240 CT=CT*R/4
2250 RETURN
2260 !
2270 !
2280 HVTA: !                                        enthalpy subroutine
2290 PB=101325
2300 GOSUB PSAT
2310 WS=.622*(PS/(PB-PS))
```

```
2320 H=.24*Temp+WS*(1061+.444*Temp)
2330 RETURN
2340 !
2350 !
2360 WETbulb: !                                               wet-bulb subroutine
2370 Temp_H=Tdb
2380 Temp_L=0
2390 h=.24*Tdb+Wi*(1061+.444*Tdb)
2400 PB=101325
2410 Temp=(Temp_H+Temp_L)/2
2420 GOSUB PSAT
2430 Twb=Temp
2440 WS=.622*(PS/(PB-PS))
2450 H=.24*Twb+WS*(1061+.444*Twb)
2460 IF ABS (H-h)<.005 THEN GOTO 2500
2470  IF H>h THEN Temp_H=Temp
2480  IF H<h THEN Temp_L=Temp
2490 GOTO 2410
2500 RETURN
2510 !
2520 !
2530 PSAT: !                                         saturated pressure subroutine
2540 A(1,1)=6.41533947
2550 A(1,2)=6.41542599
2560 A(1,3)=6.44302266
2570 A(1,4)=6.53947838
2580 A(2,1)=.0821033478
2590 A(2,2)=.0724023398
2600 A(2,3)=.0698803796
2610 A(2,4)=.0656725584
2620 A(3,1)=-.000340068991
2630 A(3,2)=-.000264361973
2640 A(3,3)=-2.0606744496E-4
2650 A(3,4)=-.000160854949
2660  TS=(Temp-32)*5/9
2670  IF TS>22.0322 THEN GOTO 2680 ELSE GOTO 2720
2680   IF TS<46.0304 THEN J=3 ELSE GOTO 2700
2690   GOTO 2780
2700    IF TS<72.0476 THEN J=4 ELSE GOTO 2760
2710    GOTO 2780
2720  IF TS>.00895 THEN J=2 ELSE GOTO 2740
2730   GOTO 2780
2740    IF TS>-19.94 THEN J=1 ELSE GOTO 2760
2750    GOTO 2780
2760 PRINT "              TEMPERATURE IS OUTSIDE OF OUR RANGE"
2770 PAUSE
2780 PS=EXP (A(1,J)+TS*(A(2,J)+TS*A(3,J)))
2790 RETURN
2800 !
2810 !
2820 HEADING:
2830  PRINT CHR$ (12)
2840 PRINT "                                                              SYSTEM"
2850 PRINT "             ---------------------------------------------------------------------------------
--------------------------"
2860 PRINT USING "45X,70A" ; " CHILLER----CHILLED WATER COILS----COOLING TOWER"
2870 PRINT USING "44X,14A,6A,15A,DD.D,6A" ; "chilled water ",wtr_cntrl$," temp. held at  ",Tcntrl_temp," d
eg.F"
2880 PRINT
2890 PRINT
2900 RETURN
2910 !
2920 !
2930 OUTPUT:
2940 P_C1=(Tco-Tc_in)/(Tco-Tci)*100
2950 P_C2=CFMt/CFM_twr*100
2960 IF no_answer=0 THEN GOTO 2990
2970 PRINT USING "5X,3D,X,2(4D.D),X,2(6D.D),X,2(4D.DD),5D.D,4D.DD,4(5D.D)" ; x,TDB,TWB,Tei,Tci,Qe,P,PLR*10
0,Qc,Tco,Tci,Tc_in,P_C1
2980 GOTO 3010
2990 PRINT USING "5X,3D,X,2(4D.D),X,2(6D.D),X,2(4D.DD),5D.D,4D.DD,4(5D.D),2(8D),6D.D" ; x,TDB,TWB,Tei,Tci,
Qe,P,PLR*100,Qc,Tco,Tci,Tc_in,P_C1,CFM_twr,CFMt,P_C2
3000 !
3010 IF x=1 THEN LOAD_annual=0 !                      initializing annual variables
3020 IF x=1 THEN P_annual=0
```

```
3030 IF x=1 THEN COIL_1_annual=0
3040 IF x=1 THEN COIL_2_annual=0
3050 IF x=1 THEN sens_annual=0
3060 IF x=1 THEN lat_annual=0
3070 IF x=1 THEN hrs_annual=0
3080 COIL_1(x)=COIL_1*12
3090 COIL_2(x)=COIL_2*12
3100 Qsens(x)=Qsens*12
3110 Qlat(x)=Qlat*12
3120 Qe(x)=Qe*12
3130 P(x)=P
3140 hrs_annual=hrs_annual+hrs(x)
3150 LOAD_annual=LOAD_annual+Qe(x)*hrs(x)
3160 P_annual=P_annual+P*hrs(x)
3170 COIL_1_annual=COIL_1_annual+COIL_1(x)*hrs(x)
3180 COIL_2_annual=COIL_2_annual+COIL_2(x)*hrs(x)
3190 sens_annual=sens_annual+Qsens(x)*hrs(x)
3200 lat_annual=lat_annual+Qlat(x)*hrs(x)
3210 RETURN
3220 !
3230 OUTPUT2:
3240  PRINT USING "2/"
3250 PRINT "                                   STEADY STATE
   ANNUAL"
3260 PRINT "          --------------------------------------------------------        ------------
-------------------------"
3270 PRINT "CASE  TDB TWB   COIL#1  COIL#2  Qsensible  Qlatent  Qtotal   Ql/Qt   POWER      hours
load/yr            power/yr  "
3280 PRINT "  #   .deg.F.     ................(Mbtuh)................     %     (kws)      (hrs)     (MBT
U)     %     (kw-hrs)       %"
3290 PRINT "----  --- ---   -----   ------  --------  -------  ------   -----  ------     ------   ------
- -----   -------   -----"
3300 FOR I=1 TO X
3310 Lpc=Qlat(I)/Qe(I)*100
3320 Qa=Qe(I)*hrs(I)
3330 Qpc=Qa/LOAD_annual*100
3340 Pa=P(I)*hrs(I)
3350 Ppc=Pa/P_annual*100
3360 CL_1=COIL_1(I)
3370 CL_2=COIL_2(I)
3380 PRINT USING "4D,X,2(4D),5(7D.D),2(6D.D),10D,11D,4D.DD,11D,6D.DD" ; I,TDB(I),TWB(I),CL_1,CL_2,Qsens(I)
,Qlat(I),Qe(I),Lpc,P(I),hrs(I),Qa,Qpc,Pa,Ppc
3390 NEXT I
3400 PRINT USING "3/"
3410 PRINT USING "30X,35A,5D.DD,18A" ; "Annual  cooling  coil # 1  load  =";COIL_1_annual/1000," MMBtu"
3420 PRINT USING "30X,35A,5D.DD,18A" ; "Annual  cooling  coil # 2  load  =";COIL_2_annual/1000," MMBtu"
3430 PRINT
3440 PRINT USING "30X,35A,5D.DD,18A" ; "Annual sensible cooling load     =";sens_annual/1000," MMBtu"
3450 PRINT USING "30X,35A,5D.DD,18A" ; "Annual latent   cooling load     =";lat_annual/1000," MMBtu"
3460 PRINT
3470 PRINT USING "30X,35A,5D.DD,18A" ; "Annual cooling load              =";LOAD_annual/1000," MMBtu"
3480 PRINT USING "30X,38A,5D,19A" ; "Annual Hours of operation       =";hrs_annual," Hours"
3490 PRINT USING "30X,35A,8D,19A" ; "Annual Chiller Power consumption =";P_annual," Kw-hrs"
3500 RETURN
3510 !
3520 !
3530 HEADING2:
3540 PRINT "                                          SYSTEM  STEADY  STATE  OPERATION "
3550 PRINT "          ----------------------------------------------------------------------
------------------"
3560 PRINT "          .ambient.      ................system..............  ....................cooling t
ower.................."
3570 PRINT "      CASE   TDB   TWB     Tcw    Tcl      Qe     P     PLR    Qc      Tco   Tci   Tc_in    %
   CFM twr   CFMt    %"
3580 PRINT "        #    ..........deg.F.........    tons   kws     %    tons    ....deg.F........    (Qtw
r)   ...cfm...    (CFMtwr)"
3590 PRINT "      ----  ----  ----   -----  ------   -----  -----  -----  -----  ------ -----  ----- ----
-- ------ ------ ------"
3600 RETURN
3610 !
3620 !
3630 OUTPUT3:
3640  PRINT USING "2/"
3650 PRINT "                                              COIL CONDITIONS"
3660 PRINT "                        -------------------------------------------------------------
```

```
"
3670 PRINT "                              .ambient.    ........ coil # 1.......       .......coil # 2.....
..."
3680 PRINT "                    CASE    TDB    TWB    Tdb    Twb    Ts     CFM      Tdb  Grains  Ts
CFM"
3690 PRINT "                     #      ..............(deg.F).........    (cfm)    deg.F  gr/lb  deg.F  (
cfm)"
3700 PRINT "                    -----  -----  -----  -----  -----  -----  -------  ------  -----  ------ --
-----"
3710 !
3720 FOR I=1 TO X
3730 PRINT USING "20X,3D,X,5(5D.D),7D,4X,3(5D.D),7D" ; I,TDB(I),TWB(I),Tdb_1(I),wb_wi_1(I),Ts_1(I),CFMe_1(
I),Tdb_2(I),wb_wi_2(I),Ts_2(I),CFMe_2(I)
3740 NEXT I
3750 !
3760 RETURN
3770 !
3780 !
3790 CONSTANTS: !                    equipment constants
3800 !
3810 Tratio=2.9
3820 A1=.9978
3830 A2=-.02943
3840 A3=.0001472
3850 B1=-.1119
3860 B2=1.5815
3870 B3=-.4702
3880 C1=.1334
3890 C2=.7439
3900 C3=.1209
3910 CYCLE=.25 !                         PLR at which system cycles
3920 C4=(C1+C2*CYCLE+C3*CYCLE^2)/CYCLE ! system cycling constant
3930 Tcl_DES=(95-32)*5/9 !               temp at which CAP_DES,FLP_DES selected
3940 Tel_DES=(46-32)*5/9 !               temp at which CAP_DES,FLP_DES selected
3950 !
3960 !               UNIT_CONSTANTS
3970 CAP_DES=21.2*3.516 !                chiller  capacity  in kws ( from mach. data )
3980 FLP_DES=20.8 !                      chiller  power     in kws ( from mach. data )
3990 !
4000 !               COIL_CONSTANTS
4010 wtr_cntrl$="supply" !               chilled water temp control--return or supply
4020 Tcntrl_temp=45 !                   temp at which above water is held at (deg.F)
4030 GPMe=42.9 !                         chilled water flow rate      ( gpm )
4040 !
4050 !               COOL_TWR_CONSTANTS
4060 C=2.15171 !                        Webb-Villacres cooling tower constant
4070 XN=.99777 !                        Webb-Villacres cooling tower constant
4080 CFM_twr=4630 !                     air flow rate of cooling tower ( cfm )
4090 GPMc=55.4 !                        water flow rate of tower,condenser ( gpm )
4100 Tc_min=75 !                        minimum entering condensing water temp  (deg.F)
4110 RETURN
4120 !
4130 !
4140 !
4150 COIL_DATA:
4160 TDB=TDB(x)
4170 TWB=TWB(x)
4180 IF DATA_coil=2 THEN GOTO 4250
4190 Tdb=Tdb_1(x) !                          first  coil
4200 wb_wi=wb_wi_1(x)
4210 UNITS$=UNITS_1$(x)
4220 Ts=Ts_1(x)
4230 CFMe=CFMe_1(x)
4240 RETURN
4250 Tdb=Tdb_2(x) !                          second coil
4260 wb_wi=wb_wi_2(x)
4270 UNITS$=UNITS_2$(x)
4280 Ts=Ts_2(x)
4290 CFMe=CFMe_2(x)
4300 RETURN
4310 !
4320 !
4330 COIL_CONDITIONS:
4340 !                                          ............coil........
4350 ! EX    hours     ambient                    inlet              outlet
```

```
4360 ! ----    -----   ---------------   ------------------------------   --------
4370 !                 dry-bulb,wet-bulb  dry-bulb,wet-bulb OR grains,cfm  dry-bulb
4380 ! DATA    504      95   ,   81        95    ,  75,WB,              ,2500    55
4390 ! DATA    504      95   ,   81        95    ,              120,G ,2500    55
4400 !
4410 !
4420 !  hrs .amb. ...#1 coil..... ....#2 coil......
4430 DATA 7,97,79,97,79,WB,55,400,78.6,23,G,55,6710
4440 DATA 142,92,78,92,78,WB,55,400,79.7,22.1,G,55,6039
4450 DATA 574,87,77,87,77,WB,55,400,80.9,21,G,55,5368
4460 DATA 1008,82,75,82,75,WB,55,400,82.3,20,G,55,4697
4470 DATA 1526,77,73,77,73,WB,55,400,84.1,18.6,G,55,4026
4480 DATA 1325,72,69,72,69,WB,55,400,85.7,17.5,G,55,3355
4490 DATA 923,67,63,67,63,WB,55,400,87.7,16.7,G,55,2684
4500 DATA 779,62,58,62,58,WB,55,400,92.2,14.2,G,55,2013
4510 DATA 701,57,53,57,53,WB,55,400,98.9,8.9,G,55,1342
4520 DATA 624,52,48,0,0,WB,0,0,99.5,6.1,G,61,1125
4530 DATA 466,47,43,0,0,WB,0,0,95.6,5.8,G,70,1125
4540 DATA 342,42,39,0,0,WB,0,0,93.4,5.4,G,79,1125
4550 !
4560 !
4570 X=12 !                                   number of performance calculations above
4580 !
4590 FOR 1=1 TO X
4600  READ hrs(I),TDB(I),TWB(I)
4610  READ Tdb_1(I),wb_wi_1(I),UN1TS_1$(I),Ts_1(I),CFMe_1(I)
4620  READ Tdb_2(I),wb_wi_2(I),UN1TS_2$(1),Ts_2(I),CFMe_2(I)
4630 NEXT I
4640 !
4650 !
4660 RETURN
```

APPENOIX F

ANNUAL PERFORMANCE DATA FOR THE CHILLER UNIT-
TWO COIL-COOLING TOWER SYSTEM

SYSTEM

CHILLER----CHILLED WATER COILS----COOLING TOWER
chilled water supply temp. held at 45.0 deg.F

SYSTEM STEADY STATE OPERATION

CASE #	.ambient. TDB	TWB	......system...... Tcw	Tci	Qe tons	P kws	PLR %	Qc tons	......cooling tower...... Tco	Tci	Tc_in	% (Qtwr)	CFM_twr	CFMt	% (CFMtwr)
	...........deg.F	....							....deg.F	....			...cfm...		
1	97.0	79.0	45.0	94.5	17.24	17.13	82.6	22.11	94.5	84.9	84.9	100.0	4630	4630	100.0
2	92.0	78.0	45.0	92.9	16.25	15.92	77.2	20.78	92.9	83.8	83.8	100.0	4630	4630	100.0
3	87.0	77.0	45.0	91.2	15.18	14.67	71.4	19.35	91.2	82.8	82.8	100.0	4630	4630	100.0
4	82.0	75.0	45.0	88.5	13.90	13.17	64.4	17.65	88.5	80.8	80.8	100.0	4630	4630	100.0
5	77.0	73.0	45.0	85.7	12.62	11.75	57.6	15.96	85.7	78.8	78.8	100.0	4630	4630	100.0
6	72.0	69.0	45.0	80.9	10.81	9.88	48.0	13.62	80.9	74.9	75.0	98.2	4630	4521	97.7
7	67.0	63.0	45.0	79.8	8.74	8.31	38.6	11.10	79.8	71.9	75.0	60.9	4630	2595	56.1
8	62.0	58.0	45.0	78.9	7.05	7.10	31.0	9.07	78.9	69.6	75.0	42.2	4630	1752	37.8
9	57.0	53.0	45.0	78.1	5.39	5.81	23.6	7.04	78.1	67.5	75.0	28.9	4630	1181	25.5
10	52.0	48.0	45.0	77.2	3.90	4.18	17.0	5.09	77.2	65.5	75.0	18.9	4630	767	16.6
11	47.0	43.0	45.0	76.5	2.60	2.77	11.3	3.38	76.5	63.8	75.0	12.1	4630	488	10.5
12	42.0	39.0	45.0	75.9	1.46	1.55	6.3	1.90	75.9	62.4	75.0	6.4	4630	259	5.6

COIL CONDITIONS

CASE #	.ambient. TDB	TWB	....coil # 1.... Tdb	Twb	Ts	CFM (cfm)	....coil # 2.... Tdb deg.F	Grains gr/lb	Ts deg.F	CFM (cfm)
	..........(deg.F)									
1	97.0	79.0	97.0	79.0	55.0	400	78.6	23.0	55.0	6710
2	92.0	78.0	92.0	78.0	55.0	400	79.7	22.1	55.0	6039
3	87.0	77.0	87.0	77.0	55.0	400	80.9	21.0	55.0	5368
4	82.0	75.0	82.0	75.0	55.0	400	82.3	20.0	55.0	4697
5	77.0	73.0	77.0	73.0	55.0	400	84.1	18.6	55.0	4025
6	72.0	69.0	72.0	69.0	55.0	400	85.7	17.5	55.0	3355
7	67.0	63.0	67.0	63.0	55.0	400	87.7	16.7	55.0	2684
8	62.0	58.0	62.0	58.0	55.0	400	92.2	14.2	55.0	2013
9	57.0	53.0	57.0	53.0	55.0	400	98.9	8.9	55.0	1342
10	52.0	48.0	0.0	0.0	0.0	0	99.5	6.1	61.0	1125
11	47.0	43.0	0.0	0.0	0.0	0	95.6	5.8	70.0	1125
12	42.0	39.0	0.0	0.0	0.0	0	93.4	5.4	79.0	1125

STEADY STATE / ANNUAL

CASE #	TDB .deg.F.	TWB	COIL#1	COIL#2	Qsensible (Mbtuh)	Qlatent	Qtotal	Ql/Qt %	POWER (kws)	hours (hrs)	load/yr (MBTU)	%	power/yr (kw-hrs)	%
1	97	79	34.8	172.1	190.5	16.4	206.9	7.9	17.1	7	1448	.15	120	.16
2	92	78	32.9	162.0	178.3	16.7	195.0	8.6	15.9	142	27688	2.88	2261	2.94
3	87	77	31.1	151.0	165.0	17.1	182.1	9.4	14.7	574	104532	10.86	8418	10.96
4	82	75	27.6	139.2	151.1	15.7	166.8	9.4	13.2	1008	168161	17.47	13276	17.29
5	77	73	24.3	127.2	136.8	14.6	151.4	9.6	11.8	1526	231048	24.00	17936	23.35
6	72	69	18.0	111.8	119.2	10.5	129.8	8.1	9.9	1325	171919	17.86	13087	17.04
7	67	63	9.6	95.2	100.5	4.3	104.8	4.1	8.3	923	96750	10.05	7671	9.99
8	62	58	3.4	81.2	84.3	.3	84.6	.4	7.1	779	65900	6.85	5523	7.20
9	57	53	.9	63.8	64.7	0.0	64.7	0.0	5.8	701	45322	4.71	4073	5.30
10	52	48	0.0	46.9	46.9	0.0	46.9	0.0	4.2	624	29236	3.04	2611	3.40
11	47	43	0.0	31.2	31.2	0.0	31.2	0.0	2.8	466	14517	1.51	1291	1.68
12	42	39	0.0	17.5	17.5	0.0	17.5	0.0	1.6	342	5992	.62	531	.69

Annual cooling coil # 1 load = 123.63 MMBtu
Annual cooling coil # 2 load = 838.88 MMBtu .

Annual sensible cooling load = 893.87 MMBtu
Annual latent cooling load = 68.64 MMBtu

Annual cooling load = 962.51 MMBtu
Annual Hours of operation = 8417 Hours
Annual Chiller Power consumption = 76801 Kw-hrs

(single bin data run)

COIL CONDITIONS

CASE	.ambient.		.... coil # 1.......				.......coil # 2........			
CASE #	TDB	TWB	Tdb	Twb	Ts	CFM (cfm)	Tdb deg.F	Grains gr/lb	Ts deg.F	CFM (cfm)
	............(deg.F).........									
1	97.0	82.0	97.0	82.0	55.0	400	79.5	22.1	55.0	6710
2	97.0	78.0	97.0	78.0	55.0	400	78.4	23.2	55.0	6710
3	92.0	86.0	92.0	86.0	55.0	400	80.6	21.1	55.0	6039
4	92.0	84.0	92.0	84.0	55.0	400	80.6	21.1	55.0	6039
5	92.0	82.0	92.0	82.0	55.0	400	80.6	21.1	55.0	6039
6	92.0	80.0	92.0	80.0	55.0	400	80.3	21.5	55.0	6039
7	92.0	78.0	92.0	78.0	55.0	400	79.7	22.1	55.0	6039
8	92.0	76.0	92.0	76.0	55.0	400	79.1	22.7	55.0	6039
9	92.0	74.0	92.0	74.0	55.0	400	78.6	23.2	55.0	6039
10	92.0	72.0	92.0	72.0	55.0	400	78.1	23.8	55.0	6039
11	87.0	84.0	87.0	84.0	55.0	400	81.9	19.9	55.0	5368
12	87.0	82.0	87.0	82.0	55.0	400	81.9	19.9	55.0	5368
13	87.0	80.0	87.0	80.0	55.0	400	81.9	20.0	55.0	5368
14	87.0	78.0	87.0	78.0	55.0	400	81.3	20.7	55.0	5368
15	87.0	76.0	87.0	76.0	55.0	400	80.6	21.4	55.0	5368
16	87.0	74.0	87.0	74.0	55.0	400	80.0	22.0	55.0	5368
17	87.0	72.0	87.0	72.0	55.0	400	79.5	22.6	55.0	5368
18	87.0	70.0	87.0	70.0	55.0	400	79.0	23.2	55.0	5368
19	87.0	68.0	87.0	68.0	55.0	400	78.6	23.7	55.0	5368
20	87.0	66.0	87.0	66.0	55.0	400	78.0	24.3	55.0	5368
21	82.0	82.0	82.0	82.0	55.0	400	83.7	18.5	55.0	4697
22	82.0	80.0	82.0	80.0	55.0	400	83.7	18.5	55.0	4697
23	82.0	78.0	82.0	78.0	55.0	400	83.4	18.9	55.0	4697
24	82.0	76.0	82.0	76.0	55.0	400	82.7	19.6	55.0	4697
25	82.0	74.0	82.0	74.0	55.0	400	81.9	20.4	55.0	4697
26	82.0	72.0	82.0	72.0	55.0	400	81.3	21.1	55.0	4697
27	82.0	70.0	82.0	70.0	55.0	400	80.7	21.7	55.0	4697
28	82.0	68.0	82.0	68.0	55.0	400	80.1	22.3	55.0	4697
29	82.0	66.0	82.0	66.0	55.0	400	79.7	23.0	55.0	4697
30	82.0	64.0	82.0	64.0	55.0	400	79.0	23.5	55.0	4697
31	82.0	62.0	82.0	62.0	55.0	400	77.7	24.4	55.0	4697
32	78.0	78.0	78.0	78.0	55.0	400	85.4	17.0	55.0	4160
33	77.0	76.0	77.0	76.0	55.0	400	85.4	17.3	55.0	4026
34	77.0	74.0	77.0	74.0	55.0	400	84.5	18.2	55.0	4026
35	77.0	72.0	77.0	72.0	55.0	400	83.7	19.0	55.0	4026
36	77.0	70.0	77.0	70.0	55.0	400	83.0	19.7	55.0	4026
37	77.0	68.0	77.0	68.0	55.0	400	82.3	20.5	55.0	4026
38	77.0	66.0	77.0	66.0	55.0	400	81.7	21.2	55.0	4026
39	77.0	64.0	77.0	64.0	55.0	400	81.3	21.9	55.0	4026
40	77.0	62.0	77.0	62.0	55.0	400	80.3	22.6	55.0	4026
41	77.0	60.0	77.0	60.0	55.0	400	78.9	23.4	55.0	4026
42	77.0	58.0	77.0	58.0	55.0	400	78.2	23.4	55.0	4026
43	74.0	74.0	74.0	74.0	55.0	400	87.7	15.1	55.0	3623
44	72.0	72.0	72.0	72.0	55.0	400	87.1	16.1	55.0	3355
45	72.0	70.0	72.0	70.0	55.0	400	86.2	17.1	55.0	3355
46	72.0	68.0	72.0	68.0	55.0	400	85.3	17.9	55.0	3355
47	72.0	66.0	72.0	66.0	55.0	400	84.6	18.8	55.0	3355
48	72.0	64.0	72.0	64.0	55.0	400	84.0	19.6	55.0	3355
49	72.0	62.0	72.0	62.0	55.0	400	83.5	20.4	55.0	3355
50	72.0	60.0	72.0	60.0	55.0	400	82.3	21.1	55.0	3355
51	72.0	58.0	72.0	58.0	55.0	400	80.7	22.1	55.0	3355
52	72.0	56.0	72.0	56.0	55.0	400	79.9	22.1	55.0	3355
53	72.0	54.0	72.0	54.0	55.0	400	79.2	22.0	55.0	3355
54	72.0	52.0	72.0	52.0	55.0	400	78.7	21.9	55.0	3355
55	70.0	70.0	70.0	70.0	55.0	400	90.8	12.5	55.0	3086
56	68.0	68.0	68.0	68.0	55.0	400	88.9	15.0	55.0	2818

(double bin data run)

COIL CONDITIONS

CASE #	.ambient. TDB	TWB	.... coil # 1....... Tdb	Twb	Ts	CFM (cfm)	.......coil # 2........ Tdb deg.F	Grains gr/lb	Ts deg.F	CFM (cfm)
	..............(deg.F).........									
57	67.0	66.0	67.0	66.0	55.0	400	89.0	15.1	55.0	2684
58	67.0	64.0	67.0	64.0	55.0	400	88.1	16.2	55.0	2684
59	67.0	62.0	67.0	62.0	55.0	400	87.4	17.1	55.0	2684
60	67.0	60.0	67.0	60.0	55.0	400	86.7	18.1	55.0	2684
61	67.0	58.0	67.0	58.0	55.0	400	85.2	19.0	55.0	2684
62	67.0	56.0	67.0	56.0	55.0	400	83.4	20.1	55.0	2684
63	67.0	54.0	67.0	54.0	55.0	400	82.3	20.1	55.0	2684
64	67.0	52.0	67.0	52.0	55.0	400	81.5	20.0	55.0	2684
65	67.0	50.0	67.0	50.0	55.0	400	80.9	19.9	55.0	2684
66	67.0	48.0	67.0	48.0	55.0	400	80.4	19.8	55.0	2684
67	64.0	64.0	64.0	64.0	55.0	400	93.0	11.7	55.0	2281
68	62.0	62.0	62.0	62.0	55.0	400	94.1	11.7	55.0	2013
69	62.0	60.0	62.0	60.0	55.0	400	93.2	13.0	55.0	2013
70	62.0	58.0	62.0	58.0	55.0	400	92.3	14.2	55.0	2013
71	62.0	56.0	62.0	56.0	55.0	400	90.3	15.4	55.0	2013
72	62.0	54.0	62.0	54.0	55.0	400	87.8	16.8	55.0	2013
73	62.0	52.0	62.0	52.0	55.0	400	86.5	16.8	55.0	2013
74	62.0	50.0	62.0	50.0	55.0	400	85.4	16.7	55.0	2013
75	62.0	48.0	62.0	48.0	55.0	400	84.7	16.5	55.0	2013
76	62.0	46.0	62.0	46.0	55.0	400	84.0	16.4	55.0	2013
77	60.0	60.0	60.0	60.0	55.0	400	97.6	9.1	55.0	1744
78	58.0	58.0	58.0	58.0	55.0	400	101.3	7.2	55.0	1476
79	57.0	56.0	57.0	56.0	55.0	400	102.9	6.8	55.0	1365
80	57.0	54.0	57.0	54.0	55.0	400	100.5	8.0	55.0	1342
81	57.0	52.0	57.0	52.0	55.0	400	96.9	10.3	55.0	1342
82	57.0	50.0	57.0	50.0	55.0	400	95.0	10.2	55.0	1342
83	57.0	48.0	57.0	48.0	55.0	400	93.3	10.1	55.0	1342
84	57.0	46.0	57.0	46.0	55.0	400	92.3	9.8	55.0	1342
85	57.0	44.0	57.0	44.0	55.0	400	91.2	9.6	55.0	1342
86	57.0	42.0	57.0	42.0	55.0	400	90.2	9.4	55.0	1342
87	54.0	54.0	0.0	0.0	0.0	0	102.6	6.6	59.0	1320
88	52.0	52.0	0.0	0.0	0.0	0	102.3	6.3	62.0	1245
89	52.0	50.0	0.0	0.0	0.0	0	101.6	6.2	61.0	1125
90	52.0	48.0	0.0	0.0	0.0	0	99.4	6.2	61.0	1125
91	52.0	46.0	0.0	0.0	0.0	0	97.3	6.2	61.0	1125
92	52.0	44.0	0.0	0.0	0.0	0	96.1	5.8	61.0	1125
93	52.0	42.0	0.0	0.0	0.0	0	94.8	5.6	61.0	1125
94	52.0	40.0	0.0	0.0	0.0	0	93.6	5.3	61.0	1125
95	50.0	50.0	0.0	0.0	0.0	0	100.8	6.3	65.0	1200
96	48.0	48.0	0.0	0.0	0.0	0	100.5	5.9	68.0	1125
97	47.0	46.0	0.0	0.0	0.0	0	98.8	5.7	70.0	1125
98	47.0	44.0	0.0	0.0	0.0	0	96.6	5.8	70.0	1125
99	47.0	42.0	0.0	0.0	0.0	0	94.9	5.7	70.0	1125
100	47.0	40.0	0.0	0.0	0.0	0	93.7	5.5	70.0	1125
101	47.0	38.0	0.0	0.0	0.0	0	92.6	5.2	70.0	1125
102	47.0	36.0	0.0	0.0	0.0	0	91.4	5.1	70.0	1125
103	44.0	44.0	0.0	0.0	0.0	0	97.6	5.7	75.0	1125
104	42.0	42.0	0.0	0.0	0.0	0	96.0	5.4	79.0	1125
105	42.0	40.0	0.0	0.0	0.0	0	94.1	5.5	79.0	1125
106	42.0	38.0	0.0	0.0	0.0	0	92.9	5.2	79.0	1125
107	42.0	36.0	0.0	0.0	0.0	0	91.7	5.1	79.0	1125
108	42.0	34.0	0.0	0.0	0.0	0	86.2	4.9	79.0	1125
109	40.0	40.0	0.0	0.0	0.0	0	94.3	5.4	83.0	1125
110	38.0	38.0	0.0	0.0	0.0	0	93.3	5.2	86.0	1125
111	37.0	36.0	0.0	0.0	0.0	0	91.8	5.1	88.0	1125

(double bin data run)

SYSTEM

CHILLER------WATER-COIL---COOLING TOWER
chilled water supply temp. held at 45.0 deg.F

SYSTEM STEADY STATE OPERATION

CASE	.ambient. TDB	TWB	...system... Tcw	Tci	Qe	P	PLR	Qc	...cooling tower... Tco	Tci	Tc_in	%	CFM_twr	CFMt	%
#	...deg.F...				tons	kws	%	tons	...deg.F...			(Qtwr)	...cfm...		(CFMtwr)
1	97.0	82.0	45.0	97.6	18.27	18.77	89.1	23.61	97.6	87.4	87.4	100.0	4630	4630	100.0
2	97.0	78.0	45.0	93.5	16.96	16.68	80.8	21.71	93.5	84.0	84.0	100.0	4630	4630	100.0
3	92.0	86.0	45.0	100.6	18.10	19.25	89.8	23.57	100.6	90.5	90.5	100.0	4630	4630	100.0
4	92.0	84.0	45.0	98.7	17.73	18.48	87.1	22.99	98.7	88.8	88.8	100.0	4630	4630	100.0
5	92.0	82.0	45.0	96.9	17.38	17.75	84.4	22.43	96.9	87.2	87.2	100.0	4630	4630	100.0
6	92.0	80.0	45.0	95.0	16.89	16.91	81.2	21.70	95.0	85.6	85.6	100.0	4630	4630	100.0
7	92.0	78.0	45.0	92.9	16.25	15.92	77.2	20.78	92.9	83.8	83.8	100.0	4630	4630	100.0
8	92.0	76.0	45.0	90.8	15.62	14.99	73.3	19.89	90.8	82.1	82.1	100.0	4630	4630	100.0
9	92.0	74.0	45.0	88.8	15.06	14.18	69.9	19.10	88.8	80.4	80.4	100.0	4630	4630	100.0
10	92.0	72.0	45.0	86.6	14.52	13.40	66.6	18.33	86.6	78.7	78.7	100.0	4630	4630	100.0
11	87.0	84.0	45.0	98.0	16.80	17.42	82.2	21.76	98.0	88.7	88.7	100.0	4630	4630	100.0
12	87.0	82.0	45.0	96.1	16.46	16.71	79.6	21.21	96.1	87.0	87.0	100.0	4630	4630	100.0
13	87.0	80.0	45.0	94.4	16.13	16.08	77.2	20.70	94.4	85.4	85.4	100.0	4630	4630	100.0
14	87.0	78.0	45.0	92.3	15.52	15.17	73.5	19.83	92.3	83.6	83.6	100.0	4630	4630	100.0
15	87.0	76.0	45.0	90.1	14.88	14.24	69.6	18.93	90.1	81.9	81.9	100.0	4630	4630	100.0
16	87.0	74.0	45.0	88.1	14.31	13.44	66.1	18.13	88.1	80.2	80.2	100.0	4630	4630	100.0
17	87.0	72.0	45.0	86.0	13.79	12.72	63.0	17.41	86.0	78.4	78.4	100.0	4630	4630	100.0
18	87.0	70.0	45.0	84.0	13.29	12.06	60.1	16.72	84.0	76.7	76.7	100.0	4630	4630	100.0
19	87.0	68.0	45.0	82.1	12.85	11.50	57.5	16.12	82.1	75.0	75.0	99.3	4630	4594	99.2
20	87.0	66.0	45.0	81.7	12.35	11.09	55.1	15.51	81.7	74.0	75.0	86.7	4630	3878	83.8
21	82.0	82.0	45.0	95.4	15.58	15.77	75.1	20.07	95.4	86.8	86.8	100.0	4630	4630	100.0
22	82.0	80.0	45.0	93.6	15.25	15.15	72.7	19.56	93.6	85.1	85.1	100.0	4630	4630	100.0
23	82.0	78.0	45.0	91.8	14.81	14.44	69.9	18.92	91.8	83.5	83.5	100.0	4630	4630	100.0
24	82.0	76.0	45.0	89.6	14.22	13.59	66.3	18.08	89.6	81.7	81.7	100.0	4630	4630	100.0
25	82.0	74.0	45.0	87.4	13.59	12.75	62.6	17.22	87.4	79.9	79.9	100.0	4630	4630	100.0
26	82.0	72.0	45.0	85.3	13.07	12.06	59.5	16.50	85.3	78.1	78.1	100.0	4630	4630	100.0
27	82.0	70.0	45.0	83.3	12.55	11.41	56.5	15.80	83.3	76.4	76.4	100.0	4630	4630	100.0
28	82.0	68.0	45.0	81.6	12.05	10.85	53.7	15.14	81.6	74.7	75.0	96.3	4630	4413	95.3
29	82.0	66.0	45.0	81.3	11.64	10.53	51.9	14.64	81.3	73.8	75.0	84.1	4630	3740	80.8
30	82.0	64.0	45.0	81.1	11.20	10.18	49.8	14.09	81.1	72.9	75.0	74.5	4630	3246	70.1
31	82.0	62.0	45.0	80.8	10.64	9.75	47.3	13.42	80.8	72.0	75.0	65.9	4630	2827	61.1
32	78.0	78.0	45.0	91.1	14.18	13.80	66.7	18.10	91.1	83.2	83.2	100.0	4630	4630	100.0
33	77.0	76.0	45.0	89.0	13.51	12.91	62.8	17.18	89.0	81.5	81.5	100.0	4630	4630	100.0
34	77.0	74.0	45.0	86.8	12.90	12.11	59.2	16.34	86.8	79.6	79.6	100.0	4630	4630	100.0
35	77.0	72.0	45.0	84.7	12.34	11.41	56.0	15.58	84.7	77.9	77.9	100.0	4630	4630	100.0
36	77.0	70.0	45.0	82.6	11.82	10.79	53.0	14.89	82.6	76.1	76.1	100.0	4630	4630	100.0
37	77.0	68.0	45.0	81.2	11.32	10.27	50.4	14.24	81.2	74.6	75.0	93.4	4630	4250	91.8
38	77.0	66.0	45.0	80.9	10.87	9.92	48.3	13.69	80.9	73.6	75.0	81.0	4630	3581	77.3
39	77.0	64.0	45.0	80.8	10.50	9.63	46.6	13.24	80.8	72.7	75.0	71.8	4630	3111	67.2
40	77.0	62.0	45.0	80.5	10.03	9.28	44.5	12.66	80.5	71.8	75.0	63.5	4630	2713	58.6
41	77.0	60.0	45.0	80.2	9.52	8.89	42.1	12.04	80.2	70.9	75.0	56.4	4630	2382	51.4
42	77.0	58.0	45.0	80.1	9.26	8.70	41.0	11.73	80.1	70.2	75.0	51.3	4630	2150	46.4
43	74.0	74.0	45.0	86.7	12.86	12.08	59.0	16.30	86.7	79.6	79.6	100.0	4630	4630	100.0
44	72.0	72.0	45.0	84.0	11.62	10.78	52.5	14.69	84.0	77.6	77.6	100.0	4630	4630	100.0
45	72.0	70.0	45.0	81.9	11.09	10.16	49.5	13.98	81.9	75.7	75.7	100.0	4630	4630	100.0
46	72.0	68.0	45.0	80.8	10.57	9.69	46.9	13.32	80.8	74.4	75.0	90.4	4630	4084	88.2
47	72.0	66.0	45.0	80.5	10.12	9.35	44.9	12.78	80.5	73.4	75.0	78.0	4630	3422	73.9
48	72.0	64.0	45.0	80.4	9.71	9.04	43.0	12.28	80.4	72.5	75.0	68.5	4630	2953	63.8
49	72.0	62.0	45.0	80.2	9.35	8.76	41.4	11.84	80.2	71.7	75.0	60.7	4630	2583	55.8
50	72.0	60.0	45.0	79.9	8.91	8.44	39.4	11.31	79.9	70.8	75.0	53.9	4630	2272	49.1
51	72.0	58.0	45.0	79.7	8.43	8.08	37.2	10.72	79.7	70.0	75.0	48.0	4630	2004	43.3
52	72.0	56.0	45.0	79.5	8.18	7.91	36.1	10.43	79.5	69.2	75.0	43.8	4630	817	17.6
53	72.0	54.0	45.0	79.4	7.97	7.75	35.1	10.17	79.4	68.5	75.0	40.3	4630	1668	36.0
54	72.0	52.0	45.0	79.3	7.81	7.64	34.4	9.99	79.3	67.8	75.0	37.5	4630	1547	33.4
55	70.0	70.0	45.0	82.3	11.60	10.59	52.0	14.61	82.3	76.0	76.0	100.0	4630	4630	100.0
56	68.0	68.0	45.0	80.5	10.01	9.26	44.4	12.64	80.5	74.3	75.0	88.1	4630	3959	85.5

(double bin data run)

SYSTEM

CHILLER-------WATER-COIL---COOLING TOWER
chilled water supply temp. held at 45.0 deg.F

SYSTEM STEADY STATE OPERATION

	.ambient.		...............system...............							.....................cooling tower..................						
CASE #	TDB	TWB	Tcw	Tci	Qe	P	PLR	Qc	Tco	Tci	Tc_in	%	CFM_twr	CFMt	%
	..........deg.F..........				tons	kws	%	tons	....deg.F........			(Qtwr)	...cfm...		(CFMtwr)
57	67.0	66.0	45.0	80.2	9.38	8.79	41.5	11.88	80.2	73.3	75.0	74.7	4630	3259	70.4
58	67.0	64.0	45.0	79.9	8.94	8.46	39.5	11.35	79.9	72.3	75.0	65.1	4630	2791	60.3
59	67.0	62.0	45.0	79.7	8.56	8.18	37.8	10.88	79.7	71.5	75.0	57.3	4630	2425	52.4
60	67.0	60.0	45.0	79.5	8.18	7.90	36.1	10.43	79.5	70.6	75.0	50.9	4630	2132	46.1
61	67.0	58.0	45.0	79.3	7.77	7.61	34.2	9.93	79.3	69.8	75.0	45.3	4630	1886	40.7
62	67.0	56.0	45.0	79.1	7.33	7.30	32.3	9.41	79.1	69.0	75.0	40.4	4630	1673	36.1
63	67.0	54.0	45.0	78.9	7.07	7.11	31.1	9.09	78.9	68.3	75.0	36.8	4630	1518	32.9
64	67.0	52.0	45.0	78.8	6.87	6.97	30.2	8.85	78.8	67.6	75.0	34.0	4630	1398	30.2
65	67.0	50.0	45.0	78.8	6.72	6.87	29.5	8.68	78.8	66.9	75.0	31.7	4630	838	18.1
66	67.0	48.0	45.0	78.7	6.60	6.78	29.0	8.53	78.7	66.3	75.0	29.8	4630	784	16.9
67	64.0	64.0	45.0	79.8	8.74	8.31	38.6	11.10	79.8	72.3	75.0	64.2	4630	2747	59.3
68	62.0	62.0	45.0	79.3	7.80	7.63	34.4	9.97	79.3	71.3	75.0	53.9	4630	2268	49.0
69	62.0	60.0	45.0	79.1	7.43	7.37	32.7	9.52	79.1	70.4	75.0	47.6	4630	1986	42.9
70	62.0	58.0	45.0	78.9	7.07	7.11	31.1	9.09	78.9	69.6	75.0	42.3	4630	1754	37.9
71	62.0	56.0	45.0	78.7	6.68	6.83	29.3	8.62	78.7	68.8	75.0	37.7	4630	1555	33.6
72	62.0	54.0	45.0	78.5	6.22	6.52	27.3	8.08	78.5	68.0	75.0	33.5	4630	887	19.2
73	62.0	52.0	45.0	78.4	5.99	6.36	26.2	7.79	78.4	67.3	75.0	30.6	4630	807	17.4
74	62.0	50.0	45.0	78.3	5.79	6.22	25.3	7.55	78.3	66.7	75.0	28.2	4630	1152	24.9
75	62.0	48.0	45.0	78.2	5.66	6.11	24.8	7.40	78.2	66.0	75.0	26.3	4630	1074	23.2
76	62.0	46.0	45.0	78.1	5.53	5.97	24.2	7.23	78.1	65.4	75.0	24.6	4630	1004	21.7
77	60.0	60.0	45.0	79.0	7.19	7.19	31.6	9.23	79.0	70.4	75.0	46.5	4630	1937	41.8
78	58.0	58.0	45.0	78.6	6.45	6.67	28.3	8.35	78.6	69.5	75.0	39.6	4630	1635	35.3
79	57.0	56.0	45.0	78.4	5.99	6.36	26.2	7.80	78.4	68.7	75.0	34.8	4630	1429	30.9
80	57.0	54.0	45.0	78.2	5.58	6.02	24.4	7.29	78.2	67.9	75.0	30.7	4630	811	17.5
81	57.0	52.0	45.0	77.9	5.15	5.54	22.5	6.72	77.9	67.1	75.0	27.0	4630	1101	23.8
82	57.0	50.0	45.0	77.8	4.92	5.29	21.5	6.42	77.8	66.4	75.0	24.5	4630	997	21.5
83	57.0	48.0	45.0	77.7	4.71	5.06	20.6	6.15	77.7	65.8	75.0	22.4	4630	911	19.7
84	57.0	46.0	45.0	77.6	4.59	4.93	20.0	5.99	77.6	65.2	75.0	20.9	4630	848	18.3
85	57.0	44.0	45.0	77.5	4.46	4.79	19.4	5.82	77.5	64.6	75.0	19.5	4630	789	17.0
86	57.0	42.0	45.0	77.5	4.33	4.65	18.9	5.66	77.5	64.0	75.0	18.2	4630	738	15.9
87	54.0	54.0	45.0	77.9	5.19	5.59	22.7	6.78	77.9	67.8	75.0	28.9	4630	1180	25.5
88	52.0	52.0	45.0	77.6	4.52	4.86	19.7	5.91	77.6	66.9	75.0	24.1	4630	980	21.2
89	52.0	50.0	45.0	77.3	4.12	4.42	17.9	5.37	77.3	66.2	75.0	20.9	4630	849	18.3
90	52.0	48.0	45.0	77.2	3.89	4.17	17.0	5.08	77.2	65.5	75.0	18.9	4630	765	16.5
91	52.0	46.0	45.0	77.1	3.68	3.94	16.0	4.80	77.1	64.9	75.0	17.1	4630	692	14.9
92	52.0	44.0	45.0	77.0	3.56	3.81	15.5	4.64	77.0	64.3	75.0	15.8	4630	641	13.8
93	52.0	42.0	45.0	77.0	3.43	3.67	14.9	4.47	77.0	63.8	75.0	15.4	4630	621	13.4
94	52.0	40.0	45.0	77.0	3.31	3.54	14.4	4.31	77.0	63.2	75.0	14.3	4630	577	12.5
95	50.0	50.0	45.0	77.2	3.87	4.15	16.9	5.05	77.2	66.1	75.0	19.8	4630	803	17.3
96	48.0	48.0	45.0	77.0	3.30	3.53	14.3	4.30	77.0	65.4	75.0	17.0	4630	686	14.8
97	47.0	46.0	45.0	76.7	2.92	3.12	12.7	3.81	76.7	64.7	75.0	14.4	4630	583	12.6
98	47.0	44.0	45.0	76.6	2.70	2.88	11.7	3.52	76.6	64.1	75.0	12.8	4630	517	11.2
99	47.0	42.0	45.0	76.5	2.52	2.69	11.0	3.29	76.5	63.5	75.0	11.5	4630	465	10.0
100	47.0	40.0	45.0	76.4	2.40	2.56	10.4	3.13	76.4	62.9	75.0	10.6	4630	427	9.2
101	47.0	38.0	45.0	76.4	2.29	2.44	9.9	2.99	76.4	62.4	75.0	9.8	4630	393	8.5
102	47.0	36.0	45.0	76.3	2.17	2.31	9.4	2.83	76.3	61.9	75.0	8.9	4630	360	7.8
103	44.0	44.0	45.0	76.4	2.29	2.44	9.9	2.99	76.4	64.0	75.0	11.0	4630	443	9.6
104	42.0	42.0	45.0	76.0	1.72	1.83	7.5	2.25	76.0	63.3	75.0	8.0	4630	323	7.0
105	42.0	40.0	45.0	75.9	1.53	1.63	6.6	1.99	75.9	62.7	75.0	6.9	4630	276	6.0
106	42.0	38.0	45.0	75.8	1.41	1.50	6.1	1.84	75.8	62.1	75.0	6.1	4630	246	5.3
107	42.0	36.0	45.0	75.8	1.29	1.37	5.6	1.68	75.8	61.6	75.0	5.4	4630	217	4.7
108	42.0	34.0	45.0	75.4	.73	.77	3.1	.95	75.4	61.0	75.0	3.0	4630	76	1.6
109	40.0	40.0	45.0	75.7	1.15	1.22	4.9	1.49	75.7	62.6	75.0	5.2	4630	208	4.5
110	38.0	38.0	45.0	75.4	.74	.78	3.2	.96	75.4	61.9	75.0	3.3	*****	*****	*****
111	37.0	36.0	45.0	75.2	.39	.41	1.7	.50	75.2	61.3	75.0	1.6	*****	*****	*****

(double bin data run)

CASE #	TDB .deg.F.	TWB .deg.F.	COIL#1 (Mbtuh)	COIL#2 (Mbtuh)	Qsensible (Mbtuh)	Qlatent (Mbtuh)	Qtotal (Mbtuh)	Ql/Qt %	POWER (kws)	hours (hrs)	load/yr (MBTU)	load/yr %	power/yr (kw-hrs)	power/yr %
			STEADY STATE							ANNUAL				
1	97	82	40.7	178.6	197.0	22.2	219.3	10.1	18.8	2	439	.04	38	.05
2	97	78	32.9	170.6	189.1	14.5	203.6	7.1	16.7	2	407	.04	33	.04
3	92	86	49.3	167.9	184.2	33.0	217.2	15.2	19.3	17	3692	.37	327	.41
4	92	84	44.9	167.9	184.2	28.6	212.8	13.5	18.5	2	426	.04	37	.05
5	92	82	40.7	167.9	184.2	24.4	208.6	11.7	17.8	19	3963	.40	337	.42
6	92	80	36.7	165.9	182.2	20.5	202.7	10.1	16.9	21	4256	.43	355	.45
7	92	78	32.9	162.0	178.3	16.7	195.0	8.6	15.9	28	5460	.55	446	.56
8	92	76	29.3	158.1	174.4	13.1	187.5	7.0	15.0	15	2812	.28	225	.28
9	92	74	25.9	154.9	171.1	9.7	180.8	5.3	14.2	9	1627	.16	128	.16
10	92	72	22.6	151.6	167.9	6.4	174.2	3.7	13.4	3	523	.05	40	.05
11	87	94	44.9	156.8	170.8	30.8	201.6	15.3	17.4	22	4436	.45	383	.48
12	87	82	40.7	156.8	170.8	26.6	197.5	13.5	16.7	68	13428	1.35	1136	1.43
13	87	80	36.7	156.8	170.8	22.7	193.5	11.7	16.1	82	15867	1.60	1319	1.65
14	87	78	32.9	153.3	167.4	18.9	186.3	10.1	15.2	202	37623	3.79	3063	3.84
15	87	76	29.3	149.3	163.3	15.3	178.6	8.6	14.2	134	23932	2.41	1909	2.40
16	87	74	25.9	145.8	159.8	11.9	171.7	6.9	13.4	74	12705	1.28	995	1.25
17	87	72	22.6	142.9	156.9	8.6	165.5	5.2	12.7	39	6455	.65	496	.62
18	87	70	19.5	140.0	154.1	5.4	159.5	3.4	12.1	6	957	.10	72	.09
19	87	68	16.5	137.7	151.7	2.5	154.2	1.6	11.5	10	1542	.16	115	.14
20	87	66	14.1	134.2	148.3	0.0	148.3	0.0	11.1	2	297	.03	22	.03
21	82	82	40.7	146.3	158.2	28.8	187.0	15.4	15.8	4	748	.08	63	.08
22	82	80	36.7	146.3	158.2	24.9	183.0	13.6	15.2	78	14276	1.44	1182	1.48
23	82	78	32.9	144.8	156.6	21.1	177.7	11.9	14.4	321	57052	5.75	4637	5.82
24	82	76	29.3	141.2	153.1	17.5	170.6	10.2	13.6	280	47765	4.81	3805	4.77
25	82	74	25.9	137.2	149.1	14.1	163.1	8.6	12.8	118	19246	1.94	1505	1.89
26	82	72	22.6	134.2	146.0	10.8	156.8	6.9	12.1	72	11289	1.14	868	1.09
27	82	70	19.5	131.1	143.0	7.6	150.6	5.1	11.4	21	3163	.32	240	.30
28	82	68	16.5	128.1	139.9	4.7	144.6	3.2	10.8	21	3037	.31	228	.29
29	82	66	13.7	126.1	137.9	1.8	139.7	1.3	10.5	13	1816	.18	137	.17
30	82	64	11.9	122.5	134.4	0.0	134.4	0.0	10.2	7	940	.09	71	.09
31	82	62	11.8	115.9	127.7	0.0	127.7	0.0	9.7	2	255	.03	19	.02
32	78	78	32.9	137.2	147.3	22.8	170.1	13.4	13.8	97	16504	1.66	1338	1.68
33	77	76	29.3	132.8	142.5	19.7	162.1	12.1	12.9	485	78634	7.92	6262	7.86
34	77	74	25.9	128.9	138.6	16.2	154.8	10.5	12.1	561	86842	8.75	6794	8.53
35	77	72	22.6	125.4	135.1	13.0	148.1	8.8	11.4	272	40271	4.06	3103	3.89
36	77	70	19.5	122.4	132.0	9.8	141.9	6.9	10.8	82	11635	1.17	885	1.11
37	77	68	16.5	119.3	129.0	6.9	135.9	5.0	10.3	72	9782	.99	740	.93
38	77	66	13.7	116.7	126.4	4.0	130.4	3.1	9.9	50	6521	.66	496	.62
39	77	64	10.9	115.0	124.7	1.3	126.0	1.0	9.6	33	4156	.42	318	.40
40	77	62	9.7	110.7	120.3	0.0	120.3	0.0	9.3	17	2045	.21	158	.20
41	77	60	9.6	104.6	114.2	0.0	114.2	0.0	8.9	4	457	.05	36	.04
42	77	58	9.6	101.5	111.1	0.0	111.1	0.0	8.7	5	556	.06	43	.05
43	74	74	25.9	128.5	136.8	17.6	154.4	11.4	12.1	120	18525	1.87	1449	1.82
44	72	72	22.6	116.8	124.3	15.2	139.4	10.9	10.8	323	45041	4.54	3483	4.37
45	72	70	19.5	113.6	121.0	12.0	133.1	9.0	10.2	203	27013	2.72	2063	2.59
46	72	68	16.5	110.3	117.8	9.1	126.8	7.1	9.7	304	38556	3.89	2945	3.70
47	72	66	13.7	107.8	115.3	6.2	121.5	5.1	9.3	163	19797	1.99	1523	1.91
48	72	64	10.9	105.6	113.1	3.5	116.6	3.0	9.0	50	5828	.59	452	.57
49	72	62	8.3	103.8	111.3	.9	112.1	.8	8.8	35	3925	.40	307	.38
50	72	60	7.5	99.5	106.9	0.0	106.9	0.0	8.4	10	1069	.11	84	.11
51	72	58	7.4	93.7	101.1	0.0	101.1	0.0	8.1	13	1314	.13	105	.13
52	72	56	7.4	90.7	98.2	0.0	98.2	0.0	7.9	12	1178	.12	95	.12
53	72	54	7.4	88.2	95.6	0.0	95.6	0.0	7.8	5	478	.05	39	.05
54	72	52	7.4	86.4	93.8	0.0	93.8	0.0	7.6	2	188	.02	15	.02
55	70	70	19.5	119.7	126.3	12.9	139.2	9.3	10.6	15	2088	.21	159	.20
56	68	68	16.5	103.6	109.3	10.8	120.1	9.0	9.3	183	21979	2.21	1695	2.13

(double bin data run)

CASE #	TDB .deg.F.	TWB .deg.F.	COIL#1 (Mbtuh)	COIL#2 (Mbtuh)	Qsensible (Mbtuh)	Qlatent (Mbtuh)	Qtotal (Mbtuh)	Ql/Qt %	POWER (kws)	hours (hrs)	load/yr (MBTU)	load/yr %	power/yr (kw-hrs)	power/yr %
			STEADY STATE							ANNUAL				
57	67	66	13.7	98.9	104.2	8.4	112.6	7.5	8.8	241	27141	2.73	2119	2.66
58	67	64	10.9	96.4	101.6	5.7	107.3	5.3	8.5	219	23498	2.37	1853	2.33
59	67	62	8.3	94.3	99.6	3.0	102.7	3.0	8.2	125	12833	1.29	1022	1.28
60	67	60	5.8	92.3	97.6	.5	98.1	.6	7.9	41	4024	.41	324	.41
61	67	58	5.3	88.0	93.2	0.0	93.2	0.0	7.6	52	4849	.49	396	.50
62	67	56	5.3	82.8	88.0	0.0	88.0	0.0	7.3	41	3609	.36	299	.38
63	67	54	5.2	79.6	84.8	0.0	84.8	0.0	7.1	34	2883	.29	242	.30
64	67	52	5.2	77.2	82.5	0.0	82.5	0.0	7.0	19	1567	.16	132	.17
65	67	50	5.2	75.5	80.7	0.0	80.7	0.0	6.9	3	242	.02	21	.03
66	67	48	5.2	74.0	79.2	0.0	79.2	0.0	6.8	2	158	.02	14	.02
67	64	64	10.9	93.9	97.9	7.0	104.8	6.7	8.3	80	8387	.85	665	.83
68	62	62	8.3	85.3	88.3	5.2	93.6	5.6	7.6	169	15817	1.59	1290	1.62
69	62	60	5.8	83.3	86.4	2.7	89.2	3.1	7.4	115	10252	1.03	847	1.06
70	62	58	3.4	81.4	84.5	.3	84.8	.4	7.1	179	15182	1.53	1273	1.60
71	62	56	3.1	77.1	80.1	0.0	80.1	0.0	6.8	121	9695	.98	827	1.04
72	62	54	3.1	71.6	74.7	0.0	74.7	0.0	6.5	51	3809	.38	332	.42
73	62	52	3.1	68.8	71.8	0.0	71.8	0.0	6.4	34	2443	.25	216	.27
74	62	50	3.1	66.4	69.4	0.0	69.4	0.0	6.2	11	764	.08	68	.09
75	62	48	3.0	64.9	67.9	0.0	67.9	0.0	6.1	12	815	.08	73	.09
76	62	46	3.0	63.3	66.4	0.0	66.4	0.0	6.0	2	133	.01	12	.01
77	60	60	5.8	80.4	82.6	3.6	86.2	4.2	7.2	17	1466	.15	122	.15
78	58	58	3.4	73.9	75.3	2.1	77.4	2.7	6.7	108	8355	.84	721	.90
79	57	56	1.1	70.7	71.6	.2	71.9	.3	6.4	154	11066	1.12	979	1.23
80	57	54	.9	66.1	67.0	0.0	67.0	0.0	6.0	165	11049	1.11	994	1.25
81	57	52	.9	60.9	61.8	0.0	61.8	0.0	5.5	129	7968	.80	715	.90
82	57	50	.9	58.1	59.0	0.0	59.0	0.0	5.3	45	2655	.27	238	.30
83	57	48	.9	55.7	56.5	0.0	56.5	0.0	5.1	45	2544	.26	228	.29
84	57	46	.9	54.2	55.1	0.0	55.1	0.0	4.9	35	1928	.19	173	.22
85	57	44	.9	52.6	53.5	0.0	53.5	0.0	4.8	16	856	.09	77	.10
86	57	42	.9	51.1	52.0	0.0	52.0	0.0	4.7	3	156	.02	14	.02
87	54	54	0.0	62.3	62.3	0.0	62.3	0.0	5.6	54	3362	.34	302	.38
88	52	52	0.0	54.3	54.3	0.0	54.3	0.0	4.9	126	6839	.69	612	.77
89	52	50	0.0	49.4	49.4	0.0	49.4	0.0	4.4	101	4990	.50	446	.56
90	52	48	0.0	46.7	46.7	0.0	46.7	0.0	4.2	151	7057	.71	630	.79
91	52	46	0.0	44.2	44.2	0.0	44.2	0.0	3.9	100	4418	.45	394	.49
92	52	44	0.0	42.7	42.7	0.0	42.7	0.0	3.8	44	1879	.19	168	.21
93	52	42	0.0	41.1	41.1	0.0	41.1	0.0	3.7	24	987	.10	88	.11
94	52	40	0.0	39.7	39.7	0.0	39.7	0.0	3.5	2	79	.01	7	.01
95	50	50	0.0	46.5	46.5	0.0	46.5	0.0	4.1	16	744	.07	66	.08
96	48	48	0.0	39.5	39.5	0.0	39.5	0.0	3.5	60	2373	.24	212	.27
97	47	46	0.0	35.0	35.0	0.0	35.0	0.0	3.1	101	3540	.36	315	.40
98	47	44	0.0	32.4	32.4	0.0	32.4	0.0	2.9	126	4078	.41	363	.46
99	47	42	0.0	30.3	30.3	0.0	30.3	0.0	2.7	103	3121	.31	277	.35
100	47	40	0.0	28.8	28.8	0.0	28.8	0.0	2.6	34	980	.10	87	.11
101	47	38	0.0	27.5	27.5	0.0	27.5	0.0	2.4	26	715	.07	63	.08
102	47	36	0.0	26.0	26.0	0.0	26.0	0.0	2.3	5	130	.01	12	.01
103	44	44	0.0	27.5	27.5	0.0	27.5	0.0	2.4	24	660	.07	59	.07
104	42	42	0.0	20.7	20.7	0.0	20.7	0.0	1.8	75	1551	.16	137	.17
105	42	40	0.0	18.4	18.4	0.0	18.4	0.0	1.6	54	992	.10	88	.11
106	42	38	0.0	16.9	16.9	0.0	16.9	0.0	1.5	82	1387	.14	123	.15
107	42	36	0.0	15.5	15.5	0.0	15.5	0.0	1.4	54	834	.08	74	.05
108	42	34	0.0	8.8	8.8	0.0	8.8	0.0	.8	11	96	.01	9	.01
109	40	40	0.0	13.7	13.7	0.0	13.7	0.0	1.2	7	96	.01	9	.01
110	38	38	0.0	8.9	8.9	0.0	8.9	0.0	.8	34	302	.03	27	.03
111	37	36	0.0	4.6	4.6	0.0	4.6	0.0	.4	45	208	.02	18	.02

Annual pre-cooling coil load = 133.66 MMBtu
Annual after-cooling coil load = 858.72 MMBtu

Annual sensible cooling load = 914.76 MMBtu
Annual latent cooling load = 77.62 MMBtu

Annual cooling load = 992.38 MMBtu
Annual hours of operation = 8537 Hours
Annual Chiller Power consumption = 79689 Kw-hrs

(double bin data run)

APPENDIX G

COMPUTER PROGRAM FOR DETERMINING THE WEBB-VILLACRES COOLING TOWER CONSANTS

```
10 !
20 !                         'TOWER_FILL'
30 !
40 ! THIS PROGRAM IS BASED ON THE MODEL DEVELOPED BY R.L.WEBB AND A.VILLACRES
50 !    AND IS USED TO CALCULATE THE FILL CHARACTERISTICS OF A COOLING TOWER
60 !
70 !
80 DIM TOWER$[30]
90 !
100 DISP " Please input the name of the cooling tower manufacture."
110 DISP
120 INPUT TOWER$
130 !
140 DISP " Please input the model number of the cooling tower."
150 DISP
160 INPUT MODEL$
170 !
180 DISP " Please input the information requested for design conditions"
190 DISP
200 !
240 DISP " Please input the design cooling tower AIR FLOW RATE in CFM "
250 DISP
260 INPUT CFM_tower
262 !
270 DISP " Please input the design ambient WET-BULB temp.,RANGE,APPROACH,& WATER FLOW RATE "
272 DISP "                                 (deg.F)      (deg.F) (deg.F)         (gpm)"
280 DISP
290 INPUT TWB,R,A1,GPM1
300 !
310 DISP " Please input another APPROACH TEMPERATURE and WATER FLOW RATE for"
320 DISP "    the COOLING TOWER to achieve that RANGE,WET-BULB---(deg.F & GPM)"
330 INPUT A2,GPM2
340 !
350 GOSUB TOWERIO  !
360 !
370 RIg=GPM1*8.33/(CFM_tower*.0712)
380 A=A1
390 RIg(1)=RIg
400 GOSUB COUNT
410 Ntur(1)=CT
420 RIg=GPM2*8.33/(CFM_tower*.0712)
430 RIg(2)=RIg
440 A=A2
450 GOSUB COUNT
460 Ntur(2)=CT
470 XN=-(LGT (Ntur(1)/Ntur(2))/LGT (RIg(1)/RIg(2)))
480 C=Ntur(1)*RIg(1)^XN
490 !
500 PRINT
510 PRINT "                 COOLING TOWER FILL CHARACTERISTICS"
520 PRINT "               ----------------------------------------"
530 PRINT
540 PRINT USING "13X,AAA,DD.DDDDD,15X,AA,DD.DDDDD" ; "XN=",XN,"C=",C
550 PRINT
560 PRINT
570 END
580 !
590 COUNT: !          SUBROUTINE TO FIND KaV/L FOR C.T.
600 !
610 CT=0
620 Y(1)=.1
630 Y(2)=.3
640 Y(3)=.2
650 Y(4)=.3
660 Temp=TWB
670 GOSUB HVTA
680 HA=H
690 TW=TWB+A
700 FOR I=1 TO 4
710    TW=TW+Y(I)*R
720    HA=HA+RIg*Y(I)*R
730    Temp=TW
740    GOSUB HVTA
750    HT=H
```

```
760   DH=HT-HA
770   CT=CT+1/DH
780 NEXT I
790 CT=CT*R/4
800 RETURN
810 !
820 !
830 HVTA: !                    SUBROUTINE TO FIND THE ENTHALPY OF AIR
840 PB=101325
850 GOSUB PSAT
860 WS=.622*(PS/(PB-PS))
870 H=.24*Temp+WS*(1061+.444*Temp)
880 RETURN
890 !
900 TOWERID: !           SUBROUTINE TO IDENTIFY THE COOLING TOWER
910 !
920 PRINT "             ";TOWER$;"    MODEL ";MODEL$
930 PRINT
940 PRINT "          COOLING TOWER  SPECIFICATIONS"
950 PRINT "       ------------------------------------------"
960 PRINT
970 PRINT "       DESIGN WET-BULB TEMPERATURE                 (deg.F)     =";TWB
981 PRINT "       DESIGN RANGE OF THE C.T.                    (deg.F)     =";R
982 PRINT "       DESIGN AIR FLOW RATE                        (cfm)       =";CFM_tower
990 PRINT "       FIRST DESIGN WATER FLOW RATE                (gpm)       =";GPM1
1020 PRINT "       FIRST DESIGN APPROACH OF THE C.T.          (deg.F)     =";A1
1030 PRINT "       SECOND DESIGN APPROACH OF THE C.T.         (deg.F)     =";A2
1040 PRINT "       SECOND DESIGN WATER FLOW RATE OF THE C.T.   (gpm)      =";GPM2
1050 PRINT
1060 PRINT
1070 RETURN
1080 !
1090 !
1100 PSAT: !               SUBROUTINE TO FIND SATURATION PRESSURE
1110 !
1120 A(1,1)=6.41533947
1130 A(1,2)=6.41542599
1140 A(1,3)=6.44302266
1150 A(1,4)=6.53947838
1160 A(2,1)=.0821033478
1170 A(2,2)=.0724023398
1180 A(2,3)=.0698803796
1190 A(2,4)=.0656725584
1200 A(3,1)=-.000340068991
1210 A(3,2)=-.000264361973
1220 A(3,3)=-2.0606744496E-4
1230 A(3,4)=-.000160854949
1240 !
1250 TT=(Temp-32)*5/9
1260 IF TT>22.0322 THEN GOTO 1270 ELSE GOTO 1310
1270   IF TT<46.0304 THEN J=3 ELSE GOTO 1290
1280   GOTO 1370
1290     IF TT<72.0476 THEN J=4 ELSE GOTO 1350
1300     GOTO 1370
1310 IF TT>.00895 THEN J=2 ELSE GOTO 1330
1320   GOTO 1370
1330   IF TT>-19.94 THEN J=1 ELSE GOTO 1350
1340   GOTO 1370
1350 PRINT " THE TEMPERATURE FALLS OUTSIDE OF OUR RANGE"
1360 PAUSE
1370 PS=EXP (A(1,J)+TT*(A(2,J)+TT*A(3,J)))
1380 RETURN
1390 !
```

ANNUAL ENERGY CONSUMPTION OF
RECIPROCATING REFRIGERATION SYSTEMS FOR
HUMIDITY CONTROL

BY

THOMAS J. MEITL

B.S., Kansas State University

AN ABSTRACT OF A MASTER'S THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY

Manhattan, Kansas

1985

ABSTRACT

Refrigeration/dehumidification systems operate the majority of the time at conditions less severe than those of design. A generalized model describing reciprocating vapor-compression refrigeraton/dehumidification system performance was developed with cylinder unloading utilized as the basis for system capacity control at partial load operation. Normalized parameters were used where possible to reduce the number of parameters required to accurately represent equipment size and operational characteristics. The procedures outlined allow the model to reproduce catalog data ($\pm$3-1/2%) for a wide range of equipment sizes. Because of the generalized nature of the model, most of the reciprocating refrigeration/dehumidification systems available are covered by the model.