HVAC SYSTEM DYNAMICS AND ENERGY USE IN EXISTING BUILDINGS

by

RICHARD JOHN HACKNER

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ABSTRACT

This thesis summarizes the findings of the research that was done for the ASHRAE Task Group on Dynamic Response. The project (ASHRAE RP321) was entitled "HVAC System Dynamics and Energy Use in Existing Buildings."

The overall objective of the project was to determine operating strategies for HVAC systems which incorporate system dynamics and interactions and which will potentially reduce energy use. In keeping with the overall objective of the project the following specific goals were established:

1) To study the process dynamics and interactions of a building HVAC system through the use of collected test data and equipment computer models;
2) To determine, via computer simulations, the effect of the time between control decisions in the dynamic control of an HVAC system;
3) To determine, via computer simulations, dynamic HVAC operating strategies that will potentially reduce the HVAC system energy consumption.

The subject building for the project was the IBM National Marketing Division Headquarter's located in Atlanta, Georgia. The building is an 11-story structure

that is used mainly for office space. A large computing center is located on the 11th floor. The load from this center represents a major portion of the cooling load throughout the entire year. The core zone load of the building is met by two variable air volume systems. The perimeter zones are handled by four constant volume air handling units. There are five water chillers available to meet the chilled water requirements for the building. Three of the chillers are centrifugal type chillers, while the remaining two chillers are reciprocating chillers with heat recovery capability. A two-celled, induced-draft cooling tower with two-speed fans is used to reject the heat from the chillers.

The computer algorithms that were developed for the modeling of the HVAC equipment (cooling tower, water chillers, air handling units, etc.) are described. TRNSYS (a component-based computer code developed by the Solar Energy Lab at the University of Wisconsin-Madison) was used to link the individual component models together in order to simulate the HVAC system of the building. Comparison between measured data and the model predictions are given. Short term tests of equipment dynamics were conducted to determine the significant dynamic effects. The results showed that the significant transient effects were:

a) the cooling tower response to fan speed changes;

b) the "flush time" of the chilled water through the system;

c) the effects of the building structure due to capacitance.

The transient effects that were insignificant, due to their very short time duration, were:

a) the chiller response to set point (supply water) changes;

b) the air handling unit response to set point (supply air) changes.

The effect of the time between control decisions was studied using the developed building computer models. The results showed a 3.5% energy savings for a system operating under dynamic control as opposed to a system in which control decisions were made on an hourly basis. The results of the optimization study showed the importance of defining the operating limits for the HVAC equipment and the building comfort zone. The end results of this proejct were the identification and investigation of potential energy saving HVAC operating strategies and the availability of "reliable" equipment models.

The groups that could benefit the most from these models are building operators, HVAC control persons, and future researchers.

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NOMENCLATURE

C	thermal capacitance
COP	coefficient of performance
C_p	specific heat
E	cooling tower effectiveness
η	efficiency
h	enthalpy of a pure substance per unit mass
H	total enthalpy of a pure substance
h_m	enthalpy of a moist air mixture per unit mass dry air
h_s	sigma energy of a moist air mixture per unit mass dry air
h_{s1*}	sigma energy evaluated at the water inlet temperature
KWD	chiller power at the design operating load
KWR	Actual chiller power consumption divided by the chiller power at design load conditions
m	mass flow rate
PLR	Part load ratio. The ratio of the actual chilled water load to the chiller design load
q	heat transfer rate per lbm substance
Q_t	total heat transfer rate
R	tower capacity factor
Range	Temperature difference
T	temperature
t	time

NOMENCLATURE (continued)

TCHWR	chilled water return temperature
TCHWS	chilled water supply temperature
TCWR	condenser water return temperature
TCWS	condenser water supply temperature
TWB	wet bulb temperature
WAR	water-air mass ratio
WARR	water-air mass ratio reference value

<u>Subscripts</u>

a	air
i	inlet condition
l	liquid water
m	moist air
o	outlet condition
v	water vapor

1.0 INTRODUCTION

Numerous companies are now marketing energy management control systems (EMCS) for use in commercial buildings. One of the goals of these systems is to reduce building energy consumption by using computer technology to help control the HVAC equipment and to optimize the HVAC "process" within a building.

This paper presents research that has been done as a joint effort between ASHRAE, IBM and the University of Wisconsin-Madison under the project ASHRAE RP321. The goals of the project ("HVAC System Dynamics and Energy Use in Existing Buildings") were:

1) To study the process dynamics and interactions of a building HVAC system through the use of collected test data and equipment computer models;
2) To determine, via computer simulations, the effect of the time between control decisions in the dynamic control of an HVAC system;
3) To determine, via computer simulations, dynamic HVAC operating strategies that will potentially reduce the HVAC system energy consumption.

Once these operating strategies have been identified they should be tested using a building EMCS. The EMCS will gather data, process and analyze it, and perform control actions based on the pre-determined operating strategies.

1.1 HVAC SYSTEM DESCRIPTION

The HVAC and control systems were evaluated on an IBM building located in Atlanta, Georgia. It is an 11-story structure that is used mainly as an office building. The building itself is primarily of masonry construction which gives the structure a large amount of thermal capacitance. It has permanent exterior sun shades and tinted glazings to help reduce the solar load on the building.

The HVAC system, as shown in Figure 1.1, consists of five water chilling units, four constant volume perimeter and two variable air volume interior air handlers, and a two-celled cooling tower. Two of the five water chilling units (chillers #1 and #2) are 550-ton centrifugal chillers which provide the bulk of the chilled water used during the cooling season. During the heating season two reciprocating chillers equipped with heat recovery systems are available to meet both the heating (perimeter zones) and the cooling (computer room and interior zone) requirements of the building. These chillers are only used when the perimeter zones require heat. Since the major portion of the annual HVAC energy

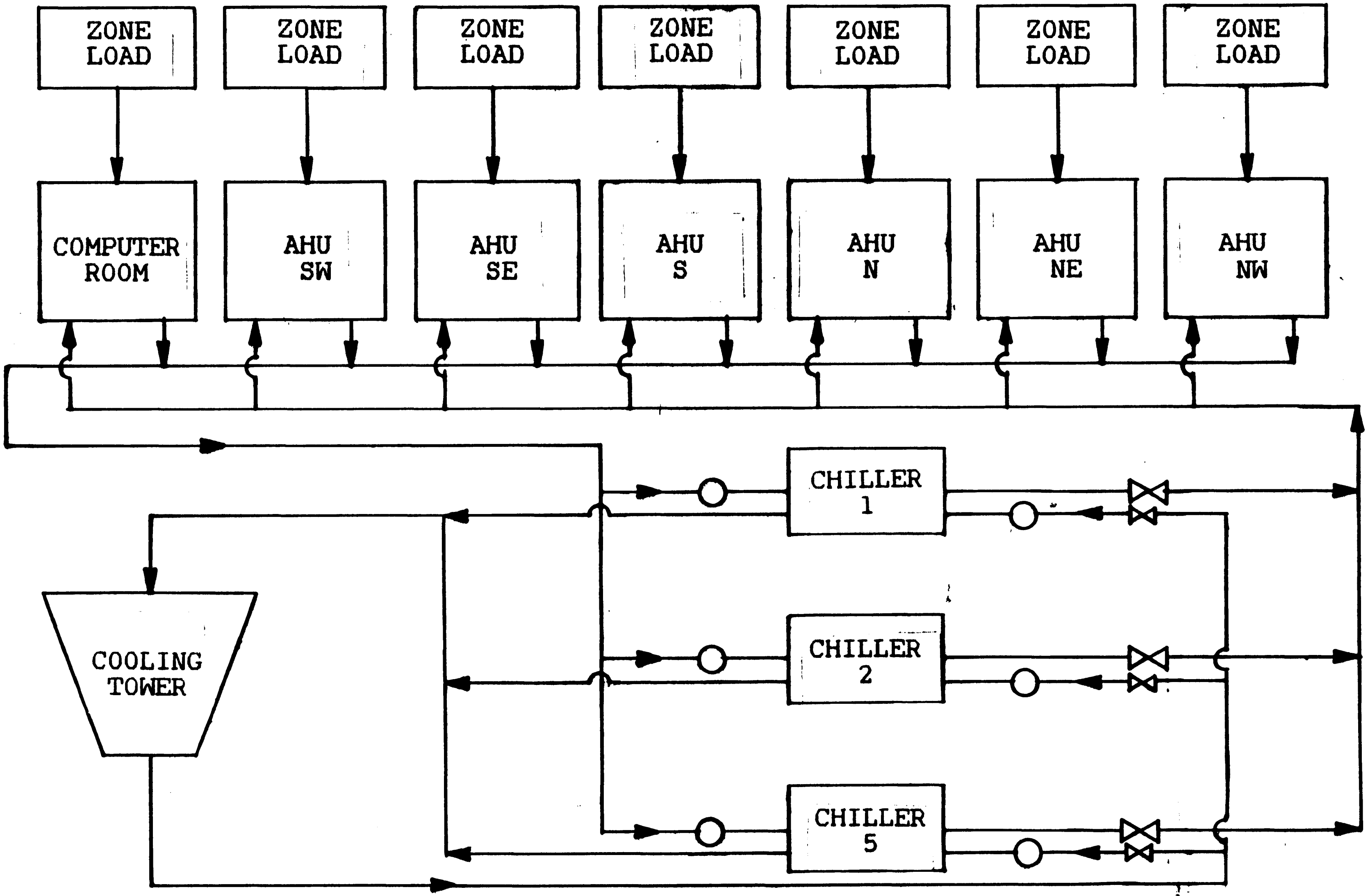

Figure 1 Schematic representation of the IBM-Atlanta HVAC system

bill for the site is directly attributable to summer cooling, the research concentrated on the cooling season. Hence, the reciprocating chillers with heat reclaim have been left out of the analysis and are not shown on the system schematic. The fifth chiller (chiller #5) is a 225-ton centrifugal water chiller equipped with a special control package that allows efficient operation at low part loads. This chiller is normally used in combination with one of the larger centrifugal units, or with one of the chillers with heat recovery, in order to provide additional chilled water capacity.

The heating or cooling loads in the perimeter zones of the building are met by four constant volume air handling units. The units, one for each perimeter zone (i.e., northwest, northeast, southwest, and southeast), are equipped with dual-purpose (heating or cooling) coils. The three modes in which the perimeter units operate are heating, cooling, and simple ventilation (i.e., the AHU is on, but it is only circulating air through the perimeter zones). The perimeter AHU's are independently controlled by the EMCS. For example, the northwest zone may require heating while the southwest zone requires cooling.

There are two variable air volume AHU's which provide the bulk of the air conditioning capacity for the site. The interior air handling units maintain the comfort conditions in the core of the building. The units, which are run only during occupied hours, can operate in three modes: "free cooling," "pay cooling," and evaporative cooling.

In the "free cooling" mode, a combination of outside air and return air is used to provide the supply air for the core zone. The "pay cool" mode means that chilled water is being supplied to the AHU in order to cool the outside air or the return air. In the evaporative cooling mode low humidity outside air is brought into the unit and is cooled by the use of humidifiers.

The two-celled cooling tower rejects the heat that is generated on the condenser side of the liquid chilling units. Each cell of the tower has its own two-speed fan. The operating status (off, low speed, high speed) of each fan is controlled by an application program in the EMCS.

1.2 THE ENERGY MANAGEMENT CONTROL SYSTEM

The EMCS that is installed in the Atlanta building employs two basic techniques to ensure that the building environment remains within the user specified "comfort zone":

1) Adjustment of the configuration of the "on-line" equipment. For example, the EMCS determines whether the type (centrifugal or reciprocal w/reheat) and number of chillers that are operating are able to meet the type (heating or cooling) and magnitude of the building load.
2) Set point adjustments for the chillers, air handlers, etc., in response to the building load. The EMCS is flexible enough to allow adjustments to be made at time increments ranging from one minute to several minutes. This is an indication of the ability of the EMCS to control dynamically.

One of the set points that the EMCS controls is the chilled water supply temperature. In this particular facility, changes in the set point are tied to the return water temperature in the chilled water loop. For example, if the return water temperature exceeds an acceptable range, the supply water set point is lowered to meet the increasing load.

Another example of set point control performed by the EMCS is that of the condenser water supply temperature (CWS). The CWS set point is a function of the wet bulb temperature of the outside air. This set point in-directly affects the operating level of the two-speed cooling tower fans. The operating level is linked to the magnitude of the difference between the ambient wet bulb temperature and the supply water set point.

2.0 HVAC EQUIPMENT MODELING

The nature of this research project required that the performance characteristics of the individual HVAC equipment components be known over their expected range of operating conditions. Fundamental modeling of the equipment was beyond the scope of the project. Manufacturer's data, however, were available in the form of tables or graphs. For a computer simulation, a more convenient way to express the manufacturer's data is in equation form. The forms of the air handling unit and chiller models were taken from standard ASHRAE algorithms (Stoecker, 1971). A more analytical approach was applied to the modeling of the cooling tower. This component was modeled using information from a paper written by Austin Whillier (1967). After modeling the equipment for steady-state operating conditions the models were modified, as necessary, to include the significant transient effects outlined in Chapter 3.

2.1 BUILDING MODEL

The Atlanta building is an 11-story office building. As such, many of the floors have identical characteristics in terms of heating and cooling requirements. The exceptions to this generalization are the top floor, which houses the computing facilities, and the third floor, which is a cafeteria area. Measured data indicate that the cooling load generated by the computers is fairly constant. This load can account for almost the entire site cooling load during the winter months and for close to one-third of the load during the peak cooling season. Since the computer room cooling load remains nearly constant, the load model for this floor is a constant value of internal generation.

The remaining ten floors of the building were modeled using a TRNSYS TYPE 19 Zone model (Klein, 1983). This zone model forms a set of heat transfer equations which describe the heat flows from and within the zone and solves the equations each timestep during the simulation. The zone model requires a number of inputs and parameters to specify the characteristics of each individual zone. Some of these parameters relate to the construction of the walls, floors, ceilings, windows, internal partitions, and walls separating zones which are at different temperatures. The "temperature level" control mode of TYPE 19 was used during the simulations. In this mode the zone temperature is affected by ambient conditions and the heating or cooling equipment input. The heating or cooling input is provided by an air ventilation stream which comes from the AHU model.

In the Atlanta building simulations, each floor was divided up into three zones: the East perimeter, the West perimeter, and the Core zones. The internal heat gains from the lights and office equipment were based on building power consumption data collected by the EMCS. Estimates for the occupancy level of the building were obtained from site personnel. Specific information on the inputs and parameters that were used can be found in the simulation deck section of Appendix A.

2.2 CHILLER MODEL

A chiller is rated for capacity according to the results of a standardized testing procedure. The chiller manufacturer presents the test results and sometimes data values at other operating conditions in the form of tables or charts. The chiller performance map (see Figure 2.1) was developed using the manufacturer's data and the methods outlined in Stoecker (1971). This map, along with the manufacturer's part load performance curve (Figure 2.2), is used to predict the performance of the chiller under various operating conditions. The part load ratio (PLR) is the measured chiller load divided by the design chiller load. The normal operating

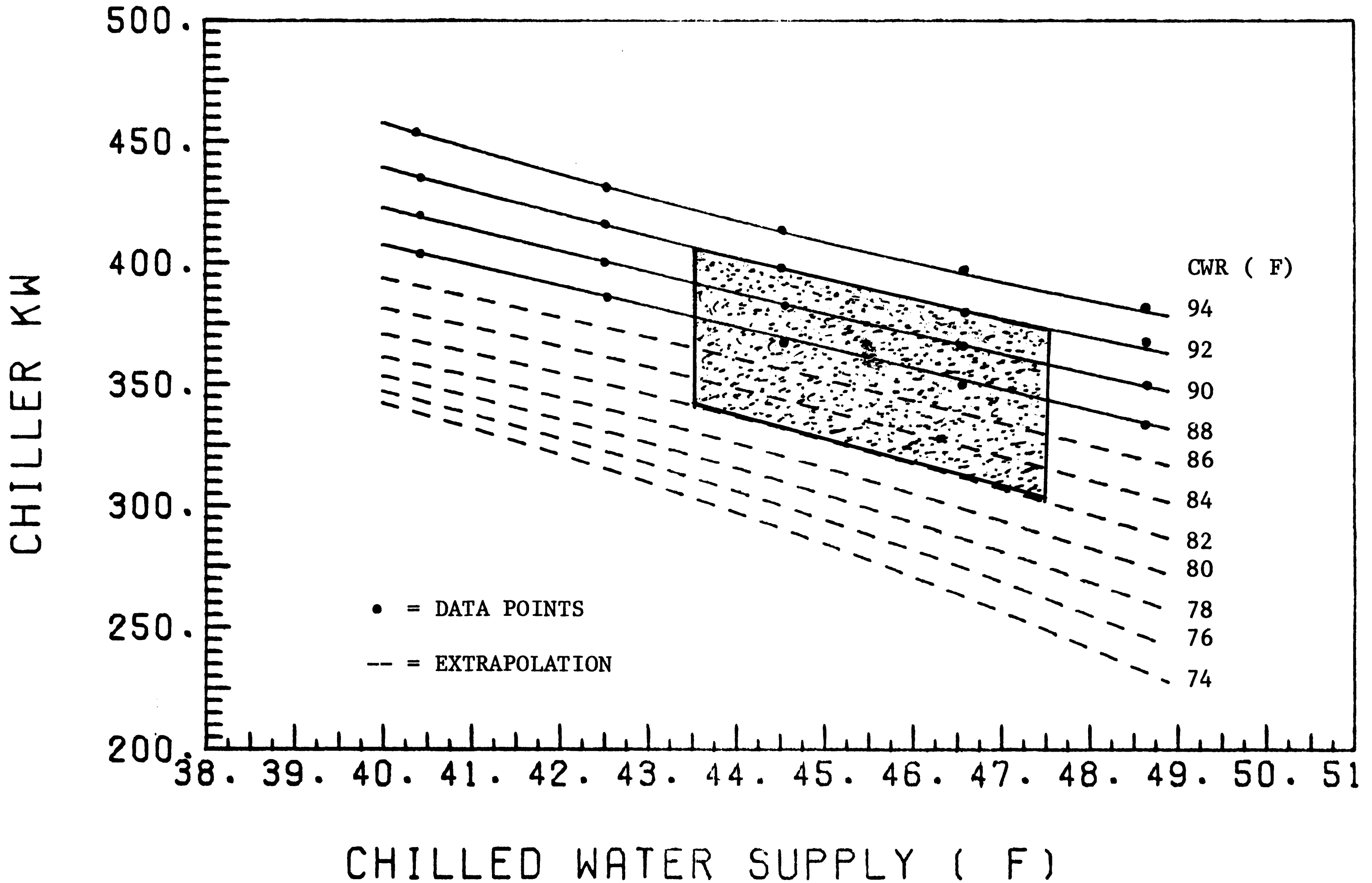

Figure 2.1 Chiller performance map for a York 550-ton centrifugal chiller

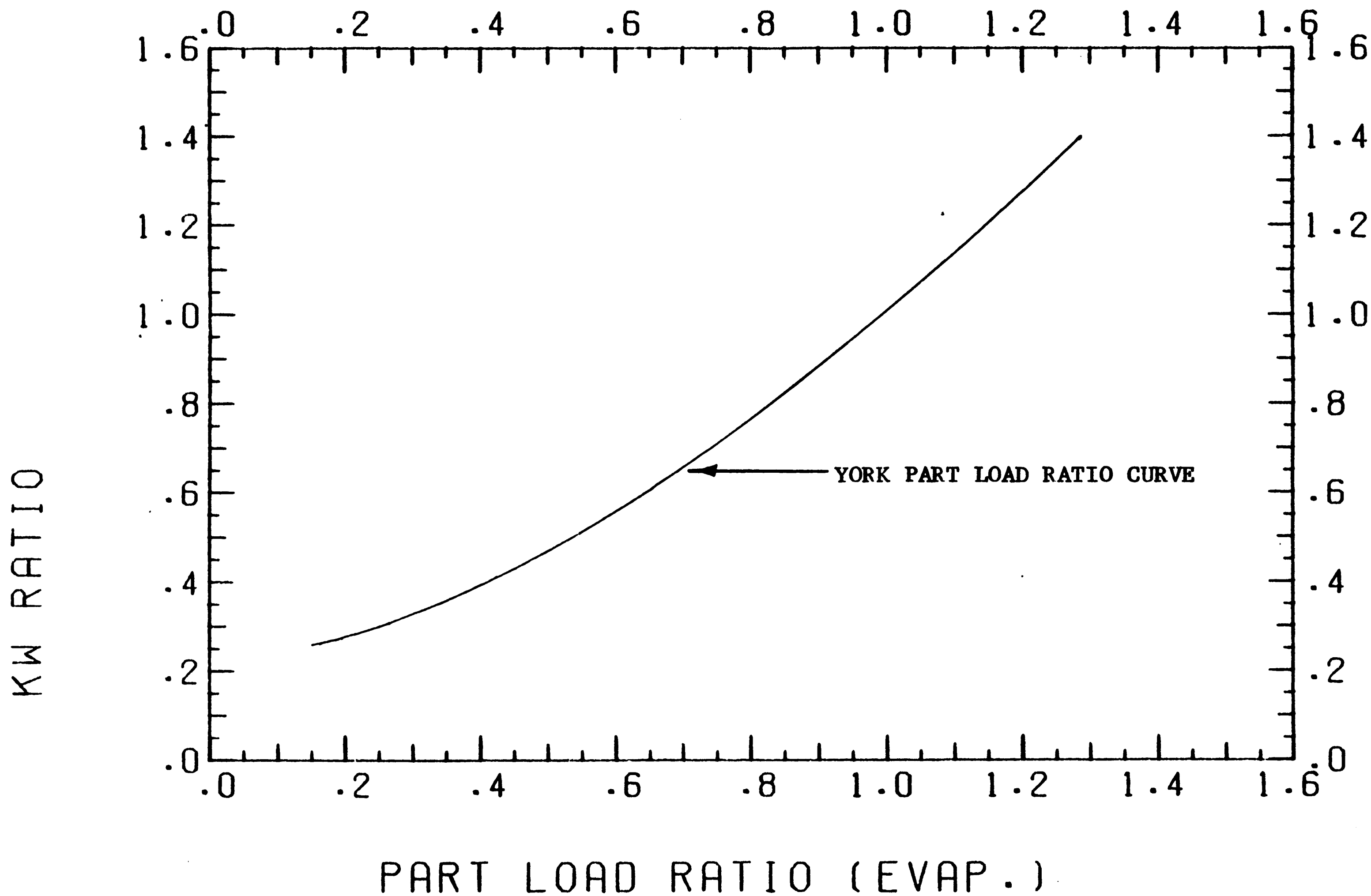

Figure 2.2 Part load ratio (PLR) curve for a York centrifugal chiller

range of the chiller is indicated by the shaded region on Figure 2.1.

As an example of the use of these two graphs, assume that on a particular day the chilled water load is 400 tons, the chilled water supply temperature is set at 46° F., and the condenser water return temperature is 80° F. These are the three parameters that are necessary in order to determine the power consumption of the chiller.

Entering Figure 2.1 with these operating conditions yields a chiller power consumption of 300 KW at the design load (550 tons) of the chiller. Using the part load ratio, the power ratio (KWR) is determined from Figure 2.2. In this particular case the PLR is .73 and the resulting KWR is .67. The KWR is now multiplied by the design power from Figure 2.1 to determine the predicted chiller power consumption. In this example the predicted chiller power consumption is 200 KW.

The performance map presented in Figure 2.1 is specific to the 550-ton centrifugal chiller manufactured by York. If it is desired to model a different chiller, a new set of chiller coefficients would have to be determined. The coefficients are the result of a biquadratic curve fit of the chiller manufacturer's data. The form of the fitted equation is:

$$KWD = C1 + C2*X + C3*X*X + C4*Y + C5*Y*Y + C6*X*Y + C7*X*X*Y + C8*X*Y*Y + C9*X*X*Y*Y \quad [2.1]$$

Where:

KWD = Chiller power consumption at design load

X = Chilled water supply temperature

Y = Condenser water return temperature

C1 - C9 = Generated coefficients

The chiller data supplied by the manufacturer can be fit to this equation by using the curve fitting routine found in Stoecker (1971) or by using one of the many curve fitting routines available on the open market.

Using data which were collected at the IBM facility and the design performance map, the KWR and PLR can be determined. Figure 2.3 shows the degree to which the steady-state data corresponds with the manufacturer's part load ratio (PLR) curve.

An error analysis was made of the measuring instruments that relate to the performance of the chiller. The relative error of each of the instruments (flow meters, temperature sensors, etc.) was then estimated. Using the root-sum-square method to determine the combined effect of the errors resulted in an overall error of

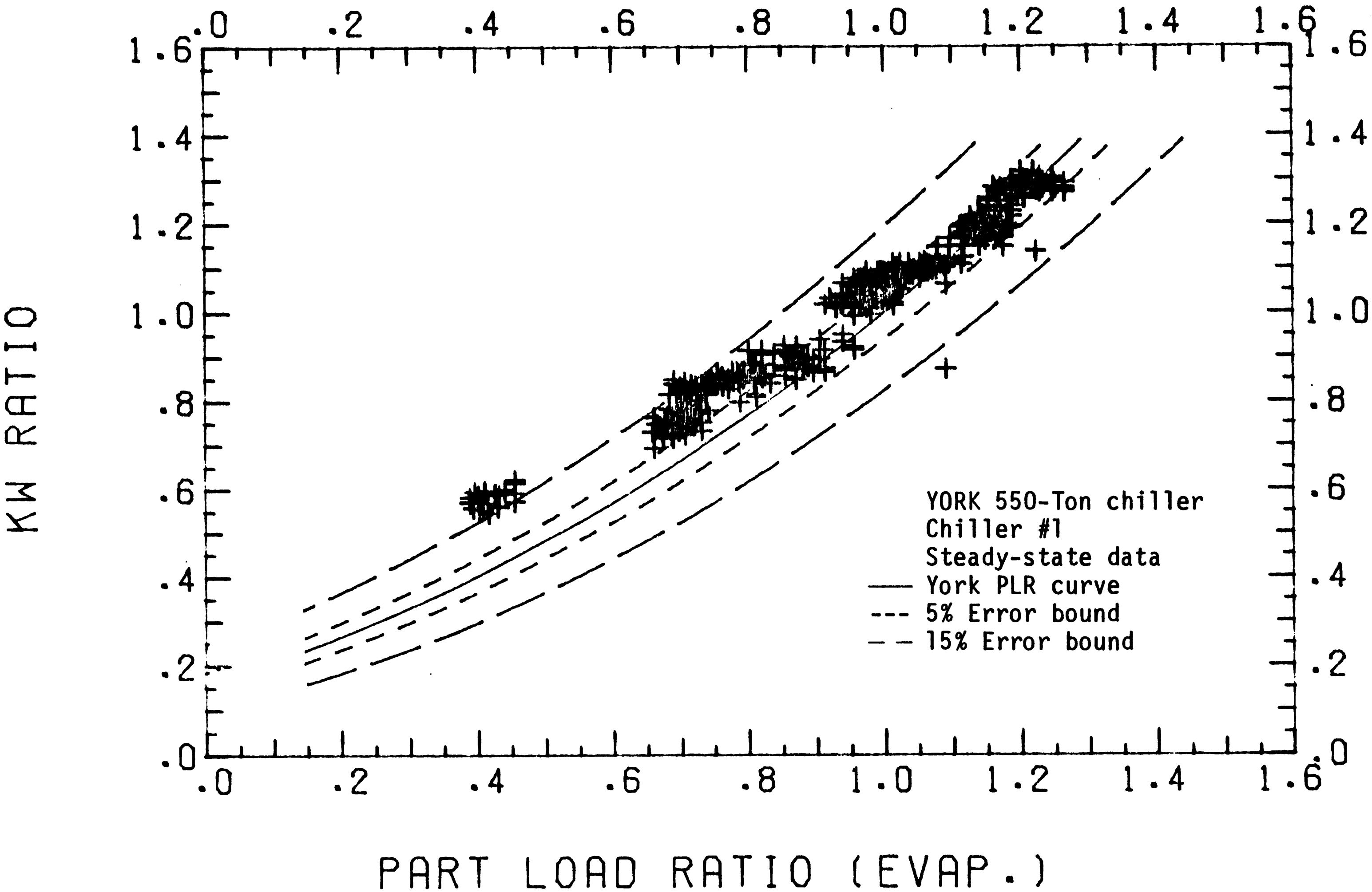

Figure 2.3 Comparison between steady-state chiller data and the York PLR curve

$\pm$ 15 percent. The data shown in Figure 2.3 are within the accuracy of the measurements except at very low part load ratios. One of the reasons for this discrepancy could be that the chiller is operating in the extrapolated region of Figure 2.1. Manufacturer's data were not available to produce this portion of the performance map.

Figure 2.4 gives another indication of the accuracy of the measured data. The data plotted in Figure 2.4 were collected at IBM-Atlanta building during June 1983. This figure shows the discrepancy between the chilled water heat transfer based on the evaporator side and the heat transfer based on the condenser side. The chilled water heat transfer based on the condenser side is defined as the condenser heat transfer minus the chiller power consumption. Theoretically, the evaporator side and the condenser side heat transfer should be equal, but, as the graph indicates, the values differ by 10 to 20 percent.

2.3 COOLING COIL MODEL

The bulk of the cooling capabilities of the Atlanta building is provided by two variable-air-volume Air Handling Units (AHU). One of the units serves the North portion of the "core zone," while the other unit serves

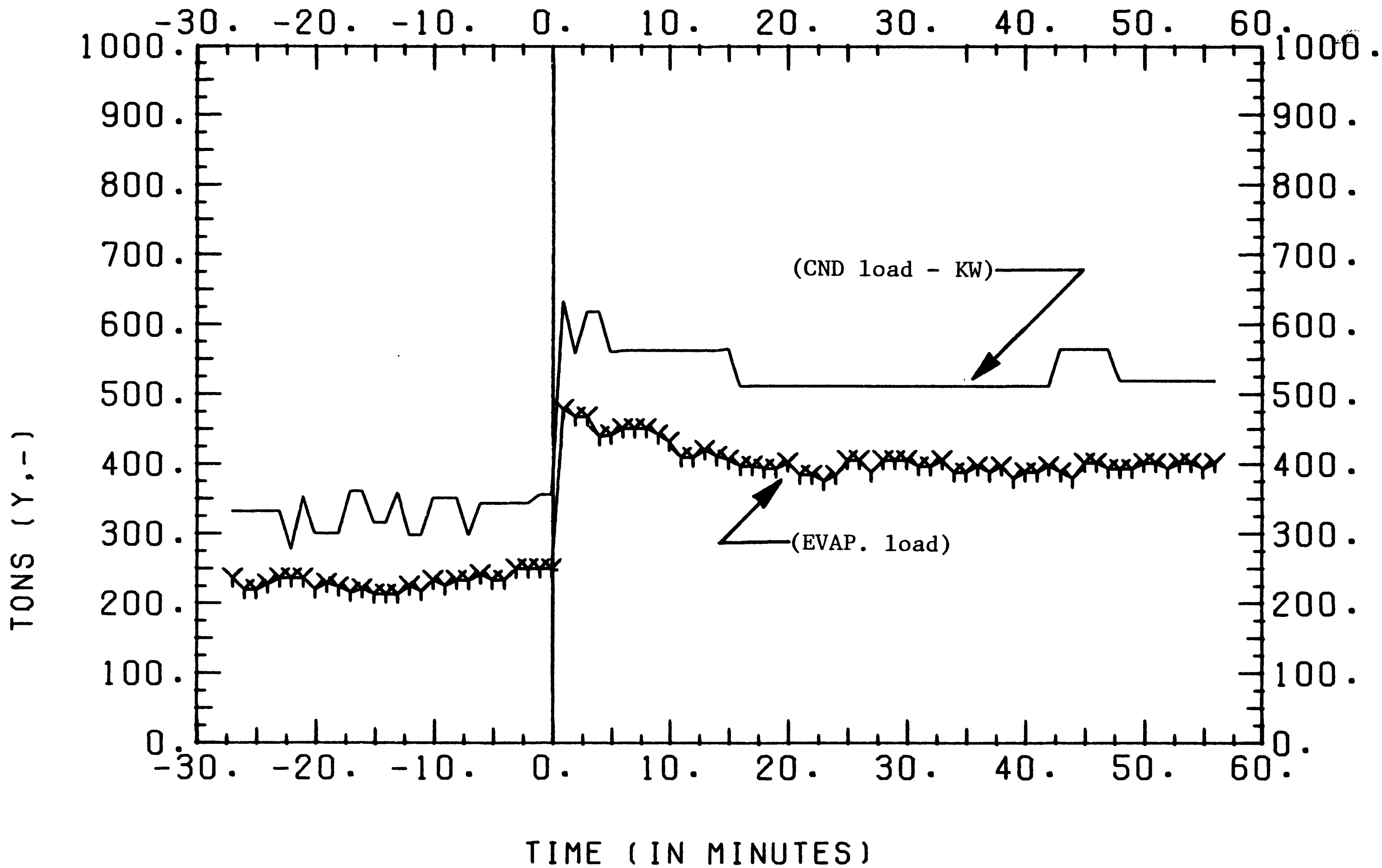

Figure 2.4 Comparison of the chilled water load based on the evaporator side and the condenser side of a chiller

the South core zone of the building. The equipment associated with each of the units consists of two supply fans, two return fans, a set of outside and return air dampers, and a pair of cooling coil banks. The specifications for the equipment are listed below:

Cooling coils (per AHU)

Two banks of coils stacked 3-coils high. The individual coils are of a full circuit design. They have a depth of six tube rows and a fin spacing of fourteen fins per inch. Each bank of coils has a face area of 120.6 square feet.

Supply fans

Type: Variable air volume using adjustable blade pitch for capacity control.

Capacity: 57,500 CFM at a static pressure of 6.2 inches of water.

Return fans

Type: Variable air volume using adjustable blade pitch for capacity control.

Capacity: 46,250 CFM at a static pressure of 1.25 inches of water.

The total cooling capacity of each AHU operating at design conditions is 5,364 MBTU/Hour.

The algorithm that was used to model the cooling coil of an AHU was based on the procedures found in

Stoecker (1971). The three main variables which govern the performance of a cooling coil are:

1) The temperature difference between the two fluid streams.
2) The design and surface arrangement of the coil.
3) The velocity and character of the individual air and water streams.

To model the performance of a cooling coil three "total" heat transfer equations are solved using an iterative procedure. The three equations are an air-side energy balance, a water-side energy balance, and an empirical heat transfer relationship. The air and water-side energy balances are simply a difference in enthalpies across either stream. The result is the following equations. For the water:

$$Q_t = M_1 * Cp_1 * (T_{1,o} - T_{1,i}) \qquad [2.2]$$

For the air:

$$Q_t = H_{a,i} - H_{a,o} \qquad [2.3]$$

The empirical relationship as described in Stoecker (1971) is:

$$Q_t = N_{row} * A_f * BRCW * WSF * LMTD \qquad [2.4]$$

Where:

N_{row} = number of coil rows deep

A_f = coil face area

BRCW = base rating of coil (heat transfer per unit area per degree LMTD)

WSF = wetted surface factor

LMTD = log mean temperature difference between the air and water fluid streams

The base rating of a coil is a function of the water-side velocity and the air-side face velocity. The form of the equation that is used to determine the BRCW can be found in Stoecker (1971). The wetted surface factor (WSF), which takes into account the possible latent heat transfer occurring at the coil, is also an empirical relationship. The form of this equation can also be found in Stoecker (1971). For both of these (WSF and BRCW) relationships manufacturers data are correlated to determine the specific coefficients to be used in each equation.

The accuracy of the cooling coil model was verified by a student working on an Independent Study project at the University of Wisconsin-Madison. In the final report for the study it was stated that the discharge temperature of the coil, using the model, was within one degree

F. of the manufacturers data. In addition, the heat transfer rate agreed to within ± 5 percent of the manufacturers reported values.

Variable volume fans are used in the North and South main AHU's. These vaneaxial fans use variable pitched fan blades to control the capacity of the fans. The internal control scheme used to control the fans is illustrated by the following situation. If a zone thermostat was reset to a lower temperature, the reduction in the thermostat setting would trigger the terminal unit dampers to open up further and provide additional cooling to the space. By opening up the damper the static pressure in the duct network would be reduced. Pressure sensors measure the static pressure at the outlet of each supply fan. These sensors would note the decreased pressure and then trigger the adjustment of the variable pitch fan blades. This maintains the desired static pressure across the supply fans.

In the AHU computer model, the flow rate of the supply fans is assumed to vary in a linear fashion according to the average temperature of the zone (see Figure 2.5). The minimum flow corresponds to the minimum ventilation requirement of the space. The maximum flow value is limited by the individual fan characteristics.

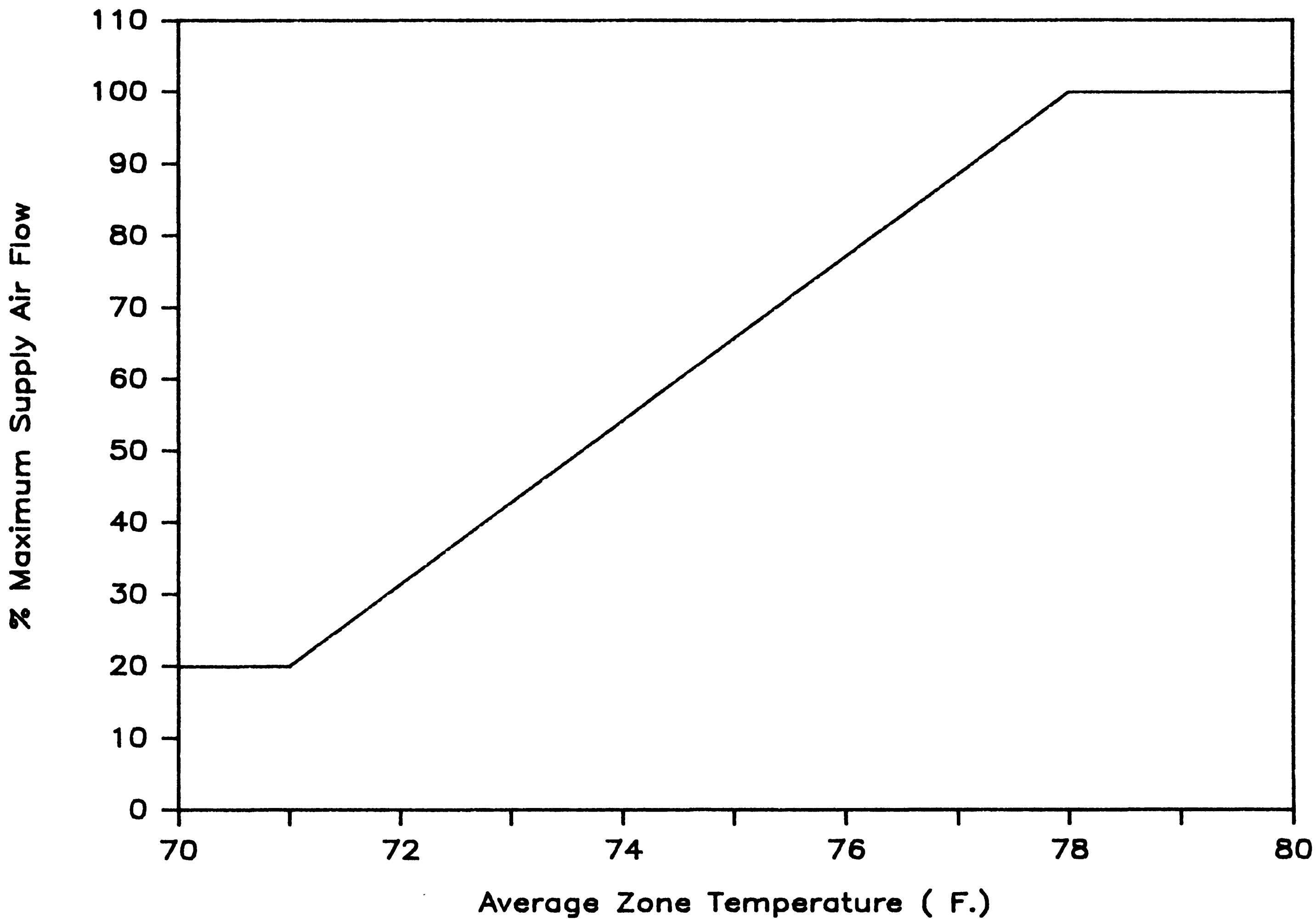

Figure 2.5 Relationship between the maximum supply fan flow and the average zone temperature

See section 4.1 for more information on the control characteristics of the AHU model.

2.4 COOLING TOWER MODEL

Cooling towers play an integral role in the operation of many large commercial HVAC systems. These towers are used to reject the heat from the condenser side of water chilling units. A schematic diagram of the induced-draft, cross-flow tower used in IBM-Atlanta is shown in Figure 2.6. The underlying principle behind the operation of a cooling tower is based on the combined effect of heat and mass transfer. The temperature and moisture difference between the air and water streams causes some of the water to evaporate. As the water evaporates it must absorb energy to change from the liquid to the vapor state. This energy is transferred from the water which has remained as a liquid. In other words, the heat of vaporization is removed from the water stream and transferred to the air stream. As a result, warmer, moist air and cooler, liquid water exit from the tower.

In general, the cooler the water that can be returned to the condenser side of the chilled water units the higher the chiller efficiency will be. However, a trade-off exists between the savings in chiller power

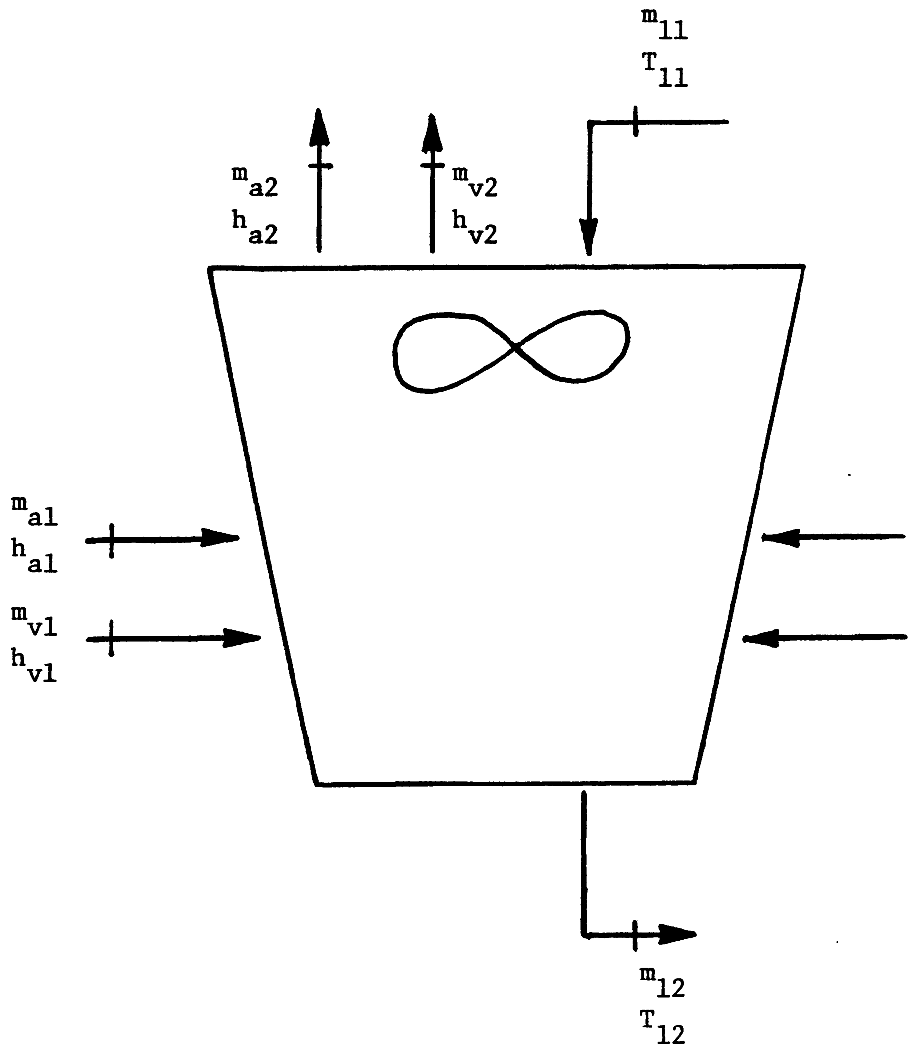

Figure 2.6 Schematic representation of a cross-flow, induced-draft cooling tower

consumption and the additional first cost of a high efficiency cooling tower or the additional operating cost of a tower. A higher fan flow rate reduces the outlet temperature; however, a higher operating cost results from the corresponding increase in cooling tower fan power consumption. Efforts to improve the efficiency of "wet" cooling towers have centered on methods of forcing air through the cooling tower and on increasing the contact surface area of the water. To increase the contact area between the water and air streams spray nozzles are used to distribute the water over baffles or fill materials. The baffles are commonly made of wooden slats, while the fill materials can be made of plastic or ceramic material. There are several methods that are used to force air through the tower. Among these are forced draft, induced draft, and natural circulation systems.

The Atlanta building uses a two-cell, induced-draft, cross-flow cooling tower to dissipate the heat from the condenser side of the water chillers. Each of the cell propeller fans can operate in the off, low speed, or high speed condition. Data from the tower manufacturer were available to determine the air flow rate at the high fan speed setting. Since the low speed was half of the high fan speed, the fan laws of performance (Tuve, 1966) were used to determine the air flow at the low speed setting. Literature from the manufacturer (Marley, 1982) indicated that in the off condition natural convection would produce an air flow equivalent to between 10 and 15 percent of the flow at the high fan speed setting. This value is highly variable depending on the wind speed and direction, and also on the presence of architectural barriers surrounding the tower.

One of the common analytical techniques used to model the performance of cooling towers is called the Merkel method. In this method the cooling tower is divided up into a number of nodes. A typical node is shown in Figure 2.7. An energy balance on the node yields the equation:

$$dq = G * dh_a = LdT \qquad [2.5]$$

The concept of enthalpy potential (Stoecker, 1982) can be used to express the rate of heat transfer as:

$$dq = h_c dA * (h_i - h_a) \qquad [2.6]$$

Where:

G = air flow

L = water flow

T = water temperature

h_c = convection coefficient

h_i = enthalpy of saturated air at the water temperature

h_a = enthalpy of air

Cp_a = specific heat of moist air

A = surface area

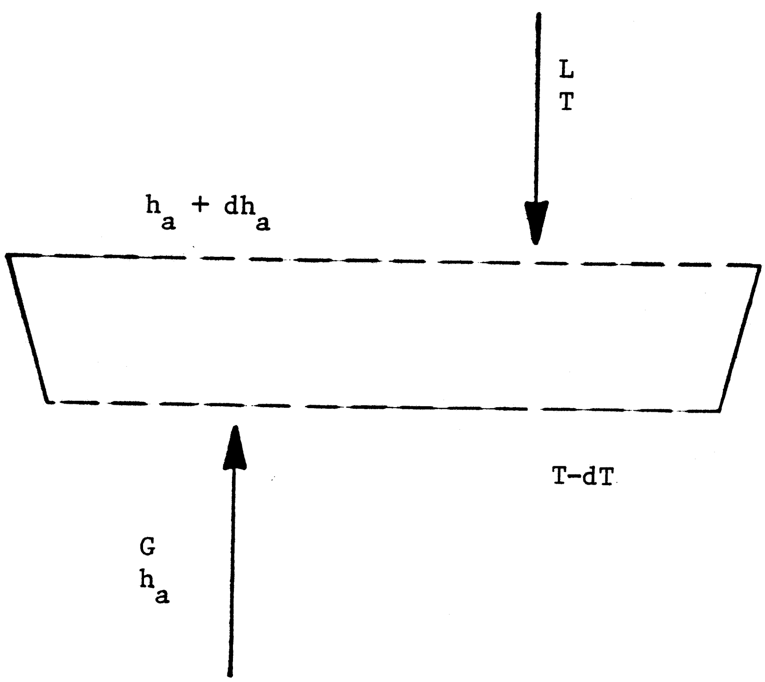

Figure 2.7 Typical node for a cooling tower

Equation 2.5 must now be integrated to find the rate of heat transferred by the whole tower. Combining Equation 2.5 and Equation 2.6 and integrating yields:

$$L \int_{T_{out}}^{T_{in}} \frac{dT}{h_i - h_a} = \frac{h_c A}{Cp_m} \qquad [2.7]$$

Where T_{in} and T_{out} are the entering and leaving water temperatures, respectively.

To predict the outlet conditions using the Merkel method an iterative procedure must be used since the leaving water temperature is initially unknown. A more straight forward approach, which does not involve integration, was proposed by Austin Whillier (Whillier, 1967). The theory behind his method is described below.

Consider a cooling tower operating under steady-state, steady-flow conditions. Applying the conservation of mass to the air stream:

$$m_{a1} = m_{a2} \qquad [2.8]$$

Similarly, to the water (liquid and vapor) streams:

$$m_{v1} + m_{11} = m_{v2} + m_{12} \qquad [2.9]$$

An energy balance on the entire system yields the following:

$$m_{a1}h_{a1} + m_{v1}h_{v1} + m_{11}Cp_1T_{11} = m_{a2}h_{a2} + m_{v2}h_{v2} + m_{12}Cp_1T_{12} \qquad [2.10]$$

Combining Equations 2.8, 2.9, and 2.10 and using the definition of the humidity ratio results in:

$$m_{a1}\ (h_{a1} + w_1 h_{v1}) - m_{a1}\ (h_{a2} + w_2 h_{v2}) + m_{a1}\ (w_2 - w_1)\ Cp_1 T_{12} = m_{11} Cp_1\ (T_{12} - T_{11}) \qquad [2.11]$$

The $(h_a + wh_v)$ terms are referred to as the enthalpy of the moist air mixture (h_m) which is one of the parameters found on a psychrometric chart. The third term on the right-hand side of Equation 2.11 is generally between 4 and 9 percent of the value of the right-hand side of the equation. In the past, analyses of cooling towers have generally neglected this term. Whillier, however, uses a different approach in order to reduce the error caused by this assumption. By using a grouping of terms called "sigma energy," introduced by William Carrier (Carrier, 1911), the analysis of processes involving changes in the moisture content of an air mixture is simplified. Sigma energy (h_s) is defined as:

$$h_s = h_m - w{*}Cp_1{*}T_m \qquad [2.12]$$

The important feature to note about sigma energy is that in an adiabatic saturation process sigma energy remains constant. By using the sigma energy approach

Equation 2.11 can be re-written as:

$$m_{11}Cp_1 (T_{12} - T_{11}) = m_{a1} (h_{s1} - h_{s2})$$
$$+ m_{a1}Cp_1 [w_1 (T_{a1} - T_{12})$$
$$- w_2 (T_{a2} - T_{11})] \qquad [2.13]$$

The bracketed term on the right-hand side of Equation 2.13 is normally between 2 and 4 percent of the value of the left-hand side and will be neglected. The use of sigma energy term instead of the standard procedure has cut the relative error in half. The resulting equation is now:

$$m_{11}Cp_1 (T_{12} - T_{11}) = m_{a1} (h_{s1} - h_{s2}) \qquad [2.14]$$

Whillier uses two ways to describe the efficiency of a cooling tower. Since the temperature of the water leaving the cooling tower can not fall below the ambient wet bulb temperature, due to thermodynamic considerations, a water efficiency is defined as:

$$\eta_1 = (T_{11} - T_{12}) / (T_{11} - T_{WB}) \qquad [2.15]$$

Likewise, the maximum possible wet bulb temperature that the exit air can achieve is the temperature of the incoming liquid water. An air efficiency can be defined as:

$$\eta_a = (h_{s2} - h_{s1}) \;/\; (h_{s1*} - h_{s1}) \qquad [2.16]$$

Whillier, further, defines a water-air ratio (WAR) and a tower capacity factor (R). The WAR reference value (WARR) is the one combination of water flow and air flow rates at which the thermal capacities of the two flow streams are equal. From this definition:

$$\mathrm{WARR} = (h_{s1*} - h_{s1}) \;/\; [Cp_l \; (T_{l1} - T_{a1})] \qquad [2.17]$$

Since the cooling tower seldom operates exactly at reference water-air ratio conditions, Whillier defines a term called the tower capacity factor. "R" is defined as the actual water-air ratio divided by WARR, or in equation form:

$$R = (m_{l1}/m_{a1}) \;/\; \mathrm{WARR} \qquad [2.18]$$

Combining Equations 2.14 through 2.18 yields the relation:

$$\eta_a = R\eta_w \qquad [2.19]$$

Normally, heat exchanger effectiveness is correlated in terms of the ratio of the thermal capacities of the fluid streams (C_{min}/C_{max}). To make the Whillier analysis consistent with established heat exchanger analysis, the cooling tower effectiveness (E) is defined:

$$E = \eta_w \qquad \text{if } R < 1 \qquad [2.20]$$

or

$$E = \eta_a \qquad \text{if } R > 1 \qquad [2.21]$$

Manufacturer's data or test data (if available) can be used to generate an effectiveness versus tower capacity curve. To simulate the outlet conditions of a cooling tower a simple 4-step computer algorithm can now be used:

1) Determine the actual water-air ratio;
2) Calculate the WARR, water-air reference value, knowing the inlet water temperature, ambient wet bulb temperature, and the barometric pressure;
3) Calculate the tower capacity factor;
4) Knowing R, determine the water efficiency ($R < 1$) or air efficiency ($R > 1$) and calculate the outlet conditions of the tower.

The "Whillier method," combined with cooling tower performance data supplied by the manufacturer, provides a relatively simple, but accurate method to determine the tower performance at varying conditions of fan speeds, condenser water flow rates, etc. In addition, the tower model has been modified to include the transient effects that were determined during the testing period.

Figure 2.8 is a schematic representation of a cross-flow, induced-draft cooling tower including the sump which receives the condenser water after it has fallen through the cooling tower. The analysis of the transient effects of the cooling tower is based on energy and mass balances. The energy balance on the sump is:

$$\text{Energy in} - \text{Energy out} = \text{Energy stored} \qquad [2.22]$$

or in terms of flow rate, specific heat, and temperature:

$$- M_1 * Cp_1 * (T_a - T_b) = M * Cp_1 * dT_a/dt \qquad [2.23]$$

solving this equation:

$$T_a = T_b + (T_{ao} - T_{bo})\exp[-M_1/(M * t)] \qquad [2.24]$$

Where:

M_1 = mass flow rate of the condenser water

M = mass of the water in the sump

Cp_1 = specific heat of water

T_a = temperature of the water leaving the sump

T_{ao} = initial temperature of the water leaving the sump

T_b = temperature of the water entering the sump.

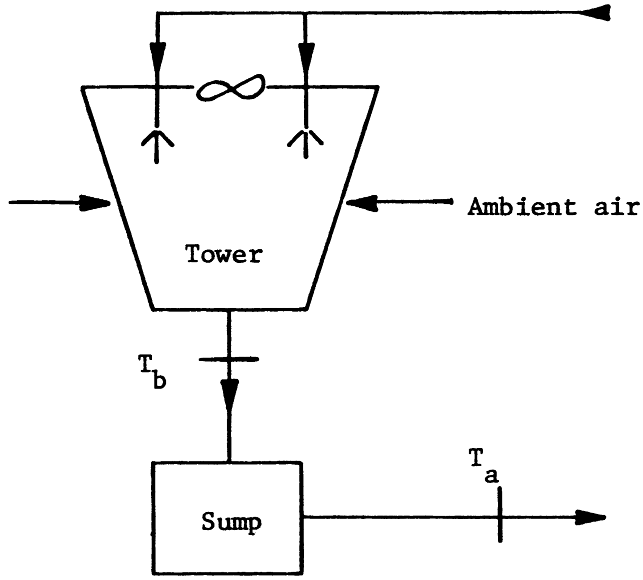

Figure 2.8 Schematic representation of a cross-flow, induced-draft cooling tower including the sump

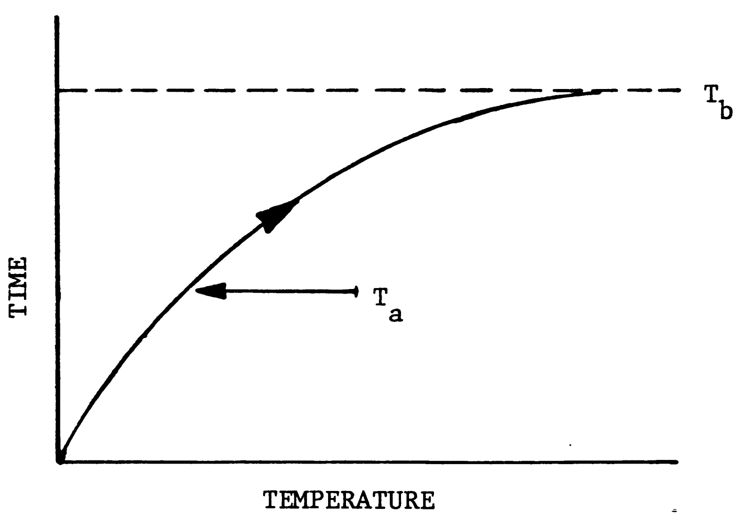

Figure 2.9 Typical response of the condenser water supply temperature to a step change in the tower fan speed

This temperature is a function of the condenser water flow rate, fan speed, and ambient wet bulb temperature.

T_{bo} = initial temperature of the water entering the sump

t = time

Figure 2.9 shows the response of the cooling tower to a step change in the tower fan speed. When one of the 550-ton chillers is operating, the time constant of the cooling tower is approximately 5 minutes.

Figures 2.10 and 2.11 show the effects of the modifications made to the cooling tower model in order to incorporate the transient effects of the water storage in the sump of the cooling tower. Figure 2.10 shows the model predictions before the transient effect of the change in the temperature of the sump is taken into account. Figure 2.11 is the same data, but, now the model has included the transient response. The measured data used in Figures 2.10 and 2.11 were taken at the IBM-Charlotte facility and are only used to show the effect of the inclusion of the transient response to the cooling tower model.

Figure 2.12 is a plot of the cooling tower model prediction versus measured data. The cooling tower

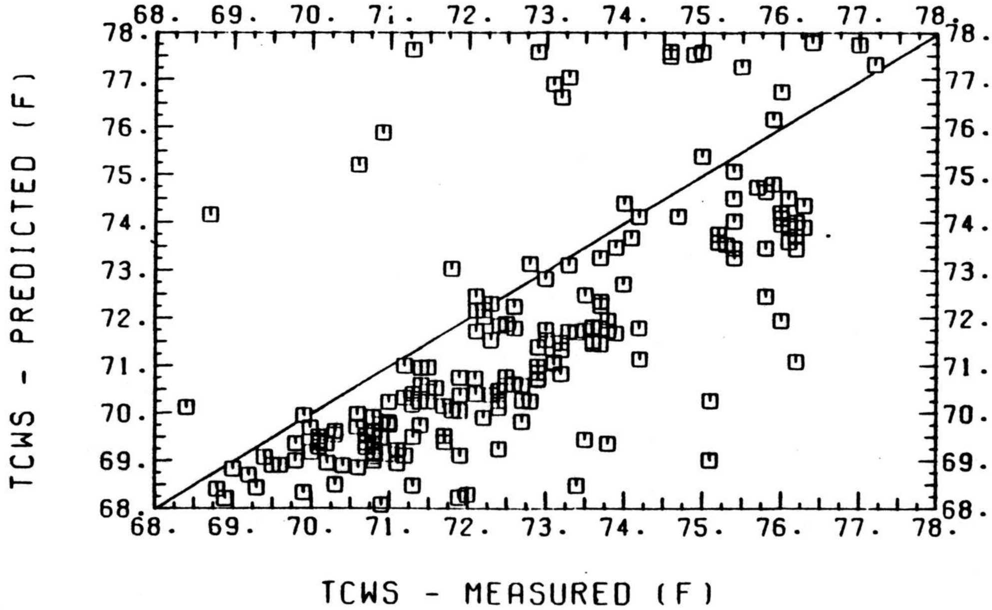

Figure 2.10 Comparison between the measured condenser water supply temperature and the predicted temperature using the "Whillier method." Data were obtained from IBM-Charlotte, N.C.

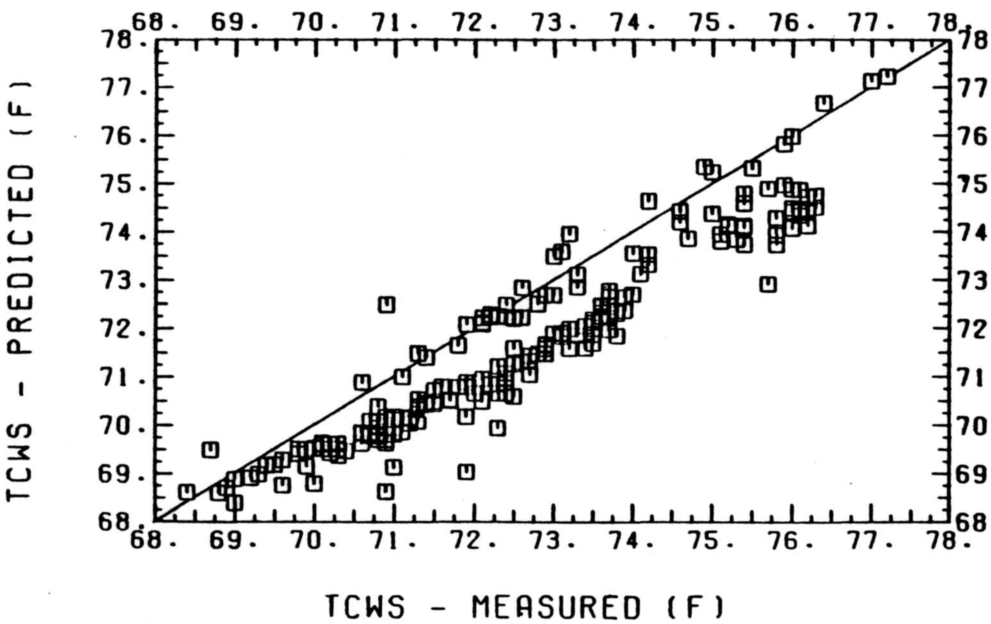

Figure 2.11 Comparison between the measured consenser water supply temperature and the predicted temperature using the "Whillier method." Model has included the transient effects of the sump. Data were obtained from IBM-Charlotte, N.C.

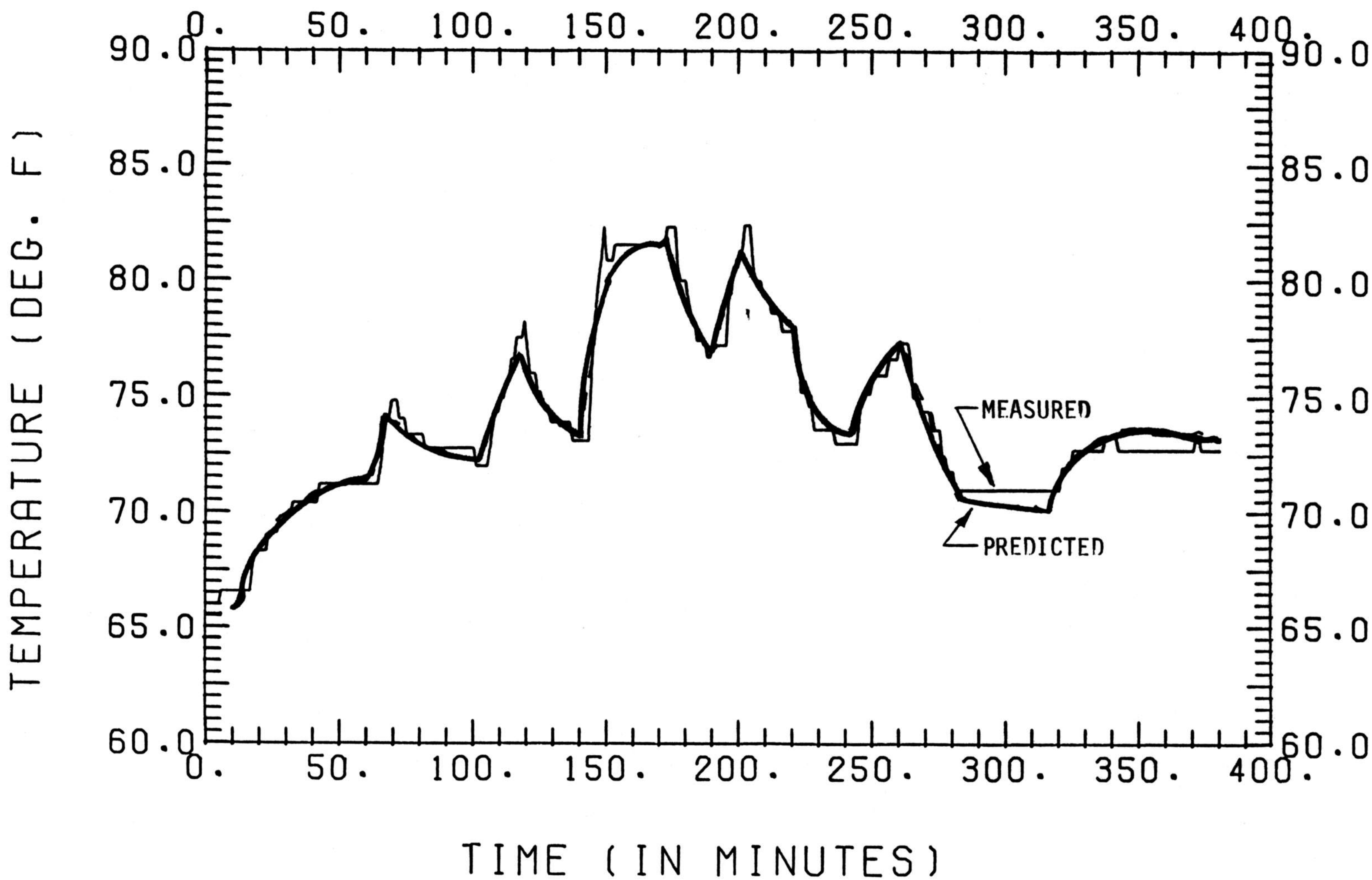

Figure 2.12 Comparison between the measured condenser water supply temperature and the predicted temperature. Measured data were collected during the cooling tower fan speed test period at IBM-Atlanta.

model with the transient effects added was used to generate the predicted condenser water supply temperature. The measured data include all of the cooling tower test data that were collected during June at the Atlanta facility.

Listings of the non-standard TRNSYS components used in the computer simulation appear in Appendix B.

3.0 EQUIPMENT TRANSIENT RESPONSE TESTS

Quasi steady-state data under normal system operation were collected during several periods in the Spring and Fall of 1983. These data were recorded by the EMCS every 15 minutes during the data collection periods. To date, 18 days of data have been collected at the site and analyzed.

Dynamic tests on the HVAC equipment and system were conducted at the Atlanta facility between June 1 and June 9, 1983. Since the EMCS was "on-line" during the testing period every effort was made to try and maintain as many controlled variables constant as possible. The reason for this was to be able to observe the system/equipment response to a single variable change during each test. Data were periodically checked following each set point change to allow the system to re-stabilize.

The collected data consist of measurements recorded by the EMCS on a minute-by-minute basis during the tests. There were approximately 230 discrete measurements which were recorded during each of the tests.

The tests that were performed include:

a) Chilled water supply set point adjustments

b) Condenser water supply set point adjustments

c) AHU supply air set point adjustments.

The chiller response to chilled water supply set point adjustments was evaluated using information recorded from a series of chiller tests. Each test consisted of raising or lowering the supply set temperature in 1, 2, or 4 degree F. increments. Sufficient time was allowed between tests to ensure that the HVAC system had returned to a quasi steady-state condition. A total of 15 tests (including some duplicate tests) were performed and data collected for each test.

To determine the response of the cooling tower to changes in the fan speed status it was necessary to work within the operating strategy of the EMCS. The EMCS uses approach control to determine the fan speed setting. Under this strategy the setting is linked to the magnitude of the difference between the actual supply water temperature and a pre-determined supply water set point. For the purpose of dynamic testing the condenser water supply set point was either raised or lowered during each test run. In this way changes in the operating status of the tower fans could be observed and test data could be collected. There were five possible combinations of fan status for the two-speed cooling tower fans. A total of 15 cooling tower tests were conducted during the June 1983 test period.

The response of the AHU to changes in the supply air set point was determined from a series of AHU tests. Each test consisted of raising or lowering the supply air set point temperature in 1, 2, or 5 degree F. increments. Again, sufficient time was allowed between tests for the HVAC system to adjust itself to the new operating condition. Fifteen AHU supply air set point tests were conducted and data recorded. Data were also collected during the following system operating conditions:

a) Morning start up;

b) Evening shut down;

c) With one chiller operation and then switching to two chillers;

d) Outside air dampers closed and then outside air dampers switched to open;

e) Building unoccupied mode.

Using the transient test data, the "significant" transient effects of the building HVAC equipment were determined. These were:

a) the cooling tower response to fan speed changes;

b) the "flush time" of the chilled water through the system;

c) the effects of the building structure due to capacitance.

The transient effects that were not significant for energy evaluations due to their very short time duration were:

a) the chiller response to set point (supply water) changes;

b) the AHU response to set point (supply air) changes.

The bases for these conclusions are discussed below.

Figures 3.1 and 3.2 show the results of a chilled water supply temperature response test. The figures show the response of the chilled water supply and return temperatures to a step change in the chilled water supply set point that occurred at Time equal to zero. As shown in Figure 3.1 the supply water temperature responds very quickly to the set point change. The response is completed in less than five minutes. This transient effect was felt to be insignificant for energy use evaluation.

Figure 3.1 also illustrates one of the significant transient effects. The effect of the "flush time," which is the time that it takes for a "slug" of water to travel through the entire chilled water loop, can be seen by observing the response of the chilled water return temperature. After the change in the supply water temperature there is an initial lag in the response of the return water temperature. After the lag, the return

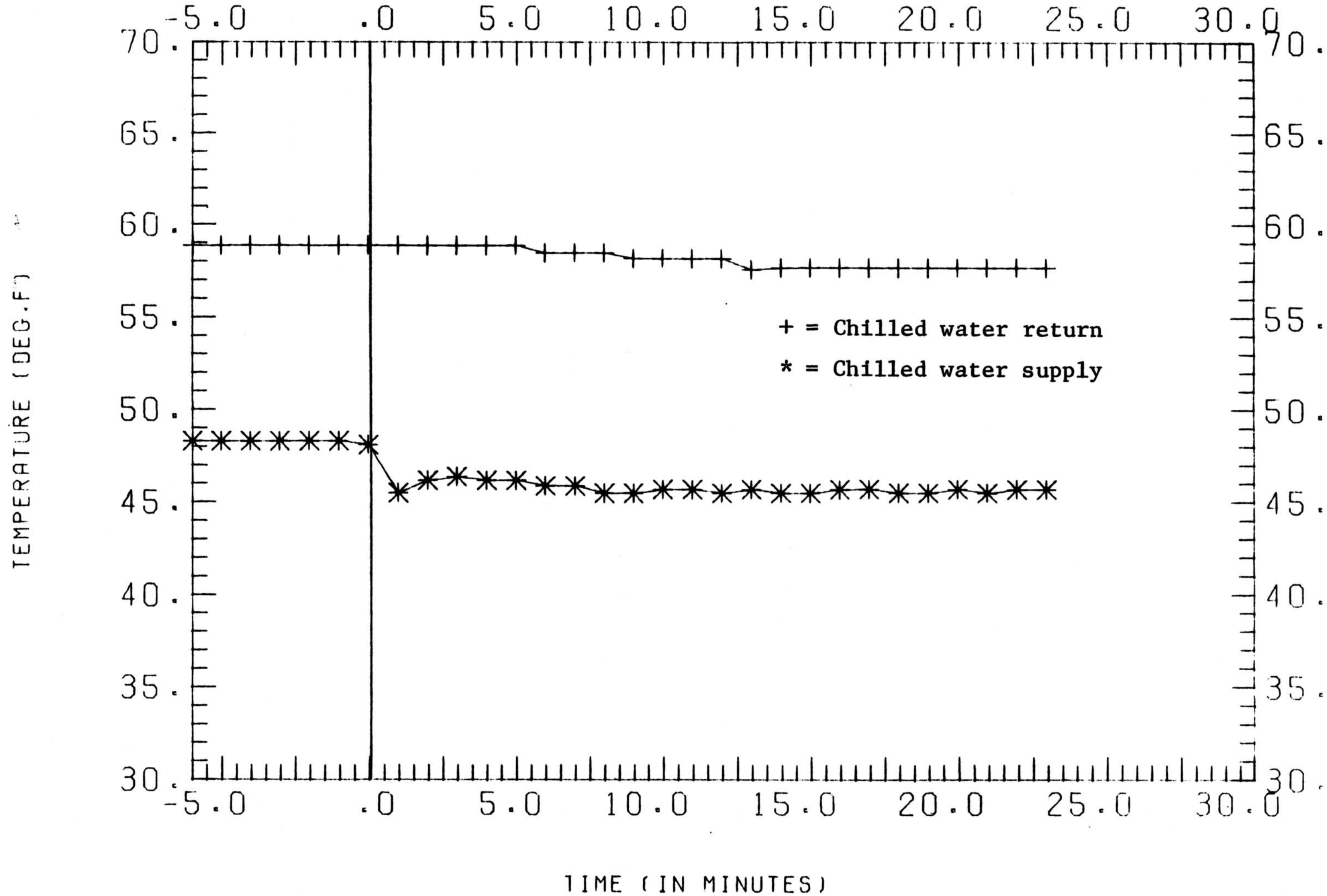

Figure 3.1 Chiller transient experiment results. Set point temperature from 48 to 46^{o} F.

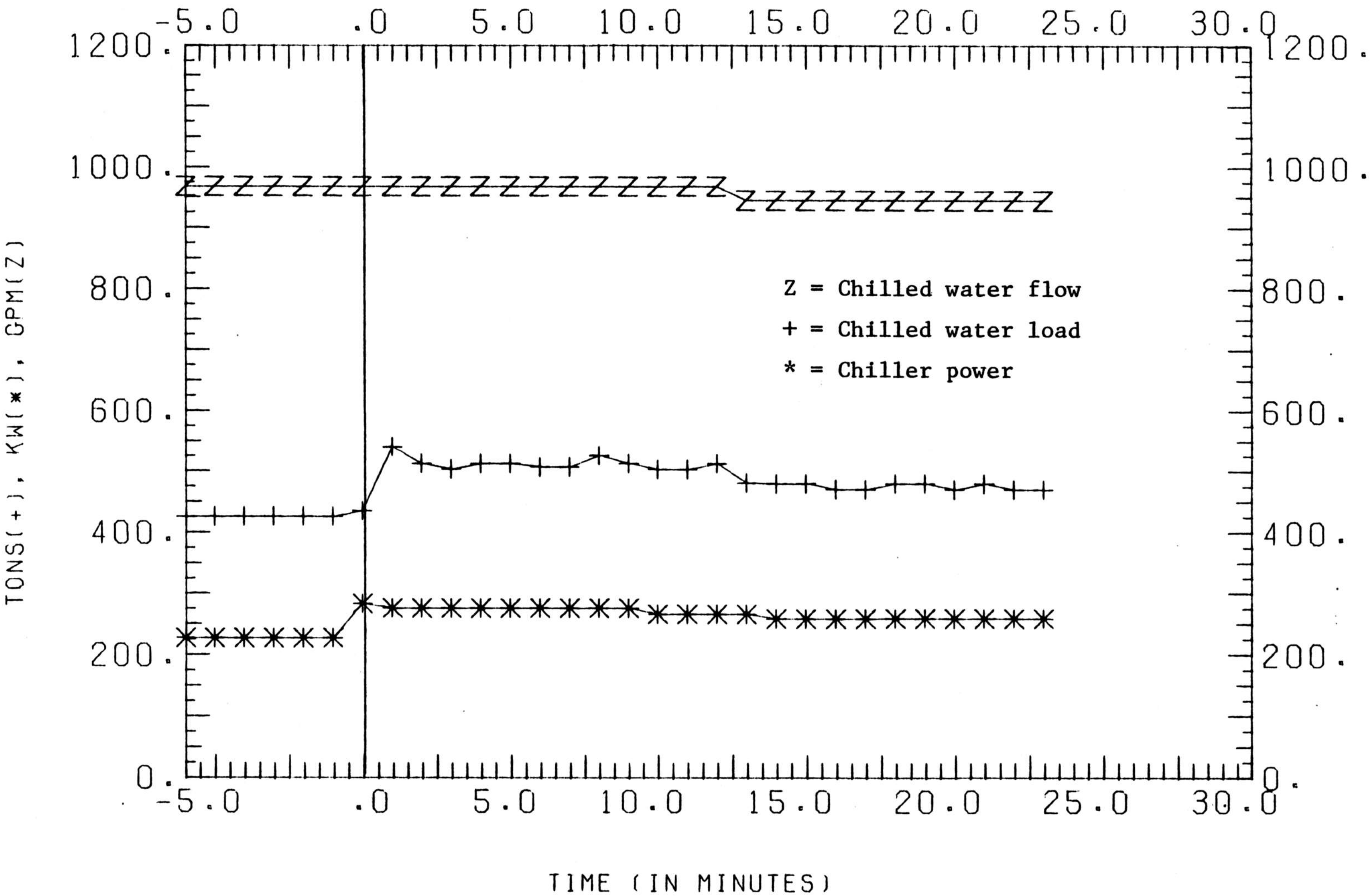

Figure 3.2 Chiller transient experiment results. Set point temperature from 48 to 46° F.

temperature begins to drop. Eventually, assuming that the building load and the chilled water flow rate remained constant, the temperature difference across the chiller at the end of the test would reach the same value as at the start of the test. The length of the flush time is on the order of five minutes.

Figures 3.3 and 3.4 illustrate the very short time duration of the chiller response. These figures use data that were collected during the entire series of response tests. In an effort to present these data on a normalized basis, the change in chiller power divided by the change in chilled water load was plotted versus time in Figure 3.3. The changes are the differences between the chiller power at the start of a test and that at points in time after the test had begun.

Figure 3.4 uses the same transient test data, but plots the measured power ratio (KWR) versus the part load ratio. The measured KWR is defined as the measured chiller power divided by the design chiller power. The design power is determined by the operating conditions of the chiller and Figure 2.1. The part load ratio is defined as the measured chilled water load (evaporator side) divided by the design load of the chiller (in this case 550 tons).

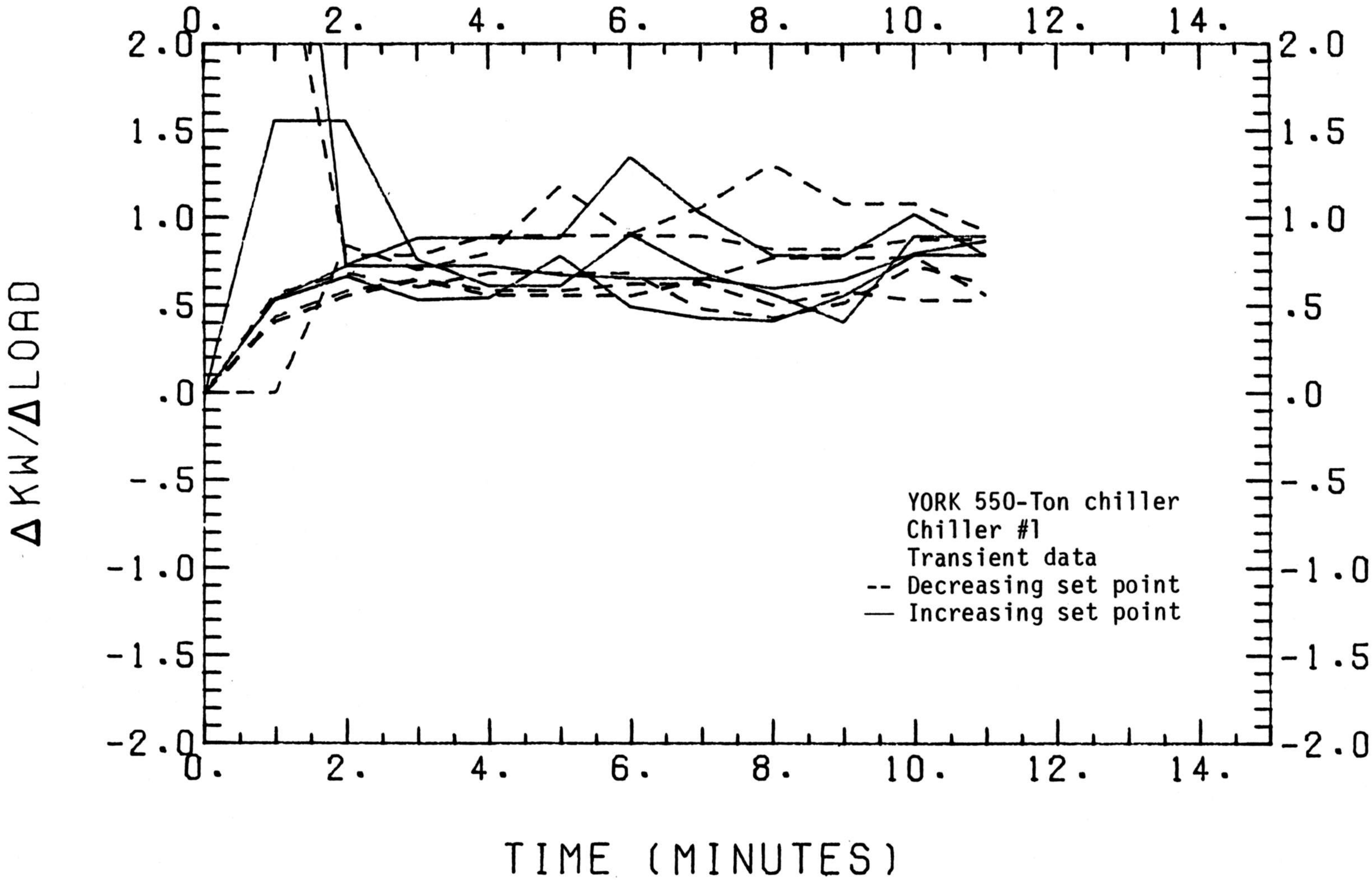

Figure 3.3 Normalized presentation of (9) independent chilled water supply set point tests.

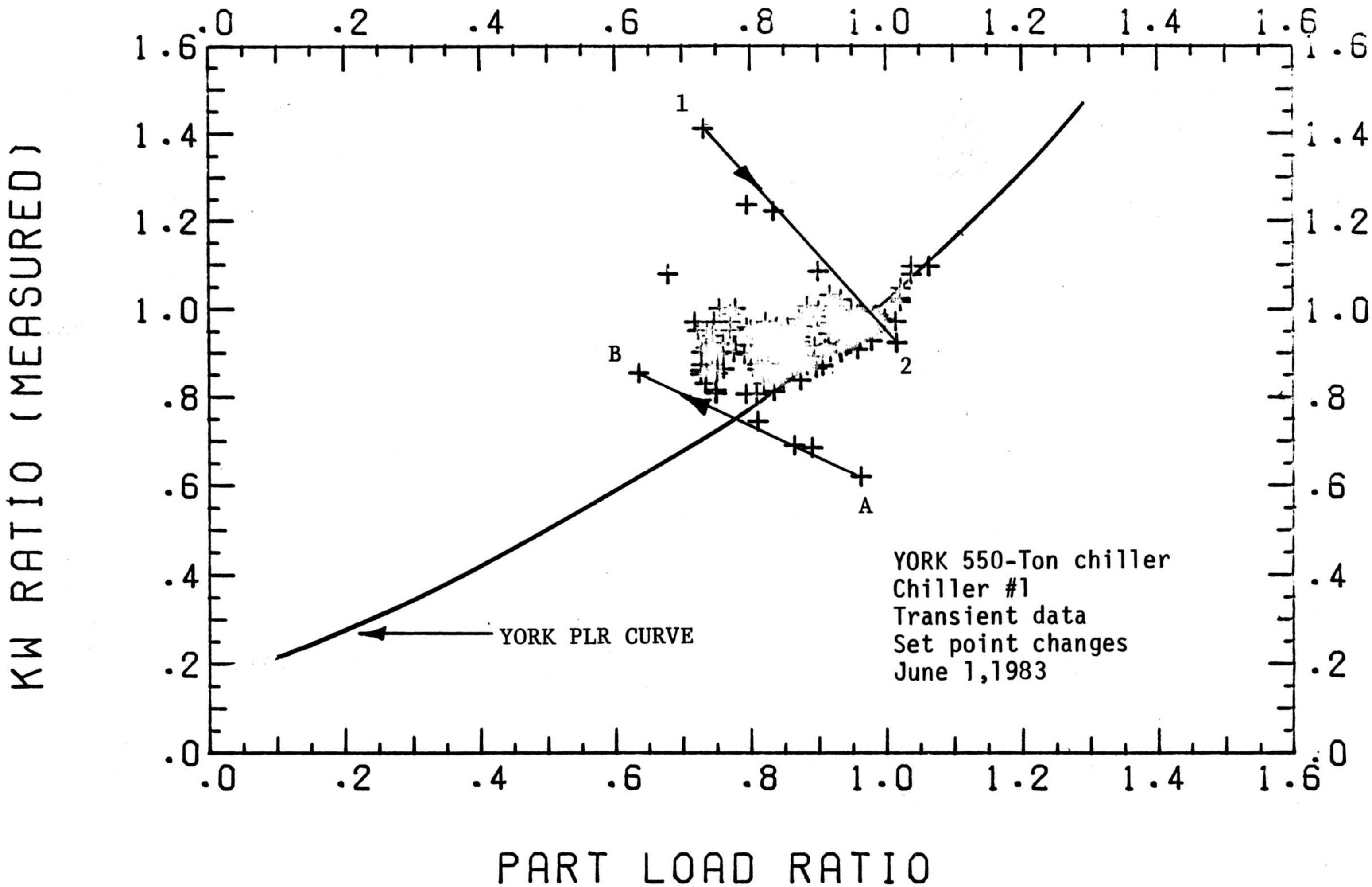

Figure 3.4 Transient chiller test data versus manufacturer PLR curve

Points 1 and A on Figure 3.4 indicate the chiller operating conditions one minute after the start of two independent tests (chilled water supply set point changes). Points 2 and B, respectively, are the conditions two minutes after the tests had begun. In both cases the three minute conditions are "lost" in the large grouping of points around the PLR curve.

The AHU response to a supply air set point change is shown in Figures 3.5 and 3.6. Figure 3.5 is an indication of the response of the chilled water supply and return temperatures to a change in the AHU supply air temperature. Figure 3.6 shows the air-side response to the set point change. Here, again, there is a rapid response (on the order of 5 minutes) to a change in the supply air set point. This transient effect was, also, felt to have an insignificant effect on energy use.

The cooling tower response to changes in the fan speeds did represent a significant transient effect. Since the transient response of the cooling tower was built in to the tower computer model the results of the cooling tower testing were presented in the EQUIPMENT MODELING section.

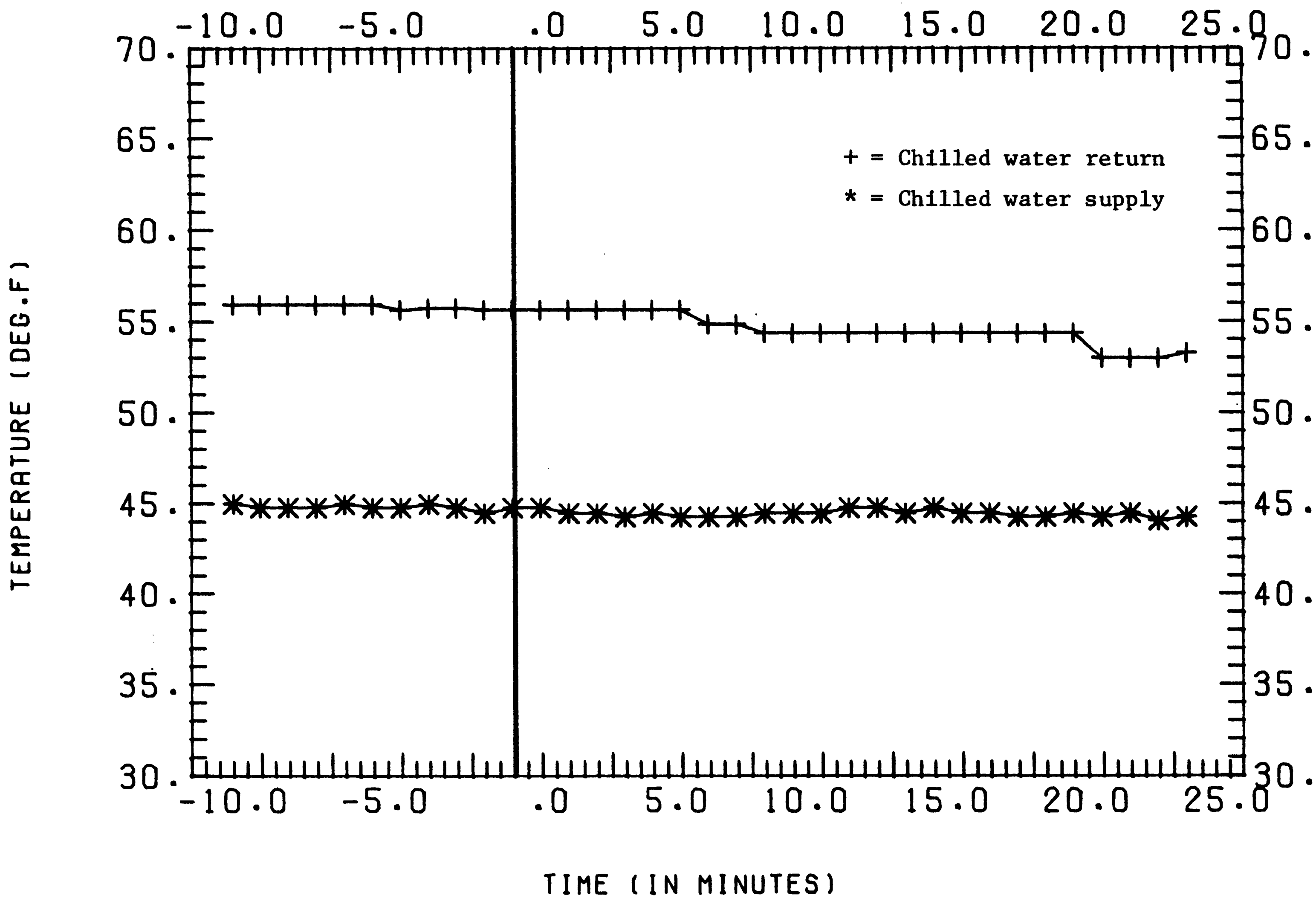

Figure 3.5 Results of a supply air temperature set point test (55° F. to 60° F.)

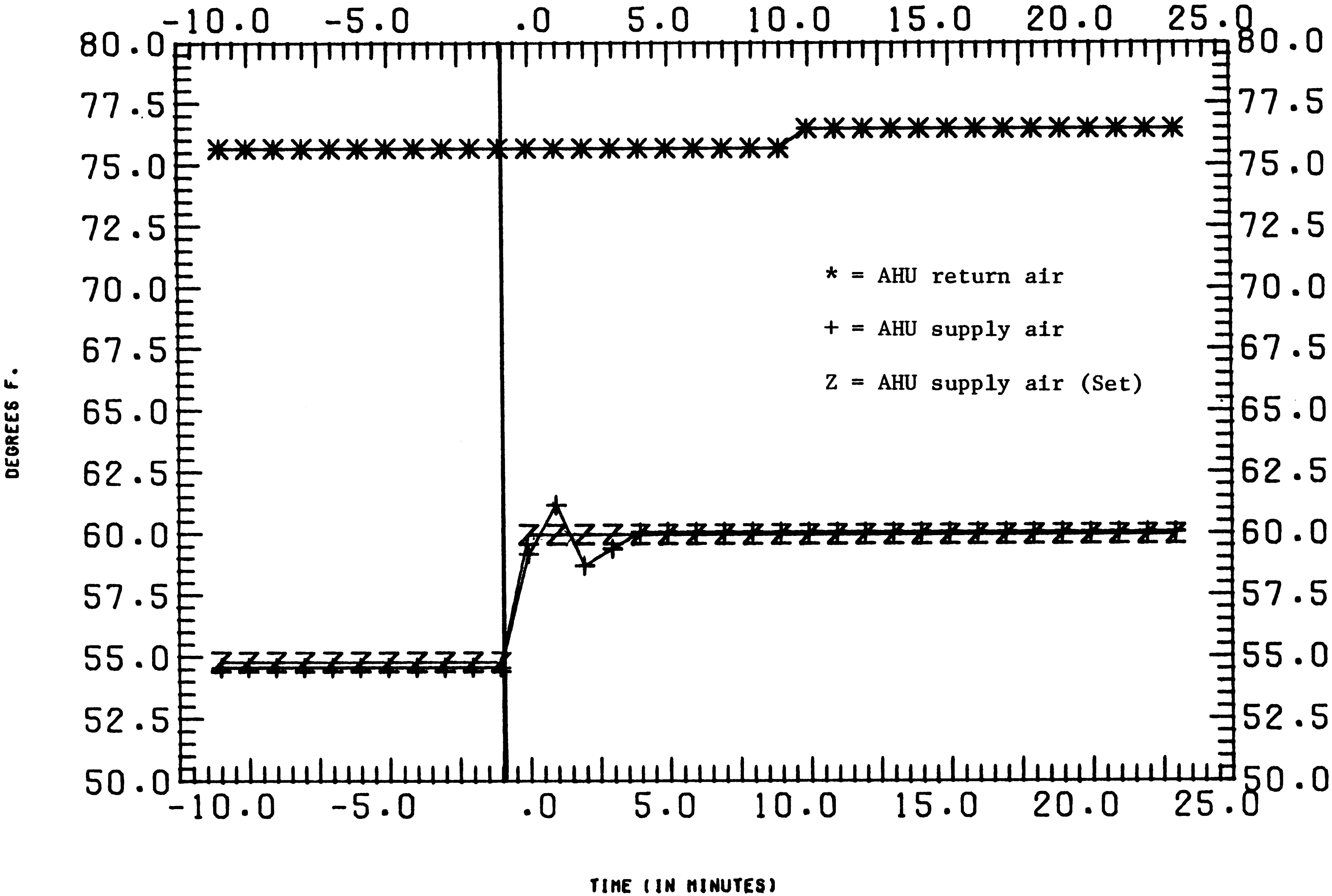

Figure 3.6 Results of a supply air temperature set point test (55^{o} F. to 60^{o} F.)

4.0 HVAC SYSTEM CONTROL STRATEGY

A building HVAC system is made up of many different components. Each component, whether it is a chiller, cooling tower, etc., has its own operating characteristics and its own internal control strategy. For an EMCS to be effective it must have information on the individual HVAC system component. Also, the interaction between the components must be known in order for the EMCS to make effective control decisions.

The following is the operating scenario for the HVAC system shown in Figure 4.1. At Time=0 the system is in steady-state operation. The following operating conditions have been set: 1) the chilled water supply set point; 2) the cooling tower fan speed; and 3) the AHU supply air temperature set point. In order to explain the operation, the following conditions hold:

1) The terminal unit supply air dampers are positioned within their throttling range, i.e., they are not at their full open or closed position.
2) The chilled water supply valves on the cooling coils are within their throttling range.

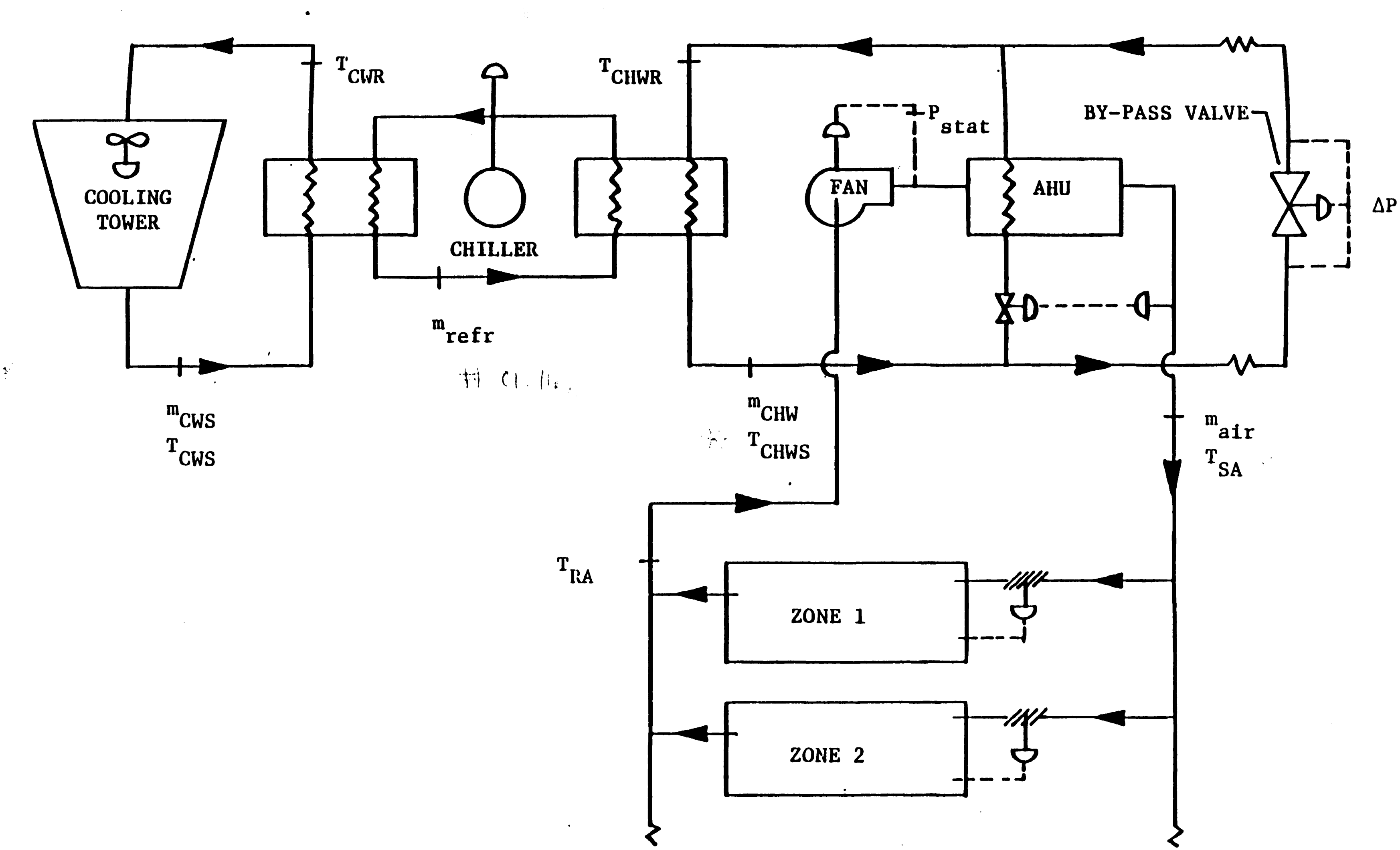

Figure 4.1 HVAC system schematic including the control points

3) The AHU fan is a VAV (Variable Air Volume) type of fan which is not operating at its maximum flow capacity. This is found in the central zones.

4) The internal control strategy of the AHU fan at this IBM facility is such that the fan seeks to maintain a set value for the static pressure in the supply air duct.

5) The condenser water and chilled water pumps are constant speed pumps.

6) The chiller is one of the 550-ton centrifugal chillers that use pre-rotation vanes for capacity control.

At Time equal 0+ the thermostat setting for the zone is reduced. This results in the following sequence of events.

1) The terminal unit supply air dampers in the zone begin to open further in response to the thermostat signal. This increases the flow of supply air.

2) The opening up of the supply air dampers causes the static pressure in the supply duct to drop. In response to the loss of static pressure, the pitch of the VAV fan blades is internally changed to bring the pressure back

to the set point value. The increase in static pressure produces a corresponding increase in the flow of air supplied by the fan and an increase in the fan horsepower.

3) The increased flow of air across the cooling coil causes the supply air temperature to rise. In response to this temperature rise a control signal is sent to the chilled water valve on the cooling coil to begin opening up.

4) The increase in the heat transferred at the cooling coils increases the chilled water return temperature, and a corresponding rise in the chilled water supply temperature. To maintain the chilled water supply set temperature at the increased water flow an increase in the refrigerant flow rate is needed. The pre-rotation vanes on the inlet to the chiller compressor begin to open up, increasing the refrigerant flow, and thus, increase the chiller capacity.

5) The increase in chiller load and power consumption increases the heat transfer in the condenser water load. The temperature of the water leaving the cooling tower begins to rise.

6) The cooling tower will now respond in one of two ways. The previous cooling tower fan

status could remain fixed. In this case the condenser water supply temperature would seek a new, higher equilibrium temperature. If the fan status changes to a higher level due to the increased condenser water temperature a new sequence of dynamic events occur.

There is an analogy between the preceding sequence of events and the propogation of a wave through a medium. The initial change in the zone thermostat setting has caused a succession of operating changes to occur in all of the HVAC system components. Now that the "wave" of disruption has traveled the length of the HVAC system it could be "reflected" back through the system. In the situation described previously, the increased cooling tower supply water temperature would affect the operating conditions of other system components. The relative magnitude of the disruption wave is greatly influenced by the damping characteristics of the HVAC system and of the individual HVAC component. The stability of a particular HVAC system is a measure of the ability of the internal control components on the equipment to minimize the effects of any disturbance on the system.

4.1 CURRENT HVAC EQUIPMENT CONTROL STRATEGY

The EMCS controls the operating status of all of the major components that comprise the HVAC system. The EMCS controls the operation of the main AHU's in three aspects. These are the on-off status of the units, the setpoint for the supply air temperature, and the settings for the return and outside air dampers. Currently, the main AHU's are brought on-line at 7:30 A.M. and are shut down at 6:00 P.M. on a normal working day. On weekends and holidays the main AHU's are usually in the off condition. The setpoint for the supply air temperature is a function of the outside air temperature. As shown in Figure 4.2, at ambient temperatures below 10^{o} F. the setpoint is constant at 68^{o} F. and at temperatures above 55^{o} F. the setpoint is constant at 55^{o} F. The choice of the 55^{o} F. limit on the supply air temperature was based in part on the concept of providing thermal comfort for the occupants. It was found that 55^{o} F. was the lowest supply temperature that could be used before people began complaining about cold drafts to the building engineers.

The position of the return and outside air dampers for the main AHU's is based on the enthalpies of the return and outside air. This "enthalpy" control allows

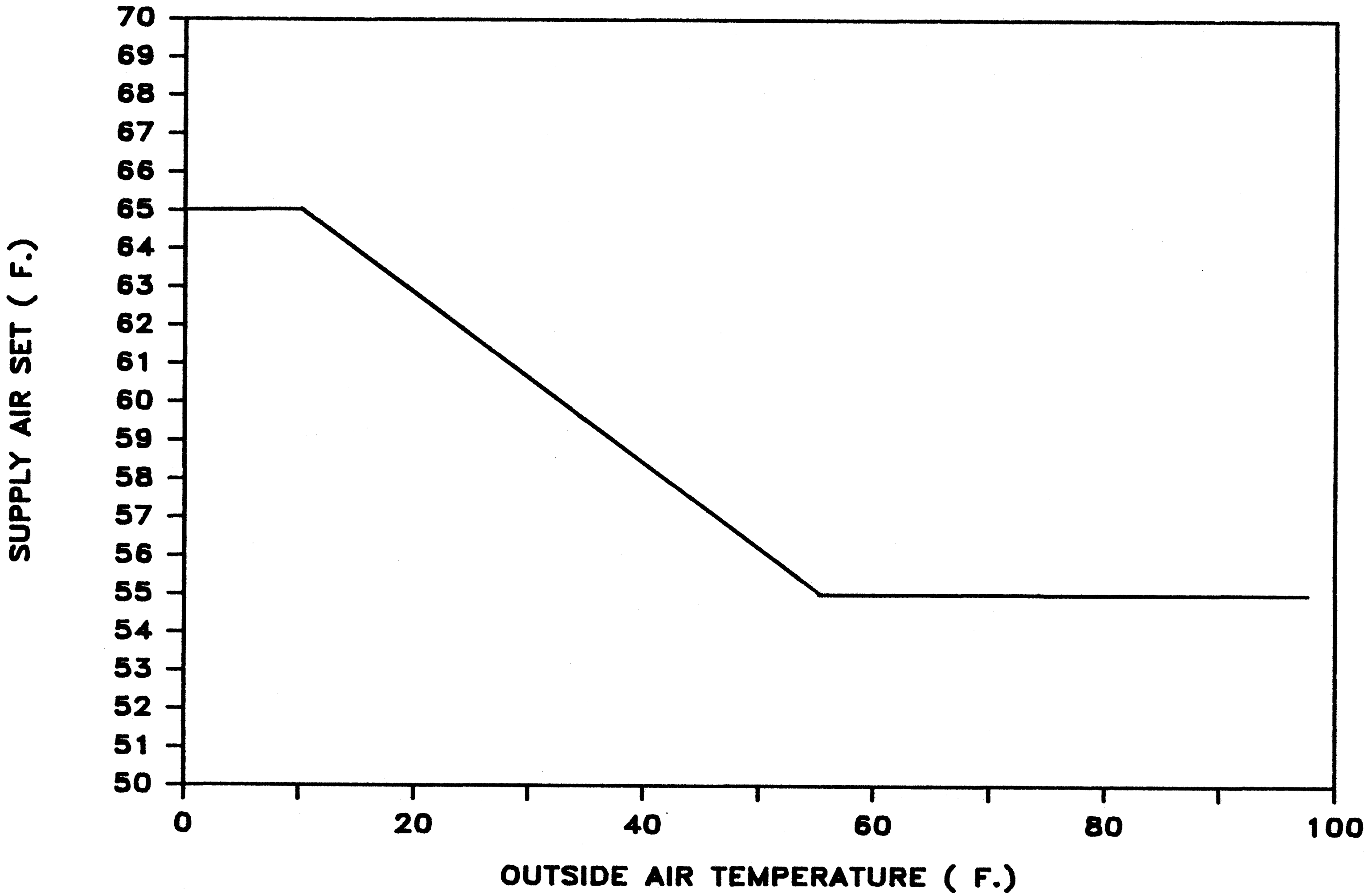

Figure 4.2 Relationship between the supply air set temperature and the ambient air temperature

the AHU to operate in either a "free cool" mode or a "pay cool" mode as shown in Figure 4.3. In the free cool mode a combination of outside air and return air is used to supply ventilation air at the supply air temperature setting. The pay cool mode uses either all outside air or a combination of return air and minimum outside air. Additional cooling in the pay cool mode is provided by the building chilled water system. The point to switch from minimum outside air to all outside air can vary depending on the enthalpies of the return and ambient air.

The EMCS also controls the chilled water supply set point temperature. At present, the EMCS uses the chilled water return temperature to determine whether a change in set temperature is desired. Figure 4.4 shows the control strategy that has been programmed into the control system. The decision to key the change in set temperature on the chilled water return temperature was made by the building operations personnel. It was felt that the return water temperature would provide a stable base on which to make control point changes. The choice of 58^{o} F. as the base-line was the result of experiments done by the operators to determine the "best" temperature upon which to key set point changes.

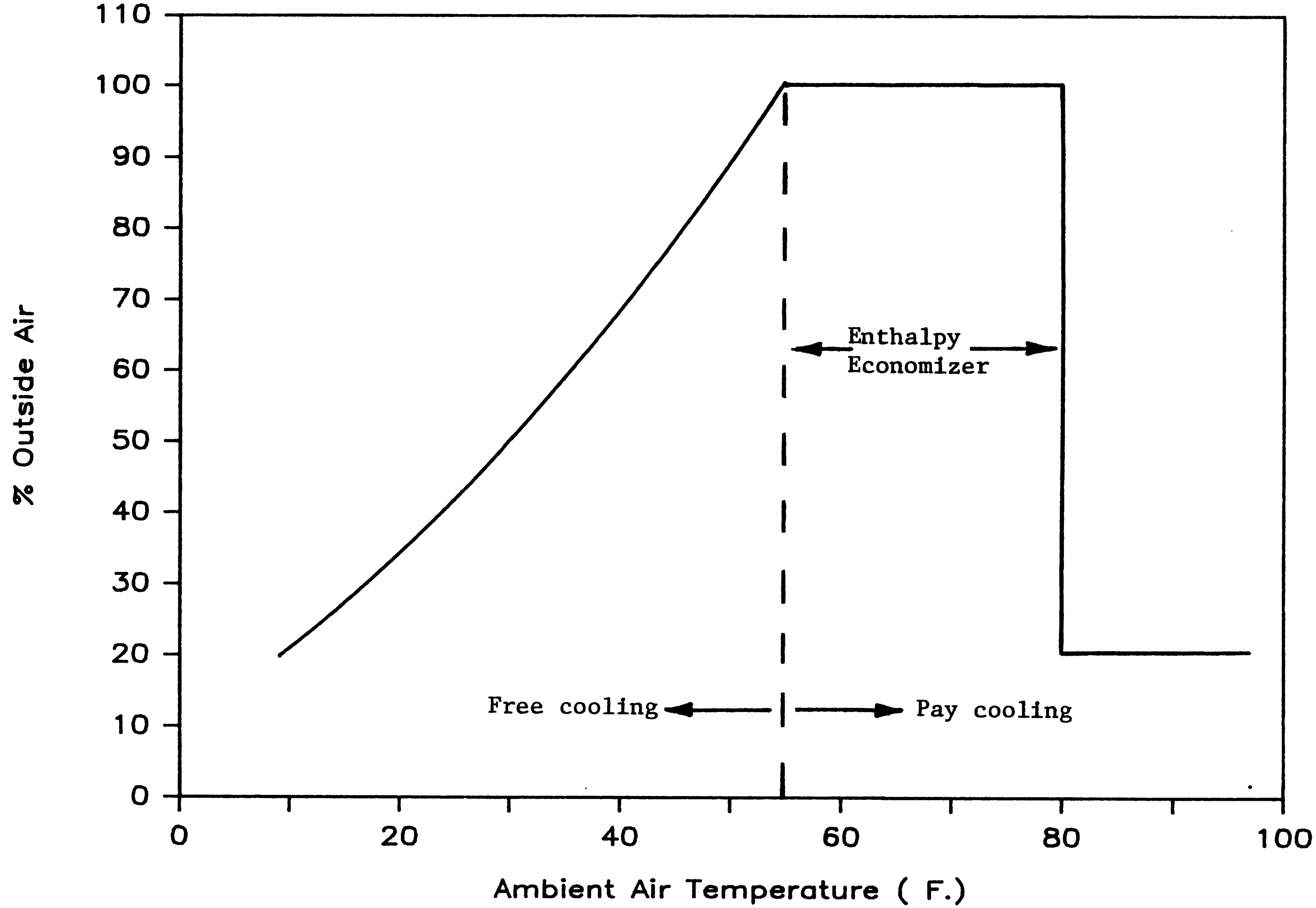

Figure 4.3 Relationship between the outside air fraction and the ambient air temperature (Economizer mode)

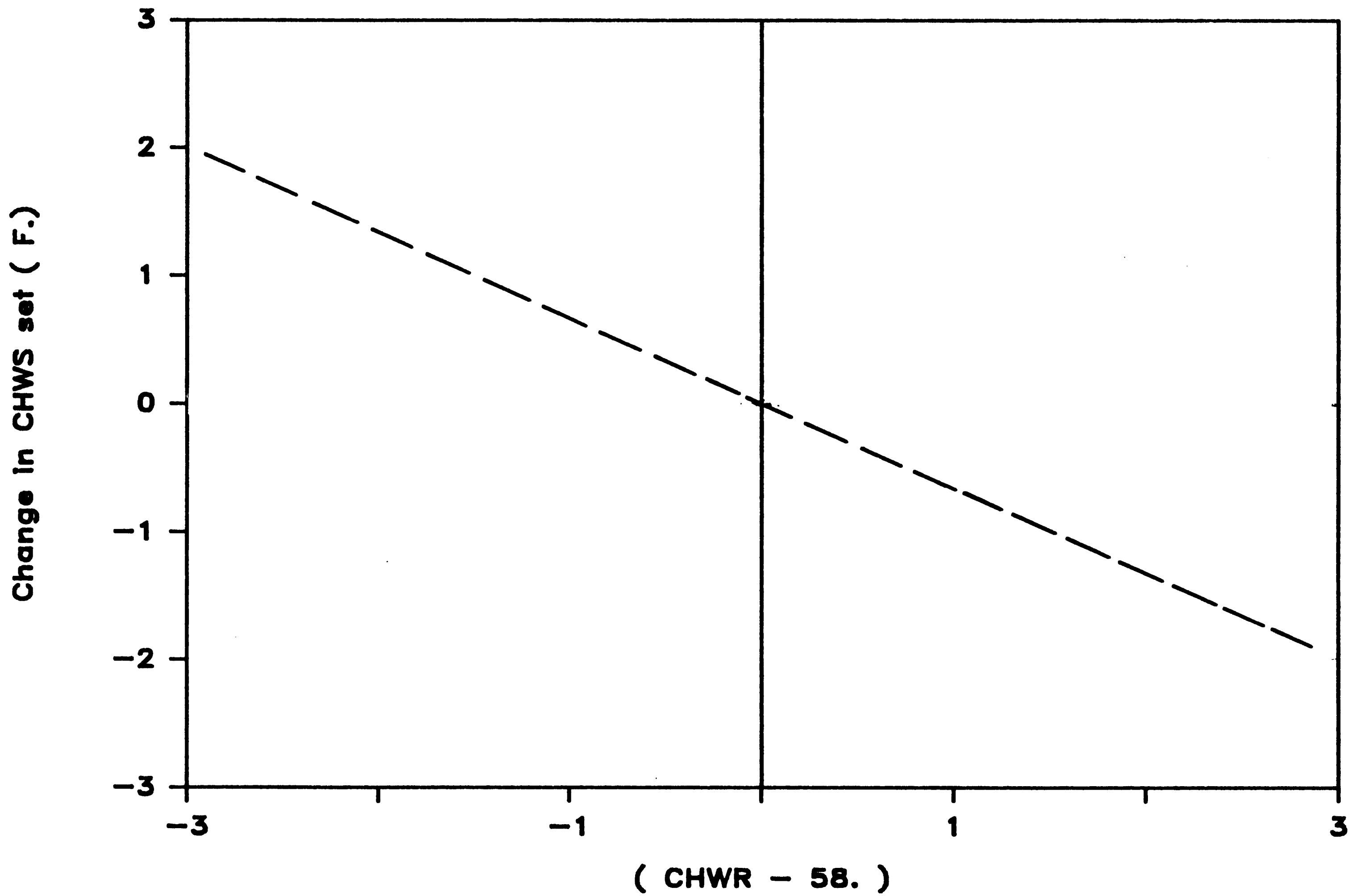

Figure 4.4 Relationship between the chilled water return temperature's deviation from a desired set point and the change in the chilled water supply set point temperature

The operating status of the cooling tower fans is also controlled by the EMCS. The method that is now used to control the fans is termed approach control. The "approach temperature" is the temperature difference between the ambient wet bulb temperature and the condenser water supply set point. The condenser water set temperature is set based on the ambient wet bulb temperature as shown in Figure 4.5. Once the approach temperature has been fixed the status of the cooling tower fans varies according to the deviation between the actual condenser water supply and the set point temperatures.

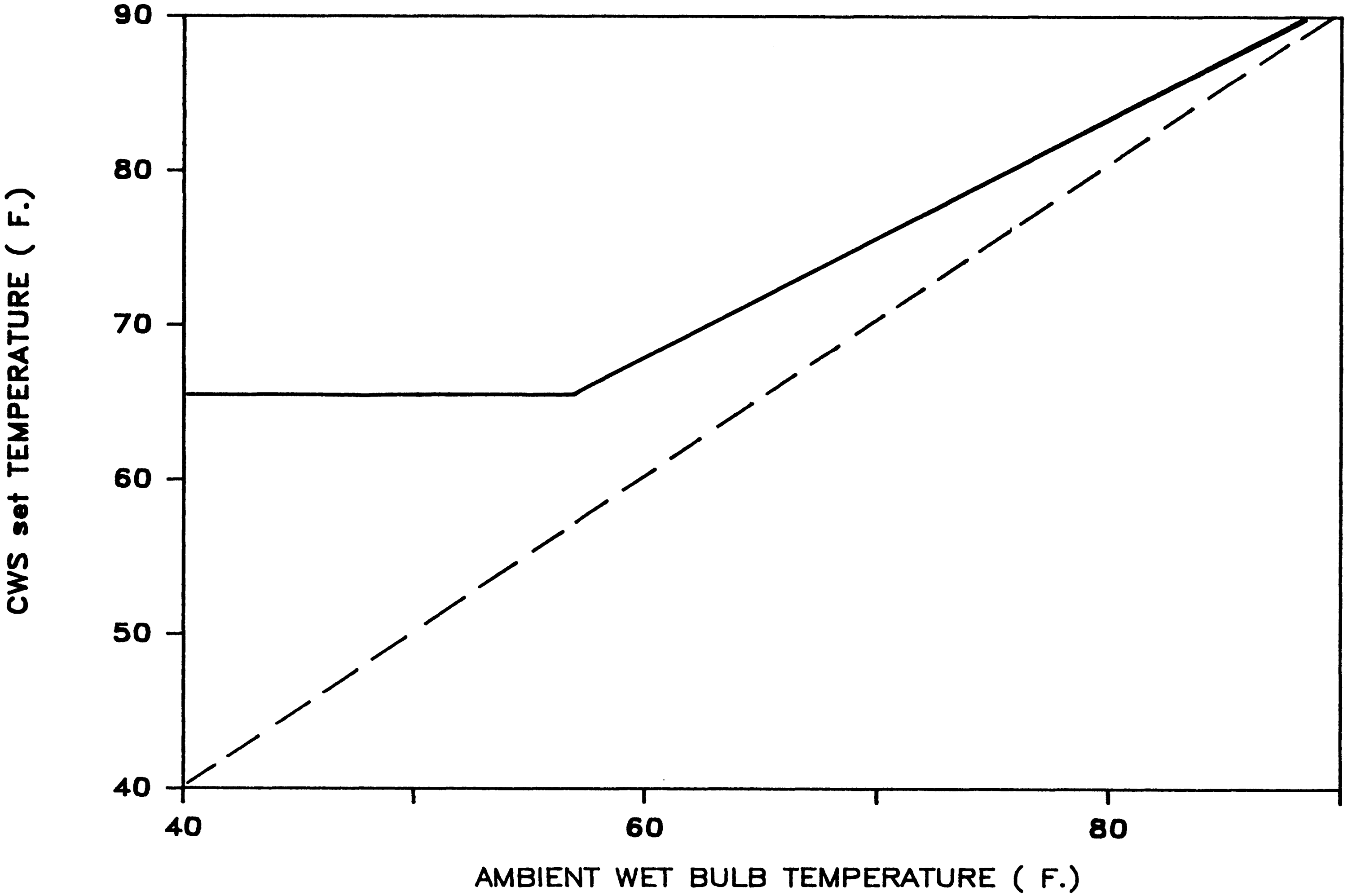

Figure 4.5 Relationship between the ambient wet bulb temperature and the condenser water supply set point temperature

5.0 SYSTEM SIMULATION AND MODEL VERIFICATION

TRNSYS (Klein, 1981) is a component based simulation program that was developed at the University of Wisconsin-Madison Solar Energy Laboratory. TRNSYS is a modular type of program. This means that a system can be modeled by connecting the appropriate system components into various operating configurations. Since the TRNSYS program was readily available, the computer models for the HVAC components were written to be compatible with the TRNSYS routine. Details and listings of the developed models can be found in Appendix B.

Using the developed models a TRNSYS simulation deck was written to verify the chiller-cooling tower subsystem. The operating parameters for the system were determined from previously collected steady-state data. The simulations were driven by the measured chilled water load and the ambient wet bulb temperature. Figures 5.1 through 5.6 show a sequence of three days in November for which the computer simulation was run. As shown by the figures, the agreement between the predicted and the measured values of the condenser water supply temperature and the chiller power consumption are very good.

The results of this simulation are only as accurate as the measured data which were used to drive the simu-

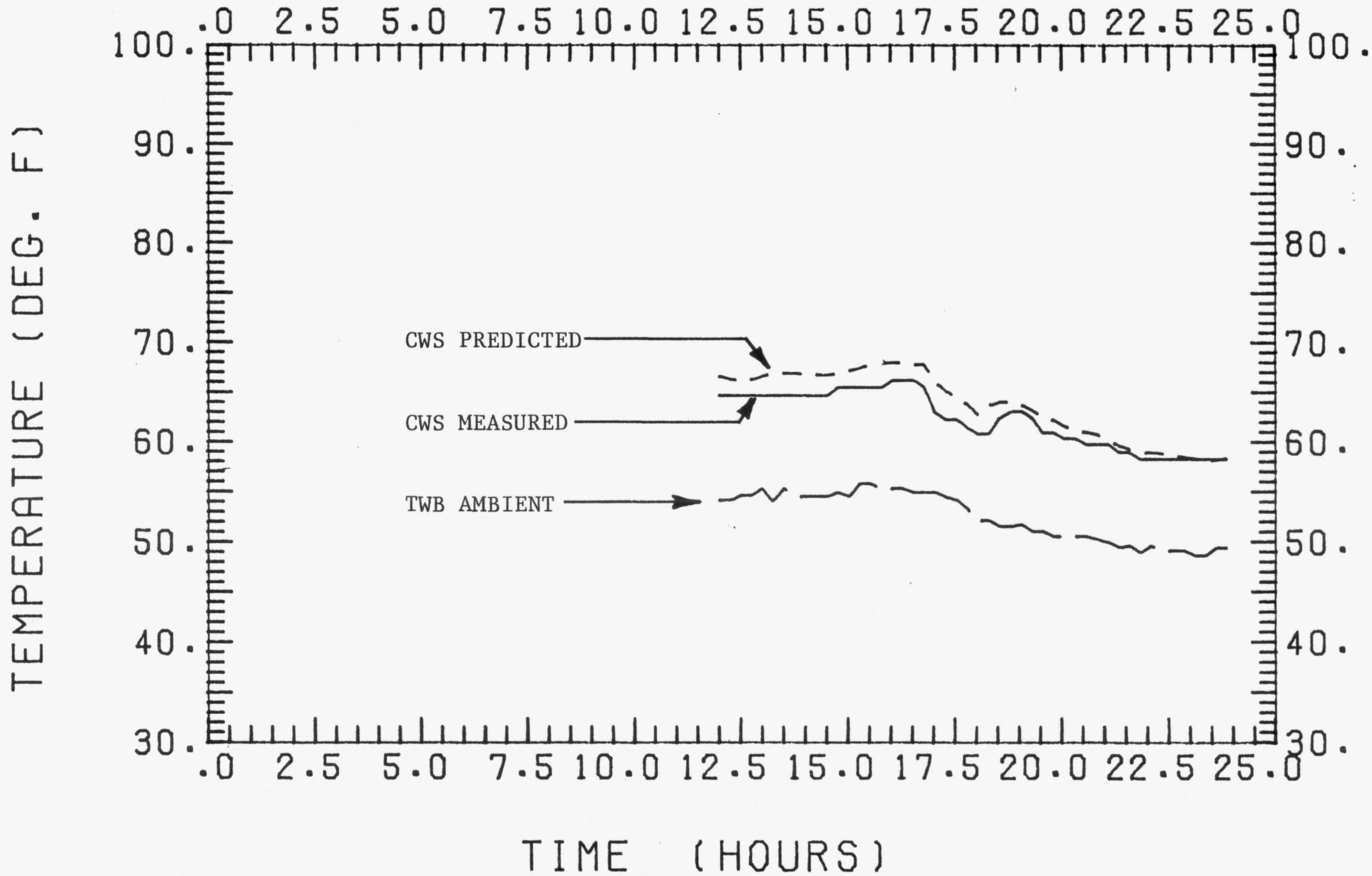

Figure 5.1 Condenser water supply temperature. Chiller-cooling tower subsystem model verification. Data were collected: November 8, 1983

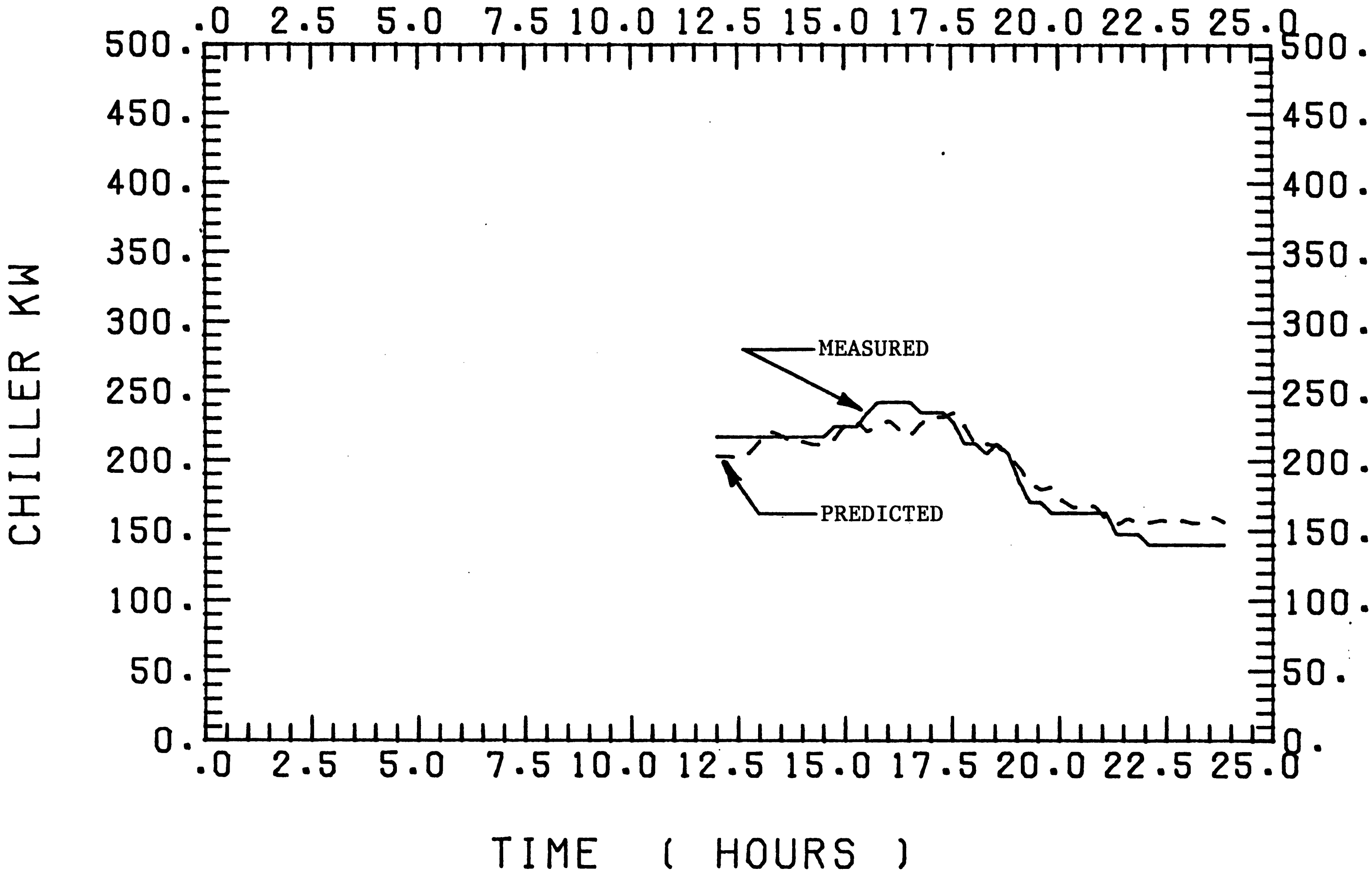

Figure 5.2 Chiller power consumption. Chiller-cooling tower subsystem model verification. Data were collected: November 8, 1983

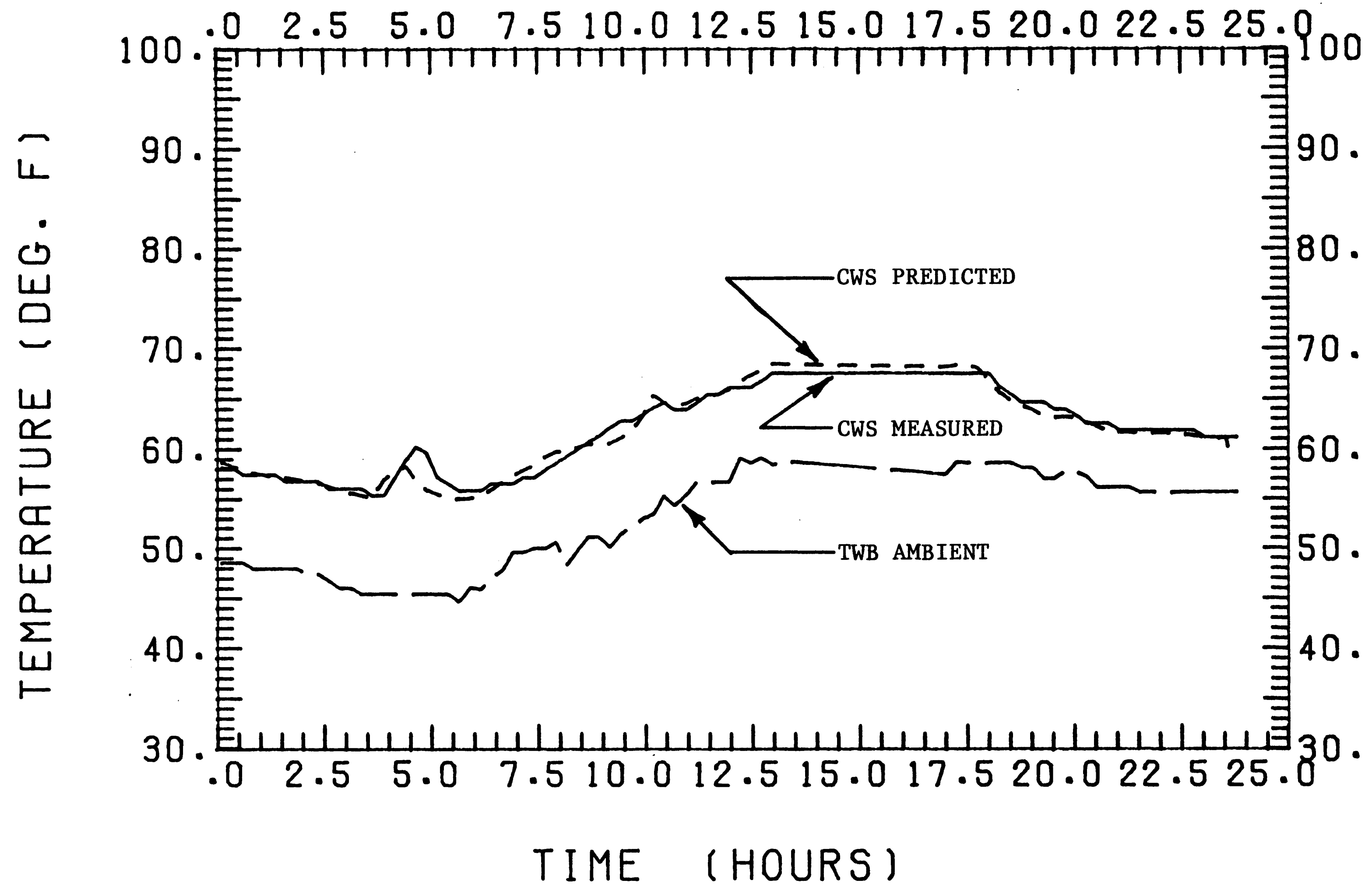

Figure 5.3 Condenser water supply temperature. Chiller-cooling tower subsystem model verification. Data were collected: November 11, 1983

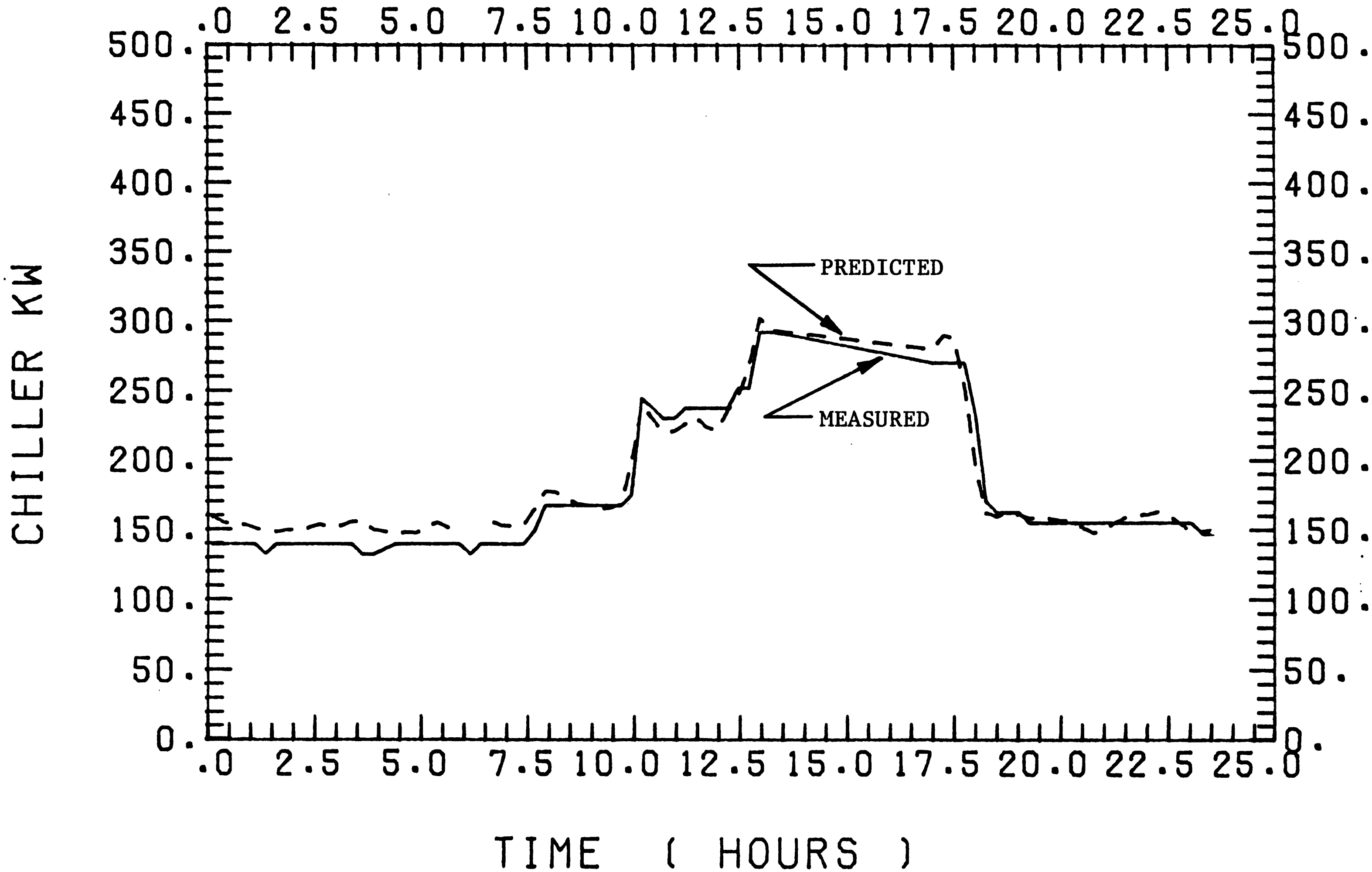

Figure 5.4 Chiller power consumption. Chiller-cooling tower subsystem model verification. Data were collected: November 9, 1983

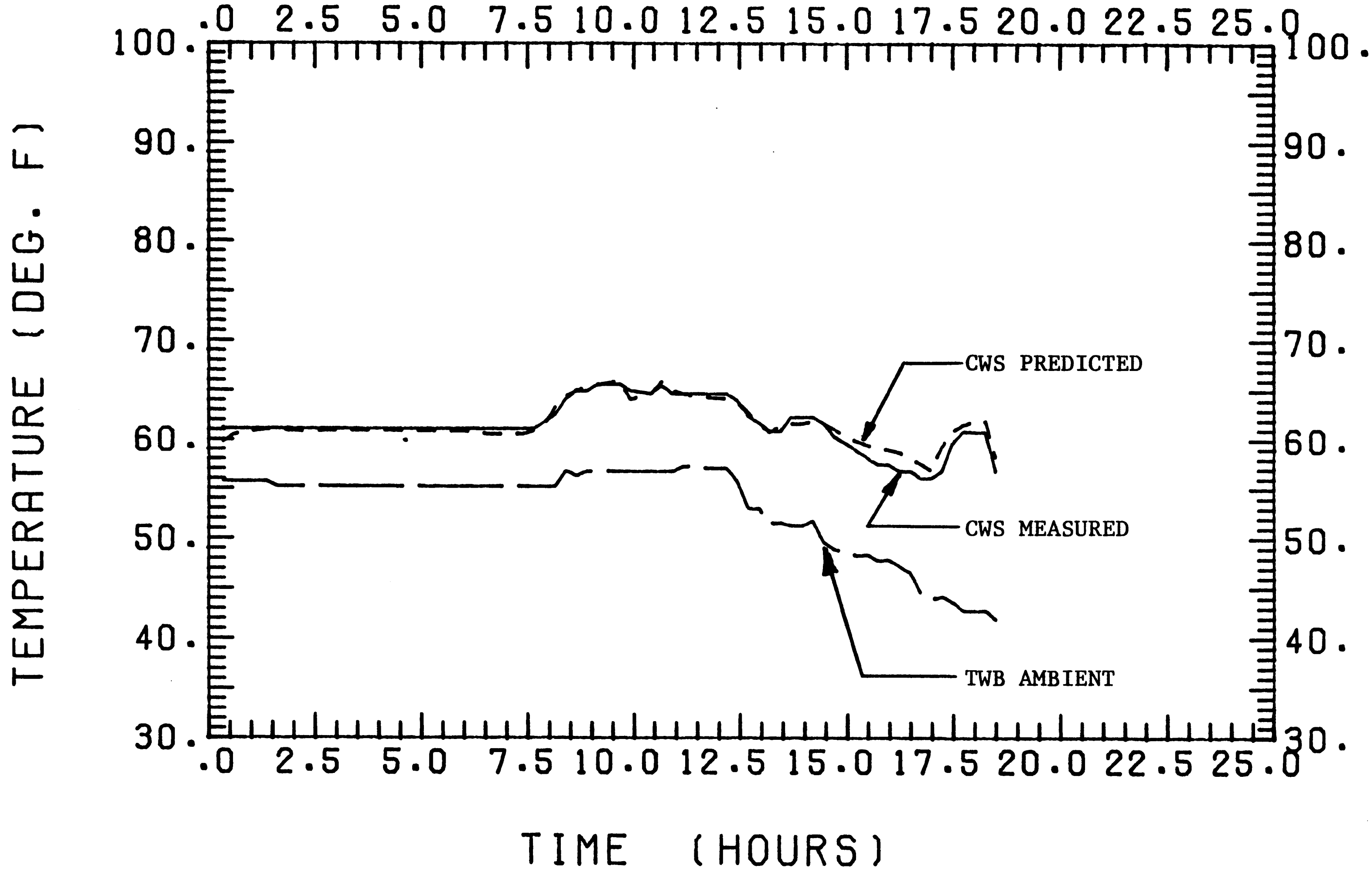

Figure 5.5 Condenser water supply temperature. Chiller-cooling tower subsystem model verification. Data were collected: November 10, 1983

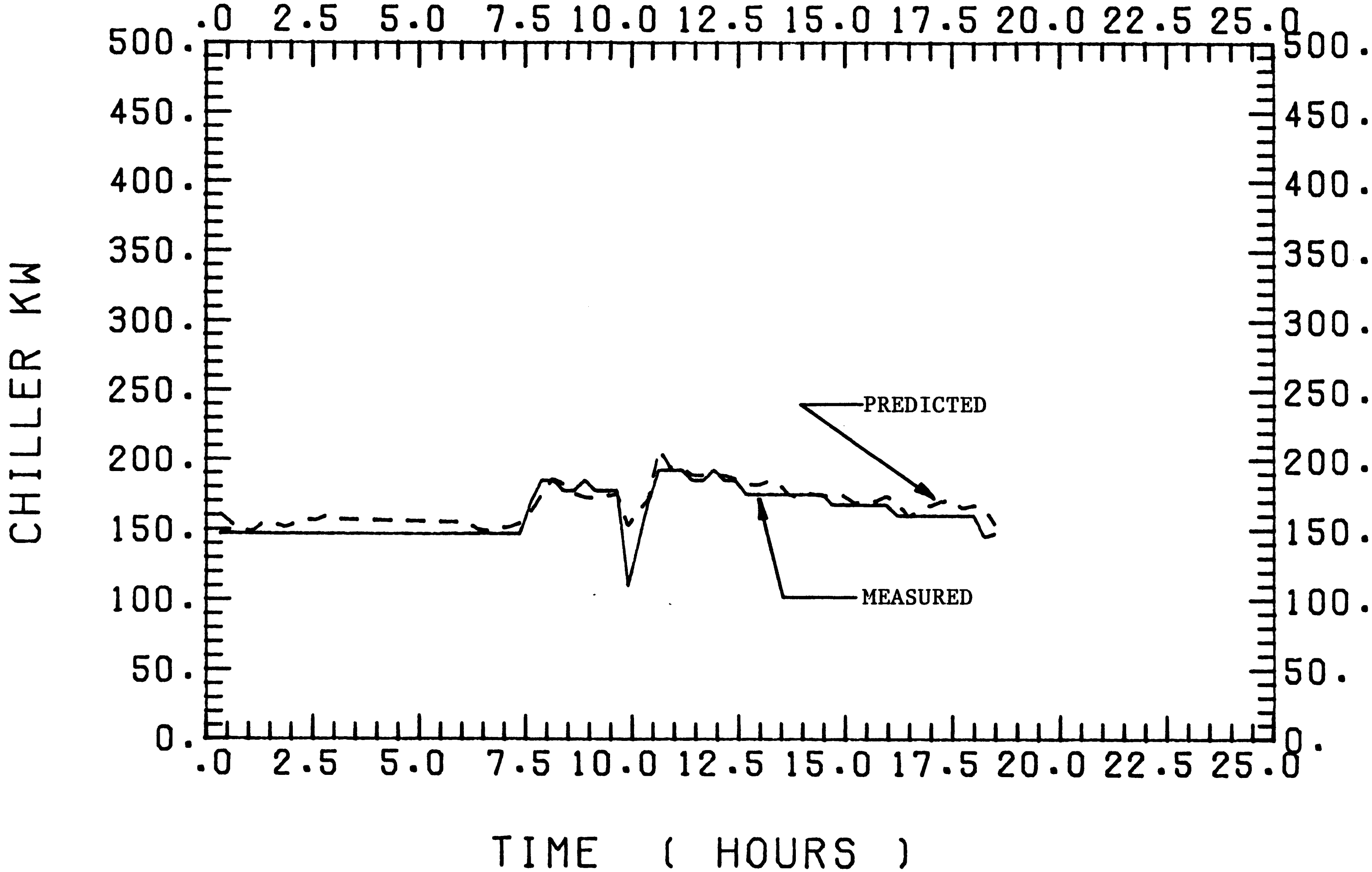

Figure 5.6 Chiller power consumption. Chiller-cooling tower subsystem model verification. Data were collected: November 10, 1983

lation. In this particular case those measured data were the actual chilled water load and the ambient wet bulb temperature.

Another TRNSYS simulation deck was written to verify the performance of the entire HVAC computer model, including a model generated building cooling load. The only inputs used to drive this simulation were the ambient wet bulb and dry bulb temperatures and horizontal solar radiation measurements. Due to the lack of continuous and consistent data from the Atlanta building an extensive performance evaluation of the computer model was not possible. However, one of the days of collected data, March 8, 1983, was simulated using the TRNSYS simulation. See Figure 5.7 through 5.10.

The simulation does reasonably well in predicting the performance of the major HVAC system components. Figure 5.7 is a plot of the predicted and measured chilled water loads. The data are within ± 15 percent for most of the day. Figure 5.8 shows the condenser water supply measured temperature versus the predicted temperature. The difference between these two values lies between ± 1.5 degrees F. for most of the time. Figure 5.9 is a plot of the measured versus the predicted AHU supply fan power consumption. Figure 5.10 shows the predicted versus measured chiller power consumption.

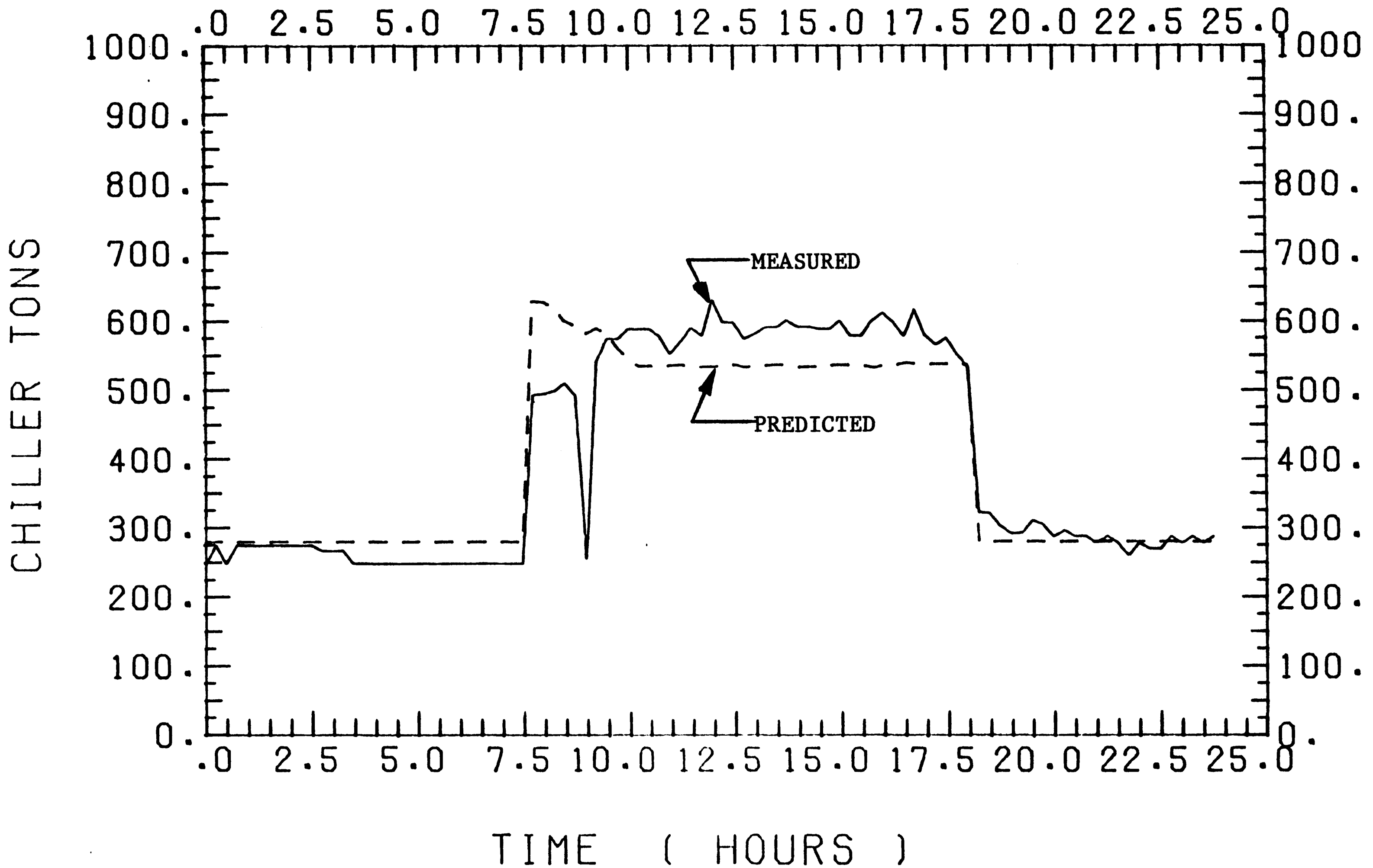

Figure 5.7 Measured versus predicted chilled water load.
Data were collected: March 8, 1983

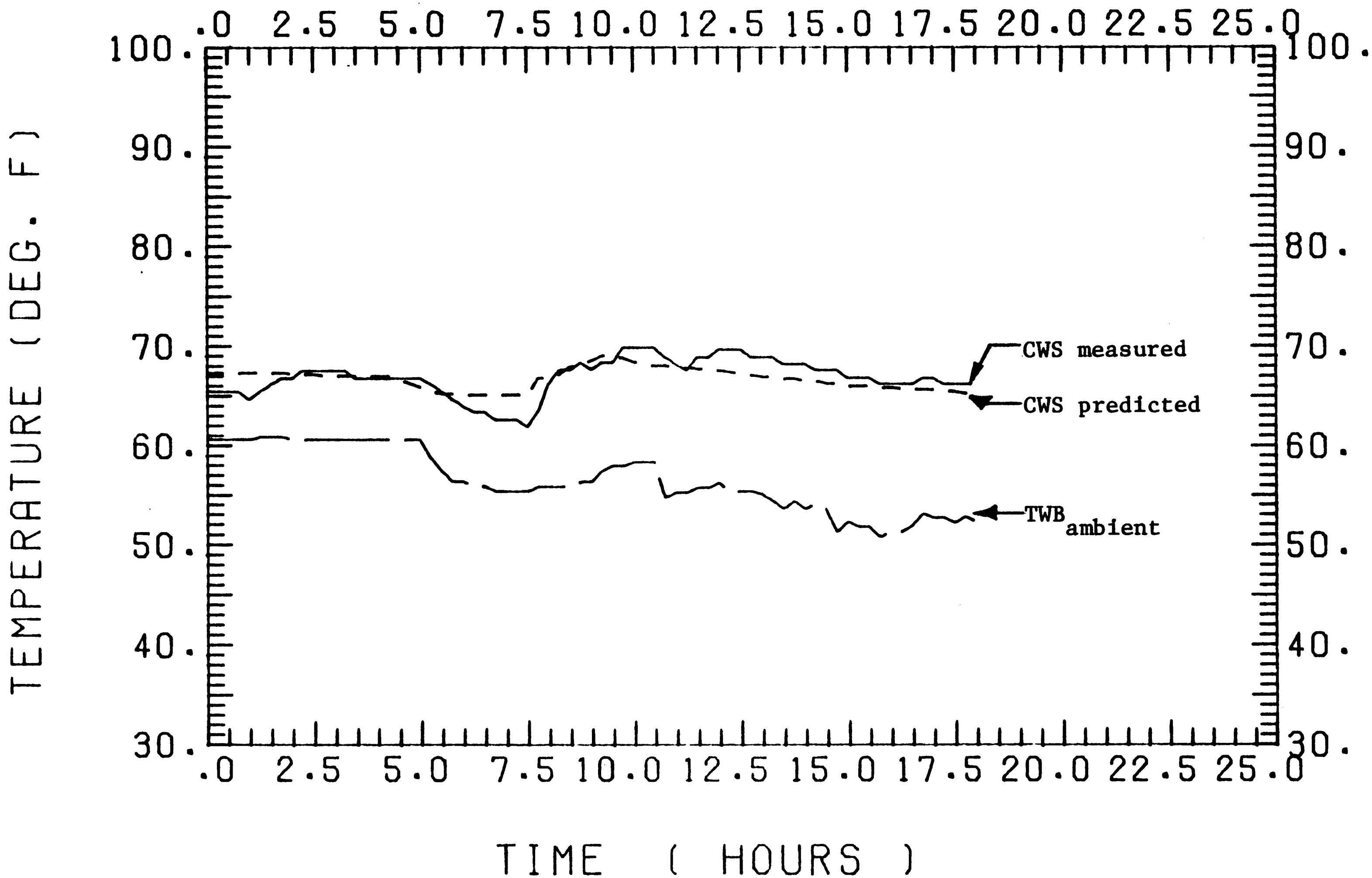

Figure 5.8 Measured versus predicted condenser water supply temperature. Ambient wet bulb temperature is also shown. Data were collected: March 8, 1983

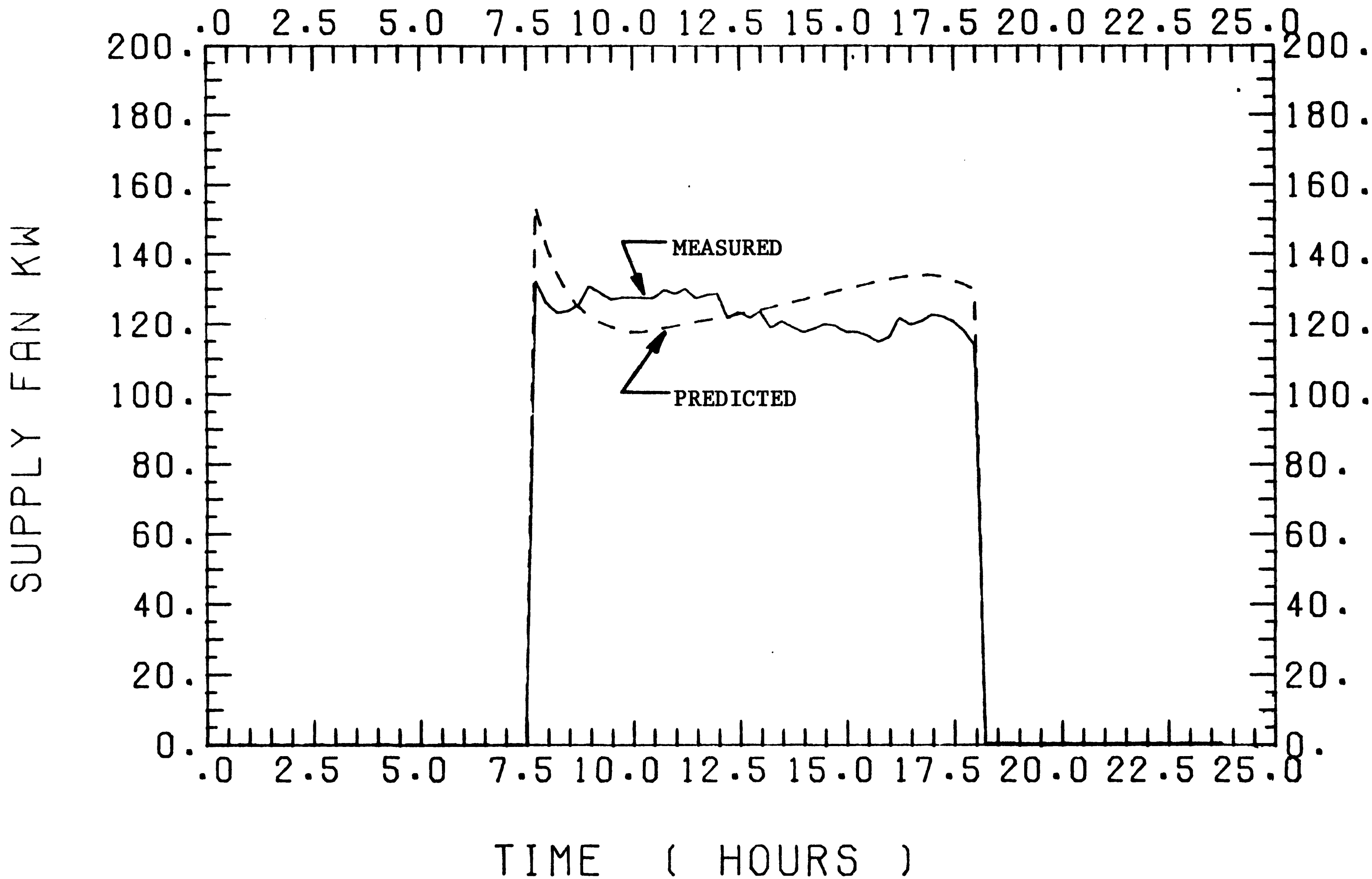

Figure 5.9 Measured versus predicted AHU supply fan power.
Data were collected: March 8. 1983

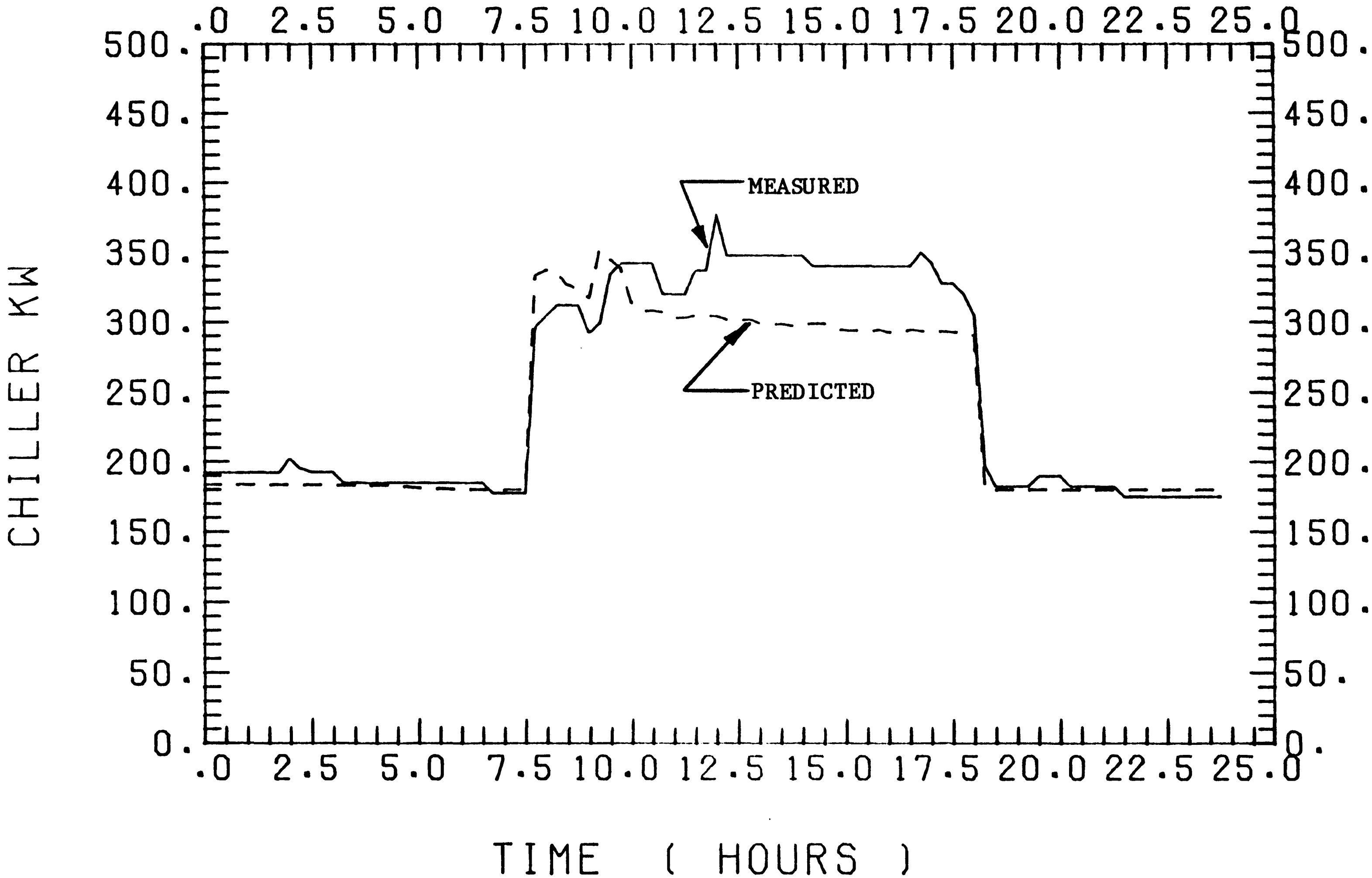

Figure 5.10 Measured versus predicted chiller power consumption.
Data were collected: March 8, 1983

The agreement between the measured and predicted values is within $\pm$ 15 percent.

Notice the large drop occurring in the chiller load (Figure 5.7) at 9:00 a.m. This drop was the result of operators switching from Chiller #1 to Chiller #2. Apparently, mechanical problems with the first chiller forced the operators to switch to the other chiller. Part of the reason for the discrepancy between the predicted and measured values can be attributed to the problems that Chiller #1 was experiencing, as well as, the subsequent switch to Chiller #2. Other reasons for the discrepancy might be the input parameters used to generate the building cooling load were incorrect or the possibility of measurement errors in the temperature and flow rate instrumentation. Since the model would ultimately be used to determine the differences in energy consumption between operating strategies it was felt that the load model of the simulation produced reasonable values for the chilled water load.

6.0 ALTERNATIVE CONTROL STRATEGIES

One of the major goals of this project was to determine and evaluate HVAC system control strategies which would potentially reduce the system energy consumption. Reducing the energy consumption of an HVAC system is not enough, though. The question of occupant comfort must also be addressed during the evaluation of the different control strategies. Therefore, a tradeoff exists between reduced energy consumption and the maintaining of occupant comfort levels.

This chapter describes alternative control strategies which were developed using TRNSYS simulation results. The refined computer models of the HVAC equipment were used in the development of the TRNSYS simulation deck. A "realistic" approach was taken towards the development of the new strategies. The control strategies have included the effect of the limitations imposed on the HVAC system due to:

1) characteristics of the individual HVAC components;
2) occupant comfort levels.

6.1 CHILLER OPERATING STATUS

One control strategy that was investigated using the equipment models and the simulation deck was the determination of the optimal point to switch from one chiller operation to two chiller operation. In Figure 6.1 the coefficient of performance (COP) versus chilled water load has been plotted for operation with one chiller and with two chillers. The upper pair of curves uses a COP calculated by dividing the chilled water load by the chiller power. The lower set of curves uses a calculated COP that includes the power to run the chilled water pump(s) and the condenser water pump(s) with the chiller power.

Using the upper curves the optimal "switch point" would be point 1 or an approproximate load of 460 tons. However, the lower set of curves indicates a switch point of approximately 700 tons. The COP that takes into account the power to run the pumps represents a more correct value on which to base control decisions.

If the decision to switch the operating status of the chiller did not include the power to run the pumps a significant energy "penalty" could be incurred. For example, at a load of 460 tons the upper curves would indicate that it was time to switch to two chiller

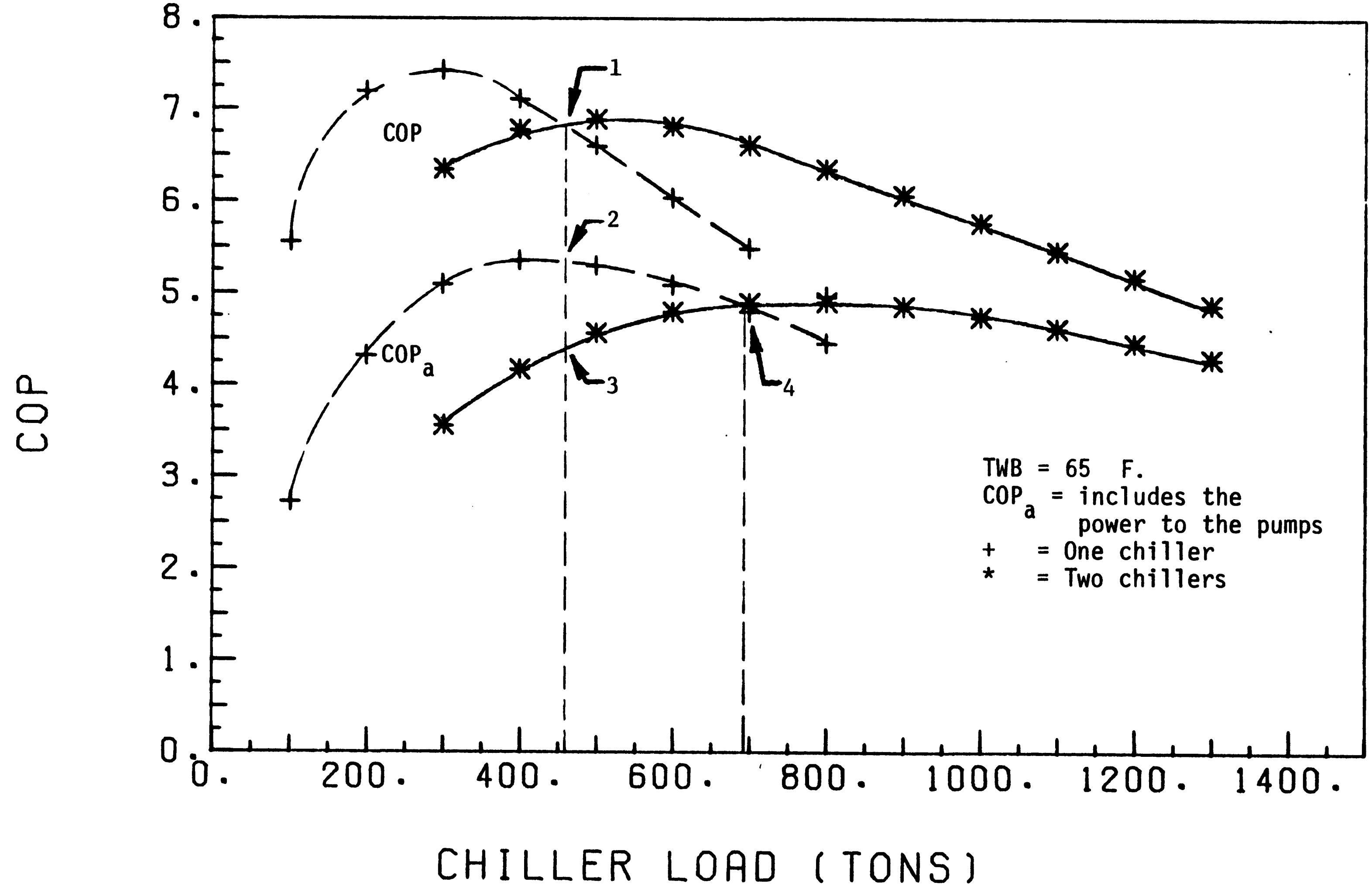

Figure 6.1 COP versus chilled water load. CHWS = 46° F.

operation. The lower curves, however, show that if such a change in chiller status was made the reduction in COP would be approximately 20 percent (from point 2 to point 3).

Figure 6.2 shows what effect the ambient wet bulb temperature has on the optimal switch point load. As the wet bulb temperature is increased from 65 to 85^{o} F. the optimal switch point load decreased from 690 to 610 tons.

The optimal points to switch from one chiller to two chiller operation occur well above the power draw limit for a single chiller. The optimal control strategy, then, is to let the chiller run up to its maximum chilled water load for the given operating conditions. If the building load increases further, an additional chiller should be brought on-line. The controller must "remember" the chilled water load that occurred just prior to the switch point. By knowing the load at which the switch point occurred the controller can make a decision as to whether:

1) the current chilled water load is too high for one chiller operation, i.e., continue using two chillers, or
2) the current load is low enough for one chiller operation, so, one of the chillers can be

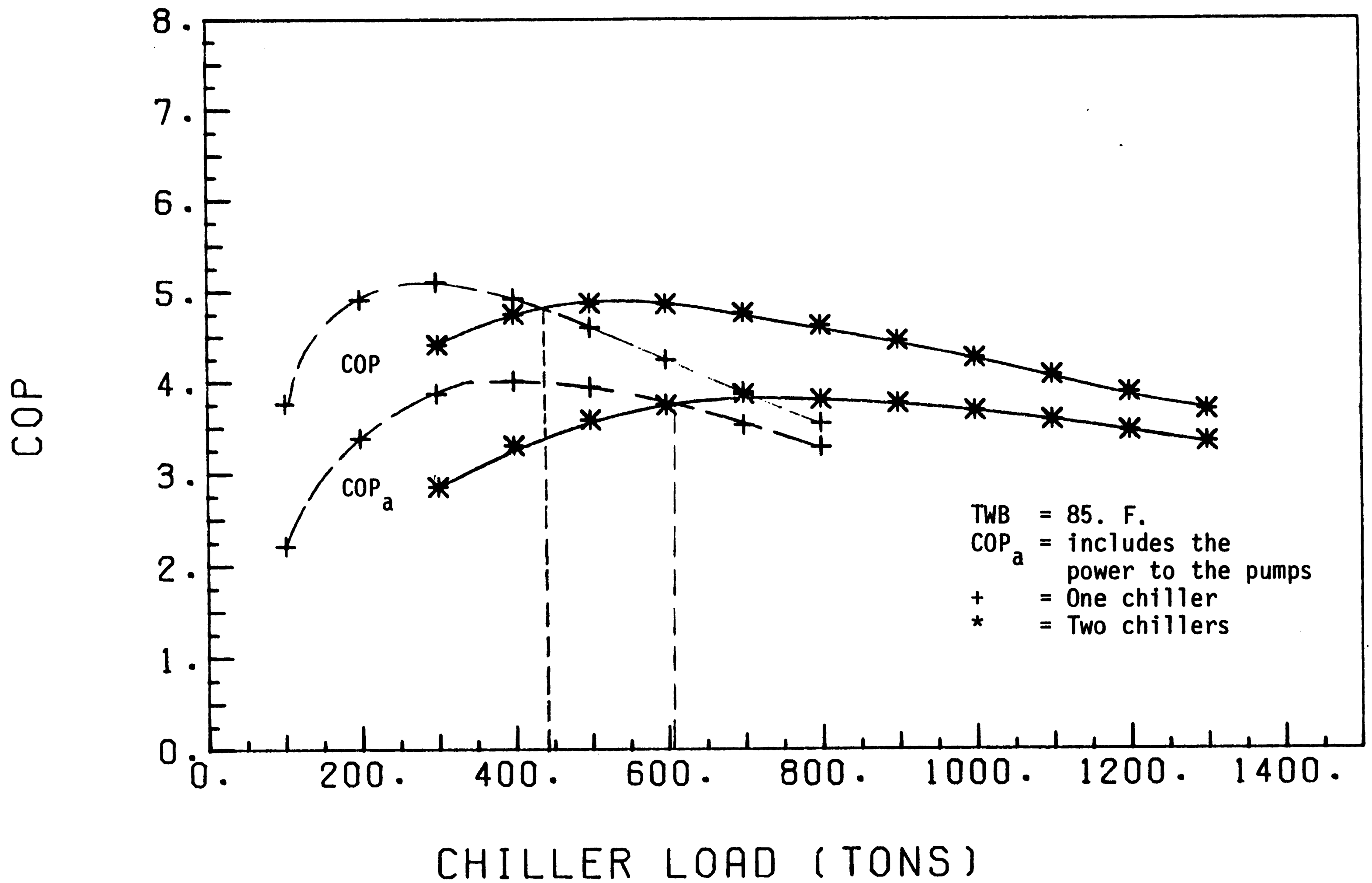

Figure 6.2 COP versus chilled water load. CHWS = 46° F.

turned off.

This happens to be the same chiller status control strategy that IBM currently uses at IBM-Atlanta.

6.2 COOLING TOWER FAN STATUS

Approach control is the technique currently used to control the operation of the cooling tower fans. This form of control inherently neglects the interaction between the cooling tower and the chiller. A more appropriate form of control would be to optimize the operation of the chiller-cooling tower subsystem so that minimum energy is used. Using the chiller and cooling tower computer models a TRNSYS simulation deck was developed. Various combinations of chilled water load, ambient wet bulb temperature, cooling tower fan speeds, and number of operating chillers were used as inputs to the simulation. The results for operation with a single chiller are shown in Figures 6.3 through 6.5 and the results for that with two chillers are shown in Figures 6.6 through 6.8. The various modes of operation for the two-speed cooling tower fans are described below:

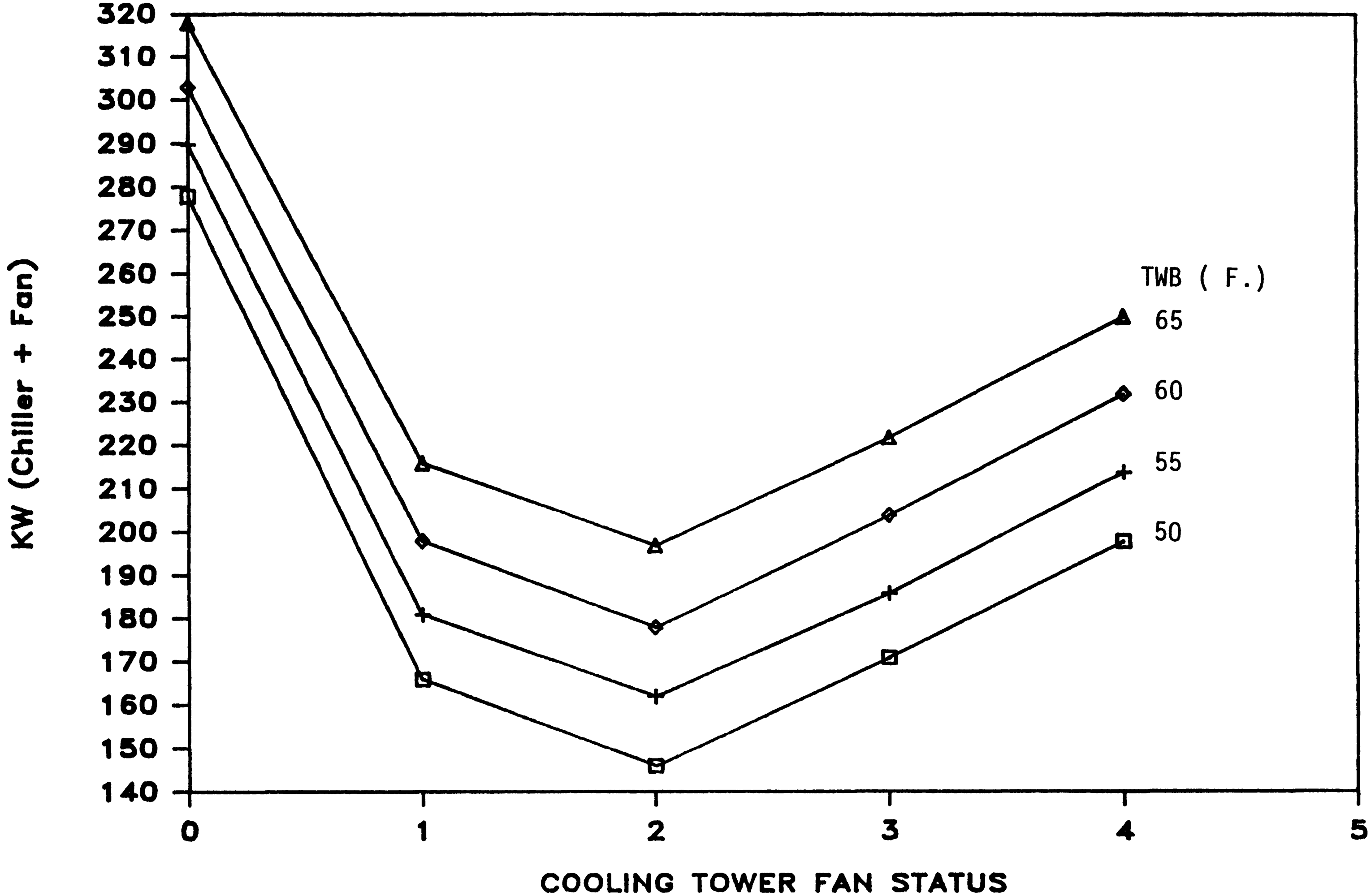

Figure 6.3 Cooling tower fan status optimization
Chilled water load = 350 tons
One chiller operation

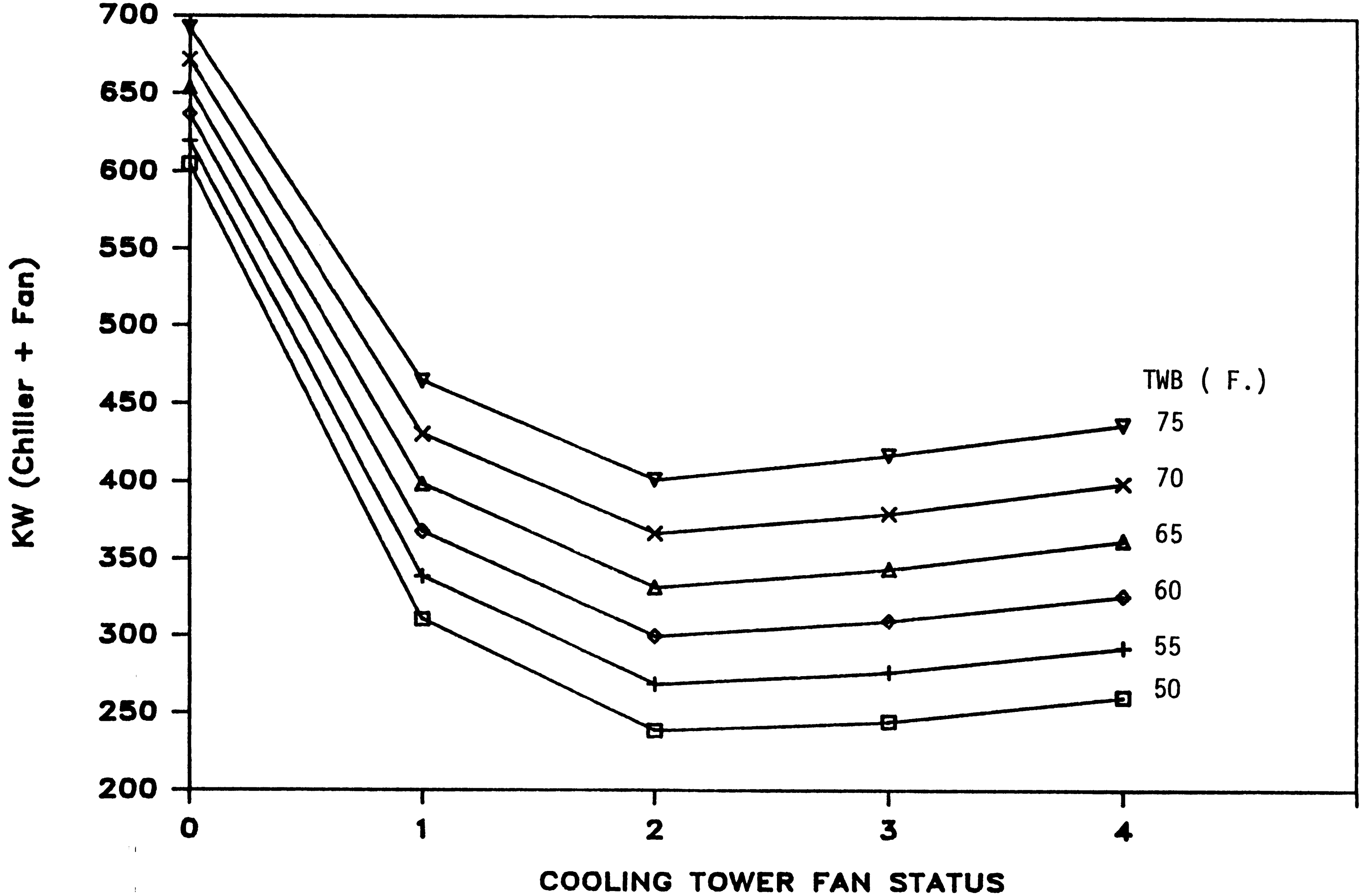

Figure 6.4 Cooling tower fan status optimization
Chilled water load = 550 tons
One chiller operation

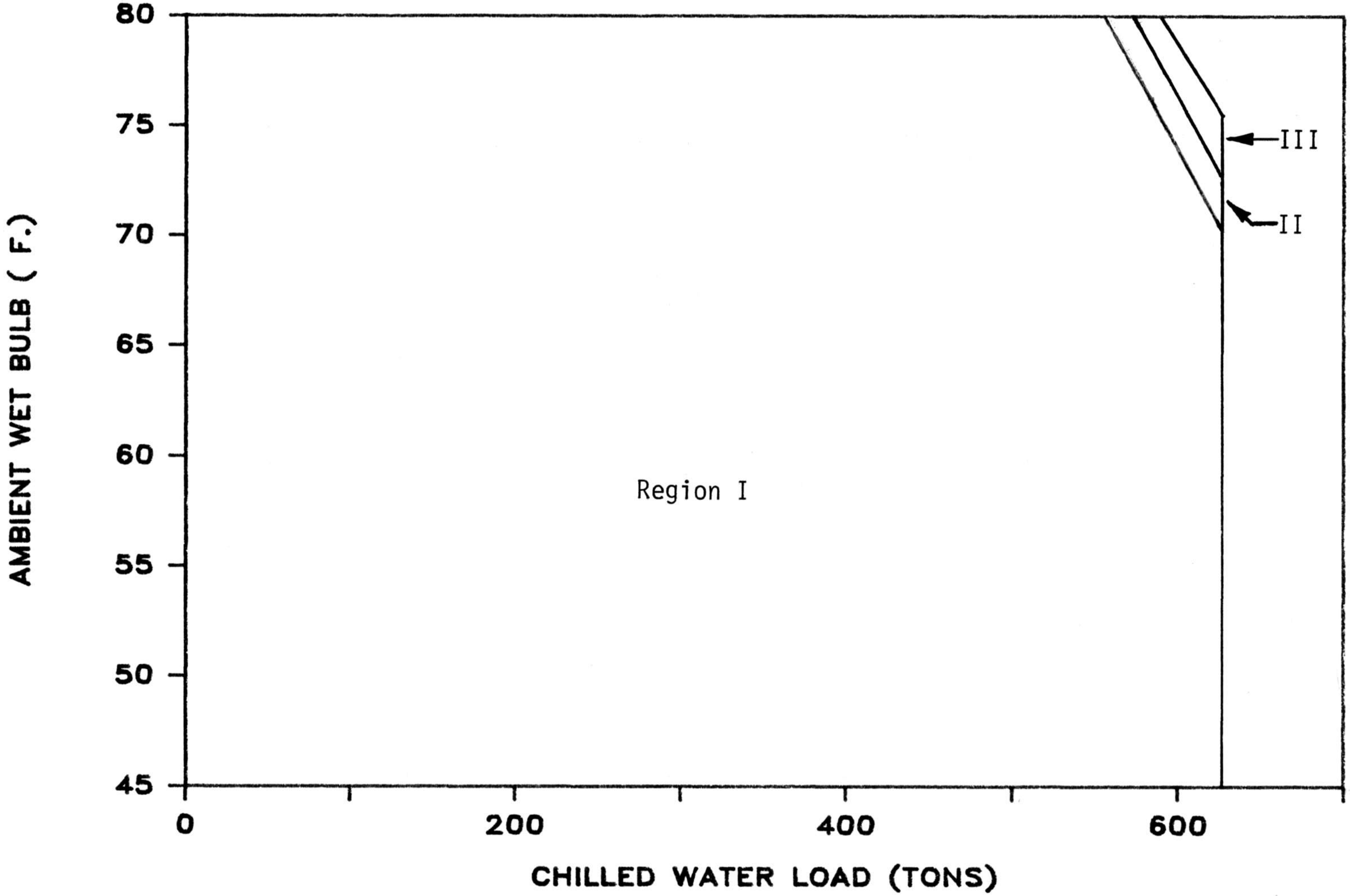

Figure 6 Optimal regions of cooling tower fan operation. Region I fan status low-low, Region II low-high, and Region III high-high. One chiller operation.

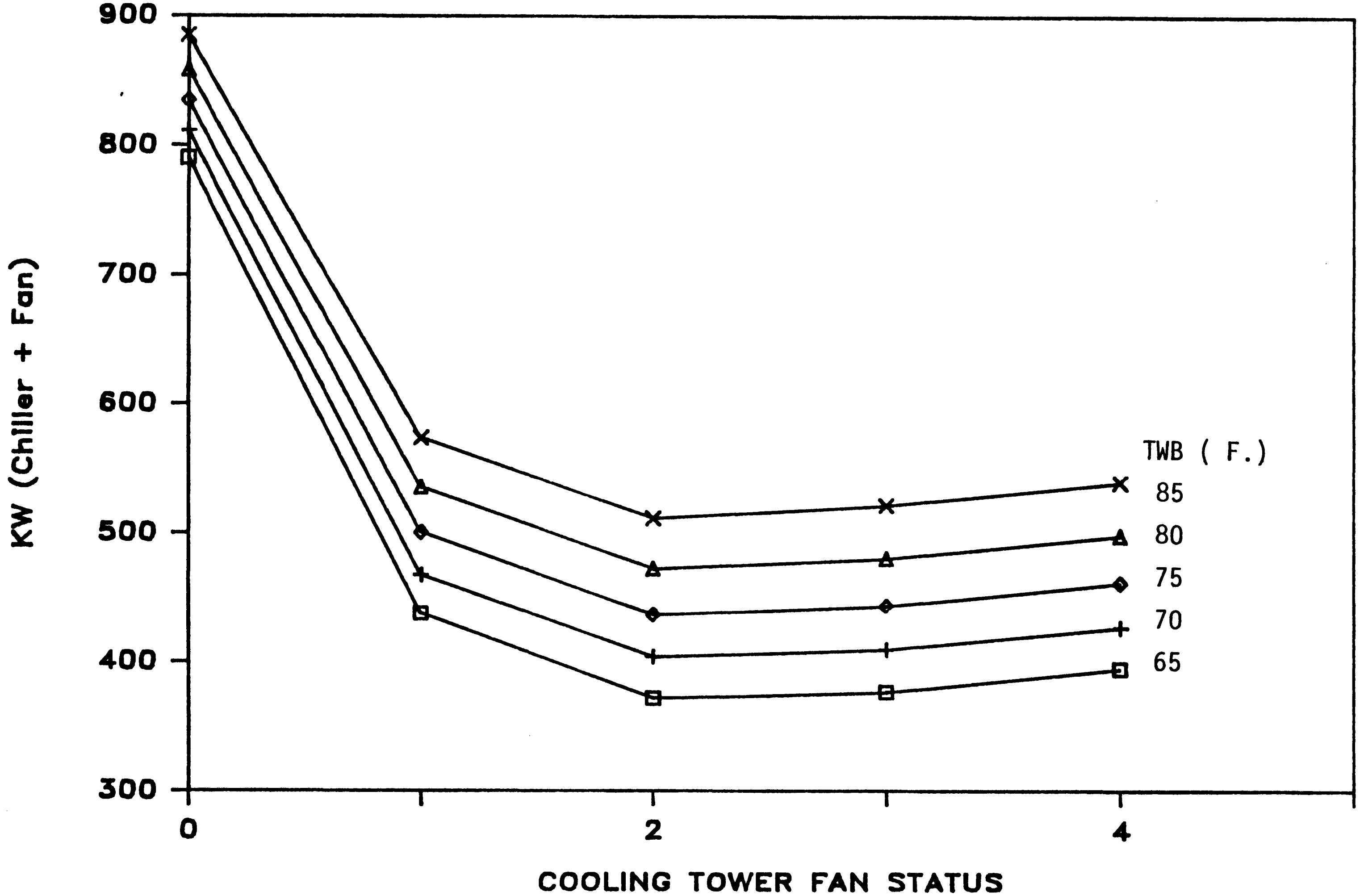

Figure 6.6 Cooling tower fan status optimization
Chilled water load = 625 tons
Two chiller operation

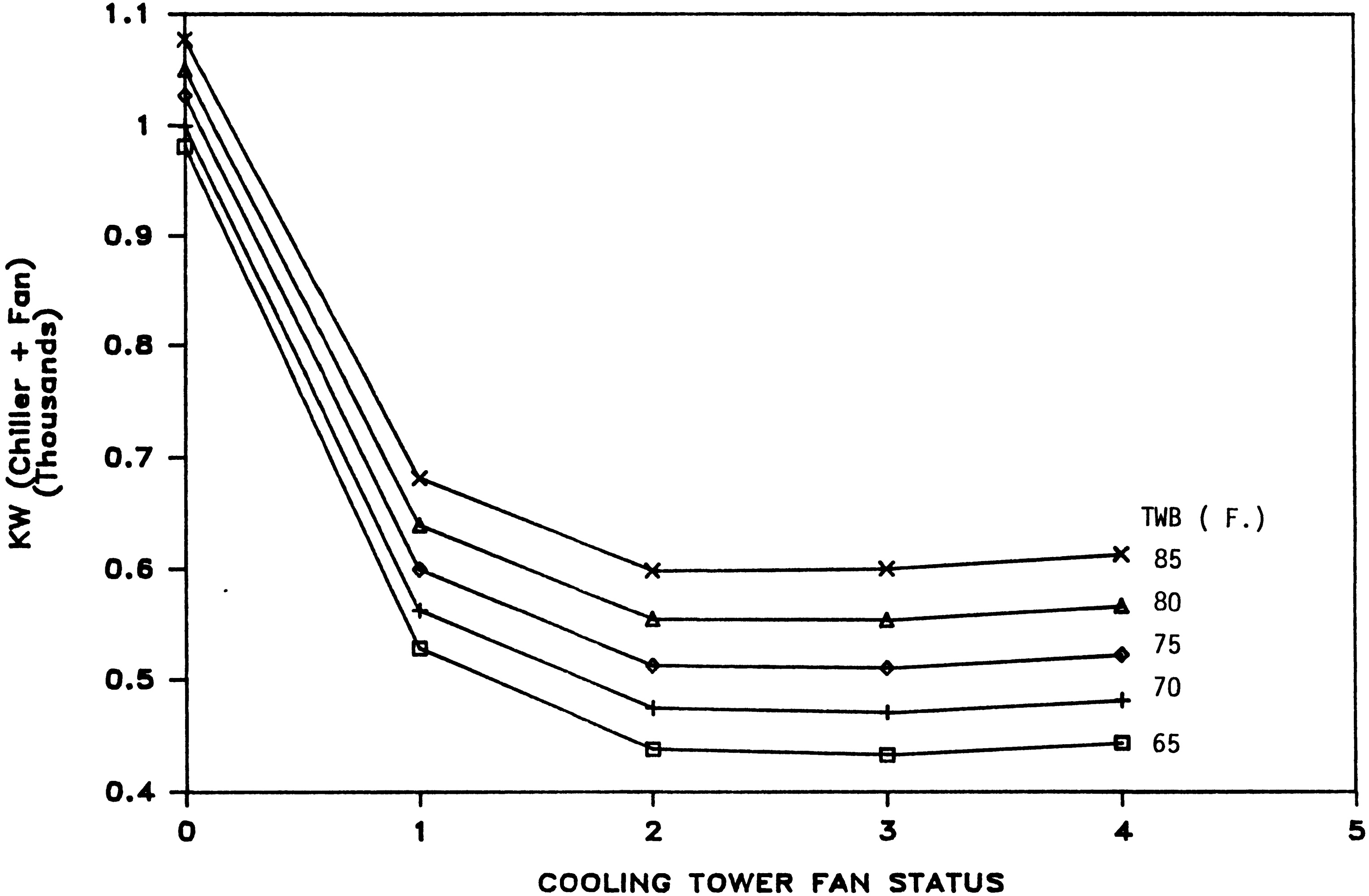

Figure 6.7 Cooling tower fan status optimization
Chilled water load = 725 tons
Two chiller operation

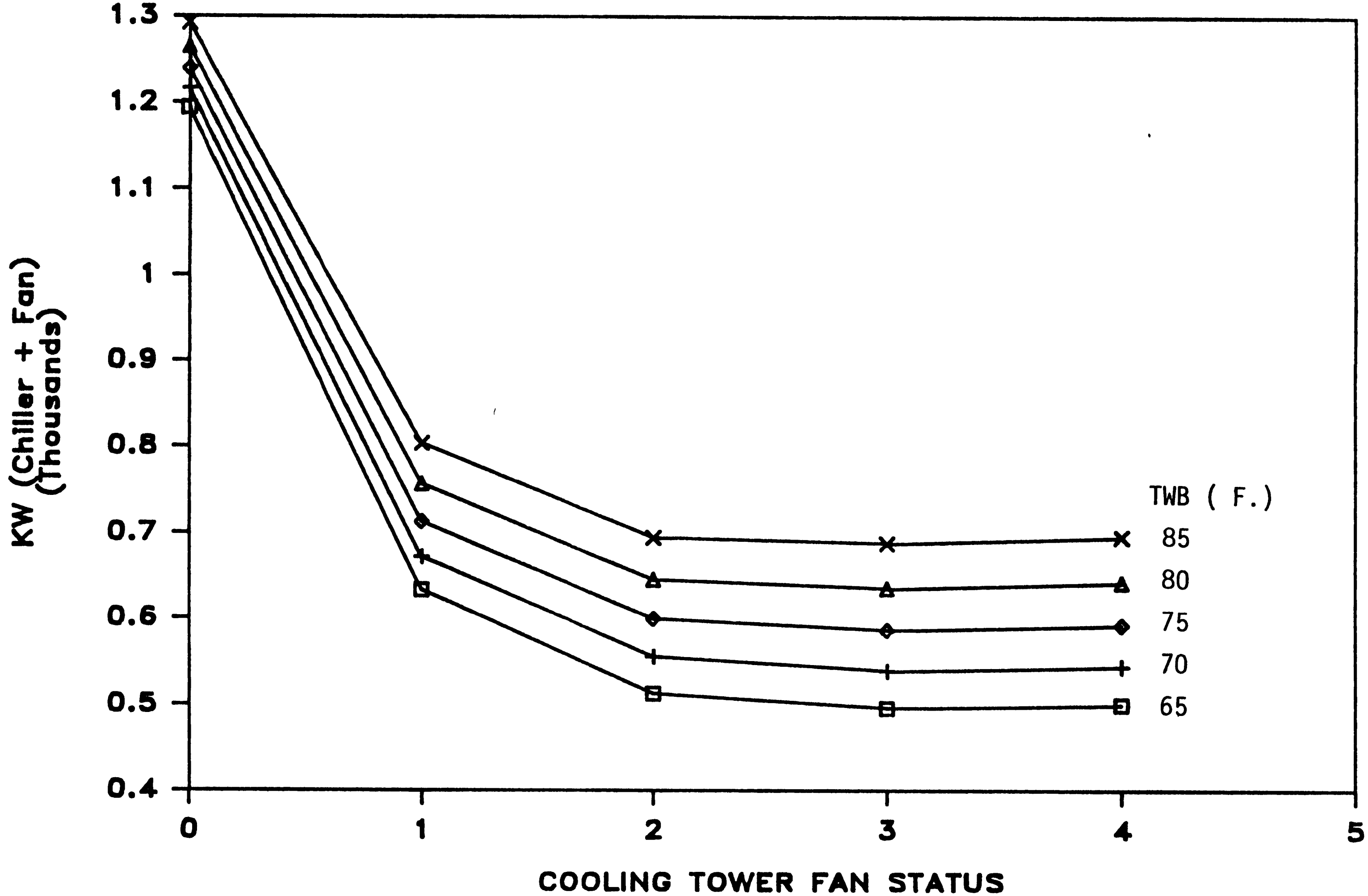

Figure 6.8 Cooling tower fan status optimization
Chilled water load = 825 tons
Two chiller operation

Mode	Fan #1	Fan #2
0	off	off
1	off	low
2	low	low
3	low	high
4	high	high

For one chiller operation the optimal fan speed status is mode 2 in almost all cases. The exception to this appears on Figure 6.5 where, at ambient wet bulb temperatures greater than 70° F., the maximum chiller power draw is exceeded. Under this condition a decision must be made whether to switch to two chiller operation or to continue operating with one chiller and increase the fan status level. In almost all cases it is better to increase the fan status level since there is a high energy "overhead" associated with turning on an additional chiller. This situation is an example of the effects of equipment limitations which have to be covered in the optimization strategies.

In the case of two chiller operation at a chilled water load of 625 tons (Figure 6.6), the optimal fan speed status remains at the mode 2 level. However, at a load of 725 tons (Figure 6.7) the optimal fan speed status is at the mode 3 level. The difference in power consumption between mode 3 and mode 2 at a load of 725 tons is slight. At a load of 825 tons (Figure 6.8), the optimal level is now mode 3, although mode 4 is very close to it.

From the results of the chiller-cooling tower simulations a map showing the optimal regions of operation for the fan levels was produced. Figure 6.9 is for one chiller operation. Region I is the mode 2 level, while regions II and III are modes 3 and 4, respectively. Regions II and III are determined by the chiller power limit. This figure is for a chilled water set point temperature of 48° F. For lower chilled water temperatures regions II and III would expand to the left on the figure. The reason behind this phenomena is, again, related to the chiller power limit. At lower chilled water supply temperatures the power limit is reached at lower chiller loads and ambient wet bulb conditions. This situation results in a more rapid switch to a new operating level.

Figure 6.10 is a map showing the optimal cooling tower fan levels for two chiller operation. Region I is, again, the mode 2 level and Region II is the mode 3 level. This figure is, also, for a chilled water set point of 48° F.

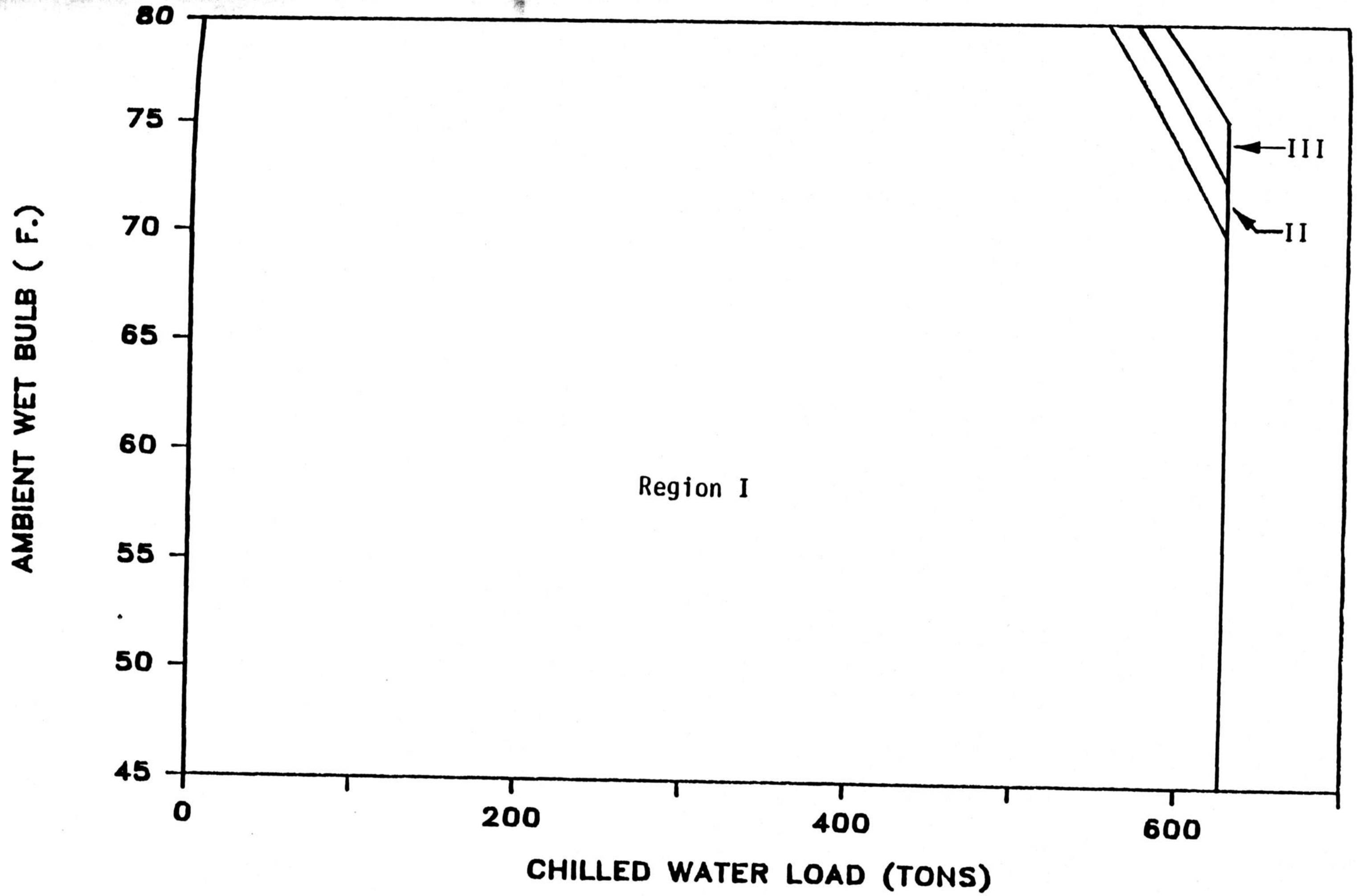

Figure 6.9 Optimal regions of cooling tower fan operation. Region I fan status low-low, Region II low-high, and Region III high-high. One chiller operation.

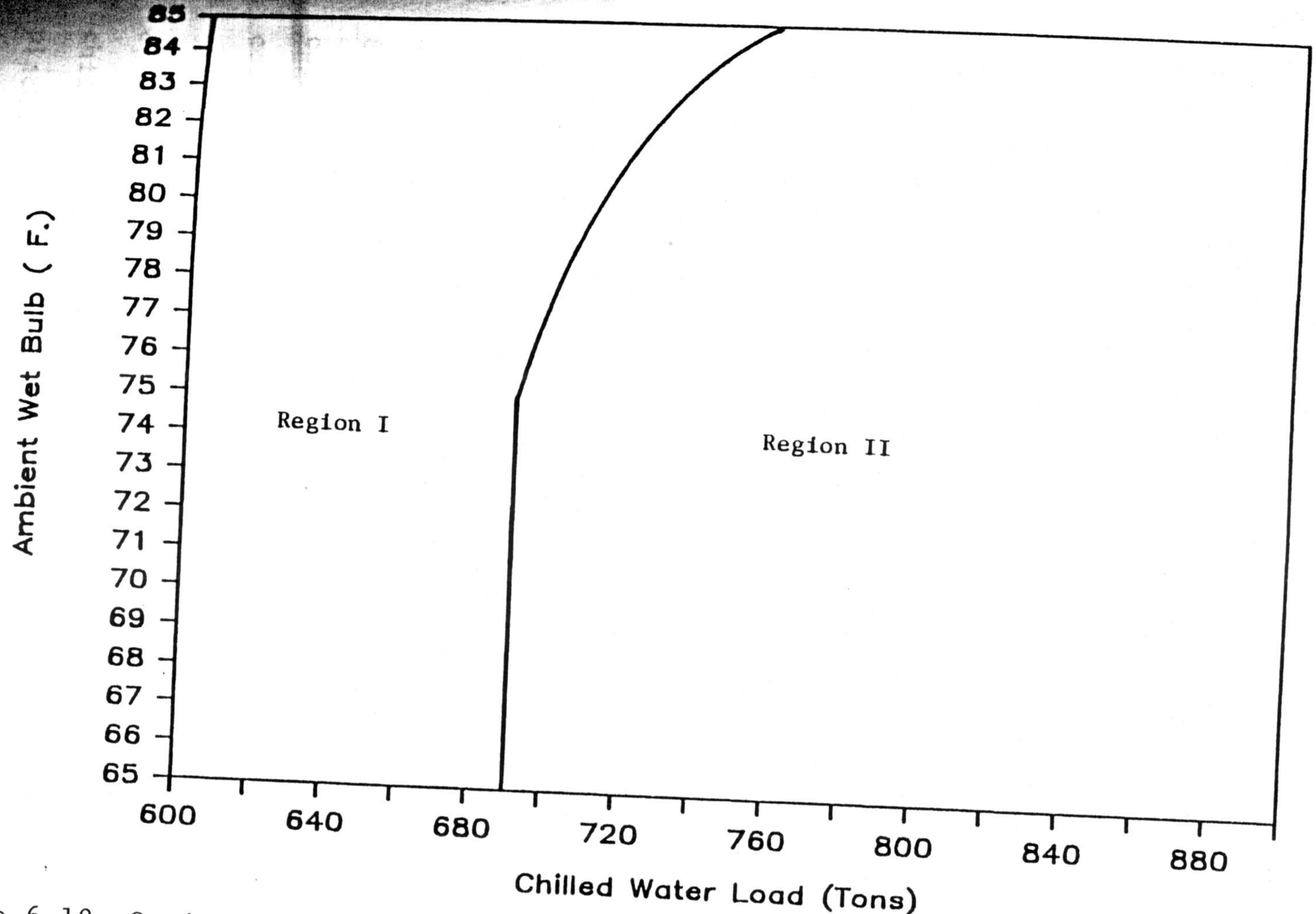

Figure 6.10 Optimal regions of cooling tower fan operation. Two chiller operation. Region I fan status low-low and Region II low-high.

6.3 CHILLED WATER SUPPLY AND AHU SUPPLY AIR TEMPERATURE

In the Atlanta building the chilled water supply set point is adjusted based on the return water temperature deviation from a desired set value. On the other hand, the AHU supply air set point is determined using a relationship involving the outdoor dry bulb temperature (see Section 4.1). The facility engineers indicate that this control scheme seems to work. However, this control scheme more-or-less neglects any interaction between the AHU supply air temperature and the chilled water supply temperature. This is the reason why the determination of the optimal temperatures of these two set points have been grouped together under one heading.

A simplified schematic showing the cooling coil and fan set-up from an air handling unit is shown in Figure 6.11. The internal control strategy of an AHU cooling coil is to modulate the flow of chilled water through the coil in order to achieve a desired supply air temperature. This control scheme introduces the first limit on the operation of the AHU. For a given air flow rate and chilled water supply temperature there is a lower limit placed on the temperature of the air that exits from the cooling coil. The reason for this limit is that the chilled water flow rate has a finite range of operation.

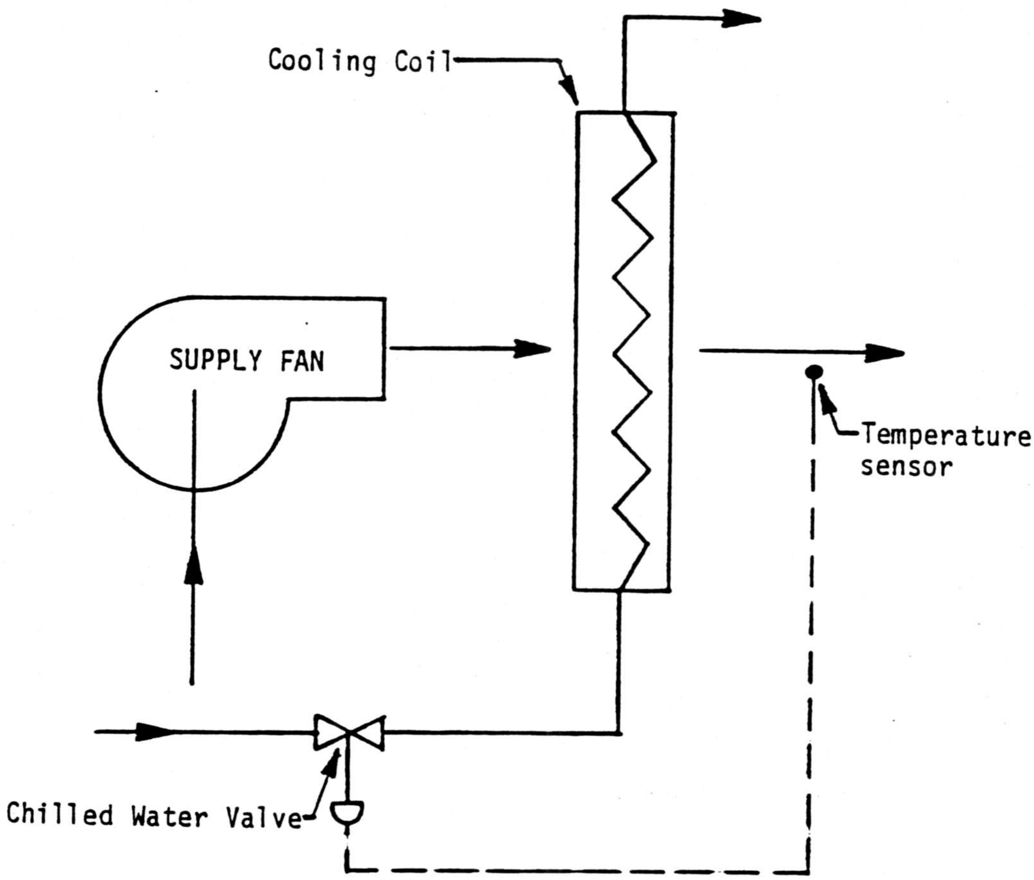

Figure 6.11 Schematic of the cooling coil and fan arrangement for an air handling unit

The range of chilled water flow rate is affected by the characteristics of the chilled water value and the chilled water distribution system. If the maximum chilled water flow rate is reached or if the supply air set point temperature is too low then the actual supply air temperature will be out of control. In the context of the current discussion "out of control" means that the desired supply air temperature can not be reached and that the actual temperature will be determined by the characteristics of the return air flow stream.

Other limits on the optimal chilled water supply and AHU supply air temperature are:

1) The chilled water supply temperature should not drop below 42^{o} F. The reason for this limit is to lessen the chance for localized freezing to occur in the chiller evaporator coils.
2) The chilled water supply temperature must not go above 48^{o} F. The 48 degree temperature was found to be the upper limit on the supply water temperature that could be tolerated by the computer room chilled water system.
3) The lower limit on the AHU supply air temperature is 55^{o} F. This was the lowest supply

temperature that could be used before people began complaining to the building engineers about cold drafts.

4) There is an upper limit on the supply air temperature that can be used and still maintain a desired humidity level inside the building. The supply air temperature must be low enough to handle the latent load from the space and possibly from the ambient ventilation air. Moisture is removed from the air stream by condensation forming on the AHU cooling coils.

5) A minimum flow rate of ventilating air must be maintained whenever the AHU system is operating. The reason for this limit is to meet the fresh air requirements of the space.

A step-by-step procedure was used to determine the optimal chilled water supply and AHU supply air temperatures subject to these constraints. A TRNSYS simulation deck was written to simulate the building operation under a range of load and ambient conditions. The simulation was run for building chilled water loads between 300 and 625 tons in 25 ton increments and the ambient wet bulb temperatures from 45 to 75^{o} F. in one degree increments. The chilled water supply temperature was varied between 42 and 48^{o} F. (.5^{o} F. increments) and the supply air temperature between 55 and 60° F. in one degree increments. A total of over 40,000 combinations of operating conditions were simulated using the TRNSYS deck. This approach was used to determine the optimal steady state operating conditions for the HVAC system. Later, these strategies would be used in the dynamic control of the system.

The first step used to process the simulation results was to determine the chilled water supply temperature limits for various supply air temperatures. Since there is a limit to the rate of the chilled water flow through the cooling coil, there is a maximum chilled water supply temperature that can be used to meet a certain supply air temperature. The limits on the chilled water supply temperature (Figure 6.12) play an important role in determining the optimal supply air temperature.

The optimal supply air temperature was determined for various building loads using the TRNSYS simulation results. Figure 6.13 is a graphical representation of the optimization results. At low cooling loads (less than 450 tons) the minimum flow rate of ventilating air is the dominant factor in determining the optimal supply air temperature. A higher supply air temperature can be used at these low load conditions because a minimum

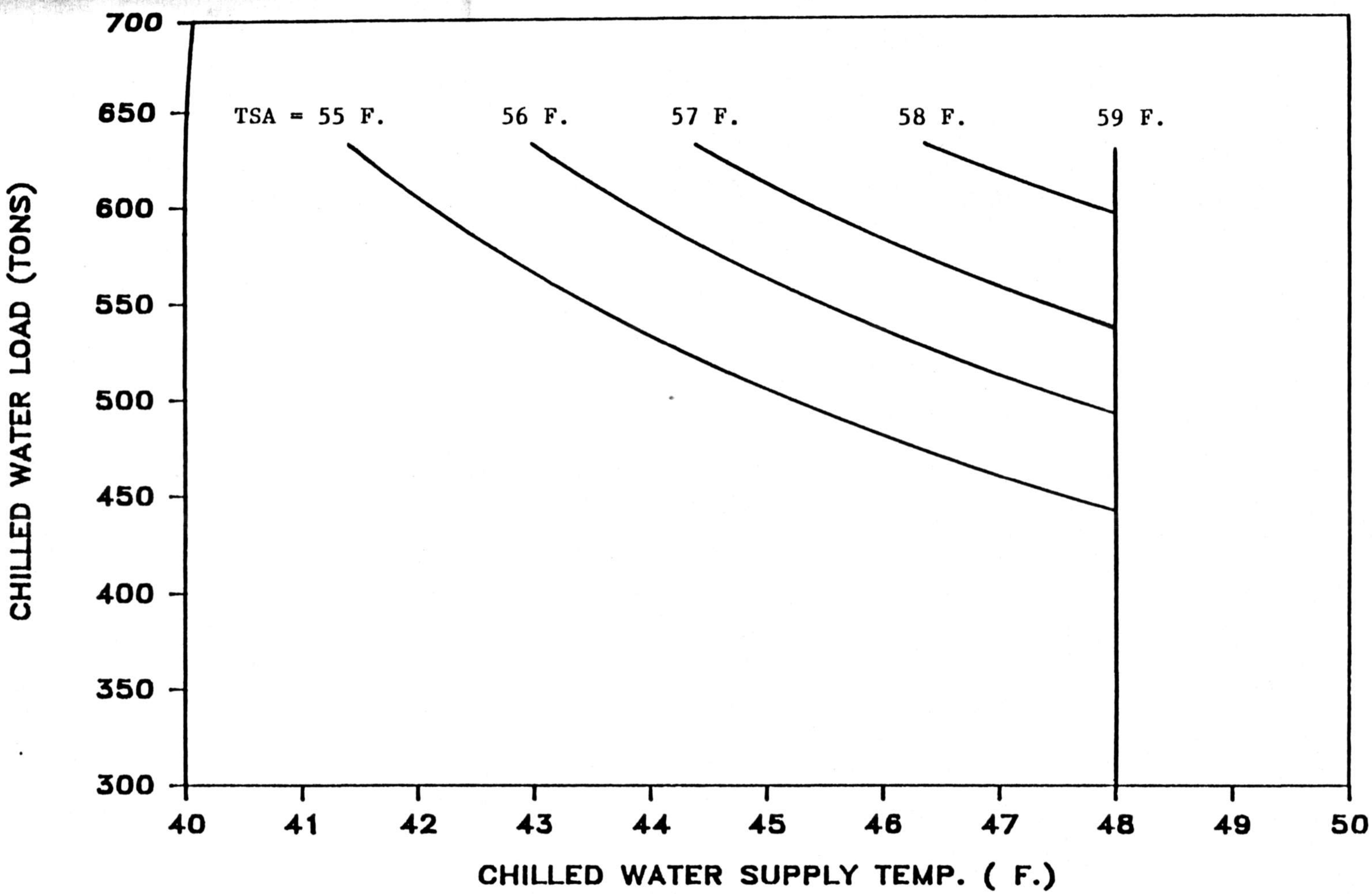

Figure 6.12 Chilled water supply temperature limits for various chilled water loads and AHU supply air temperatures

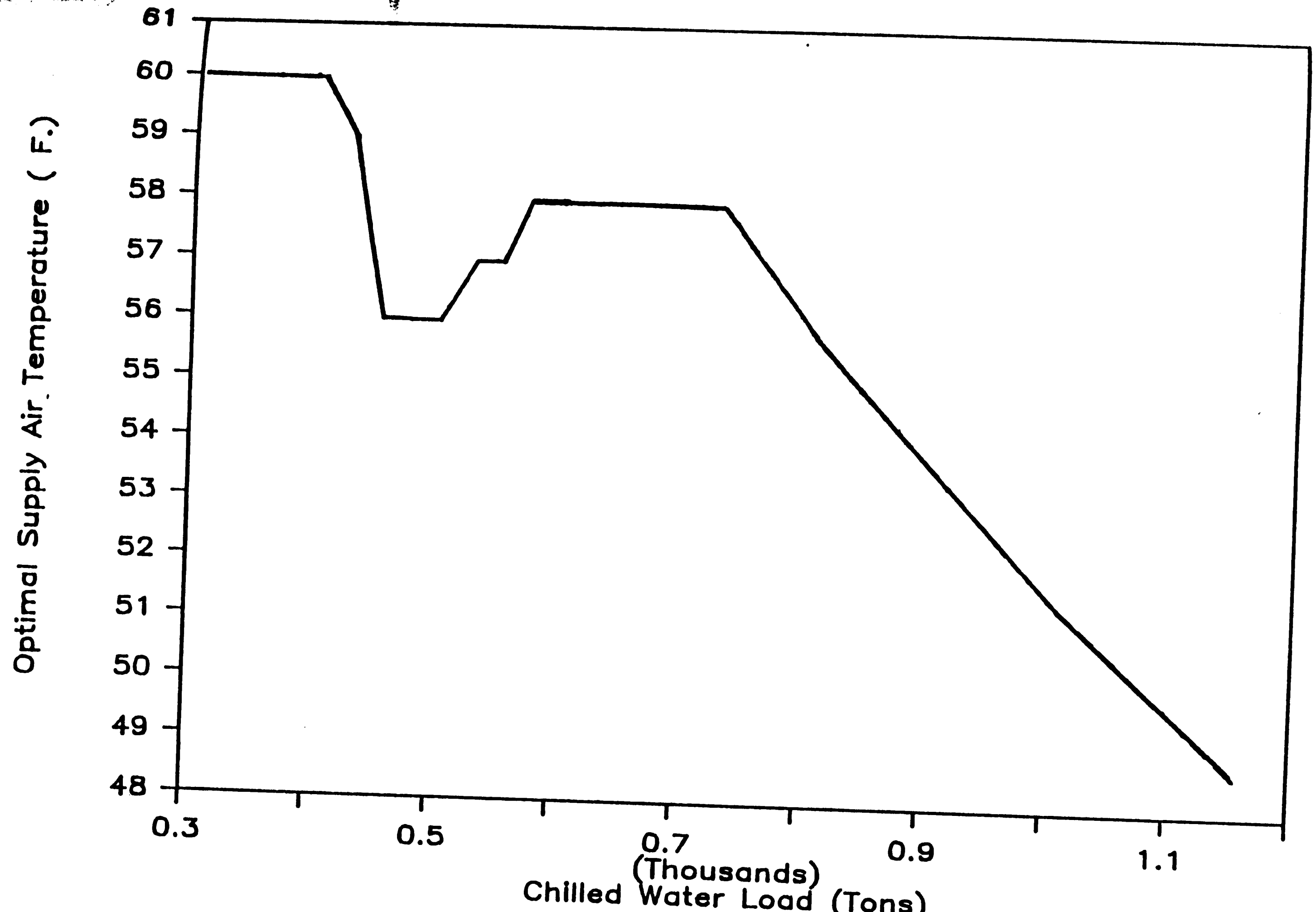

Figure 6.13 Optimization results for the AHU supply air temperature versus chilled water load

amount of supply air must be delivered to the building zones. For cooling loads between 450 and 600 tons the optimal temperature is determined by the chilled water supply temperature limits shown in Figure 6.12. For loads between 450 and 500 tons, supply water at 48° F. is able to yield a supply air temperature of 56° F. As the cooling load increases a trade-off between the chiller power and the AHU fan power must be established. The results of the optimization show that the additional fan power necessary to meet the load with a higher supply air temperature is much less than the additional chiller power necessary to produce a lower chilled water temperature and a lower supply air temperature. In other words, as the building load increases it is better to maintain the upper limit on the water temperature and to meet the load with a higher mass flow rate of warmer supply air. Figure 6.14 illustrates this tradeoff between fan power and chiller power. For a chilled water load of 550 tons the optimal supply air temperature is 57° F.

At loads above 600 tons another important limiting factor comes into effect. The limit on the humidity level for the building now has an effect on determining the supply air temperature that can be used. A number of calculations were performed to find the limit on the

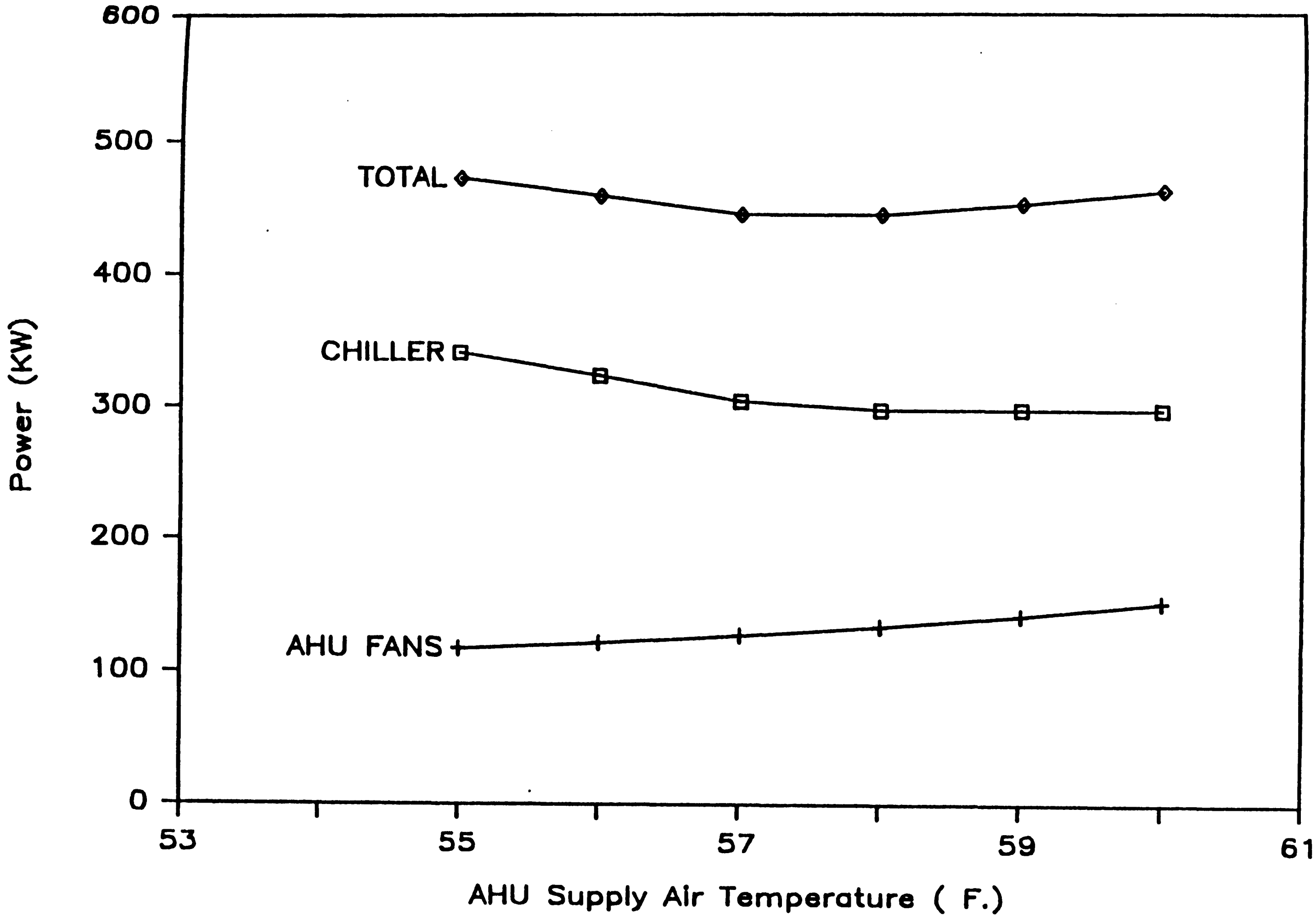

Figure 6.14 Power use distribution versus AHU supply air temperature. Load = 550 tons and $T_{WB} = 65^{o}$ F.

supply air temperature for various cooling loads. The following assumptions were made:

1) The indoor conditions of the zone were to be maintained at Tdb = 76° F. and RH = 60%.
2) The latent load for the zone was a constant value of 40 tons. This figure was based on estimates of the building occupancy level and other sources of latent loads (the cafeteria area, etc.).
3) The outdoor conditions were such that a minimum amount of outside air was being used. To account for a possible increase in the coil load caused by this assumption an additional 1 Btu/lbm dry air was added to the enthalpy of the return air to calculate a "mixed" air enthalpy.
4) 10 percent of the mixed air was assumed to bypass the cooling coil.

From these assumptions and knowledge of the maximum air flow rate for the AHU supply fans it was possible to determine the limiting supply air temperature. As the supply air temperature begins to decrease, the chilled water temperature will also have to be decreased so that the lower air temperature can be reached.

7.0 SIMULATION RESULTS

7.1 SIMULATION RESULTS--OPTIMAL CONTROL

A TRNSYS simulation deck of the entire HVAC system was used to compare the system energy use under the optimal control strategies identified in Chpater 6 to the current control strategies. TMY (Typical Meterological Year) data provided the necessary input data to drive the simulation. TMY data is a single year data set for various geographical locations which contains hourly solar radiation and meterological data for an average year. The inputs used were the ambient wet bulb temperature, the ambient dry bulb temperature, and the horizontal solar radiation measurement.

The results for a typical simulation run under optimal and the current method of control are shown in Figure 7.1. The data used to drive this simulation were for one week in the month of May. The timestep used for the simulation and the control decision time step was one-eighth hour. Notice that for both methods of control the chiller is the major energy user. This leads to the conclusion that the chiller is the area in which to concentrate on for any energy saving control alternatives.

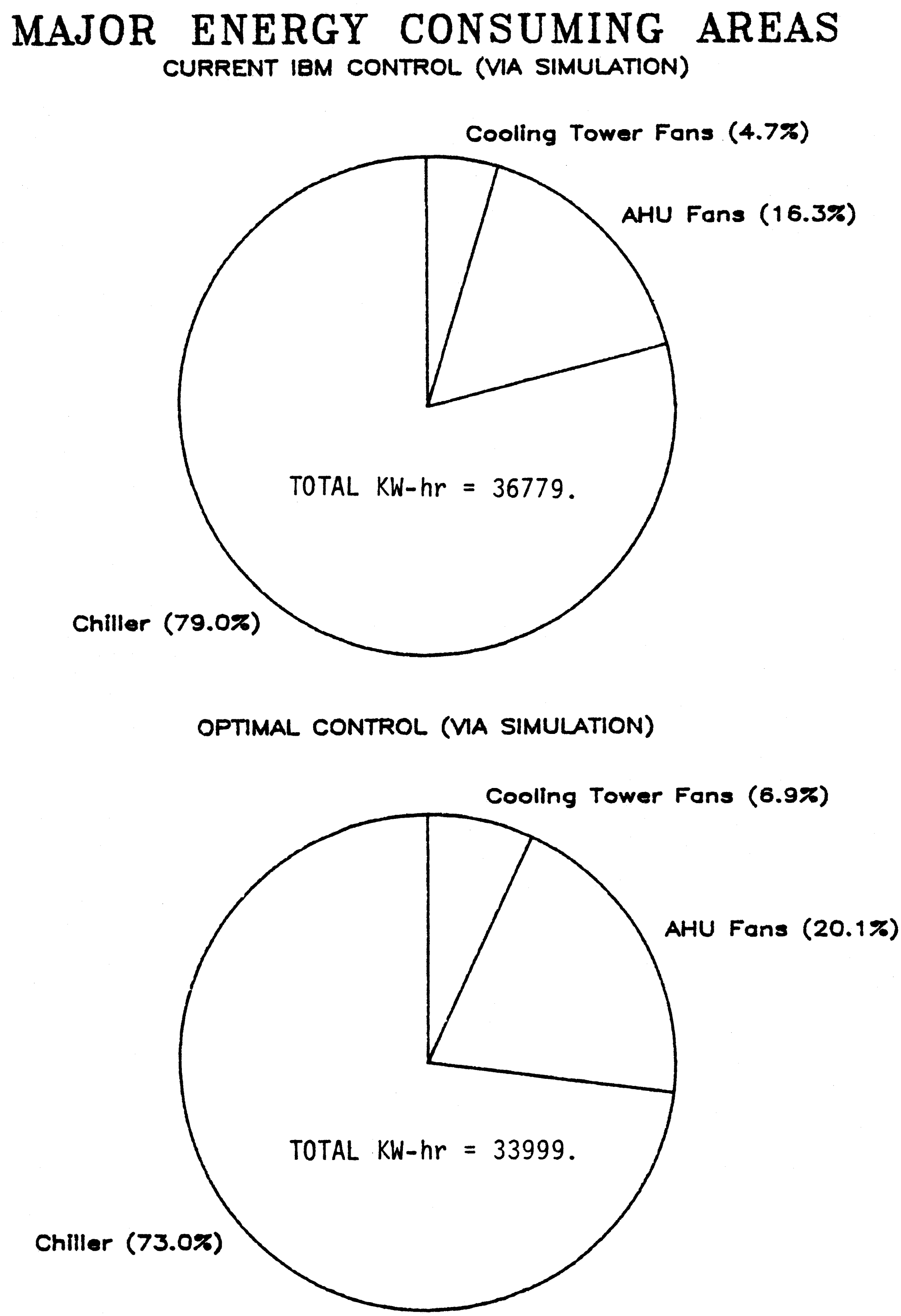

Figure 7.1 Energy use distribution for current control versus optimal control via simulation. One week in May.

Figure 7.1 also shows a large difference in the distribution of energy use between the two control strategies. For the optimal control case the power consumption has been increased (relatively and absolutely) in the cooling tower and AHU supply fan areas. However, the additional energy use in these areas is more than offset by the reduction in chiller energy use. An overall energy use reduction of 7.56 percent was achieved when the optimal control strategies were used in the simulation.

Figure 7.2 shows the results for longer term simulation. The current control and optimal control simulations were run from mid-May to mid-September, the major portion of the cooling season. A decrease in energy use of 8.73 percent was obtained by using the optimal control strategies. A dollar value can be placed on this decrease by assuming the cost of electricity is \$.06/KW-hr. The savings that results from optimal control is then \$3,240 for this particular period of operation.

7.2 SIMULATION RESULTS--DYNAMIC CONTROL

A major goal of this project was to determine the impact of dynamic control on the energy consumption of an HVAC system. For this project, dynamic control means

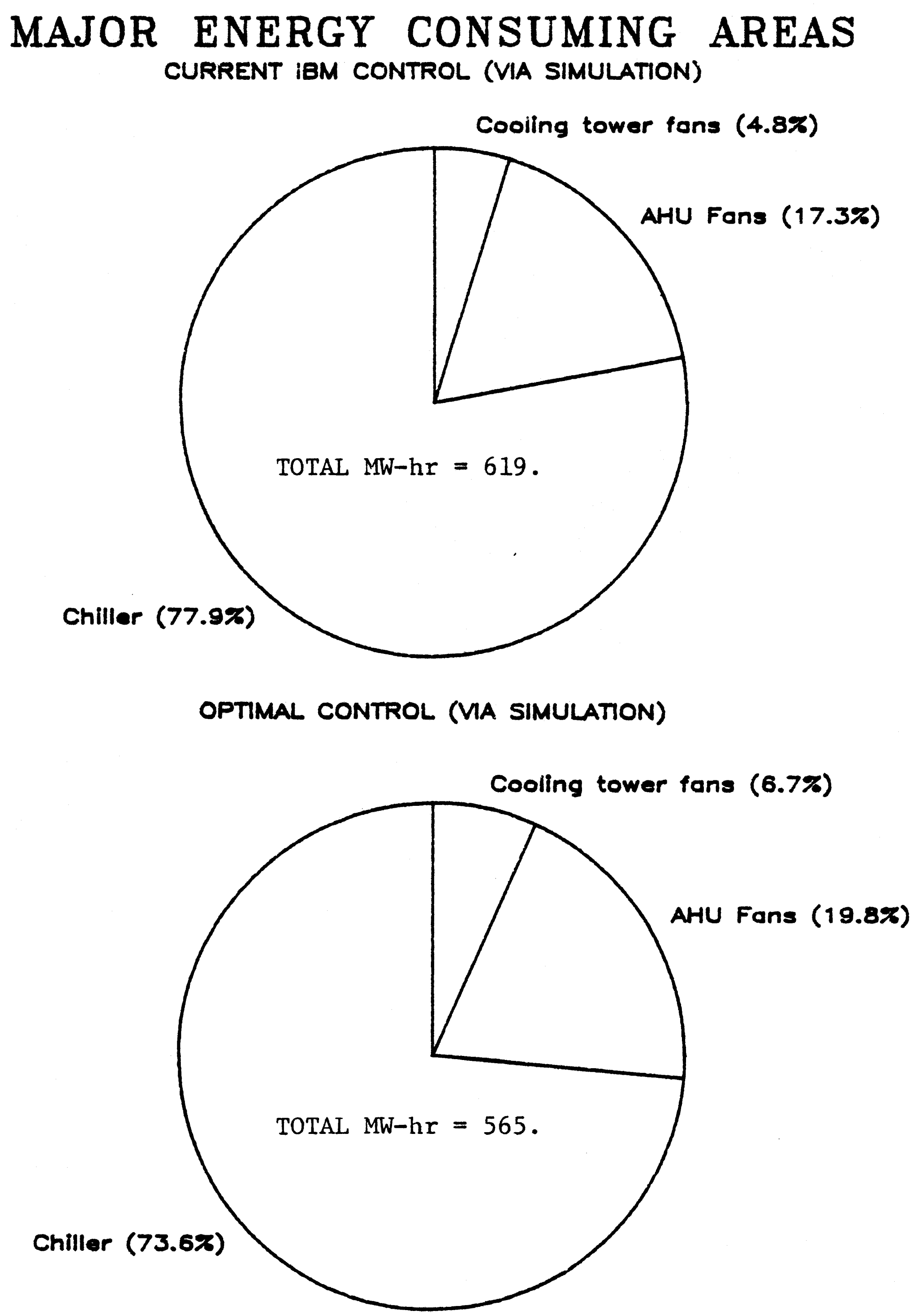

Figure 7.2 Energy use distribution for current control versus optimal control via simulation. Mid-May to mid-September.

the resetting of control points in response to changes in chilled water load and ambient conditions. To study this effect the TRNSYS simulation deck for the entire HVAC system was used. Each simulation was run using 1/64th hour timesteps. The timestep used to make control decisions was then varied from 1/64th hour up to 8 hour steps. The large amount of computing time that was necessary to generate each data value required that the simulations be run for one week intervals.

The simulation results for control decision timesteps up to one hour are shown in Figure 7.3. The current IBM control strategy was used to generate Figure 7.3. The percent power increase is defined in terms of the energy consumption of a simulation using the 1/64th hour time step as the base case. The solid line is a curve fit of the data points. An R-squared of 98.4% was obtained with this curve fit. The dashed lines indicate approximate error limits for the results. The TRNSYS simulation used a convergence tolerance of $\pm$ .1%. Three separate TRNSYS-generated values were used to determine the total power consumption. These were the cooling tower fan, AHU supply fan, and chiller power consumption. Using the root-sum-square method an absolute error tolerance of $\pm$ .17% power increase was determined. These error bounds are shown as dashed lines in

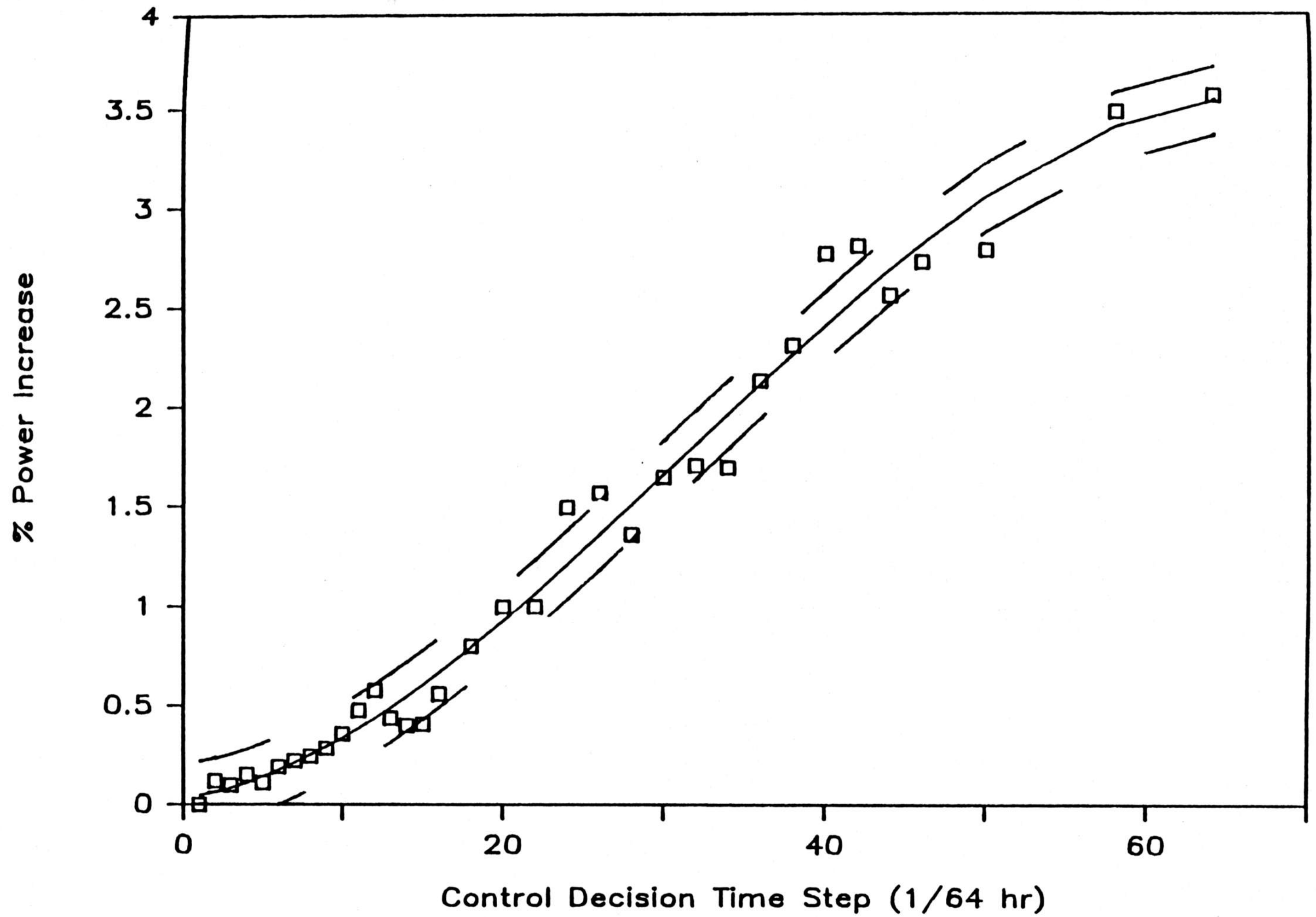

Figure 7.3 Percent power increase versus the control decision time step. Results via computer simulation.

Figure 7.3.

The results for control decision timesteps between 1/64th hour and 8 hours are shown in Figure 7.4. The data points no longer follow any particular pattern. The randomness of the points can be attributed to two major factors. The first is that as the decision time-step increases the control of the system becomes less and less dynamic. A control decision is made based on the current status of the system even though the operating conditions of the system may change during the timestep. The second factor is related to the first, but, it involves the major switch points in the daily operation of the building. The AHU's are turned on at 7:30 a.m. and shut off at 6:00 p.m. Depending on the timestep of the control decision a decision based on one of the switch points would persist for the entire timestep. For these reasons an error envelope was placed around the data points in Figure 7.4. The envelope indicates that the possible power increase for "non-dynamic" control could range between 2 and 7 percent.

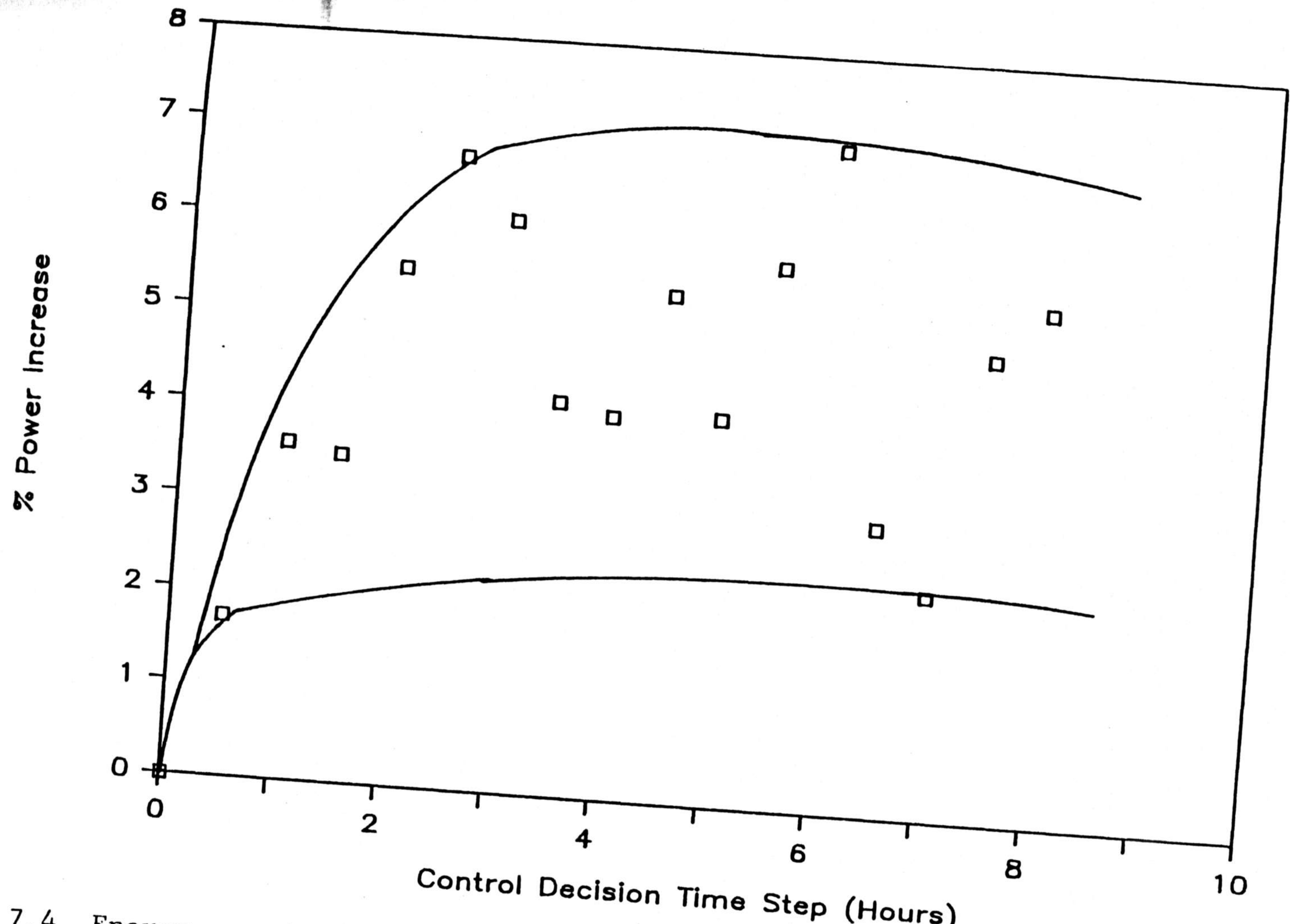

Figure 7.4 Energy use distribution for current control versus optimal control via simulation.

8.0 CONCLUSION

The overall objective of the project was to determine operating strategies for HVAC systems which incorporate system dynamics and interactions and which will potentially reduce energy use. In keeping with the overall objective of the project the following specific goals were achieved:

1) To study the process dynamics and interactions of a building HVAC system through the use of collected test data and equipment computer models;
2) To determine, via computer simulations, the effect of the time between control decisions in the dynamic control of an HVAC system;
3) To determine, via computer simulations, dynamic HVAC operating strategies that will potentially reduce the HVAC system energy consumption.

The computer algorithms that were developed to model the HVAC equipment were described. Comparison between measured data and the model predictions were given. Short term tests of equipment dynamics were conducted to determine the significant dynamic effects. The results showed that the significant transient effects were:

a) the cooling tower response to fan speed changes;

b) the "flush time" of the chilled water through the system;

c) the effects of the building structure due to capacitance.

The transient effects that were insignificant, due to their very short time duration, were:

a) the chiller response to set point (supply water) changes;

b) the air handling unit response to set point (supply air) changes.

The effect of the time between control decisions was studied using the developed building computer models. The results showed a 3.5% energy savings for a system operating under dynamic control as opposed to a system in which control decisions were made on an hourly basis. The results of the optimization study showed the importance of defining the operating limits for the HVAC equipment and the building comfort zone. Optimal control strategies were developed using the TRNSYS routines. An overall energy use reduction of 8.73% was achieved when the optimal control strategies were used as opposed to the current control methods. The end results of this project were the identification and investigation of potential energy saving HVAC operating strategies and the availability of "reliable" equipment models. The groups that could benefit the most from these models are building operators, HVAC controls persons, and future researchers.

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APPENDIX A

TRNSYS SIMULATION DECK

IBM Atlanta, Georgia

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**************************************************************
*            IBM ATLANTA - BUILDING SIMULATION               *
**************************************************************
*
*
SIMULATION   0   168   .015625
TOLERANCES .001 .001
LIMITS 500 0
NOLIST
*
UNIT 1 TYPE 9 DATA READER
*
      PARAMETERS 7
      13 1 -8 1 0 0 1
(7F3.0,6F7.2)
*
* DATA FROM TMY (TYPICAL METEORLOGICAL YEAR) DATA TAPE
*
**************************************************************
*                  SITE LOAD DESCRIPTION                     *
**************************************************************
*
UNIT 2 TYPE 16 RADIATION PROCESSOR
*
      PARAMETERS 7
      3 1 182 33 428.9 0 -1
      INPUTS 12
      1,8 1,19 1,20  0,0 0,0 0,0 0,0 0,0 0,0 0,0 0,0 0,0
       0  0     1   .25 90  180  90 270  90  0   90  90
*OUTS: 1=NV, 2=EV, 3=SV, 4=WV
*
UNIT 3 TYPE 33 PSYCH CALCS.
*
      PARAMETERS 2
      1 2
      INPUTS 2
      1,11 1,12
      70   50
* OUT(1)=W, OUT(2)=TWB
*
**************************************************************
* ZONE 1 & 2 SCHEDULES--PEOPLE,LIGHTS,ELECTRIC,AHU START-UP*
**************************************************************
*
UNIT 4 TYPE 14 PEOPLE SCHEDULE --- WEEKDAYS
     PARAMETERS 12
     0 0 8 0 9 20 16.5 20 17.5 0 24 0
UNIT 5 TYPE 14 LIGHT SCHEDULE --- WEEKDAYS
     PARAMETERS 12
     0 3.06E04 7.5 3.06E04 9.0 6.41E04 16.5 6.41E04 18
```

```
     3.06E04  24 3.06E04
UNIT 6 TYPE 14 ELECTRIC SCHEDULE --- WEEKDAYS
     PARAMETERS 12
      0 2.60E04 7.5 2.60E04 9 3.89E04 18 3.89E04 18.5
      2.60E04  24 2.60E04
UNIT 7 TYPE 14 PEOPLE SCHEDULE --- WEEKEND
     PARAMETERS 4
     0 0 24 0
UNIT 8 TYPE 14 LIGHT SCHEDULE --- WEEKEND
     PARAMETERS 4
     0 3.06E04 24 3.06E04
UNIT 9 TYPE 14 ELECTRIC SCHEDULE --- WEEKEND
     PARAMETERS 4
     0 2.60E04 24 2.60E04
*
**************************************************************
*   ZONE 3 SCHEDULES--PEOPLE,LIGHTS,ELECTRIC,AHU START-UP  *
**************************************************************
*
UNIT 10 TYPE 14 PEOPLE SCHEDULE --- WEEKDAYS
     PARAMETERS 12
     0 0 8 0 9 40 16.5 40 17.5 0 24 0
UNIT 11 TYPE 14 LIGHT SCHEDULE --- WEEKDAYS
     PARAMETERS 12
     0 9.79E04 7.5 9.79E04 9.0 2.37E05 16.5 2.37E05 18
     9.79E04 24 9.79E04
UNIT 12 TYPE 14 ELECTRIC SCHEDULE --- WEEKDAYS
     PARAMETERS 12
      0 7.44E04 7.5 7.44E04 9 1.11E05 18 1.11E05 18.5
      7.44E04 24 7.44E04
UNIT 13 TYPE 14 PEOPLE SCHEDULE --- WEEKEND
     PARAMETERS 4
     0 0 24 0
UNIT 14 TYPE 14 LIGHT SCHEDULE --- WEEKEND
     PARAMETERS 4
     0 9.79E04 24 9.79E04
UNIT 15 TYPE 14 ELECTRIC SCHEDULE --- WEEKEND
     PARAMETERS 4
     0 7.44E04 24 7.44E04
UNIT 16 TYPE 14 MAIN AHU SCHEDULE --- WEEKDAYS
     PARAMETERS 12
     0 0 7.5 0 7.5 1 18 1 18 0 24 0
UNIT 17 TYPE 14 MAIN AHU SCHEDULE --- WEEKEND
     PARAMETERS 4
     0 0 24 0
UNIT 18 TYPE 41 LOAD SEQUENCER - PEOPLE,LIGHTS,&ELECTRIC
     PARAMETERS 8
     7 1 1 1 1 1 2 2
     INPUTS 14
     4,1 5,1 6,1 10,1 11,1 12,1 16,1
```

```
     7,1 8,1 9,1 13,1 14,1 15,1 17,1
     0 0 0 0 0 0 0 0 0 0 0 0 0 0
*
************************************************************
*                ZONE 1 ---- WEST PERIMETER                *
************************************************************
*
UNIT 19 TYPE 19  ZONE 1
     PARAMETERS 10
     2 2 4.704E04 0 0 0 2.82E04 7 73. .009
     INPUTS 11
     1,11  3,1  25,15  25,12  25,16  0,0  18,1  0,0  18,2
     18,3  0,0
     75.  .01  55.  5.64E04  .0083  0  0  4  0  0  7.5
*NORTH WALL
     PARAMETERS 7
     1  1  120.  .4  .6  2  11
     INPUTS 1
     2,6
     0
*SOUTH WALL
     PARAMETERS 7
     2 1  120.  .4  .6  2  11
     INPUTS 1
     2,14
     0
*EAST WALL
     PARAMETERS 7
     3  3  3920.  .74  .26  3  23
     INPUTS 3
     21,12  21,1  0,0
     0   75.  0
*WEST WALL
     PARAMETERS 7
     4  1  3920.  .4  .6  2  11
     INPUTS 1
     2,17
     0
*CEILING
     PARAMETERS 7
     5  2  4704.  .74  .26  3  39
*FLOOR
     PARAMETERS 7
     6  2  4704.  .4  .6  3  35
*WEST GLAZING
     PARAMETERS 8
     7  5  1372.  1  .4  1.  1  6
     INPUTS 5
     2,17  2,18  0,0 0,0 0,0
     0  0  .4  .56  1
```

```
*VIEW FACTOR BOX
     PARAMETERS  11
     1  10  12  392.  1  3  2  4  6  5  1
     PARAMETERS 6
     7   4  0  3.5  3.5  392.
*
*************************************************************
*                ZONE 2 --- EAST PERIMETER                  *
*************************************************************
*
UNIT 20 TYPE 19  ZONE 2
     PARAMETERS 10
     2 2 4.704E04 0 0 0 2.82E04 7 73. .009
     INPUTS 11
     1,11  3,1  25,17  25,13  25,18  0,0  18,1  0,0  18,2
     18,3  0,0
     75.  .01   55.  5.64E04  .0083  0  0  4  0  0  7.5
*NORTH WALL
     PARAMETERS 7
     1  1  120.  .4  .6  2  11
     INPUTS 1
     2,6
     0
*SOUTH WALL
     PARAMETERS 7
     2 1  120.  .4  .6  2  11
     INPUTS 1
     2,14
     0
*WEST WALL
     PARAMETERS 7
     3  3  3920.  .74  .26  3  23
     INPUTS 3
     21,11  21,1  0,0
     0  75.  0
*EAST WALL
     PARAMETERS 7
     4  1  3920.  .4  .6  2  11
     INPUTS 1
     2,11
     0
*CEILING
     PARAMETERS 7
     5  2  4704.  .74  .26  3  39
*FLOOR
     PARAMETERS 7
     6  2  4704.  .4  .6  3  35
*EAST GLAZING
     PARAMETERS 8
     7  5  1372.  1  .4  1.  1  6
```

```
     INPUTS 5
     2,11  2,12  0,0 0,0 0,0
     0  0  .4  .56  1
*VIEW FACTOR BOX
     PARAMETERS  11
     1  10  12  392.  1  4  2  3  6  5  1
     PARAMETERS 6
     7   4  0  3.5  3.5  392.
*
************************************************************
*                ZONE 3 --- BUILDING CORE                  *
************************************************************
*
UNIT 21  TYPE 19 CORE ZONE
     PARAMETERS 10
     2  2  2.352E05  0  0  0  1.411E05  6  73.  .009
     INPUTS  11
     1,11  3,1  24,1  25,14  31,1  0,0  18,4  0,0  18,5
     18,6  0,0
     75.   .01   55.   3.97E04   .0083  0  0   4   0   0
     7.5
*NORTH WALL
     PARAMETERS 7
     1  1  600.  .4  .6  2  11
     INPUTS  1
     2,6
     0
*SOUTH WALL
     PARAMETERS  3
     2  -1  600.
     INPUTS  1
     2,14
     0
*EAST WALL
     PARAMETERS  7
     3  3  3920.  .74  .26  3  23
     INPUTS  3
     20,11  20,1  0,0
     75.  73.  0
*WEST WALL
     PARAMETERS  7
     4  3  3920.  .74  .26  3  23
     INPUTS  3
     19,11  19,1  0,0
     75.  73.  0
*CEILING
     PARAMETERS  7
     5  2  2.352E04  .74  .26  3  39
*FLOOR
     PARAMETERS  7
```

```
     6  2  2.352E04  .4  .6  3  35
*VIEW FACTOR BOX
     PARAMETERS 11
     1  10.  60.  392.  1  3  2  4  6  5  0
*
UNIT 22  TYPE 32   COOLING COIL WEST
*
     PARAMETERS 6
     4  32  35.  62.4  2  2
     INPUTS 6
     19,1  32,2  25,8  26,6  25,10  0,0
     55.   45.  9.E04  47.   2.5E04  .009
*
UNIT 23  TYPE 32   COOLING COIL EAST
*
     PARAMETERS 6
     4  32  35.  62.4  2  2
     INPUTS 6
     20,1  33,2  25,9  26,6  25,11  0,0
     55.    45.  9.E04  47.  2.5E04  .009
*
UNIT 24  TYPE 32  COOLING COIL MAIN
*
     PARAMETERS 6
     6  28  40.2  62.4  2  2
     INPUTS 6
     21,1  34,2  25,5  26,6  25,6  0,0
     55.   45.  9.E04  47.  2.5E04  .009
*
UNIT 25  TYPE 6  HVAC  CONTROLLER
*
     PARAMETERS 5
     .15  .15  .7  60.  .5
     INPUTS 24
     1,11  21,1  21,2  3,1  19,1  20,1  22,1  29,1  23,1
     30,1  24,1  31,1  22,4  23,4  24,4  18,7  19,2
     20,2  26,6  26,9  0,0  28,4  28,5  28,6
     70. 75. .009 .009 75. 75. 55. .008 55. .008
     55.  .008  58.  58.  58.  0.  .007  .007  47. 3.E06
     245.  55.  55.  55.
*
UNIT 26  TYPE 37  YORK 550-TON CHILLER
*
     PARAMETERS 29
      .66E07 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0
      .16027725E05 -.54109760E03 .426480E01 -.42346677E03
      .28358068E01
      .15273896E02 -.133075 -.10464374 .9597943E-03
      .194 .516 .356 1. 3413. .001  50.
      1.  .20  .7  0.
```

```
     INPUTS 6
     28,7  25,20  25,19  27,1  28,3  28,8
     1     59.    2.5E05 76.   48.   9.E05
*
UNIT 27  TYPE 22  TWO CELL COOLING TOWER
*
     PARAMETERS 12
     -.393 1.12  1.60E04  1.60E05   8.10E04  40.  7. 2.
      0.  1.2E04  35.  1.
     INPUTS 8
     26,7  26,8   3,2  1,11    0,0  28,8   28,1  28,2
     65.  1.08E07  55.  70.   14.7  9.E05  1    1
*
UNIT 28  TYPE 7  HVAC  CONTROLLER
*
      PARAMETERS 5
      1.  58.  8.00  430.  1.1
      INPUTS 14
      1,12  27,1  1,11 25,20 26,6 26,9
      28,1 28,2 28,3 28,4 28,5 28,6 26,2 28,7
       70.  70.   75.  58.  47.  3.E06
        1    1   48.  55.  55.  55.  150. 1.
*
UNIT 29 TYPE 33 PSYCH CALCS.
*
      PARAMETERS 2
      1 2
      INPUTS 2
     22,1 22,2
      70   50
* OUT(1)=W, OUT(2)=TWB
*
UNIT 30 TYPE 33 PSYCH CALCS.
*
      PARAMETERS 2
      1 2
      INPUTS 2
     23,1  23,2
      70   50
* OUT(1)=W, OUT(2)=TWB
*
UNIT 31 TYPE 33 PSYCH CALCS.
*
      PARAMETERS 2
      1 2
      INPUTS 2
     24,1  24,2
      70   50
* OUT(1)=W, OUT(2)=TWB
*
```

```
UNIT 32 TYPE 33  PSYCH CALCS.
*
     PARAMETERS 2
     4  2
     INPUTS  2
     19,1  19,2
     55.   .008
* OUT(1) = RH  OUT(2) = TWB
*
UNIT 33 TYPE 33  PSYCH CALCS.
*
     PARAMETERS 2
     4  2
     INPUTS  2
     20,1  20,2
     55.  .008
* OUT(1) = RH  OUT(2) = TWB
*
UNIT 34 TYPE 33  PSYCH CALCS.
*
     PARAMETERS 2
     4  2
     INPUTS  2
     21,1  21,2
     55.  .008
* OUT(1) = RH  OUT(2) = TWB
*
UNIT 36  TYPE 25  PRINTER 1
*
     PARAMETERS 4
     1.00  0 1 13
     INPUTS 6
     25,7  25,19  26,2  27,1  28,3   25,20
     FANKW MWTR   CHKW  CWS  CHWSET  TRET
*
UNIT 37  TYPE 25  PRINTER 2
*
     PARAMETERS 4
     1.00  0 1 13
     INPUTS 6
     28,1 28,2  27,2  28,7 1,11  28,10
     FN1   FN2  CTFKW NCHIL TDB  LOAD
*
UNIT 38  TYPE 25   PRINTER 3
*
      PARAMETERS 4
     1.00  0 1 13
      INPUTS 6
     19,1    20,1    21,1    26,6  1,12  28,4
     TZONE1  TZONE2  TZONE3  CHWSUP TWB  TSET3
```

```
*
UNIT 39  TYPE 28  SIMULATION SUMMARY
*
      PARAMETERS 13
      24  0   672 13  2  0 -4 0 -4 0 -4 0 -4
      INPUTS 4
      27,2  25,7  26,2 28,10
      LABELS 4
      TOWER  FANKW  CHILLER LOAD
END
```

APPENDIX B

--HVAC Controller Component

--Building to HVAC Equipment
Linking Component

--Centrifugal Chiller Component

--Cooling Tower Component

```
C*************************************************************
C*     HVAC CONTROLLER FOR MAKING SET POINT CHANGES AND      *
C*     OTHER CONTROL DECISIONS                               *
C*************************************************************
C
      SUBROUTINE TYPE7(TIME,XIN,OUT,TT,DTDT,PAR,INFO)
C
      DIMENSION XIN(14),OUT(10),TT(1),DTDT(1),PAR(5)
      DIMENSION INFO(10)
      REAL LOAD
      INFO(6) = 10
      MODE = IFIX(PAR(1))
      CHWRFX = PAR(2)
      VARYT = PAR(3)
      MAXKW = PAR(4)
      LIMIT = PAR(5)
      TWB = XIN(1)
      TCWS = XIN(2)
      TDB = XIN(3)
      CHWR = XIN(4)
      CHWS = XIN(5)
      LOAD = XIN(6)/12000.
      NFAN1 = XIN(7)
      NFAN2 = XIN(8)
      CHWST = XIN(9)
      TSET3 = XIN(10)
      TSET2 = XIN(11)
      TSET1 = XIN(12)
      CHKW  = XIN(13)
      NCHIL = XIN(14)
C
C     PARAMETER AND VARIABLE DEFINITIONS
C
C     MODE   = CONTROL MODE FOR THE SIMULATION
C     CHWRFX = IBM STRATEGY FOR THE SUPPLY TEMPERATURE ON
C              THE RETURN TEMPERATURE ( F.)
C     VARYT  = TIME INCREMENT AT WHICH CONTROL DECISIONS
C              ARE MADE ( HOURS OR FRACTIONS THEREOF)
C     TWB    = AMBIENT WET BULB TEMPERATURE ( F.)
C     TCWS   = CONDENSER WATER SUPPLY TEMPERATURE ( F.)
C     TDB    = AMBIENT DRY BULB TEMPERATURE ( F.)
C     CHWR   = CHILLED WATER RETURN TEMPERATURE ( F.)
C     CHWS   = CHILLED WATER SUPPLY TEMPERATURE ( F.)
C     LOAD   = SITE CHILLED WATER LOAD (TONS)
C     NFAN1  = COOLING TOWER FAN STATUS (FAN #1)
C     NFAN2  = COOLING TOWER FAN STATUS (FAN #2)
C     CHWST  = CHILLED WATER SET POINT TEMPERATURE ( F.)
C     TSET3  = ZONE 3 AHU SUPPLY AIR SET TEMPERATURE ( F.)
C     TSET2  = ZONE 2 AHU SUPPLY AIR SET TEMPERATURE ( F.)
C     TSET1  = ZONE 1 AHU SUPPLY AIR SET TEMPERATURE ( F.)
```

```
C     APP    = APPROACH TEMPERATURE ( F.)
C     CHKW   = CHILLER POWER CONSUMPTION (KW)
C     NCHIL  = NUMBER OF OPERATING CHILLERS
C     LIMIT  = RATIO OF THE SWITCH POINT LOAD TO THE CURRENT
C              LOAD ABOVE WHICH TWO CHILLER OPERATION MUST
C              CONTINUE
C     CFLOW  = CONDENSER WATER FLOW RATE (LBM/HR)
C
C     VARIABLE TIME INCREMENT FOR MAKING CONTROL
C     DECISIONS
C
      DELT = AMOD(TIME,24.)
      IF(DELT .GE. 7. .AND.  DELT .LE. 8. ) GO TO 30
      IF(AMOD(DELT,VARYT).GT. 0.001) GO TO 500
30    CONTINUE
C
C     END TIME CHECK
C
      IF(INFO(7) .GT. 10) GO TO 500
      IF(MODE .EQ. 1) GO TO 100
      IF(MODE .EQ. 2) GO TO 200
      IF(MODE .EQ. 3) GO TO 300
      IF(MODE .EQ. 4) GO TO 400
      IF(MODE .EQ. 5) GO TO 200
C
C     DEFINITIONS OF THE VARIOUS MODES OF OPERATION
C
C     MODE 1 --- Current IBM control
C     MODE 2 --- Optimal Supply Air Set
C     MODE 3 --- Optimal Chilled Water Supply Temperature
C     MODE 4 --- Optimal Cooling Tower Fan Status
C     MODE 5 --- Combination of MODE 2,3,&4
C
C     FAN STATUS DEFINITIONS
C
C     0 = FAN OFF
C     1 = FAN LOW SPEED
C     2 = FAN HIGH SPEED
C
100   APP = -.333333 * TWB  +  28.3333
      TAPP = TWB + APP
      DIF = TCWS - TAPP
      IF(DIF.GT.3.)THEN
      NFAN1 = 2
      NFAN2 = 2
      GO TO 10
      END IF
      IF(DIF.LT. -3.)THEN
      NFAN1 = 0
      NFAN2 = 0
```

```
      GO TO 10
      END IF
      IF(DIF.GT. 1.5)THEN
      NFAN1 = 1
      NFAN2 = 2
      GO TO 10
      END IF
      IF(DIF.LT. -1.5)THEN
      NFAN1 = 1
      NFAN2 = 0
      GO TO 10
      END IF
      NFAN1 = 1
      NFAN2 = 1
10    CONTINUE
C
C     SUPPLY AIR SET TEMPERATURE
C
      IF(TDB .GT. 54.) TSET3 = 55.
      IF(TDB .GT. 10. .AND.  TDB .LE. 54.)
     +TSET3 = -.2955 * TDB + 70.955
      IF(TDB .LE. 10.) TSET3 = 68.
      TSET1 = -.5625 * TDB + 118.4375
      TSET2 = TSET1
C
C     CHILLED WATER SUPPLY TEMPERATURE
C
      DELR = CHWR - CHWRFX
      DELS = -.67 * DELR
      CHWST = CHWS + DELS
      IF(CHWST .GT. 48.) CHWST = 48.
      IF(CHWST .LT. 42.) CHWST = 42.
      GO TO 500
C
C     MODE 2
C
C     SUPPLY AIR SET TEMPERATURE
C
200   IF(LOAD .LT. 400.) TSET3 = 59.
      IF(LOAD .GE. 400. .AND.  LOAD .LT. 425.)
     +TSET3 = -.04 * LOAD + 76.
      IF(LOAD .GE. 425. .AND.  LOAD .LT. 450.)
     +TSET3 = -.12 * LOAD + 110.
      IF(LOAD .GE. 450. .AND.  LOAD .LT. 500.) TSET3 = 56.
      IF(LOAD .GE. 500. .AND.  LOAD .LT. 425.)
     +TSET3 = .04 * LOAD + 36.
      IF(LOAD .GE. 525. .AND.  LOAD .LT. 550.
     +    .AND.  TWB .GT. 64.) TSET3 = 57.
      IF(LOAD .GE. 525. .AND.  LOAD .LT. 550.
     +    .AND.  TWB .LT. 64.) TSET3 = .04 * LOAD + 36.
```

```
      IF(LOAD .GE. 550. .AND.  LOAD .LT. 575.
     +   .AND.  TWB .GT. 64.) TSET3 = .04 * LOAD + 35.
      IF(LOAD .GE. 550. .AND.  LOAD .LT. 575.
     +   .AND.  TWB .LE. 64.)
     +TSET3 = 58.
      IF(LOAD .GE. 575. .AND.  LOAD .LT. 600.) TSET3 = 58.
      IF(LOAD .GE. 600.) TSET3 = .04 * LOAD + 34.
      IF( TSET3 .GT. 59. ) TSET3 = 59.
      TSET1 = -.5625 * TDB + 118.4375
      TSET2 = TSET1
      IF( MODE .EQ. 2 ) GO TO 500
C
C     MODE 3
C
C     CHILLED WATER SUPPLY TEMPERATURE
C
300   CHWST = 48.
      IF( MODE .EQ. 3 ) GO TO 500
C
C     MODE 4
C
C     COOLING TOWER FAN STATUS
C
400   NFAN1 = 1
      NFAN2 = 1
      TWB1 = -.138 * LOAD + 157.25
      TWB2 = -.14  * LOAD + 160.5
      TWB3 = -.16  * LOAD + 176.
      IF ((TWB2 - TWB) .GT. 0. .AND.  (TWB1 - TWB) .LT. 0.)
     +THEN
      NFAN1 = 1
      NFAN2 = 2
      END IF
      IF ((TWB3 - TWB) .GE. 0. .AND.  (TWB2 - TWB) .LE. 0.)
     +THEN
      NFAN1 = 2
      NFAN2 = 2
      END IF
C
C
C DETERMINATION OF THE NUMBER OF OPERATING CHILLERS
C
500   IF(NCHIL .EQ. 2.) GO TO 550
      IF(DELT .GT. 7. .AND.  DELT .LT. 8.)
       NCHIL = 1.
       CFLOW = 9.E05
       GO TO 600
      END IF
      IF(CHKW .LT.  MAXKW)
       NCHIL = 1.
```

```
         CFLOW = 9.E05
         GO TO 600
        END IF
        IF(COUNT1 .EQ. 1) LMAX = LOAD
        COUNT1 = COUNT1 + 1
        COUNT2 = 1
        NCHIL = 2.
        CFLOW = 1.8E06
        GO TO 600
550     IF((LMAX/LOAD) .GT.  LIMIT)
         COUNT1 = 1
         COUNT2 = COUNT2 + 1
         GO TO 600
        END IF
        COUNT2 = COUNT2 + 1
        COUNT1 = 1
        NCHIL = 1.
        CFLOW = 9.E05
C
C OUTPUTS
C
600     OUT(1) = NFAN1
        OUT(2) = NFAN2
        OUT(3) = CHWST
        OUT(4) = TSET3
        OUT(5) = TSET2
        OUT(6) = TSET1
        OUT(7) = NCHIL
        OUT(8) = CFLOW
        OUT(9) = LMAX
        OUT(10) = LOAD
        RETURN
        END
```

```
C***********************************************************
C*       THIS PROGRAM IS USED IN A BUILDING TRNSYS          *
C*       SIMULATION TO LINK THE BUILDING ZONE COMPONENTS    *
C*       AND THE COOLING COILS.  INCLUDED IN THE ALGORITHM  *
C*       ARE PROVISIONS FOR THE CONTROL OF AIR AND WATER    *
C*       FLOW RATES TO THE ZONES AND THE COOLING COILS.     *
C***********************************************************
C
C        VARIABLES AND PARAMETER DEFINITIONS:
C
C     PER(1,2,3)  = FRACTION OF SUPPLY AIR TO EACH ZONE
C     FLPWR       = FULL LOAD AHU SUPPLY FAN POWER (KW)
C     TOL         = INTERNAL CONVERGENCE TOLERANCE
C     PLR(1,2,3)  = PART LOAD RATIO FOR EACH ZONE
C     TOA         = OUTSIDE AIR DRY BULB TEMPERATURE (F)
C     WOA         = OUTSIDE AIR HUMIDITY RATIO
C     HZONE3      = ZONE3 AIR ENTHALPY (BTU/LBM)
C     HOUT        = OUTSIDE AIR ENTHALPY (BTU/LBM)
C     TZ1         = RETURN AIR TEMP ZONE 1 (F)
C     TZ2         = RETURN AIR TEMP ZONE 2 (F)
C     TZ3         = RETURN AIR TEMP ZONE 3 (F)
C     WZ1         = RETURN AIR HUMIDITY RATIO ZONE 1
C     WZ2         = RETURN AIR HUMIDITY RATIO ZONE 2
C     WZ3         = RETURN AIR HUMIDITY RATIO ZONE 3
C     TMIXR       = MIXED RETURN AIR TEMPERATURE ( F.)
C     WMIXR       = MIXED RETURN AIR HUMIDITY RATIO
C     TSET1       = SUPPLY AIR SET FOR PERIMETER AHU ZONE 1
C     TSET2       = SUPPLY AIR SET FOR PERIMETER AHU ZONE 2
C     TSET3       = SUPPLY AIR SET FOR CORE AHU - ZONE 3
C     MWTR(1,2,3) = CHILLED WATER MASS FLOW TO ZONE AHU
C                   COOLING COIL (PER COIL LBM/HR)
C     MWTOT       = TOTAL CHILLED WATER MASS FLOW (LBM/HR)
C     MAIR(1,2,3) = AIR MASS FLOW TO ZONE (LBM/HR)
C     MFAN(1,2,3) = AIR MASS FLOW TO ZONE AHU (LBM/HR)
C     CONTRL      = OPERATING STATUS OF MAIN AHU
C                   (0 = OFF, 1 = ON)
C     TWSET       = CHILLED WATER SUPPLY TEMPERATURE ( F.)
C     QCHIL       = SITE CHILLED WATER LOAD (BTU/HR)
C     FANPWR      = TOTAL AHU SUPPLY FAN POWER (KW)
C
      SUBROUTINE TYPE6(TIME,XIN,OUT,TT,DTDT,PAR,INFO)
C
      DIMENSION XIN(24),OUT(20),TT(1),DTDT(1),PAR(5)
      DIMENSION INFO(10)
      REAL MWTR1,MWTR2,MWTR3,MWTOT,MAIR1,MAIR2,MAIR3,MCRM
      REAL MFAN1,MFAN2,MFAN3
      INFO(6) = 21
      PER1 = PAR(1)
      PER2 = PAR(2)
      PER3 = PAR(3)
```

```
      FLPWR = PAR(4)
      TOL = PAR(5)
9     TOA = XIN(1)
      TZ3 = XIN(2)
      WZ3 = XIN(3)
      WOA = XIN(4)
      TZ1 = XIN(5)
      TZ2 = XIN(6)
      TSA1 = XIN(7)
      WSA1 = XIN(8)
      TSA2 = XIN(9)
      WSA2 = XIN(10)
      TSA3 = XIN(11)
      WSA3 = XIN(12)
      TRZ1 = XIN(13)
      TRZ2 = XIN(14)
      TRZ3 = XIN(15)
      CONTRL = XIN(16)
      WZ1 = XIN(17)
      WZ2 = XIN(18)
      TWSET = XIN(19)
      QCHIL = XIN(20)
      TSET3 = XIN(22)
      TSET2 = XIN(23)
      TSET1 = XIN(24)
C
C
C     ENTHALPY CONTROLLER AND DAMPER SET
C
      PLR1 = (.1314 * TZ1 - 9.2514) * PER1
      PLR2 = (.1314 * TZ2 - 9.2514) * PER2
      PLR3 = (.1371 * TZ3 - 9.6971) * PER3
      IF(PLR1.LT.(.2*PER1)) PLR1 = .2 * PER1
      IF(PLR2.LT.(.2*PER2)) PLR2 = .2 * PER2
      IF(PLR3.LT.(.2*PER3)) PLR3 = .2 * PER3
      IF(PLR1.GT.PER1) PLR1 = PER1
      IF(PLR2.GT.PER2) PLR2 = PER2
      IF(PLR3.GT.PER3) PLR3 = PER3
      PLR = PLR1 + PLR2 + PLR3
      TMIXR = (TZ1*PLR1 + TZ2*PLR2 + TZ3*PLR3)/PLR
      WMIXR = (WZ1*PLR1 + WZ2*PLR2 + WZ3*PLR3)/PLR
      HZONE3 = .24*TMIXR + WMIXR*(1061. + .444*TMIXR)
      HOUT = .24*TOA + WOA*(1061. + .444*TOA)
      DIFF = (HZONE3 - HOUT)/HZONE3
      IF(TOA.LT.55.) MODE = 3
      POA = (TSET3 - TMIXR)/(TOA - TMIXR)
      IF(MODE.EQ.3) GO TO 10
      IF(DIFF.LT.-.03) MODE = 1
      IF(DIFF.GT..03) MODE = 2
      IF(DIFF.GE.-.03.AND.DIFF.LE..03) MODE = MODE
```

```
      IF(MODE.EQ. 1) POA = .20
      IF(MODE.EQ. 2) POA = 1.00
10    TMIX3 = POA*TOA + (1.-POA)*TMIXR
      WMIX3 = POA*WOA + (1.-POA)*WMIXR
      VFAN3 = 2.52E-02 * (TMIX3 + 460.) *(1. + 1.6078*WMIX3)
      QFAN3 = PLR * 57500.
      MFAN3 = QFAN3 * 60./VFAN3/3.
      IF(CONTRL.EQ.1.) AHU3ON = AHU3ON + 1.
      IF(CONTRL.EQ.0.) AHU3ON = 0.
      IF(AHU3ON.GT.1.) GO TO 20
      QWTR3 = PLR * 240.
      IF(PLR .GT. 1.) QWTR3 = 240.
      MWTR3 = QWTR3 * 500./3.
      GO TO 30
20    IF(ABS(TSA3-TSET3).LT.TOL) GO TO 30
      DELWTR = -((TSET3 - TSA3)*MFAN3*.2404)/(TRZ3 - TWSET)
      MWTR3 = MWTR3 + DELWTR
      IF(MWTR3 .GT. 4.E04) MWTR3 = 4.E04
30    CONTINUE
C
C     NOTE: THE COEFFICIENTS IN THE FOLLOWING EQUATION
C           ARE SPECIFIC TO A VAV FAN.  SOURCE BLAST II.
C
      FANPWR = 4.*(.51650734 - .78387254*PLR +
     +          1.2598*PLR**2) * FLPWR
C
C
C
C AIR AND WATER FLOW CALCULATIONS FOR PERIMETER ZONES
C
C
      IF((TZ1-78.).LT.TOL) GO TO 50
      IF(INFO(7).GE.1) GO TO 40
         MFAN1 = 9.0E04
         MWTR1 = 2.5E04
         GO TO 60
40    IF(ABS(TSA1 - TSET1).LT.TOL) GO TO 60
         MFAN1 = 9.0E04
      DELWTR = -((TSET1 - TSA1)*MFAN1*.2404)/(TRZ1 - TWSET)
         MWTR1 = MWTR1 + DELWTR
         GO TO 60
50    MFAN1 = 1000.
      MWTR1 = 0.0
60    CONTINUE
      IF((TZ2-78.).LT.TOL) GO TO 80
      IF(INFO(7).GE.1) GO TO 70
         MFAN2 = 9.0E04
         MWTR2 = 2.5E04
         GO TO 90
70    IF(ABS(TSA2 - TSET2).LT.TOL) GO TO 90
```

```
         MFAN2 = 9.0E04
      DELWTR = -((TSET2 - TSA2)*MFAN2*.2404)/(TRZ2 - TWSET)
         MWTR2 = MWTR2 + DELWTR
         GO TO 90
80    MFAN2 = 1000.
      MWTR2 = 0.0
90    CONTINUE
      IF(CONTRL.EQ.0.) MFAN3 = 1.
      IF(CONTRL.EQ.0.) FANPWR = 0.
      IF(CONTRL.EQ.0.) MWTR3 = 0.
      MAIR1 = MFAN1*2. + PLR1*12.*MFAN3
      MAIR2 = MFAN2*2. + PLR2*12.*MFAN3
      TSZ1 = (MFAN1*2.*TSA1 + 12.*PLR1*MFAN3*TSA3)/MAIR1
      WSZ1 = (MFAN1*2.*WSA1 + 12.*PLR1*MFAN3*WSA3)/MAIR1
      MAIR2 = MFAN2*2. + PLR2*12.*MFAN3
      TSZ2 = (MFAN2*2.*TSA2 + 12.*PLR2*MFAN3*TSA3)/MAIR2
      WSZ2 = (MFAN2*2.*WSA2 + 12.*PLR2*MFAN3*TSA3)/MAIR2
      MAIR3 = PLR3*12.*MFAN3
      MCRM = 2.5E05
      MWTOT = MCRM + MWTR1*2. + 2.*MWTR2 + 12.*MWTR3
      TCRM = XIN(21) * 24.*500./MCRM + TWSET
      TRET = (MCRM*TCRM + 2.*MWTR1*TRZ1 + 2.*MWTR2*TRZ2 +
     +        MWTR3*12.*TRZ3) / MWTOT
      IF(MFAN1.EQ.1.0) TSZ1 = 55.
      IF(MFAN1.EQ.1.0) WSZ1 = .008
      IF(MFAN2.EQ.1.0) TSZ2 =  55.
      IF(MFAN2.EQ.1.0) WSZ2 = .008
      IF(MFAN3.EQ.1.) TMIX3 = TZ3
      IF(MFAN3.EQ.1.) WMIX3 = WZ3
      IF(MFAN3.EQ.1.) MFAN3 = 9.0E04
      QHVAC = MWTOT * (TRET - TWSET)
      IF(ABS((QHVAC - QCHIL)/QHVAC) .LT. .01) GO TO 100
      MWTR3 = (QCHIL/QHVAC) * MWTR3
      MWTR2 = (QCHIL/QHVAC) * MWTR2
      MWTR1 = (QCHIL/QHVAC) * MWTR1
      MWTOT = MCRM + MWTR1*2. + 2.*MWTR2 + 12.*MWTR3
      TRET = (MCRM*TCRM + 2.*MWTR1*TRZ1 + 2.*MWTR2*TRZ2 +
     +        MWTR3*12.*TRZ3) /MWTOT
100   OUT(1) = MODE
      OUT(2) = POA
      OUT(3) = TMIX3
      OUT(4) = WMIX3
      OUT(5) = MFAN3
      OUT(6) = MWTR3
      OUT(7) = FANPWR
      OUT(8) = MFAN1
      OUT(9) = MFAN2
      OUT(10) = MWTR1
      OUT(11) = MWTR2
      OUT(12) = MAIR1/9.
```

```
OUT(13) = MAIR2/9.
OUT(14) = MAIR3/9.
OUT(15) = TSZ1
OUT(16) = WSZ1
OUT(17) = TSZ2
OUT(18) = WSZ2
OUT(19) = MWTOT
OUT(20) = TRET
RETURN
END
```

```
C***********************************************************
C*   THIS ROUTINE MODELS THE PERFORMANCE OF CENTRIFUGAL    *
C*   CHILLERS USING EMPIRICAL CURVE FITS AS DESCRIBED IN   *
C*   STOECKER(1971).                                       *
C***********************************************************
C
      SUBROUTINE TYPE37(TIME,XIN,OUT,TT,DTDT,PAR,INFO)
C
      DIMENSION XIN(6),OUT(9),TT(1),DTDT(1),PAR(29),INFO(10)
      REAL LCH,LCHMX,MODE,KWA,KWFL,KWR,LTOT,LCOND,KWTOEN
      IF(INFO(7) .GT. 0) GO TO 9
      PLRO = OUT(4)
      KWO = OUT(2)
9     IF(INFO(7) .GT. -1)GO TO 10
      INFO(6) = 9
C COEFFICIENTS C1-C9 ARE FROM A BIQUADRATIC CURVE FIT
C OF THE CHILLER CAPACITY AS A FUNCTION OF (CHWSET,TCWR).
C THE GENERAL FORM OF THE BIQUADRATIC CURVE FIT IS:
C     Y = C1 + C2*X1 + C3*X1**2 + C4*X2 + C5*X2**2 +
C         C6*X1*X2 + C7*X1**2*X2 + C8*X1*X2**2 +
C         C9*X1**2*X2**2
C
      C1 = PAR(1)
      C2 = PAR(2)
      C3 = PAR(3)
      C4 = PAR(4)
      C5 = PAR(5)
      C6 = PAR(6)
      C7 = PAR(7)
      C8 = PAR(8)
      C9 = PAR(9)
C PARAMETERS W1-W9 ARE FROM A BIQUADRATIC CURVE FIT
C OF THE CHILLER POWER CONSUMPTION (KW) AS A
C FUNCTION OF (CHWSET,TCWR).
      W1 = PAR(10)
      W2 = PAR(11)
      W3 = PAR(12)
      W4 = PAR(13)
      W5 = PAR(14)
      W6 = PAR(15)
      W7 = PAR(16)
      W8 = PAR(17)
      W9 = PAR(18)
C PARAMETERS E1-E3 ARE FROM A QUADRATIC FIT OF KWR
C (ACTUAL KW/FULL-LOAD KW) AS A FUNCTION OF
C PLR (ACTUAL LOAD/FULL LOAD).
      E1 = PAR(19)
      E2 = PAR(20)
      E3 = PAR(21)
      CP = PAR(22)
```

```
C THE PARAMETER "KWTOEN" CONVERTS KW TO ENERGY UNITS
C FOR DETERMINING THE CONDENSOR LOAD.
      KWTOEN = PAR(23)
C PARAMETER "TOL" IS THE CONVERGENCE TOLERENCE FOR
C DETERMINING THE LEAVING CONDENSOR TEMPERATURE.
      TOL = PAR(24)
C
C VARIABLE DEFINITIONS:
C        CHWSET = CHILLED WATER SUPPLY TEMPERATURE SETTING
C        CHWSUP = ACTUAL CHILLED WATTER SUPPLY TEMPERATURE
C        TCWR   = LEAVING CONDENSOR WATER TEMPERATURE
C        TCWS   = ENTERING CONDENSOR WATER TEMPERATURE
C        NCH    = NUMBER OF CHILLERS
C        TCHIN  = ENTERING CHILLED WATER TEMPERATURE
C        FLOWP  = FLOWRATE OF CHILLED WATER
C        FLOWC  = FLOWRATE OF CONDENSOR WATER
C        KWA    = TOTAL CHILLER POWER DEMAND
C        KWFL   = FULL-LOAD CHILLER POWER DEMAND
C        KWR    = RATIO OF ACTUAL DEMAND TO FULL-LOAD
C        CAP    = FULL-LOAD CHILLER CAPACITY
C        PLR    = PART-LOAD RATIO:ACTUAL LOAD/FULL LOAD
C        LCH    = LOAD PER CHILLER
C        LTOT   = TOTAL CHILLER LOAD
C        LCOND  = CONDENSOR HEAT REJECTION RATE
10    NCH = INT(XIN(1))
      TCHIN = XIN(2)
      FLOWP = XIN(3)
      TCWS = XIN(4)
      CHWSET= XIN(5)
      FLOWC = XIN(6)
      KWA = 0.
      KWFL = 0.
      CAP = 0.
      PLR = 0.
      IF(FLOWP .LT. 1. .OR.  NCH .LT. 1 .OR.  FLOWC .LT. 1.)
     +GO TO 100
C DETERMINE CHILLER LOADING.
      LCH = (FLOWP/NCH) * CP * (TCHIN-CHWSET)
C
C DETERMINE THE OFFSET FROM THE DESIRED SET POINT.
C OFFSET = F( PLR )
C
C     NIT  = NUMBER OF ITERATIONS
C     BP   = CHILLER OFFSET MODE BY-PASS (BY-PASS FOR BP
C            NOT EQUAL TO 0.)
C     TLIM = CHILLER STARTUP TRANSIENT TIME LIMIT
C     PLIM = CHANGE IN CHILLER PLR NECESSARY TO INITIATE
C            CHILLER THROTTLING
C     A    = THROTTLE LIMIT
C
```

```
      NIT = IFIX(PAR(25))
      TLIM = PAR(26)
      PLIM = PAR(27)
      A = PAR(28)
      BP = PAR(29)
      K = 0
      IF(BP .NE. 0.) GO TO 20
      DELSET = 0.
      GO TO 21
20    PLR1 = LCH/(550. * 12000.)
      IF(PLR1 .LT. 1.) DELSET = 4.657*PLR1 - 4.657
      IF(PLR1 .GE. 1. .AND.  PLR1 .LE. 1.1) DELSET = 0.
      IF(PLR1 .GT. 1.1) DELSET = 20.*PLR1 - 22.
21    CHWSUP = CHWSET + DELSET
      LCHN = (FLOWP/NCH) * CP * (TCHIN-CHWSUP)
      IF(ABS((LCHN - LCH)/LCH) .LT. .01) GO TO 30
      LCH = LCHN
      K = K + 1
      IF(K.LT.NIT) GO TO 20
      LCH = (FLOWP/NCH)*CP*(TCHIN-CHWSET)
      CHWSUP = CHWSET
      WRITE(*,*) 'THERE MAY BE A PROBLEM WITH THE
     +            CONVERGENCE OF CHWSUP'
      WRITE(*,*) TIME
30    CONTINUE
      LCH = (FLOWP/NCH)*CP*(TCHIN-CHWSET)
      LTOT = LCH * NCH
C
C FIND CHILLER FULL-LOAD CHARACTERISTICS.
C
      T = CHWSUP
C
C AN ITERATIVE ROUTINE IS NEEDED TO FIND THE CHILLER
C CHARACTERISTICS BECAUSE OF THE DEPENDENCE ON THE CONDENSOR
C WATER RETURN TEMPERATURE, WHICH IS ITSELF DEPENDENT ON
C THE CHILLER OPERATION.
C
      TC = TCWS + 15.
      DO 50 I=1,100
      CAP = C1 + C2*T + C3*T**2 + C4*TC + C5*TC**2 +
     +      C6*T*TC + C7*T**2*TC + C8*T*TC**2 +
     +      C9*T**2*TC**2
      KWFL = W1 + W2*T + W3*T**2 + W4*TC + W5*TC**2 +
     +       W6*T*TC + W7*T**2*TC + W8*T*TC**2 +
     +       W9*T**2*TC**2
      PLR = LCH / CAP
      KWR = E1 + E2*PLR + E3*PLR**2
      KWA = KWR * KWFL * NCH
      LCOND = LTOT + KWA*KWTOEN
      TCWR =  TCWS + LCOND/(FLOWC*CP)
```

```
      RDIFF = ABS((TC-TCWR)/TCWR)
C BRANCH OUT IF CONVERGED WITHIN SPECIFIED TOLERENCE.
      IF(RDIFF .LT.  TOL)GO TO 100
      TC = TCWR
50    CONTINUE
      IF(PLR.LT.1.) GO TO 100
70    K = K + 1
      IF(K.EQ.1) START = TIME
       IF(K.EQ.1) PLRMAX = PLR
      IF(ABS(START-TIME).GT.TLIM) GO TO 90
      DELPLR = (PLR-PLRO)/PLR
      IF(DELPLR.LT.PLIM) GO TO 90
      PLR = (PLRMAX - A)*(TIME-START)/TLIM + A
      LTOT = PLR * NCH * CAP
      T = CHWSUP
      TC = TCWS + 15.
      DO 80 I=1,100
      CAP = C1 + C2*T + C3*T**2 + C4*TC + C5*TC**2 +
     +      C6*T*TC + C7*T**2*TC + C8*T*TC**2 +
     +      C9*T**2*TC**2
      KWFL = W1 + W2*T + W3*T**2 + W4*TC + W5*TC**2 +
     +       W6*T*TC + W7*T**2*TC + W8*T*TC**2 +
     +       W9*T**2*TC**2
      PLR = LCH/CAP
      KWR = E1 + E2*PLR + E3*PLR**2
      KWA = KWR * KWFL * NCH
      LCOND = LTOT + KWA*KWTOEN
      TCWR =  TCWS + LCOND/(FLOWC*CP)
      RDIFF = ABS((TC-TCWR)/TCWR)
C BRANCH OUT IF CONVERGED WITHIN SPECIFIED TOLERENCE.
      IF(RDIFF .LT.  TOL)GO TO 100
      TC = TCWR
80    CONTINUE
      GO TO 100
90    K = 0
      IF(ABS(START-TIME) .GT.  TLIM) GO TO 70
100   OUT(1) = KWFL
      OUT(2) = KWA
      OUT(3) = CAP
      OUT(4) = PLR
      OUT(5) = FLOWP
      OUT(6) = CHWSUP
      OUT(7) = TCWR
      OUT(8) = LCOND
      OUT(9) = LTOT
      RETURN
      END
```

```
C************************************************************
C*    THIS PROGRAM CALCULATES THE PERFORMANCE OF A           *
C*    COOLING TOWER(INLET AND OUTLET WATER TEMPERATURES)     *
C*    USING A METHOD DEVELOPED BY AUSTIN WHILLIER.  THE      *
C*    METHOD WAS PRESENTED IN AN ARTICLE ENTITLED "A FRESH *
C*    LOOK AT THE CALCULATION OF PERFORMANCE OF COOLING      *
C*    TOWERS", ASHRAE TRANSACTIONS, V.82, PART 1, 1967.      *
C************************************************************
C
      SUBROUTINE TYPE22(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
      DIMENSION XIN(16),OUT(15),PAR(12),INFO(10),T(1),
     +          DTDT(1), NFS(10),TT(10),TTI(10),TW(10),
     +          TF(10)
      COMMON /A/F,FLOW,TWB,TDB,TCWR,P,RNGE,AA,BB
      REAL LTOWER,MAKEUP,MAKUPH,MAKUPL,MAKUPO,LOFANP,MC,
     +     MSUMP
      IF(INFO(7).GT.-1) GO TO 10
C
C DEFINITION OF INPUT PARAMETERS:
C
C        FANOF  = FAN FLOW PER CELL WITH FAN OFF (CFM)
C        FANHI  = FAN FLOW PER CELL ON HIGH SPEED (CFM)
C        FANLO  = FAN FLOW PER CELL ON LOW SPEED (CFM)
C        HIFANP = FAN POWER ON HIGH (KW)
C        LOFANP = FAN POWER ON LOW (KW)
C        CP     = SPECIFIC HEAT OF WATER (BTU/LBM-F)
C        NCELLS = TOTAL NUMBER OF CELLS IN TOWER
C        MC     = MASS OF WATER IN CELL SUMP (LBM)
C        MSUMP  = MASS OF WATER IN COMMON SUMP (LBM)
C        AA,BB  = COEFFICIENTS OF LINEAR REGRESSION OF
C                 EFFECTIVENESS VERSUS R-FACTOR DATA
C                 E = AA(R) + BB
C        TCWMIN = MINIMUM CONDENSER WATER SUPPLY
C                 TEMPERATURE ( F.)
      AA = PAR(1)
      BB = PAR(2)
      FANOF = PAR(3)*60.
      FANHI = PAR(4)*60.
      FANLO = PAR(5)*60.
      HIFANP = PAR(6)
      LOFANP = PAR(7)
      NCELLS = INT(PAR(8))
      MC = PAR(9)
      MSUMP = PAR(10)
      TCWMIN = PAR(11)
      CP = PAR(12)
      INFO(9) = 0
      INFO(6) = 15
C
```

```
C INPUT DEFINITIONS:
C        TCWR   = CONDENSOR WATER RETURN TEMPERATURE ( F.)
C        LTOWER = TOTAL TOWER HEAT REJECTION RATE (BTU/HR)
C        TWB    = AMBIENT WET-BULB TEMPERATURE ( F.)
C        TDB    = AMBIENT DRY-BULB TEMPERATURE ( F.)
C        P      = AMBIENT BAROMETRIC PRESSURE(PSIA ABSOLUTE)
C        NCELL  = NUMBER OF CELLS WITH WATER FLOW, ASSUMED
C                 THAT FLOW IS EQUALLY DISTRIBUTED AMONG
C                 CELLS.
C        NFANHI = NUMBER OF CELLS ON HIGH FAN SPEED
C        NFANLO = NUMBER OF CELLS ON LOW FAN SPEED
C
C EXPLANATION OF OTHER VARIABLES:
C
C        FLOWC  = TOTAL CONDENSOR WATER FLOW (LBM/HR)
C        FLOW   = PER CELL CONDENSOR WATER FLOW (LBM/HR)
C        RNGE   = CONDENSOR WATER RANGE (TIN - TOUT)
C        FANPWR = TOTAL FAN POWER DEMAND (KW)
C        MAKEUP = TOTAL MAKEUP WATER FLOW (GPM)
C
10    IF(INFO(7).NE.0)GO TO 15
      LIMIT = 0
      TSI = OUT(1)
      DO 12 I=1,NCELLS
12       TTI(I) = OUT(5+I)
15    TCWR = XIN(1)
      LTOWER = XIN(2)
      TWB = XIN(3)
      TDB = XIN(4)
      P    = XIN(5)
      FLOWC = XIN(6)
      IF(INFO(7).EQ.-1)THEN
         TSI = TDB
         DO 18 I=1,NCELLS
18          TTI(I) = TDB
      END IF
      NFANHI = 0
      NFANLO = 0
      NFANOF = 0
C POSSIBLE CONTROL INPUTS FOR EACH CELL:
C     -1 = NO FLOW
C      0 = FLOW, FAN OFF
C      1 = FLOW, LO FAN
C      2 = FLOW, HI FAN
C
      DO 20 I=1,NCELLS
         NFS(I) = INT(XIN(6+I))
         IF(NFS(I).EQ.2)NFANHI = NFANHI+1
         IF(NFS(I).EQ.1)NFANLO = NFANLO+1
         IF(NFS(I).EQ.0)NFANOF = NFANOF+1
```

```
20    CONTINUE
      NCELL = NFANHI+NFANLO+NFANOF
      FNC = FLOAT(NCELL)
      FLOW = FLOWC/FNC
      RNGE = LTOWER/(FLOWC*CP)
C
C FIND CURRENT TOWER WATER TEMPERATURES BASED ON CURRENT
C TCWR
C
50    IF(NFANOF.GT.0)CALL TOWER(FANOF,TW(1),MAKUPO)
      IF(NFANLO.GT.0)CALL TOWER(FANLO,TW(2),MAKUPL)
      IF(NFANHI.GT.0)CALL TOWER(FANHI,TW(3),MAKUPH)
C
C FIND THE STEADY-STATE SUPPLY TEMPERATURE FOR LIMIT CHECK.
C
      TCWSSS = (FLOAT(NFANLO)*TW(2)+FLOAT(NFANHI)*TW(3)+
     +       FLOAT(NFANOF)*TW(1))/FNC
C
C IF MINIMUM TEMPERATURE LIMIT HAS NOT BEEN HIT, GO ON
C TO THE SUMP CALCULATIONS
C
      IF(TCWSSS.GT.TCWMIN.AND.LIMIT.EQ.0)GO TO 300
C
C IF MINIMUM IS HIT THE FIRST TIME, ALL CELLS ARE STARTED
C WITH FAN OFF (LOWEST LEVEL).  THEN LEVEL IS INCREASED STEP
C BY STEP UNTIL THE TCWS AGAIN DROPS BELOW THE MINIMUM.
C THIS INDICATES THAT THE TOWER SHOULD CYCLE BETWEEN THE
C CURRENT LEVEL AND THE PREVIOUS ONE.
C
      IF(TCWSSS.LT.TCWMIN.AND.LIMIT.EQ.0)THEN
         LIMIT = 1
         NFANOF = NCELL
         NFANLO = 0
         NFANHI = 0
         GO TO 50
      END IF
C
C CHECK TO SEE IF LATEST LEVEL IS BELOW MINIMUM - BRANCH
C TO CYCLING CALCULATIONS IF SO.
C
      IF(TCWSSS.LT.TCWMIN.AND.LIMIT.EQ.1)GO TO 150
C
C OTHERWISE, RAISE LEVEL A STEP AND CONTINUE.
C
      MKUPOP = MAKUPO
      MKUPLP = MAKUPL
      MKUPHP = MAKUPH
      NFANOP = NFANOF
      NFANLP = NFANLO
      NFANHP = NFANHI
```

```
      TCWSP = TCWSSS
      IF(NFANOF.GT.0)THEN
      NFANOF = NFANOF-1
      NFANLO = NFANLO+1
      GO TO 50
      ELSE
      IF(NFANHI.EQ.NCELL)GO TO 300
      NFANLO = NFANLO-1
      NFANHI = NFANHI+1
      END IF
      GO TO 50
C
C CURRENT LEVEL IS TOO COLD, PREVIOUS LEVEL WAS NOT: MIX
C THE TWO TO OBTAIN THE DESIRED TEMPERATURE.
C
150   RUNL = (TCWMIN-TCWSP)/(TCWSSS-TCWSP)
      FANPL = FLOAT(NFANHI)*HIFANP+FLOAT(NFANLO)*LOFANP
      MAKEL = FLOAT(NFANHI)*MAKUPH+FLOAT(NFANLO)*MAKUPL+
     +        FLOAT(NFANOF)* MAKUPO
      FANPL1 = FLOAT(NFANHP)*HIFANP+FLOAT(NFANLP)*LOFANP
      MAKEL1 = FLOAT(NFANHP)*MKUPHP+FLOAT(NFANLP)*MKUPLP+
     +         FLOAT(NFANOP)*MKUPOP
      FANPWR = RUNL*FANPL + (1.-RUNL)*FANPL1
      MAKEUP = RUNL*MAKEL + (1.-RUNL)*MAKEL1
      DO 170 K=1,3
170      TW(K) = TCWMIN
300   IF(MC.GT.0.)THEN
         A = -FLOW/MC
         DO 310 I=1,NCELLS
            IF(NFS(I).GE.0)THEN
               B = (FLOW/MC)*TW(NFS(I)+1)
               CALL DIFFEQ(TIME,A,B,TTI(I),TF(I),TT(I))
            ELSE
               TT(I) = TTI(I)
            END IF
310      CONTINUE
      ELSE
         DO 320 I=1,NCELLS
            IF(NFS(I).GE.0)THEN
               TT(I) = TW(NFS(I)+1)
               TF(I) = TT(I)
            ELSE
               TT(I) = TTI(I)
               TF(I) = TT(I)
            END IF
320      CONTINUE
      END IF
C
C FIND AVERAGE WATER TEMP LEAVING THE CELLS
C
```

```
      TTA = 0.
      DO 330 I=1,NCELLS
         IF(NFS(I).EQ.-1)GO TO 330
         TTA = TTA + (TT(I)/FNC)
330   CONTINUE
      IF(MSUMP.GT.0.)THEN
         A = -FLOWC/MSUMP
         B = (FLOWC/MSUMP)*TTA
         CALL DIFFEQ(TIME,A,B,TSI,TSF,TCWS)
      ELSE
         TCWS = TTA
         TSF = TCWS
      END IF
      IF(LIMIT.EQ.0)THEN
         FANPWR = FLOAT(NFANHI)*HIFANP + FLOAT(NFANLO)*
     +            LOFANP
         MAKEUP = FLOAT(NFANHI)*MAKUPH + FLOAT(NFANLO)*
     +            MAKUPL + FLOAT(NFANOF)*MAKUPO
      END IF
      OUT(1) = TCWS
      OUT(2) = FANPWR
      OUT(3) = MAKEUP
      OUT(4) = TSF
      OUT(5) = LIMIT
      DO 450 I=1,NCELLS
450      OUT(5+I) = TF(I)
      END
      SUBROUTINE TOWER(FAN,TCWS,MAKEUP)
      COMMON /A/F,FLOW,TWB,TDB,TCWR,P,RNGEC,A,B
C
C EXPLANATION OF PARAMETERS:
C
C        FLOW   = WATER FLOW (LBM/HR)
C        FAN    = AIR FLOW (CU FT/HR)
C        MAKEUP = MAKEUP WATER FLOW (GPM)
C        WARA   = ACTUAL RATIO OF WATER FLOW TO AIR FLOW
C        WARR   = REFERENCE WATER TO AIR RATIO
C        ETAW   = TOWER WATER EFFICIENCY
C
      REAL MAKEUP
      RNGE = RNGEC
      CALL  SIG(TWB,TDB,P,SI,VI,WI)
      DO 50 I=1,100
      TO= TCWR
      DO 40 M=1,100
C
C    LOOP FOR DETERMINING THE STATE OF THE OUTLET AIR.
C    USED TO DETERMINE THE MASS FLOW OF AIR.
C
         CALL SIG(TO,TO,P,SO,VO,WO)
```

```
          FUN = FAN*(SO-SI)/VO-FLOW*1.003*RNGE
          CALL SIG(TO+.1,TO+.1,P,SH,VH,WH)
          CALL SIG(TO-.1,TO-.1,P,SL,VL,WL)
          FH = FAN*(SH-SI)/VH-FLOW*1.003*RNGE
          FL = FAN*(SL-SI)/VL-FLOW*1.003*RNGE
          DFDT= (FH-FL)/.2
          TN = TO - FUN/DFDT
          IF(ABS(TN-TO).LT..01) GO TO 500
          IF(M.EQ.100)WRITE(*,1)
1         FORMAT('0','*** COOLING TOWER AIR LOOP DID NOT
     +                        CONVERGE ***')
          TO= TN
40        CONTINUE
500    G = FAN/VO
       WARA= FLOW/G
C
C     ACTUAL WATER-TO-AIR RATIO.
C
C LOOP FOR DETERMINING THE ACTUAL TOWER WATER TEMPERATURES
C
         CALL SIG(TCWR,TCWR,P,S3,V3,W3)
         WARR= (S3-SI)/(TCWR-TWB)
         R    = WARA/WARR
         RR   = R
         IF(RR.LT.1.) GO TO 520
         RR   = 1./RR
520      E = A*RR + B
         IF(E.GT.1.)E=1.
         IF(R.LE.1.) GO TO 530
C WATER EFFICIENCY IS "ETAW".
         ETAW= E/R
         GO TO 540
530      ETAW=E
540    RNGEN = ETAW*(TCWR-TWB)
         IF(ABS(RNGEN-RNGE).LT..01) GO TO 550
          RNGE = RNGEN
         IF(I.EQ.100)WRITE(*,2)
2        FORMAT('0','*** COOLING TOWER WATER LOOP DID NOT
     +                        CONVERGE ***')
50       CONTINUE
550    MAKEUP = G/500.*(WO-WI)
       TCWS = TCWR - RNGE
       RETURN
       END
        SUBROUTINE SIG(TWB,TDB,P,S,V,W)
          T = (TWB-32.)/1.8 + 273.15
          PWS= EXP(-5800.2206/T + 1.3914993 - .04860239*T +
     +         .41764768E-4*T**2 - .14452093E-7*T**3 +
     +         6.545967*LOG(T))
          PWS = PWS*.000145
```

```
       WSST= .62198*PWS/(P-PWS)
      IF(ABS(TDB-TWB).LT..01)GO TO 10
      W = ((1093.-.556*TWB)*WSST - .24*(TDB-TWB))/
     1     (1093.+.444*TDB-TWB)
      V = .3705*(TDB+460.)*(1.+1.6078*W)/P
      H = .24*TDB + W*(1061.+.444*TDB)
      S = H - W*1.003*TWB
      GO TO 20
10     V= .3705*(TWB+460.)*(1.+1.6078*WSST)/P
       H= .24*TWB + WSST*(1061. + .444*TWB)
       S= H - WSST*1.003*TWB
      W = WSST
20    RETURN
      END
```

APPENDIX C

DATA TAPE READING INFORMATION

--IBM to UW Correspondence 3/17/83

--IBM to UW Correspondence 6/21/83

--Listing of Tape Read Program

International Business Machines Corporation

P.O. Box 2150
Atlanta, Georgia 30055
404/238-2000

March 17, 1983

Mr. Richard Hackner
University of Wisconsin - Madison
College of Engineering
Engineering Experiment Station
1500 Johnson Drive
Madison, Wisconsin

Dear Richard,

The enclosed data listings cover a period of February 23, 1983 to March 16,1983. During this period the building was operating in various modes; heating, pay cool, ventilation, occupied and unoccupied. The assortment of data should provide an overview of the HVAC operation during the various modes.

The data consists of the 240 data points requested by Jim Madsen in his letter of January 4, 1983.

The data will be copied to tape and forwarded to you shortly. Each data point record consists of 36 bytes and is formatted in the following way:

DATE:	6 Bytes	3I2	Month, Day, Year
TIME:	4 Bytes	2I2	Hour, Minute
LABEL:	16 Bytes	Alphanumeric	Data Point Name
VALUE:	10 Bytes	Alphanumeric	Point Value/Status

If the data point is an analog point value will be NNNNNNNN.N where N = Numeric Character

If the data point is a status point value will be bbssssssss where b = Blank (HEX 40); s = Alphabetic character

International Business Machines Corporation

P.O. Box 2150
Atlanta, Georgia 30055
404/238-2000

June 21, 1983

Mr. Rich Hackner
College of Engineering
University of Wisconsin - Madison
1500 Johnson Drive
Madison, WI 53706

Dear Rich,

The three tapes contain data copied from the diskettes, that was collected from the experiments that were conducted during June 1 through June 9, 1983.

The system used to copy the data was set up for bypass label processing. The files on the tape are image copies of the diskette with a tapemark between each file. Block size is 256 bytes. The data starts at Block 33 and continues until Block 2174. Records 1-32 contain information relating to the diskette. Use the file description enclosed with this letter and disregard the previous one I sent you. There was a diskette error on the diskette for 6/3/83 for the time between 13:54 and 15:00. This diskette was not copied to tape.

I hope this information will help you in dumping the tapes.

I enjoyed working with you during the experiments and hope the data collected will be meaningful in the analysis portion of the project.

Sincerely,

Carl E Carlsen

Carl E. Carlsen

CEC:jh

```
C*********************************************
C*                                           *
C*       PROGRAM FOR READING IBM DATA        *
C*               UNIVAC 1100                 *
C*                                           *
C*********************************************
C
C
      IMPLICIT INTEGER (A-Z)
      EXTERNAL IOTPIN
      LOGICAL MATCH
      CHARACTER DATA*1(10,236),FIRST(15),LABEL
      DIMENSION BLOCK(4096),UNBLK(16384),TIME(5)
C
        DATA FIRST/'A','H','U',' ','C','R','N',' ','C',
     +   'H','W','V',' ','S','T'/
      NWORDS = 960
      NVAL   = 236
      LU=10
      LFILE=10
      IL=0
      DREC=NVAL
      IP=36
      NTIME=96
5     CALL FLK$8(IOTPIN,LU,1,BLOCK,NWORDS,LENGTH,$1000,
     +           $2000,$2000)
C
C  MOVE BYTES FROM PACKED FORM TO UNPACKED FORM
C
      DO 100 CHR=1,LENGTH
      J=CHR-1
      IF(J.EQ.0) J=0
      BIT=9*BITS(J,35,2)+1
      WORD=1+(CHR-1)/4
      UNBLK(CHR)=BITS(BLOCK(WORD),BIT,9)
C
      IP=MOD(IP,36)+1
      IF(IP.EQ.1) DREC=MOD(DREC,NVAL)+1
      IF(DREC.EQ.1 .AND.  IP.EQ.1) MATCH=.FALSE.
      IF( IP.GT.10) GO TO 30
C
C  TIME STAMP WRITTEN IN I2 FORMAT, ONLY EVERY
C  SECOND BYTE MEANINGFULL
C
      K=(IP-1)/2+1
      TIME(K)=UNBLK(CHR)
C
C  CONVERSION FROM EBCDIC TO ASCII
C
30    IF(UNBLK(CHR).LT.240 .OR.  UNBLK(CHR).GT.249) GO TO 35
```

```
C
C  NUMBERS
C
      UNBLK(CHR)=UNBLK(CHR)-192
      GO TO 75
35    IF(UNBLK(CHR).LT.193 .OR.  UNBLK(CHR).GT.201) GO TO 40
C
C  LETTERS A-I
C
      UNBLK(CHR)=UNBLK(CHR)-128
      GO TO 75
40    IF(UNBLK(CHR).LT.209 .OR.  UNBLK(CHR).GT.217) GO TO 45
C
C  LETTERS J-R
C
      UNBLK(CHR)=UNBLK(CHR)-135
      GO TO 75
45    IF(UNBLK(CHR).LT.226 .OR.  UNBLK(CHR).GT.233) GO TO 50
C
C  LETTERS S-Z
C
      UNBLK(CHR)=UNBLK(CHR)-143
      GO TO 75
C
C  CHECK FOR BLANK OR DECIMAL POINT
C
50    IF(UNBLK(CHR).EQ.64 .OR.  UNBLK(CHR).NE.75)
     + UNBLK(CHR)=32
      IF(UNBLK(CHR).EQ.75) UNBLK(CHR)=46
C
C  CHECK FOR NEW DATA RECORD
C
75    LABEL=CHAR(UNBLK(CHR))
      IF(LABEL.EQ.FIRST(IL)) GO TO 76
C
      IL=1
      IF(IP.GT.26) GO  TO 80
      GO TO 85
C
76    IF(IL.EQ.15) GO TO 77
      IL=IL+1
      IF(IP.GT.26) GO TO 80
      GO TO 85
C
77    MATCH=.TRUE.
      IL=1
      DREC=1
      IP=25
      GO TO 85
C
```

```
80     DATA(IP-26,DREC)=CHAR(UNBLK(CHR))
C
85     IF(DREC.LT.NVAL .OR.  IP.LT.36 .OR. .NOT.(MATCH))
      +GO TO 100
C
C   OUTPUT TIME STAMP AND DATA VALUES TO FILE
C
       NTIME=MOD(NTIME,96)+1
       IF(NTIME.EQ.1) LFILE=LFILE+1
       WRITE(LFILE,86) (TIME(I),I=1,5),
      +                ((DATA(I,J),I=1,10),J=1,NVAL)
86     FORMAT(1X,5I2,6(1X,10A1)/33(7(1X,10A1)/))
100    CONTINUE
       GO TO 5
1000   PRINT 1001
1001   FORMAT(' END OF FILE DETECTED')
       STOP
2000   PRINT 2001
2001   FORMAT(' SOME SORT OF READ ERROR')
       STOP
       END
```

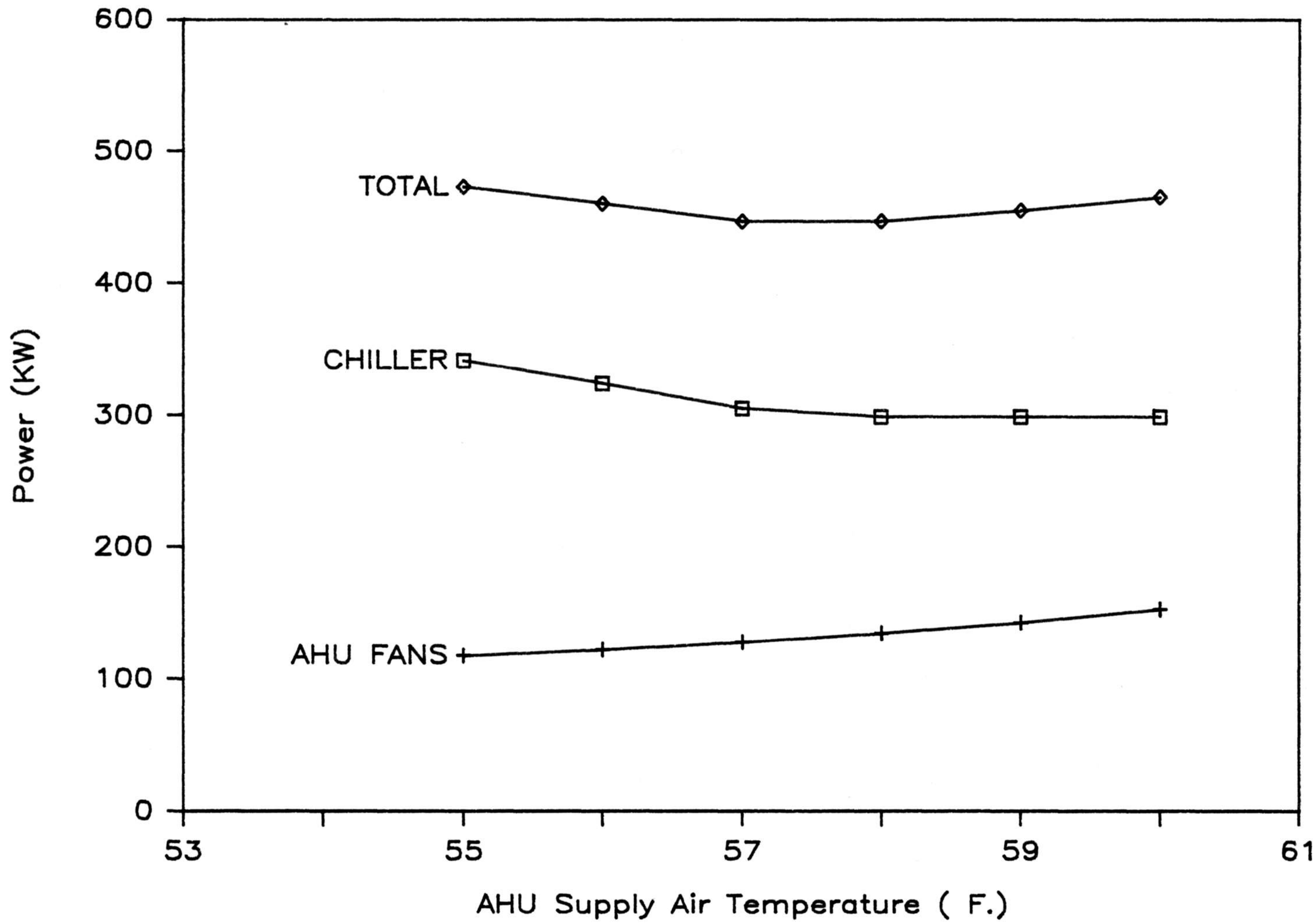


Figure 11 Power use distribution versus AHU supply air temperature. Load = 550 tons and $T_{WB} = 65°F$

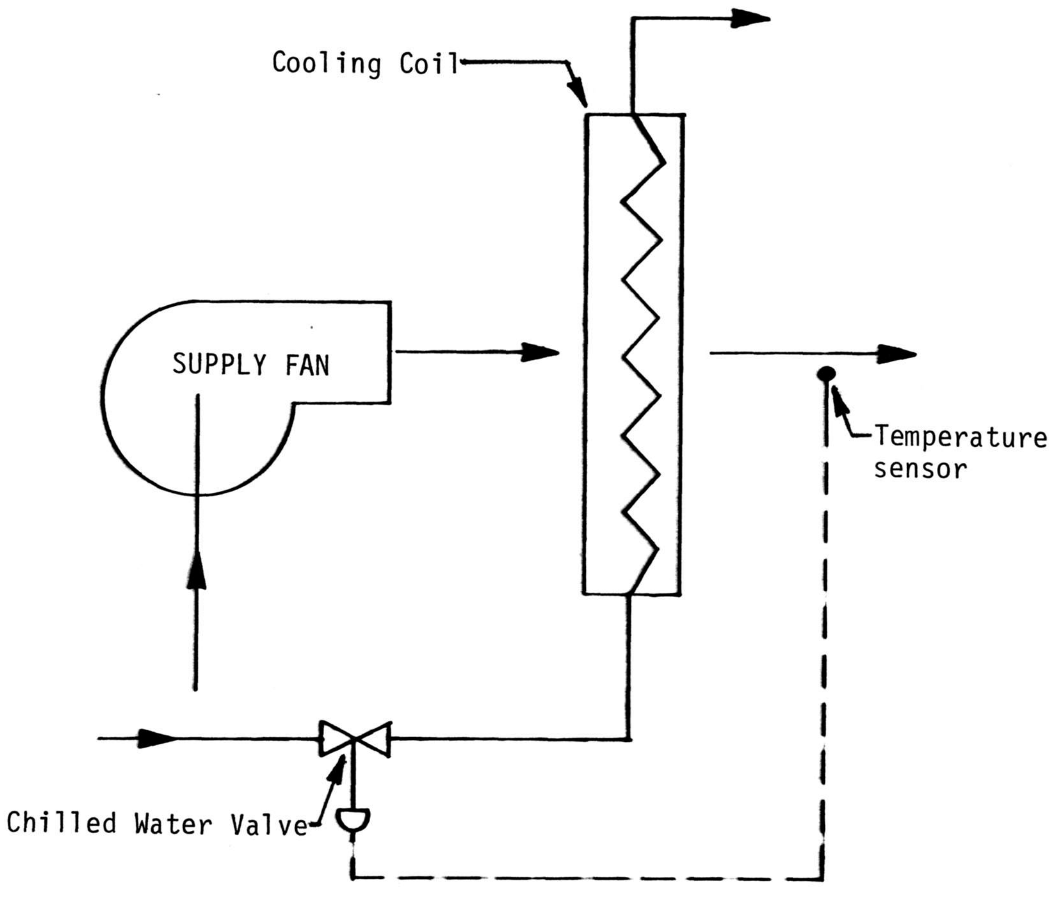


Figure 8 schematic of the cooling coil and fan arrangement for an air handling unit

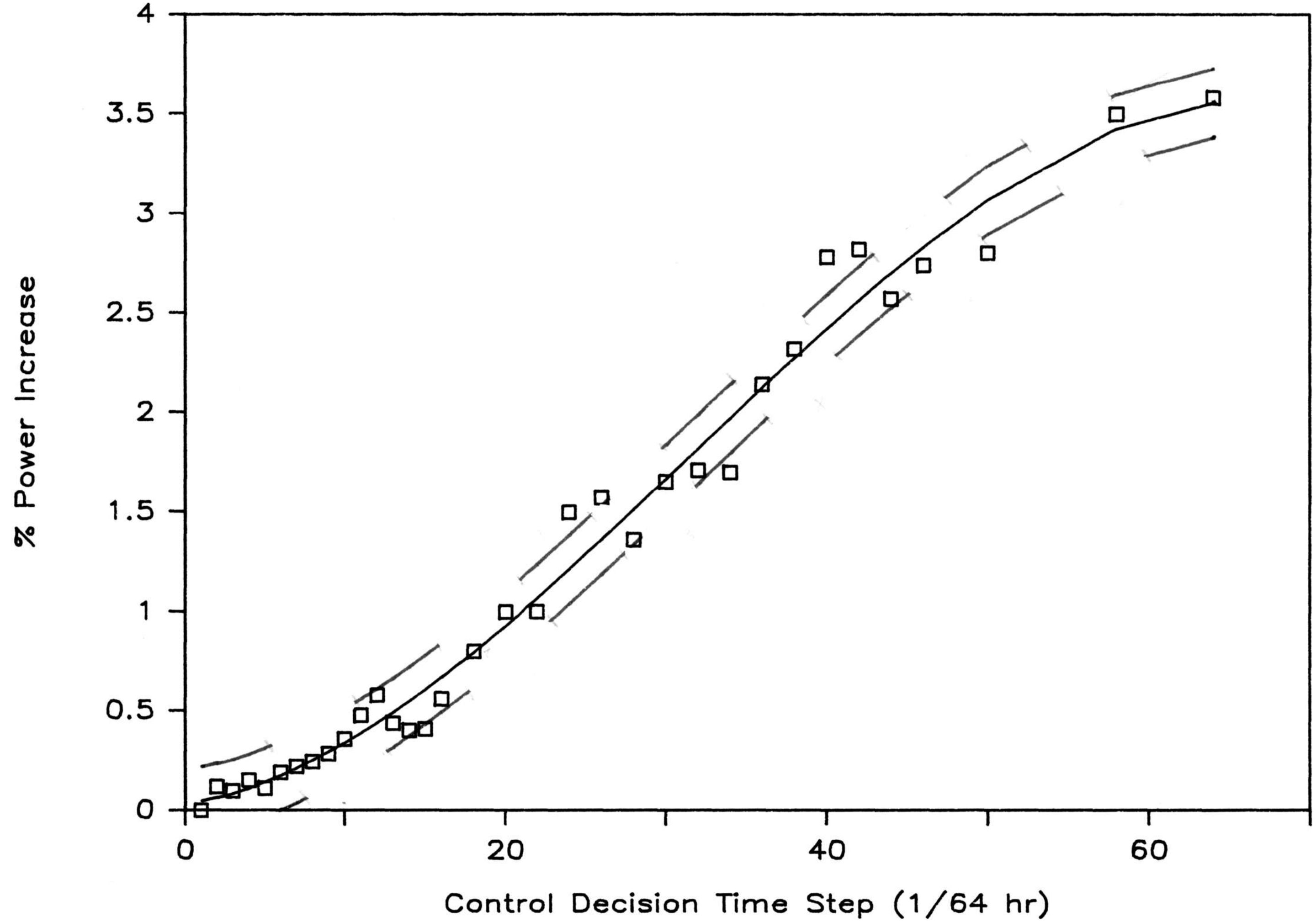


Figure 14 Percent power increase versus the control decision time step size. Results via computer simulation.

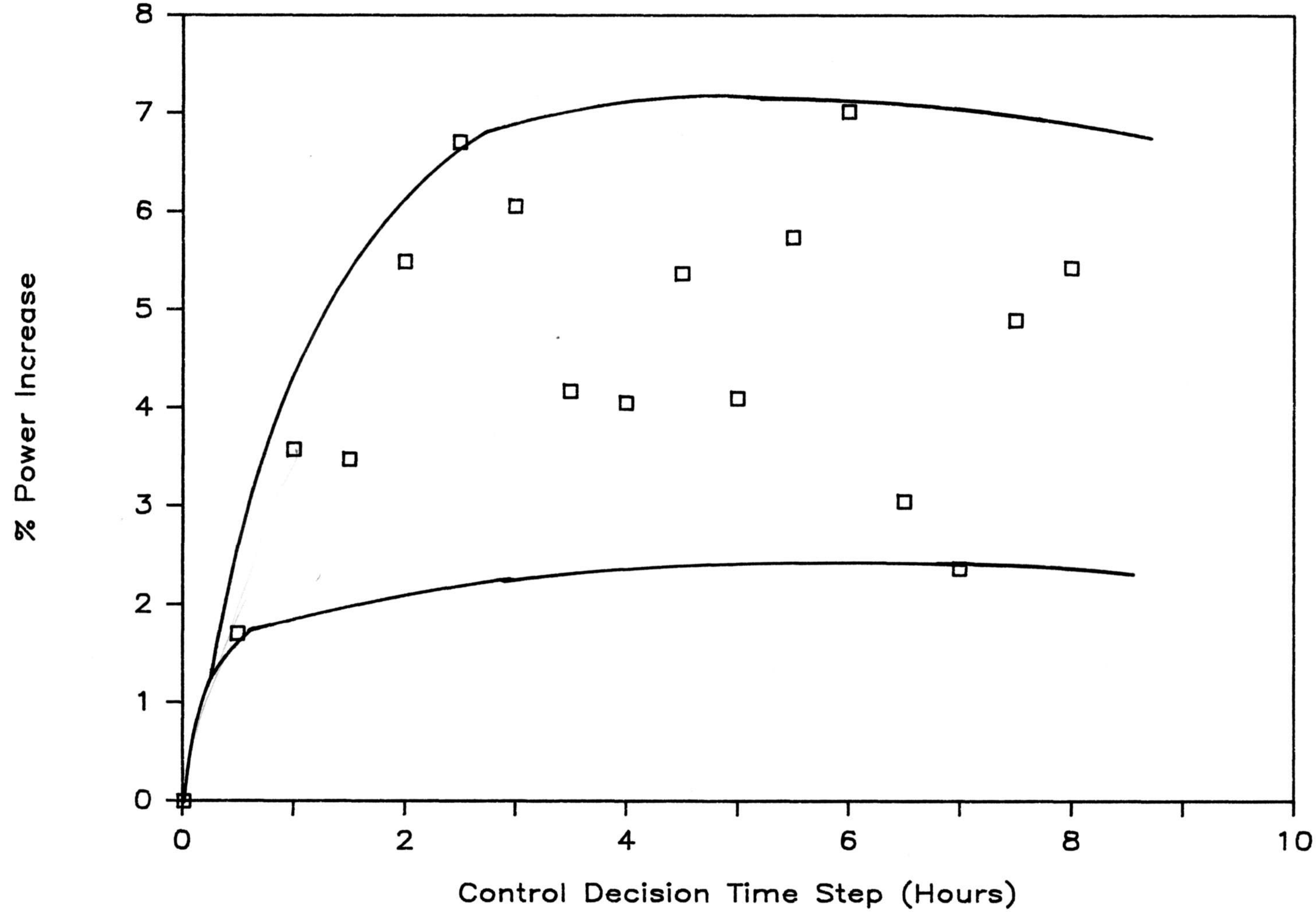


Figure 15 Energy use distribution for current control versus optimal control via ~~sim~~ simulation.

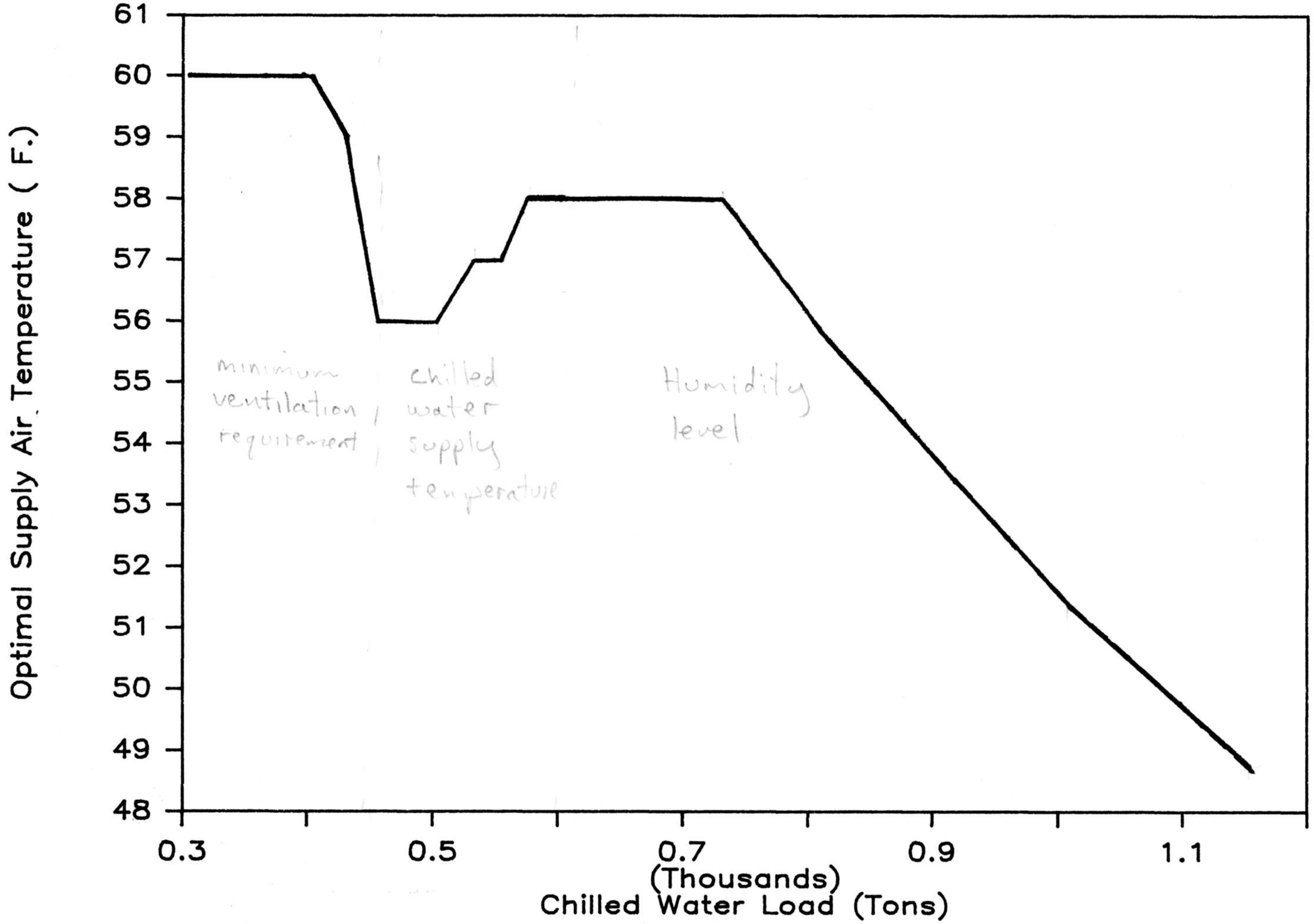
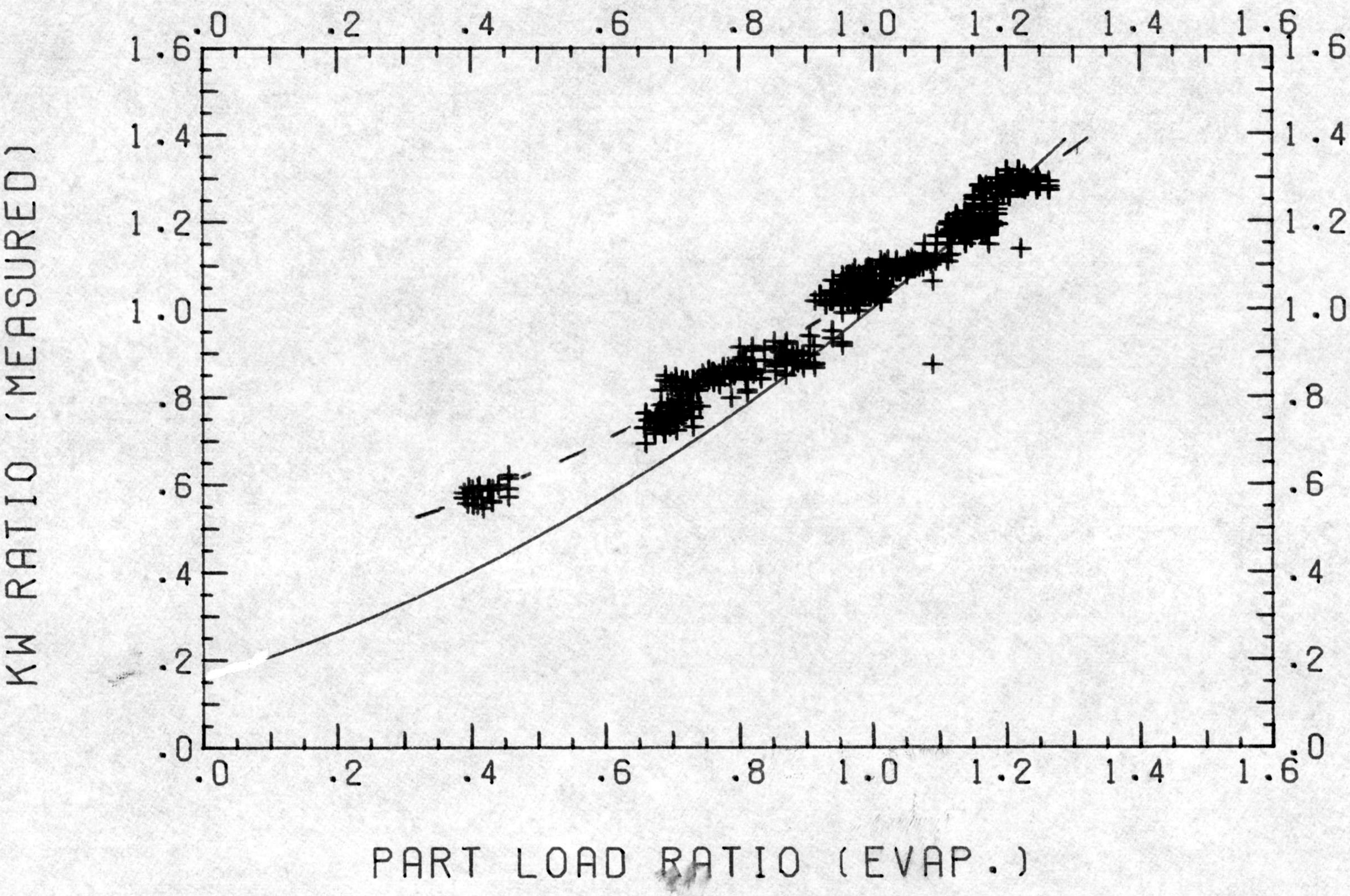
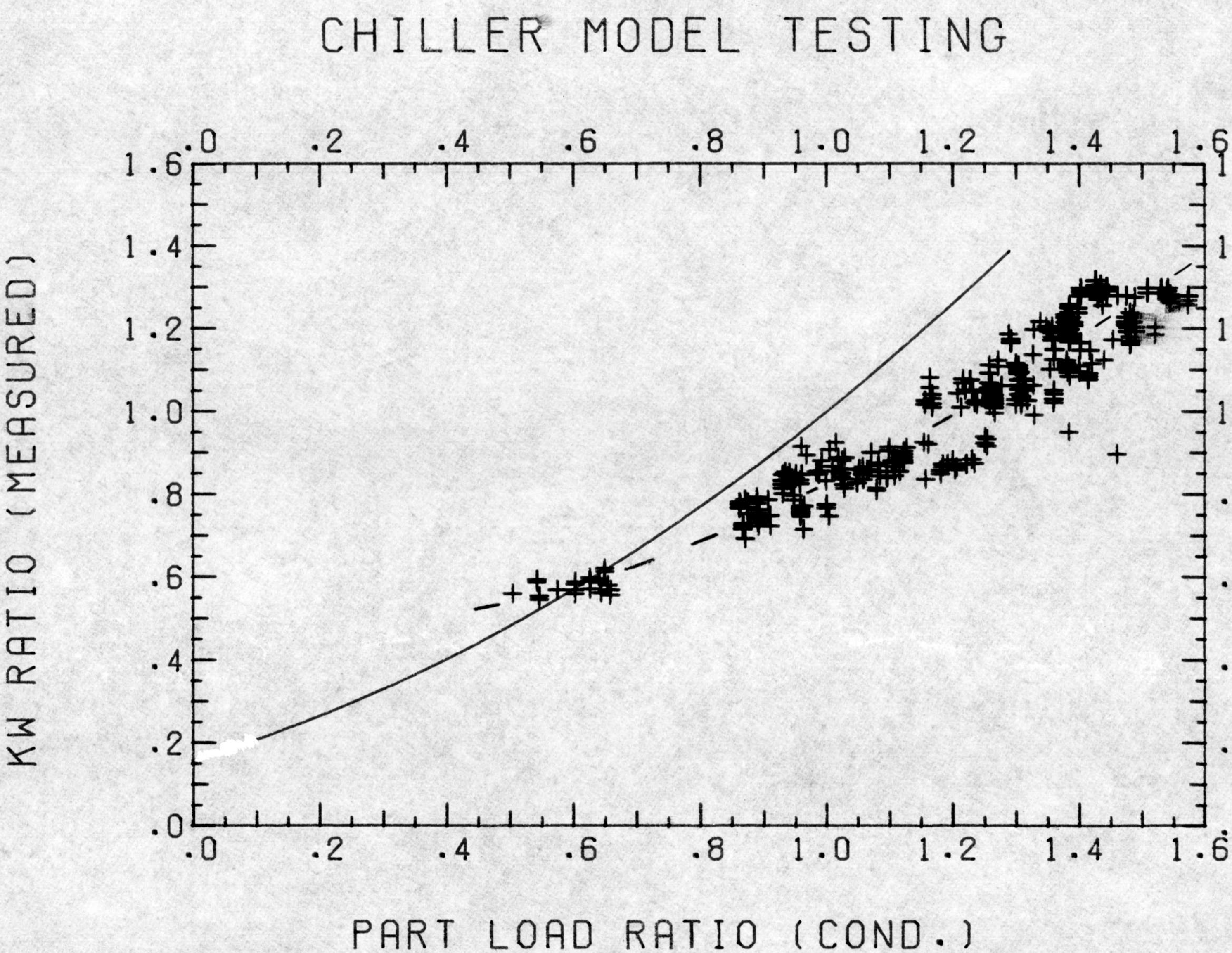


Figure 10 Optimization resUlts for the AHU supply air temperature versus chilled water load. The dominant factor ~~for each division~~ is indicated for each region.

CHILLER MODEL TESTING




CHILLER MODEL TESTING
KW RATIO (MEASURED)
PART LOAD RATIO (COND.)
.0
.2
.4
.6
.8
1.0
1.2
1.4
1.6

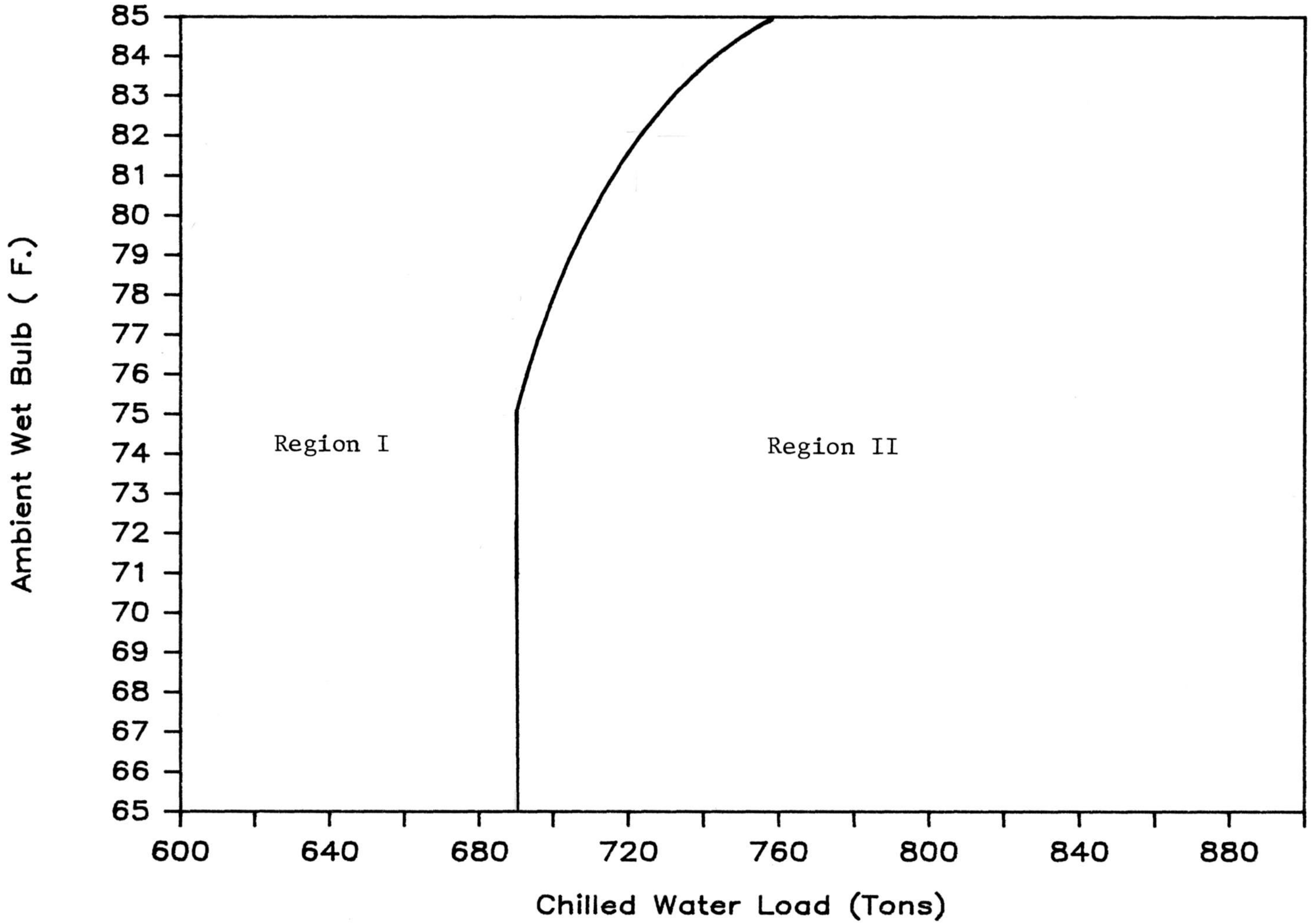


Figure 7 Optimal regions of cooling tower fan operation. Two chiller operation. Region I fan status low-low and Region II low-high.