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HVAC System Dynamics and Energy Use in Existing Buildings—Part I

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ABSTRACT

This paper summarizes the initial findings of the research that was done for the ASHRAE Task Group on Dynamic Response. The project ASHRAE RP 321 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.

The computer algorithms that were developed for the modeling of HVAC equipment (cooling tower, water chiller, air handling unit, etc.) are described. TRNSYS (a computer code developed by the Solar Energy Laboratory at the University of Wisconsin-Madison) was used to link the individual component models together in order to simulate the HVAC system of a 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:

--the cooling tower response to fan speed changes;

--the "flush time" of the chilled water through the system;

--the effects of the building structure due to capacitance.

The transient effects that were not significant (due to their very short time duration) were:

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

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

The end results of this project are the identification and investigation of potential energy saving HVAC operating strategies and the availability of "reliable" equipment models. The final results on dynamic control will be the subject of a following paper. The groups that could benefit the most from these models are building operators, HVAC controls persons, and future researchers.

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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.

In an effort to evaluate aspects relating to the dynamics and control of HVAC systems, ASHRAE and the University of Wisconsin-Madison collaborated on the ASHRAE project RP 321. The goals of the project ("HVAC System Dynamics and Energy Use in Existing Building") are:

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, HVAC operating strategies that will potentially reduce the HVAC system energy consumption.

This paper presents the initial finding of ASHRAE RP 321. Included in the paper are the description of the building HVAC system that was studied, a brief description of the computer models that were developed for the study, a comparison of these models with existing data, and the results of the HVAC equipment response tests. The final results on dynamic control will be the subject of a following paper.

HVAC SYSTEM DESCRIPTION

The HVAC and control systems were evaluated on a 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 sunshades to help reduce the solar load on the building and also tinted glazings.

The HVAC system. as shown in figure 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 (1930 KW) 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 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 (790 KW) 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 (AHU). 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 AHUs are independently controlled. For example, the northwest zone may require heating while the southwest zone requires cooling.

There are two variable-air-volume AHUs (labeled AHU-N and AHU-S in figure 1) that 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 cooling" 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, "dry" outside air is brought into the unit and is cooled by the use of evaporative coolers.

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.

THE ENERGY MANAGEMENT CONTROL SYSTEM (EMCS)

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 with 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 dynamic nature of the EMCS.

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 ranges beyond an acceptable level, 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 indirectly 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.

HVAC EQUIPMENT MODELING

Algorithms and computer codes were developed to model the HVAC system, which consisted of chillers, air-handling units, cooling tower, controllers, etc. The forms of the equipment models were taken from standard ASHRAE algorithms (Stoecker, 1971) and were modified as necessary to include the significant transient effects that were determined from the dynamic response tests.

The chiller performance map shown in figure 2 was developed using the methods outlined in Stoecker (1971). This map, along with the manufacturer's part-load performance curve shown in figure 3, is used to predict the performance of the chiller under various operating conditions. The normal operating range of the chiller is indicated by the shaded region on figure 2.

As an example of the use of these two graphs, suppose that on a particular day the chilled water load is 400 tons (1405 KW). The chilled water supply temperature is set at 46°F (8.1°C) and the condenser water return temperature is 80°F (27°C). These are the three parameters that are necessary in order to determine the power consumption of the chiller.

Entering figure 2 with these operating conditions yields a chiller power consumption of approximately 300 KW at the design load (550 tons (1930 KW)) of the chiller. Using the part-load ratio, PLR, which is the measured chiller tons divided by the design chiller tons, the chiller power ratio (KWR) is determined from figure 3. 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 to determine the predicted chiller power consumption. In this example the predicted chiller power is approximately 200 KW.

The performance map presented in figure 2 is specific to the 550 ton (1930 KW) centrifugal chiller in the Atlanta facility. If it is desired to model a different chiller, a new set of chiller coefficients would have to be determined. The coefficents are the result of a biquadratic curve fit of the chiller manufacturer's data. The form of the fitted equation, Stoecker (1971) is:

$$KWD = C1 + C2*X + C3*X*X + C4*Y + C5*Y*Y + C6*Y*Y + C7*X*X*Y + C8*X*Y*Y + C9*X*X*Y*Y \quad [1]$$

where:

KWD = Chiller power consmpmtion 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 fitted 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.

Figure 4 shows the degree to which the steady-state data correspond 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 (flowmeters, 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 ±15%. The data shown in figure 4 are within the accuracy of the measurements except at the very low part-load ratios. One of the reasons for this discrepancy could be that the chiller is operating in the extrapolate region of figure 2. Manufacturer's data were not available to produce this portion of the performance map.

Figure 5 gives another indication of the accuracy of the measured data. This figure shows the discrepancy between the chilled water load based on the evaporator side and the load based on the condenser side. The chilled water load based on the condenser side is defined as the condenser load minus the chiller power consumption. Theoretically, the evaporator side and the condenser side loads should be equal, but, as the graph indicates, the values differ by 10% to 20%.

The algorithm developed to model the cooling coil of an air-handling unit was also based on the procedures found in Stoecker (1971). The cooling tower model, however, was developed using the method proposed by Austin Whillier (1967). The "Whillier method" combined with cooling tower performance data supplied by the manufacturer provide a relatively simple but accurate method to determine the tower performance at varying conditions of the 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 6 is a schematic representation of a cross-flow, induced draft cooling tower including the sump that 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:

$$\text{Energy in} - \text{Energy out} = \text{Energy stored} \quad [2]$$

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

$$-Mr * cp * (Ta - Tb) = M * cp * dTa/dt \quad [3]$$

solving this equation:

$$Ta = Tb + (Tao - Tbo)\exp[-Mr/(M * t)] \quad [4]$$

where:

Mr = mass flow rate of the condenser water
M = mass of the water in the sump
cp = specific heat of water
Ta = temperature of the water leaving the sump
Tao = initial temperature of the water leaving the sump
Tb = temperature of the water entering the sump. This temperature is a function of the condenser water flow rate, fan speed, and ambient wet bulb temperature.
Tbo = initial temperature of the water entering the sump
t = time

Figures 7 and 8 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 7 shows the model predictions before the transient effect of the change in the temperature of the sump is taken into account. Figure 8 is the same data, but, now the model has included the transient response. The measured data used in figures 7 and 8 were taken at the Charlotte facility and are only used to show the effect of the inclusion of the transient response to the cooling tower model.

Figure 9 is a plot of the cooling tower model prediction versus measured data for the Altanta facility. The cooling tower 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 collected during June at the Atlanta facility. An extensive discussion of the theory and development of the cooling tower model can be found in Lau (1983).

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.

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 set point test. Data were periodically checked following each set-point change to allow the system to restabilize.

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 recorded during each of the tests. The tests that were performed include the response to:

--Chilled water supply set-point adjustments
--Condenser water supply set-point adjustments
--AHU supply air set-point adjustments.

Data were collected during the following system operating conditions:

--Morning start-up;
--Evening shutdown;
--One-chiller operation switching to two chiller operation;
--Outside air dampers closed versus outside air dampers open;
--Building unoccupied mode.

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

--The cooling tower response to fan speed changes;
--The "flush time" of the chilled water through the system;
--The effects of the building structure due to capacitance

The transient effects that are not significant due to their very short time duration are:

--The chiller response to set-point (supply water) changes
--The AHU response to set-point (supply air) changes

The bases for these conclusions are discussed below.

Figures 10 and 11 show the results of a 2°F (1.1°C) step change in the chilled water set-point temperature. As shown in figure 10 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 not felt to be significant for energy use evaluation.

Figure 10 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 temperature begins to drop. Since the building load and the chilled water flow rate remained constant, eventually the temperature difference across the chiller at the end of the test would reach the same value as at the start of the test.

Figures 12 and 13 illustrate the very short time duration of the chiller response. In an effort to present the data on a normalized basis, the change in power divided by the change in chilled water load was plotted versus time in figure 12. The changes are the differences between the chiller power (or load) at the start of a test and the power (or load) at points in time after the test had begun.

Figure 13 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. 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 (1930 KW).

Points 1 and A on figure 13 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 14 and 15. Figure 14 is an indication of the response of the chilled water supply and return temperatures to a change in the AHU supply air temperature. Figure 15 shows the air-side response to the set point change. Here, again, there is a rapid response (on the order of five 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. The results of the cooling tower testing were presented in the previous section on equipment modeling.

SYSTEM SIMULATION AND CONTROL

Simulations were performed with TRNSYS, a component based simulation program, Klein (1981). Using the developed models, a TRNSYS simulation deck was written to verify the chiller-cooling tower subsystem. The simulations were driven by the measured chilled water load and the ambient wet-bulb temperature. Figures 16 through 21 show a sequence of three days in November for which the computer simulation was run. As shown, 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, used to drive the simulation. In this particular case, those measured data were the actual chilled load and the ambient wet-bulb temperature.

Another set of results that was generated using the equipment models and the simulation deck was the determination of the optimum point to switch from one-chiller operation to two-chiller operation. In figure 22 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 optimum "switch point" would be point 1 or an approximate load of 460 tons (1620 KW). However, the lower set of curves indicates a switch point of approximately 700 tons (2460 KW). 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 (1620 KW) the upper curves would indicate that it was time to switch to two-chiller operation. The lower curves, however, show that if such a change in chiller status was made the reduction in COP would be approximately 20% (from point 2 to point 3).

Figure 23 shows what effect the ambient wet-bulb temperature has on the optimum switch-point load. As the wet bulb temperature is increased from 65°F (18.7°C) to 85°F (29.8°C), the optimal switch point load decreased from 690 (2430 KW) to 610 tons (2150 KW).

CONCLUSION

These results present some of the research done for ASHRAE RP 321 which is entitled "HVAC System Dynamics and Energy Use in Existing Buildings."

The preliminary work for this project has included the development of computer algorithms and codes to model the HVAC equipment, the modification of the models to incorporate the significant transient effects, and the verification of the models using measured load to drive the computer simulation.

The end results of this project will be the identification and investigation of potential energy saving HVAC operating strategies and the availability of reliable equipment models. The groups that could potentially benefit the most from these models are building operators, HVAC controls persons and future researchers.

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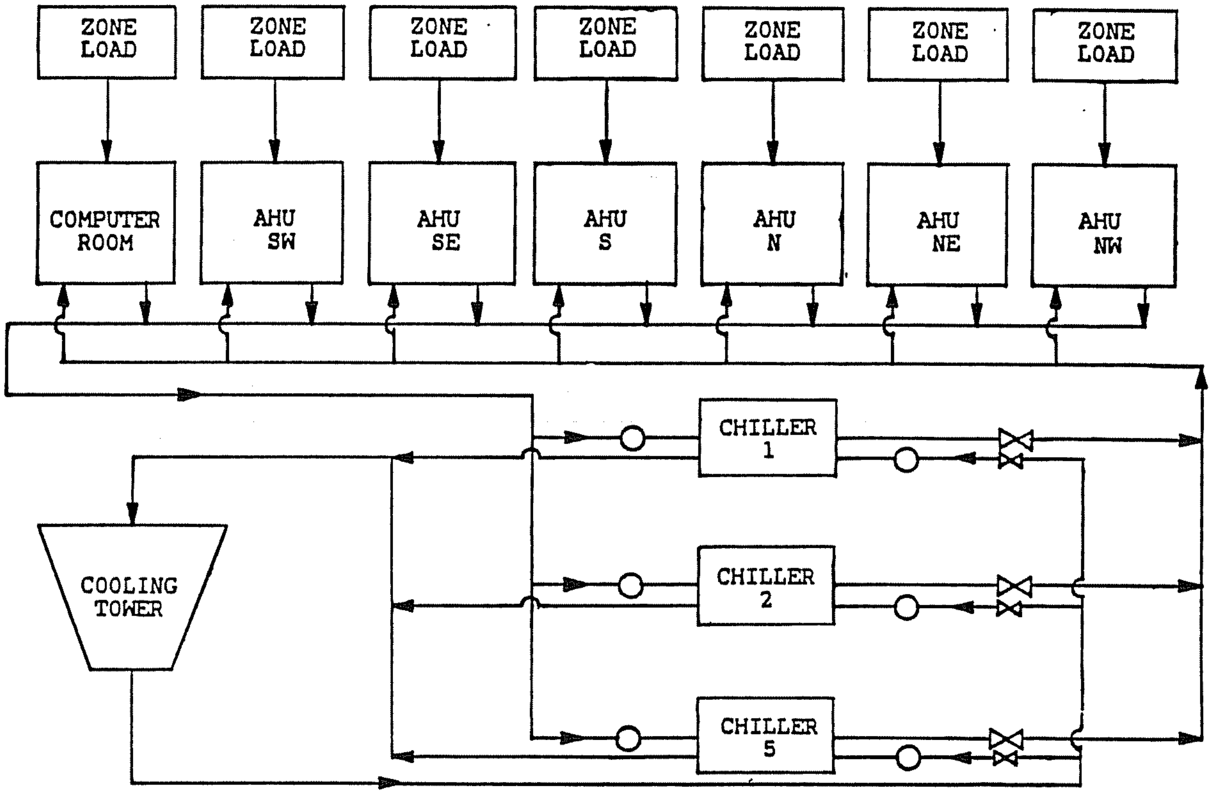

Figure 1. Schematic representation of the IBM Atlanta HVAC system

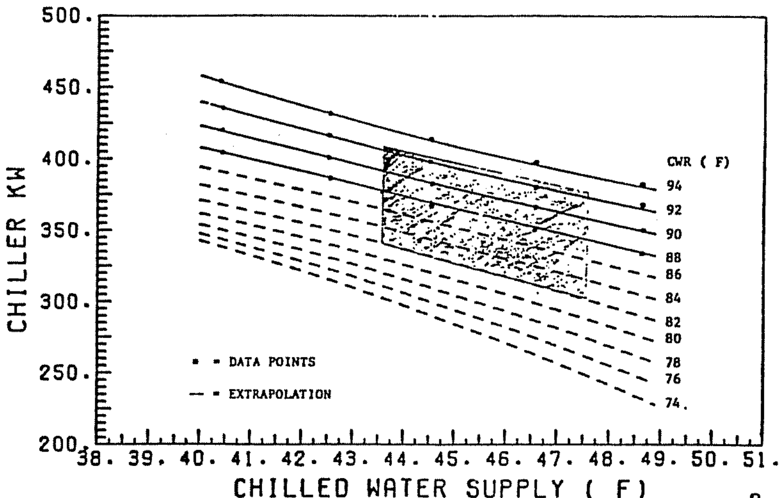

Figure 2. Performance map for a 550-ton centrifugal water chiller. Data points are from the manufacturer. Shaded region indicates the normal operating range

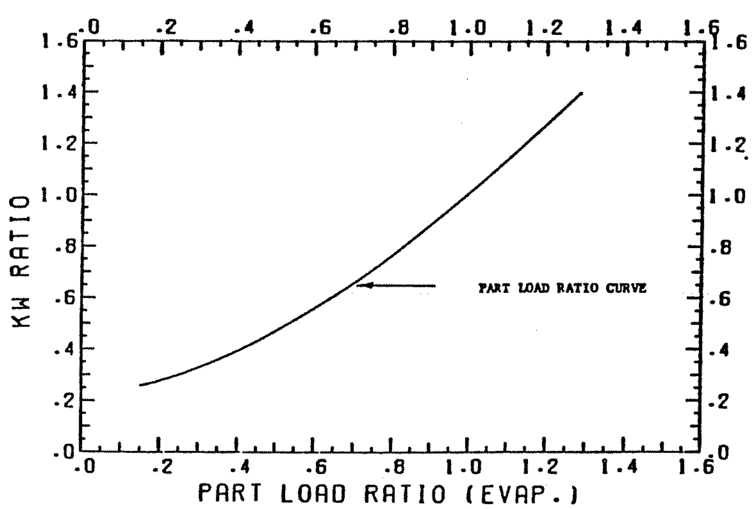

Figure 3. Manufacturer part load ratio curve for a centrifugal chiller

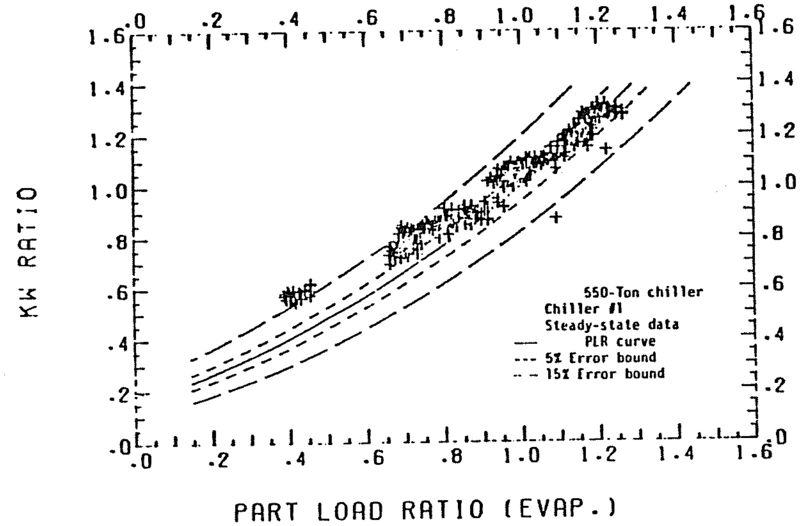

Figure 4. *Comparison between steady state chiller data and the PLR curve*

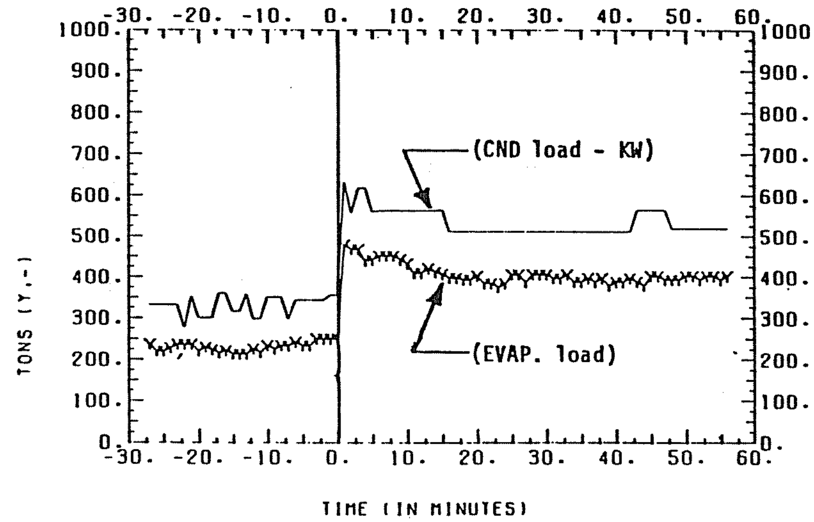

Figure 5. *Comparison of the chilled water load based on the evaporator side and the condenser side of a chiller*

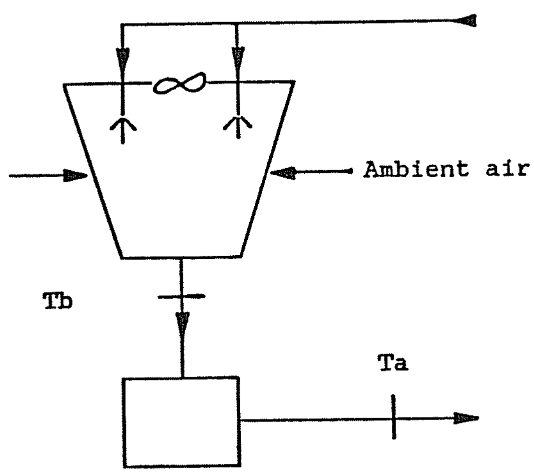

Figure 6. *Schematic representation of a cross-flow, induced-draft cooling tower*

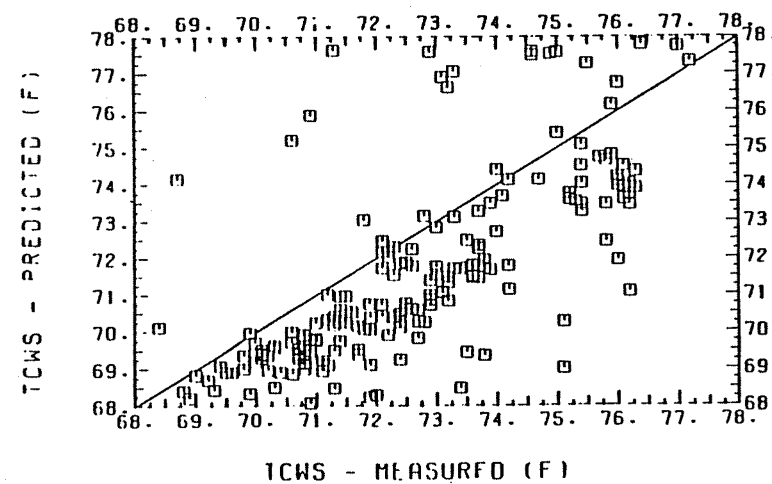

Figure 7. *Comparison between the measured condenser water supply temperature and the predicted temperature using the "Whillier method." Data were obtained from IBM, Charlotte, NC*

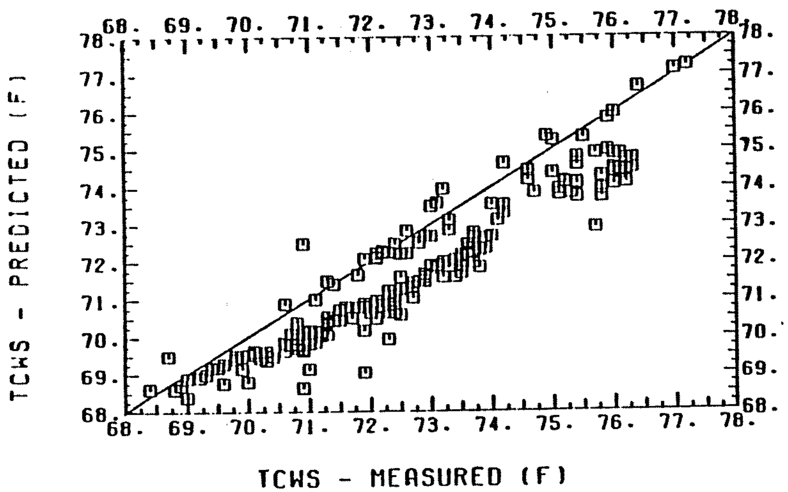

Figure 8. *Comparison between the measured condenser water supply temperature and the predicted temperature using the "Whillier method." Model has included the transient effects of the sump. Data from Charlotte, NC*

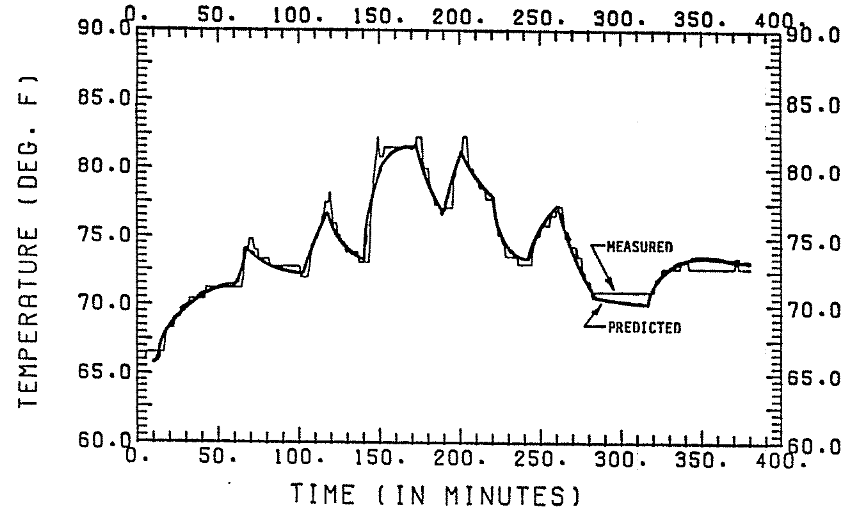

Figure 9. *Comparison between the measured condenser water supply temperature and the predicted temperature. Measured data were collected during the cooling tower fan speed test period at Atlanta*

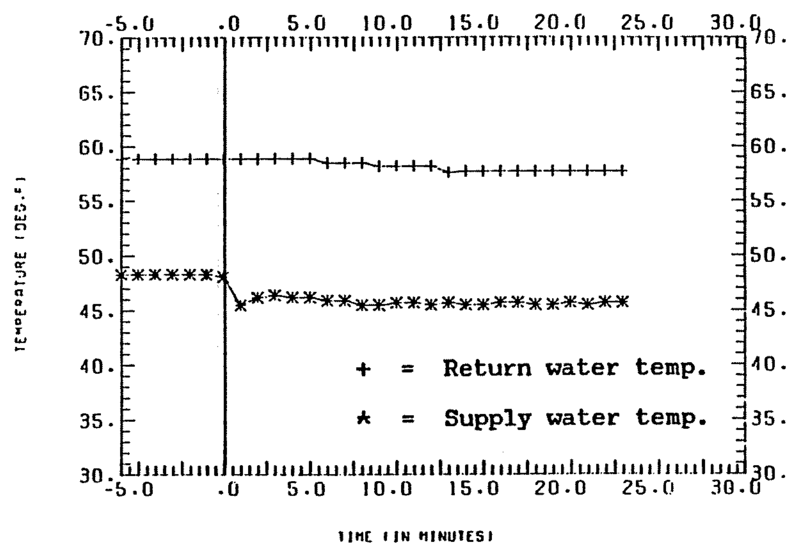

Figure 10. *Results of a chilled water supply set point test (48 F to 46 F)*

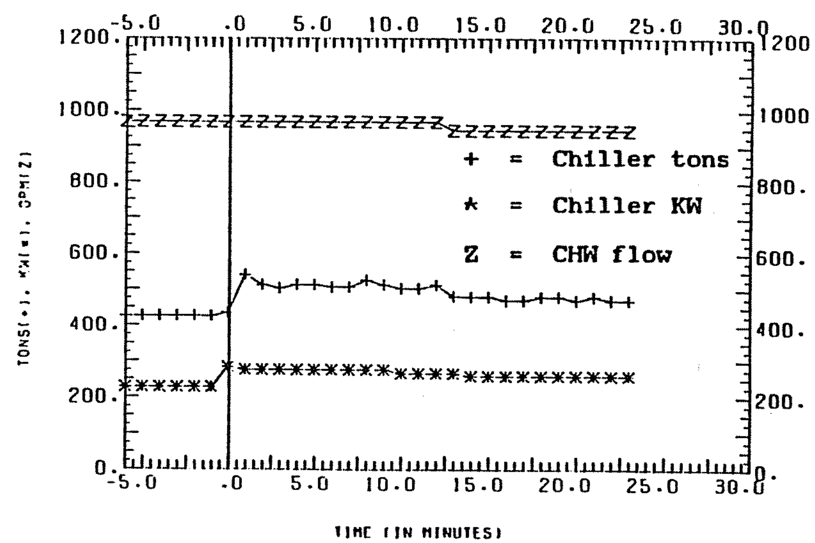

Figure 11. *Results of a chilled water supply set point test (48 F to 46 F)*

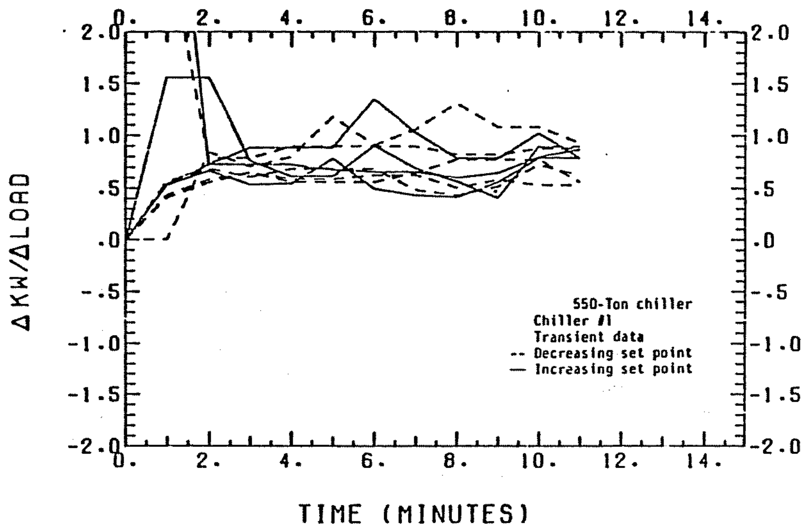

Figure 12. *Normalized presentation of 9 independent chilled water supply set point tests*

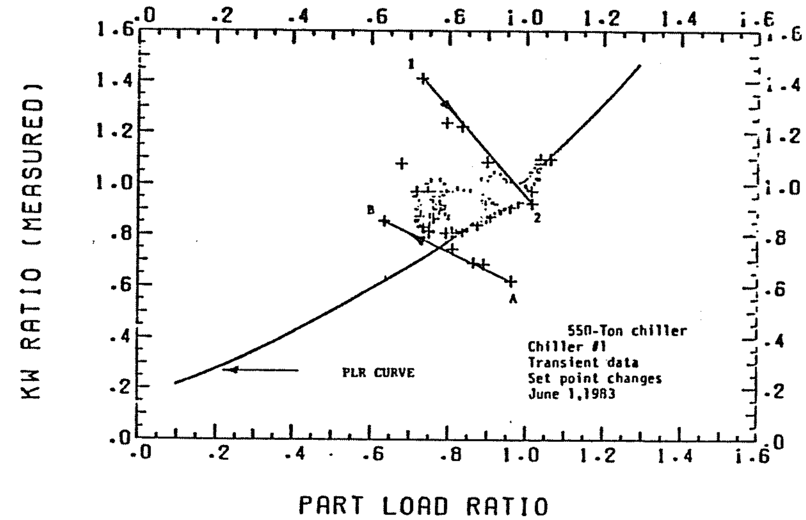

Figure 13. *Transient chiller test data versus the manufacturer PLR curve*

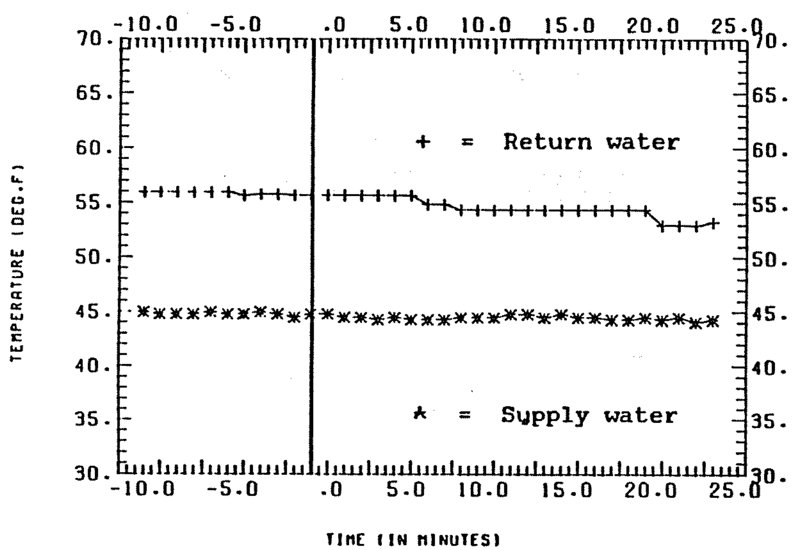

Figure 14. *Results of a supply air temperature set point test (55 F to 60 F)*

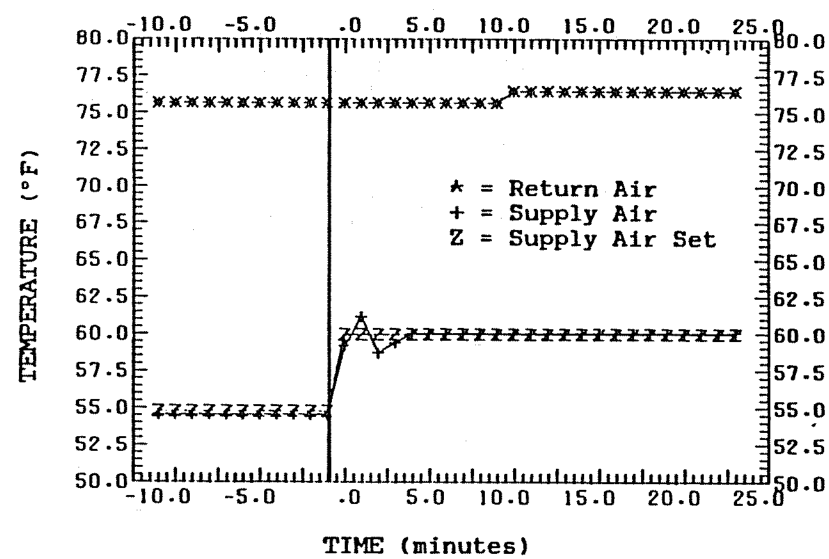

Figure 15. *Results of a supply air temperature set point test (55 F to 60 F)*

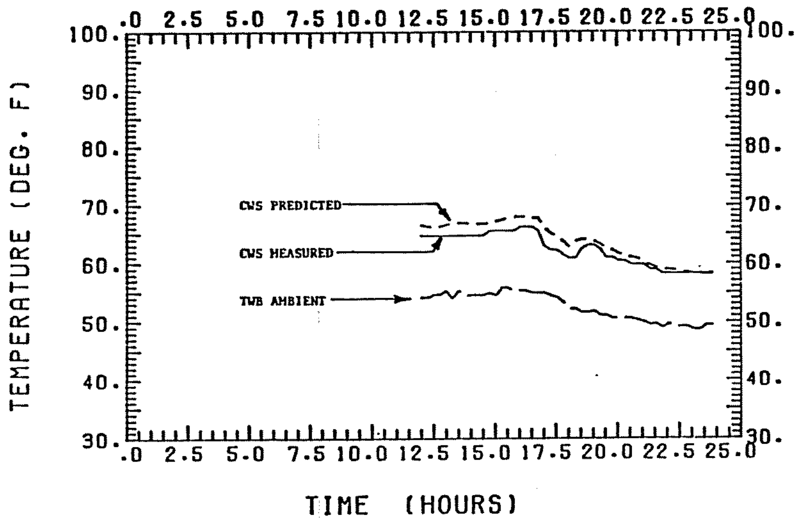

Figure 16. *Condenser water supply temperature. Chiller cooling tower subsystem model verification. Data collected November 8, 1983*

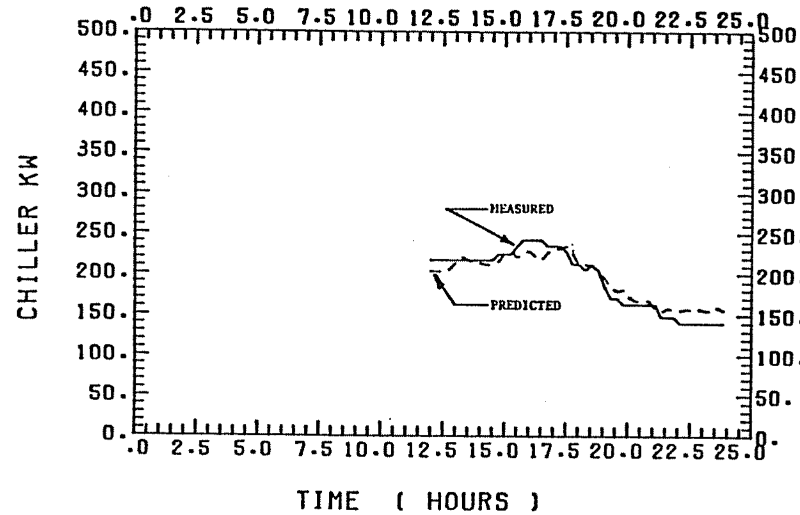

Figure 17. *Chiller power consumption. Chiller-cooling tower subsystem model verification. Data collected November 8, 1983*

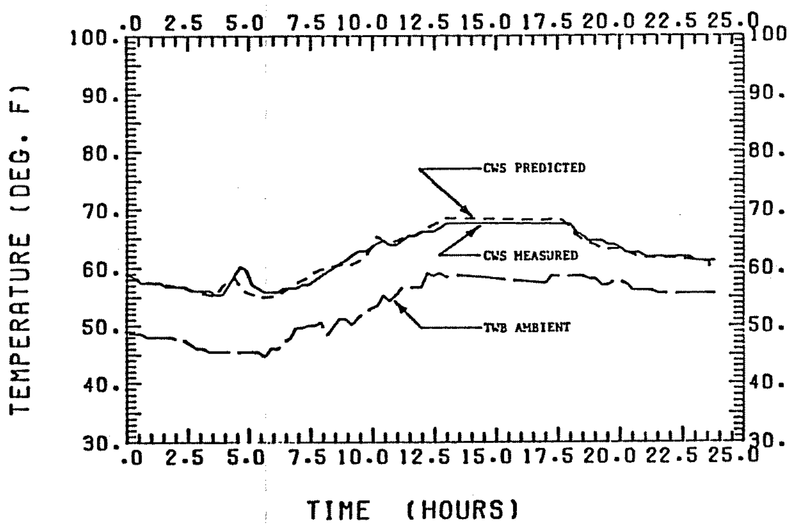

Figure 18. *Condenser water supply temperature. Chiller tower subsystem model verification. Data collected November 9, 1983*

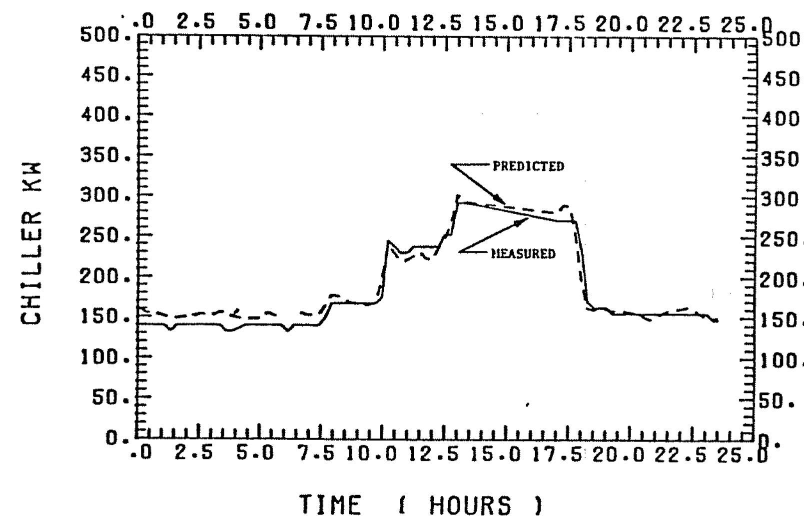

Figure 19. *Chiller power consumption. Chiller-cooling tower subsystem verification. Data collected November 9, 1983*

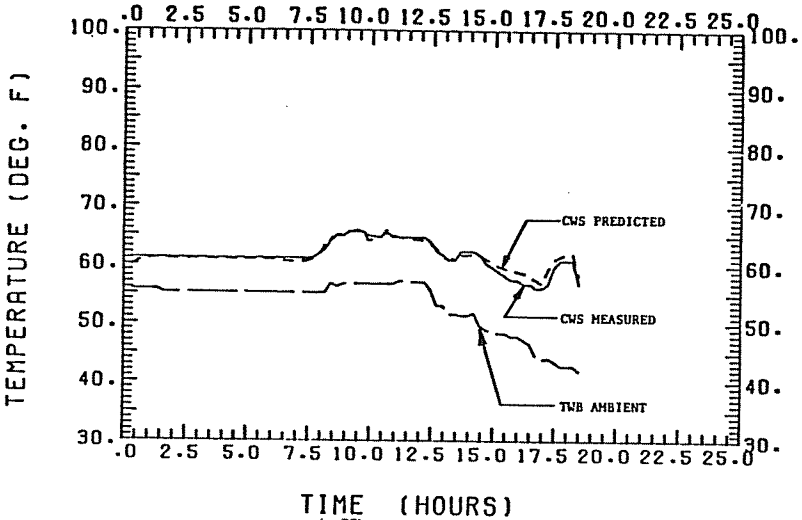

Figure 20. Condenser water supply temperature. Chiller cooling tower subsystem model verification. Data collected November 10, 1983

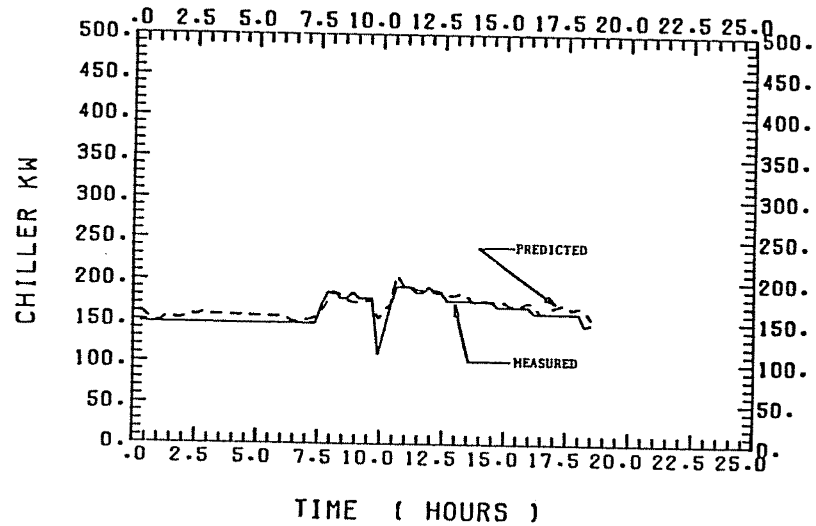

Figure 21. Chiller power consumption. Chiller-cooling tower subsystem model verification. Data collected November 10, 1983

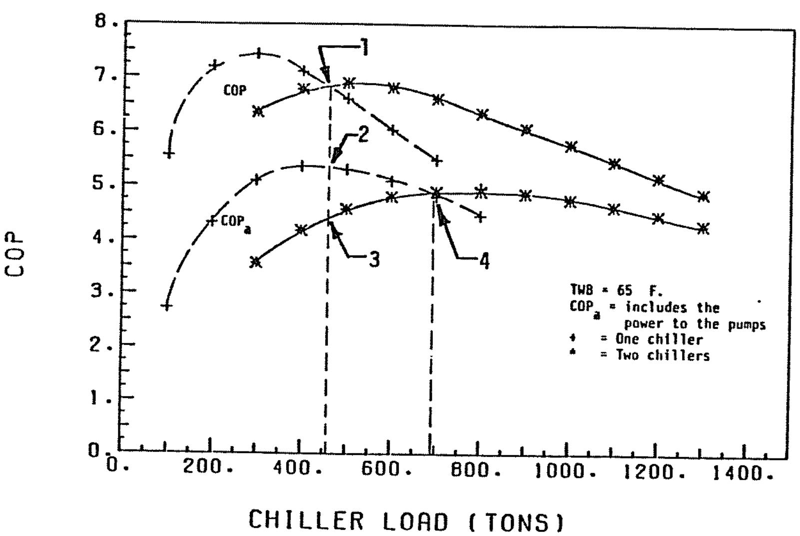

Figure 22. Chiller operating status optimization. Design load is 550 tons

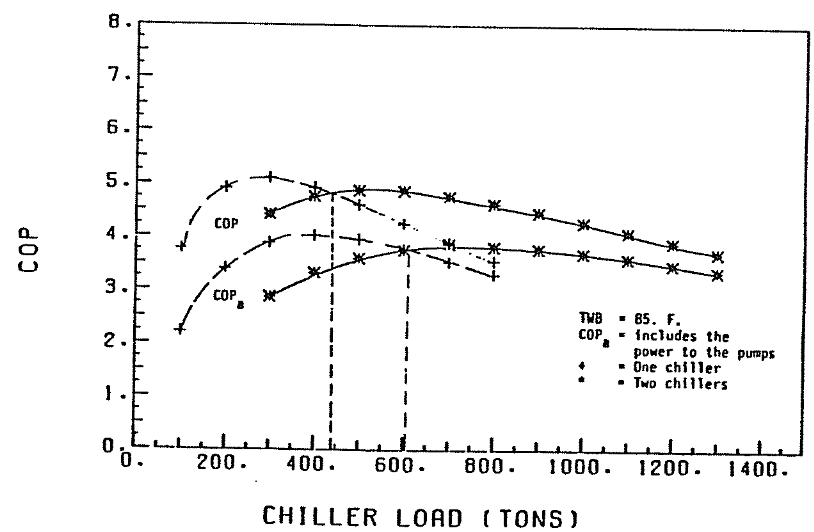

Figure 23. Chiller operating status optimization. Design load is 550 tons