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Development of Computerized Control Strategies for a Large Chilled Water Plant

A.S. Lau, P.E.
ASHRAE Member

W.A. Beckman, Ph.D., P.E.

J.W. Mitchell, Ph.D., P.E.
ASHRAE Member

ABSTRACT

A minicomputer monitoring and control system is part of the HVAC system of a 1.4 million square foot commercial facility. The chiller system at the control plant consists of four 1250 ton centrifugal chillers, a six-cell cooling tower, five 100,000 gallon chilled water storage tanks, and associated pumps and piping.

Computer models of the chilled water system were developed to study the energy conservation potential of control strategies intended for the minicomputer control system. Empirical curve fits were used for all of the components except the cooling tower. An effectiveness model based on manufacturer's data was used for the cooling tower. Comparisons of modeled performance with measured data showed good agreement both at the component and system levels.

Optimal control strategies for the number of chillers, cooling tower fan speeds, and condenser pump flow rates were developed. These resulted in an estimated combined savings of $4,400 a year. Demand limiting using the chilled water storage reduced annual peak demand by 161 kW and saved an additional $5,500. Reset of the chilled water set point saved an additional $4,400 for a total combined savings of $14,300.

INTRODUCTION

The primary goal of this project was to identify computer control strategies to reduce the cost of electricity consumed by the chilled water plant of a large commercial facility. This project deals with a facility that has a minicomputer system designed to monitor and control many of the HVAC control functions. In order to study the many different control options and to assess the long-range impact on energy bills, the HVAC system was simulated using TRNSYS (Klein, et al, 1981), a component-based transient simulation program developed at the University of Wisconsin Solar Energy Laboratory. Since TRNSYS has mainly been used for solar system analysis, several new component models were developed to represent the plant equipment and control functions. Performance data were available on the complete system and on individual components, which allowed comparison of model component with actual data and facilitated model development.

The facility is located in Charlotte, NC, on a site comprising about 700 acres. Ten interconnected buildings with a total of 1.4 million square feet (130 thousand square meters) of floor space are served by the central chilled water plant that has a capacity of 6000 tons

A.S. Lau, Assistant Professor of Engineering, Capitol Campus, The Pennsylvania State University.
W.A. Beckman, Professor of Mechanical Engineering, University of Wisconin-Madison.
J.W. Mitchell, Professor of Mechanical Engineering, University of Wisconsin-Madison.

(21,100 kW). Chilled water is provided by four 1250-ton (4400 kW) electrically driven centrifugal chillers. A unique aspect of the chilled water system is the presence of five 100,000-gallon (378,500 liter) storage tanks. Water chilled and stored at night can be used during the day to reduce the peak electrical demand and chiller load. The site is multi-function, divided between offices, manufacturing, and warehouse. Energy efficient practices have been used throughout the site and include high-efficiency lighting, variable-volume fans, and enthalpy economizers for the air-handling units.

The chilled water system is shown schematically in Figure 1. Heat from the chiller condensers is rejected to the ambient at the cooling tower. The tower has six individual cells and a two-speed fan that can be controlled by computer. The six condenser pumps and four secondary pumps have variable-speed motors on half of the pumps, the other half being fixed-speed. The secondary pumps are controlled by sensors in the buildings, while the condenser pumps can be controlled by computer. The primary pumps are fixed-speed, but their flow can be regulated through computer control of an automatic valve on each chiller evaporator.

There are five possible active modes of chilled water system operation. The numbers assigned to each mode are those used in the actual system. In mode 1, the storage tanks are not used and the site load is met entirely by the chillers.

In mode 2, which is the demand-limiting mode, the storage tanks are used in parallel with the chillers to meet the load. In mode 7, the chillers are used both to meet the site load and to supply chilled water to the tanks. The storage flow in both these modes is controlled indirectly via the evaporator flow valves.

Mode 3 is similar to mode 2, except the chillers are turned off and storage meets the entire load. This mode can be used in cold weather when the site load and water flow rate are below the limit on storage.

Mode 5 is a strainer cycle or water economizer. In this mode, chilled water from the cooling tower is filtered and sent directly to the site, bypassing the need for the chillers. This mode is viable in cold weather when the tower alone can produce the desired chilled water temperatures. Mode 5 operation was not part of this study.

SIMULATION METHODOLOGY

The simulation models were separated into the plant model and the chilled water load model. The buildings and air-handling equipment were simulated first and the individual zone loads totaled to define the chilled water load as a function of time. Typical Meterological Year (TMY) weather data for Charlotte, North Carolina were input to this part of the simulation (TMY 1982). The chilled water load, along with the site electrical loads and pertinent weather data, were stored for use as input to the plant model. In this manner, multiple plant runs were carried out without requiring simulation of chilled water load each time. This results in significant computer time savings because the chilled water load simulation was quite expensive.

PLANT MODEL

The components in the TRNSYS plant model are chillers, pumps, cooling tower, valves, piping, and controls, together with their input and output features. The new components developed for this project are the chiller, cooling tower, pumps, site chilled water circuit, and all of the control functions. Two output components were also developed to obtain the average and peak day electric demand profiles and to determine the utility bill.

Measured performance data were compared with simulation results in order to establish accuracy. First, each major component was examined and adjustments made where necessary to improve the agreement. After all components were evaluated, the entire plant simulation was run using only the two measured inputs of total chilled water load and ambient wet-bulb temperature. The simulated plant electric demand was then compared with measured data as the test of the validity of the simulation.

The measurements were taken from four separate data tapes. Each tape has about two and one-half days of continuous operation with 218 points recorded every 15 minutes. The dates covered by the four tapes are 4/30-5/2, 5/30-6/1, 6/17-6/19, and 7/19-7/21, all in 1983. The models for chillers and cooling tower are described below.

Chillers

The four electric-driven chillers in this study have centrifugal, two-stage compressors. As the evaporator load varies, the built-in chiller controls attempt to maintain a constant leaving water temperature. This is accomplished by automatic adjustment of the refrigerant flow rate with prerotation vanes at the compressor inlet. The evaporator load and electrical input are rejected at the condenser to a cooling tower by a separate water stream. Figure 2 shows a chiller system schematic and the steady-state energy balance.

A technique for modeling centrifugal chillers is described by Stoecker (1975) and was used as the basis for this model. The model assumes steady-state behavior and is a combination of curve fits to manufacturer's data. The chiller performance is represented as the power consumption as a function of the chilled water load, leaving chilled water temperature (chilled water supply temperature), and leaving condenser water temperature (condenser water return temperature). The chilled water supply temperature setting can be computer controlled and is the main control parameter directly input to the chiller. The internal controls vary the chilled water supply temperature with load as shown in Figure 3. At part loads, (actual chilled water load divided by design chilled water load) the actual chilled water temperature is lower than the desired set point. This control was included in the model.

To verify the chiller model, measurements taken at the plant were used. The accuracy of the data was evaluated by performing an energy balance on the chiller to yield:

$$\text{Cooling Load} + \text{Electrical Input} = \text{Cooling Tower Load} \tag{1}$$

The energy flows should balance, with any measuring errors showing up as an unbalance of energy. The distributions of the calculated energy unbalances are shown in Figure 4 for two of the chillers. Both chillers have energy unbalances as high as 30% to 40%, which indicates the magnitude of instrument uncertainty.

The predicted chiller demands for chillers 2 and 3 are compared with measurements in Figures 5 and 6, respectively. There appears to be a bias toward overprediction for chiller 2, although the discrepancy is within the error in the measurements. For chiller 3, the predicted demands agree well with the measurements, with 95% of the data within ±5%. Since the simulation model, based on manufacturer's performance curves, results in predicted chiller demand within the range of energy unbalances based on measurements, the manufacturer's curves are assumed to be valid for simulation purposes.

Cooling Tower

The cooling tower has six cells, each rated at a water flow of 3125 gpm (12.2 L/sec). At a temperature drop of 15 F (8.3°C), the entire tower can exhaust heat at a rate of 11,800 tons (41,500 kW). A two-speed fan is used to control airflow with low speed equal to half the high speed. In the model, it is assumed that the fan laws (Tuve and Dumholdt 1966) prevail and at low speed the volumetric flow rate is half the flow at high speed.

The heat and mass transfer processes occurring in a cooling tower have been studied (Baker and Shyrock 1961) and the results form the basis of a commonly accepted analytical technique described by ASHRAE (1983). The solution of the governing equations for each unique set of operating conditions involves repetitive numerical integration.

Whillier (1967) recognized the complexity of this common technique and proposed a new method that does not require integration yet is nearly as accurate as the former method. Manufacturer's data are used to define an effectiveness as a function of the air and water flow rates and entering conditions. The primary assumption in the model is that the air leaves the tower saturated.

The data from all four data tapes were used for comparison with the Whillier model (1967) and the results are illustrated in Figure 7. There is a slight bias toward overprediction by 0.62 F (0.34°C) on the average. The RMS error for all of the data points is 1.36 F (0.76°C).

The water temperature measurements are recorded to a few tenths of a degree Fahrenheit. However, the wet-bulb temperature is derived from a dewpoint sensor, and the error can be as high as a few degrees Fahrenheit. Since the leaving water temperature is highly dependent on the wet-bulb temperature, the RMS error of 1.36 F (0.76°C) seems acceptable.

As a check of the plant simulation model, the total plant electric demand was predicted for a two-day period in July for which good measurements were available. The two measured inputs used to force the plant simulation are the total chilled water load and ambient wet-bulb temperature. The results are shown in Figure 8. The agreement is quite good. The difference between the total consumption for the period shown is 1.1%, with the predicted being lower than the measured. The RMS error for the 15-minute points is 158 kW, which is on the order of 10%. However, errors in the measured cooling load, wet-bulb and/or electric demand can account for the larger 15-minute discrepancies.

SITE CHILLED WATER LOAD MODEL

The site is comprised of about ten main buildings, with a total of more than 30 air-handling units (AHU) and zones. Many of the buildings have perimeter zones with induction units for both cooling and heating. In order to model this, the site was broken into 15 zones. Each zone has scheduled inputs for lighting, people, and miscellaneous electric loads, as well as dynamic solar and envelope gains. To meet the heating and cooling loads, each zone is coupled to an AHU with unique characteristics such as fan capacity, temperature set points, outside air requirements and enthalpy economizer cycle. All of the AHUs modeled have variable-volume fans, which adjust flow by varying the pitch of the fan blades. All but two of the zones use an economizer mode when applicable. When economizer operation is not used or applicable, outside airflow is controlled to be a 10% fraction of the supply airflow rate.

There was an insufficient amount of data to compare the site load model with measured loads. Preliminary indications were that the model predicted the peak chilled water loads well, but there were some discrepancies between the hourly load profiles. The modeled load profile was flatter than the measured loads during the day. Therefore, the chilled water storage will have less demand-limiting capability as modeled.

INVESTIGATION OF CONTROL STRATEGIES

The control strategies that were investigated are described in this section. The month of July was used as indicative of the peak cooling season since the peak cooling load occurs then. A few of the strategies were simulated for an entire year to obtain a better estimate of the annual impact. The change in the electric utility bill was considered the measure of a strategy's merit. The effects on peak demand and consumption were distinguished due to the separate billing for energy and demand. The peak electric demand is paid for every month for 12 months, and this must be considered when comparing the July results alone.

Chiller, Tower, and Pump Strategies

A subsystem of the entire chilled water system, which could be separated from the larger system for control studies was identified. By studying the interaction of the possible control settings and independent variables in this subsystem, the optimum control strategies were determined directly. Therefore, these strategies are referred to as deterministic.

The subsystem is shown in Figure 9 and includes the chiller, primary pumps, cooling tower, and condenser pumps. There are three controlled variables in this subsystem and two independent variables. The object was to find the combination of the three control states that results in minimum power consumption for each combination of wet-bulb temperature and total chiller load.

A steady-state model was developed that contained only the specified components. The combinations of wet-bulb temperature, chilled water load, cooling tower fan speeds, condenser pump flow, and number of chillers were input to determine the steady-state system performance. The entire set of results was searched for the minimum power consumption condition for each combination. This defined the optimum control states for cooling tower fan speeds, condenser pump flow rate, and number of chillers as a function of total chilled water load and wet-bulb temperature.

As a basis for comparison for all of the strategies, a base case plant was defined. Condenser pump flow rate was constant at the measured value of 2800 gpm (10.9 L/sec) per pump. Cooling tower fan status was controlled to maintain the approach temperature within 5 to 15 F (2.8 to 8.3°C) of the wet-bulb. The number of operating chillers was based on operation up to 100% of design load if the site supply temperature is below 46 F (7.8°C). These controls are similar to those used in the actual system at the time of this modeling.

The three optimum strategies were simulated in various combinations and the results are summarized in Table 1. Cases 2, 3, and 4 are for each strategy implemented singly, while case 5 is for all three in combination. Case 5 shows the savings from the combined strategies to be $495 for the month of July. Both electrical consumption and demand are reduced by the more efficient plant operation. The individual changes show that of the three strategies, condenser flow has the largest impact by an order of magnitude.

The variable condenser flow strategy is compared to that for a constant flow in cases 6a, b, and c with the other two optimum strategies implemented. The benefit of reducing condenser flow from the typical value of 2800 gpm (8.6 L/sec) per pump is large. Case 6 also shows that a constant flow of 2200 gpm (8.6 L/sec) results in savings ($484) very near that obtained with the optimum, variable-flow control ($495).

For the base case, the plant power consumption is about 19% of the total consumption, or about 987 MWh. This fraction was applied to the monthly utility bill to obtain the plant portion, $35,400. The savings from the three optimized strategies of $495 are about 1.4% of the plant bill. An implication of this relatively minor savings is that the actual controls being used are not far from being optimum from the standpoint of overall plant efficiency and purchased energy.

Storage Mode Regulation

Storage flow in mode 2 is controlled to maintain the chiller flow rate constant in order to limit the chiller load and electrical demand. The control investigated here is the time of activation of mode 2 during the day to limit the peak demand. In the changing of storage, mode 7 was activated at midnight and was completed by 4-5 a.m.

If mode 2 is activated too late in the day, chilled water will remain in the tanks when the peaking period ends in late afternoon. As a result, the peak demand limiting capability would not be fully utilized. Conversely, if mode 2 is activated too early in the day, the chilled water tanks would be depleted before the end of the peak period. This too would result in the demand limiting capability of storage being underutilized.

The results of the start-time simulations for July, with the previous three optimized strategies, are summarized in Table 2. For times later than 9:00 a.m., peak demands were higher and savings were lower. The results in the table show that the best time for activating mode 2 is 8:00 a.m. Peak demand was reduced by 257 kW and a savings of $711 is obtained for July. This is about 2.0% of the base case plant bill.

Reset of the Chilled Water Set point

The chilled water set point is controlled by the minicomputer system. Because the actual temperature is typically below that required, the chiller power consumption is higher than necessary. This deviation from the set point can be greatly reduced by resetting the set point via the minicomputer system. The reset frequency should be greater than the response time of the chiller to set point changes. Preliminary results from another project at the University of Wisconsin show that chiller dynamic response time is typically less than two minutes.

Without set point reset, the actual chilled water temperature decreases 4 F (2.2°C) as the load drops from 100% to 0%. It was assumed that the temperature drop could be reduced to 0.5 F (0.3°C). The simulation was run for July and case 2c from Table 4.2. The results are 5056.9 MWh consumption, 9485.5 kW peak, and a bill of $184,739. This represents a reduction of $488 below case 2c, or about 1.4% of the base case plant bill. This savings is significant given the relative ease of resetting the set point with the minicomputer system.

Storage Tank Subcooling

The effective storage capacity and cooling rate from storage can be increased by reducing the temperature of the chilled water stored in the tanks. However, running the chillers at a lower set point in charging mode 7 will also increase the chiller power consumption. The simulation was used to examine this trade-off between reduced demand and increased consumption resulting from storage tank subcooling.

The best control modes discussed thus far were used in the simulation to compare four levels of subcooling for July. The results are compared in the left half of Table 3. A temperature of 44 F (6.7°C) is the base case.

In all cases, the reduced demand charge is more than offset by the increase in the consumption charge. This can be misleading, because the demand savings are effectively multiplied by 12 on an annual basis. On the other hand, subcooling is only required for those months in which demand is near the peak. An examination of the annual simulations indicated that subcooling would be required for about four months, May through August. Therefore, in order to compare the annual impact of storage subcooling, the consumption charge increases for July were multiplied by 4 while the demand charge decreases were multiplied by 12.

Even with these adjustments, the net effect of all the levels of subcooling on the annual utility bill was an increase, as shown in the last column of Table 3. The utility bill increase is larger as the subcooling temperature is reduced. Therefore, subcooling of the storage tanks does not appear profitable with the current utility rate structure. If the demand charge was higher, relative to the consumption charge, or if time-of-day rates applied, subcooling could save money. Subcooling might be profitable if it was only needed for the days when the demand is near the peak, which would require a forecasting strategy.

Annual Results

Selected strategies were simulated for an entire year and compared with the base case. Table 4 lists the monthly and annual results for the base case. The next to last column is the plant power consumption and includes the chillers, pumps, and cooling tower fans, amounting to $275,000.

The first strategy selected for comparison was the activation of mode 2 at 8:00 a.m., without any of the three optimized strategies. The results are shown in Table 5. Use of storage results in a reduction in the annual utility bill of $5450 or 2.0% of the base case plant bill.

The June and August demands in Table 5 are greater than the July demand. The annual demand reduction is 161 kW, compared to 249 kW for July. For greater demand reduction and more savings, the optimum start time for mode 2 should be studied for these other months. A start time dependent on the month could then be programmed into the minicomputer system.

The addition of the three optimized strategies, together with the previous case with storage activated at 8:00 a.m., was studied next. The annual utility bill is further reduced by $4440 or 1.6% of the base case plant bill. The total reduction relative to the base case is $9890 or 3.6%. Most of the reduction in the utility bill resulting from the three optimized strategies is due to reduced consumption and more efficient plant operation. The more efficient operation also reduces the peak demand by 30 kW.

The final case studied on an annual basis was the use of automatic reset of the chilled water set point. Reset results in an additional annual savings of $4350, or 1.6% of the base case plant bill.

For all of the strategies simulated on an annual basis, the total reduction in the utility bill is $14,240 or 5.2% of the annual plant portion of the electric bill. The total reduction in consumption is 252.2 MWh, and the total reduction in peak demand is 197 kW. The utility rate schedule has a direct impact on the monetary savings accrued by these and any other conservation strategies. The local rates are relatively low compared to the rest of the country. Peak demand charge is about $2.71/kW and the consumption charge is about $0.031/kWh for typical summer conditions. Higher future rates and higher rates in other parts of the country would increase the savings substantially. The presence of time-of-day rates would also add to the benefit of the storage system on a year-round basis.

CONCLUSIONS

The major part of this project was the development of new component models. Both the chiller and cooling tower models were designed to be general models, and were made specific for this study using manufacturer's data.

The other major part of the developmental effort was the modeling of the control functions. Most of the control strategies required for the simulation were performed manually in the actual system when this project started. The operators were cooperative in relating their general control concepts, but many of the details required were not discussed. These details were discovered by implementing the general concepts in the simulation and then observing the system behavior.

The complexity and uniqueness of innovative heating and cooling systems would likely require a similar trial and error procedure for development of control strategies. This is a benefit of a detailed computer simulation tool like TRNSYS. Otherwise, the strategies would have to be tried in the actual system where an error or oversight could have disastrous consequences. The following specific control strategies are considered.

Chiller, Tower, and Pump Strategies: The three optimized strategies in combination saved $495 in July and $4440 for the annual simulation. The July savings resulting from the optimum condenser flow rate were an order of magnitude higher than the savings from either the optimum cooling tower fan control or the optimum number of chiller control. Simply reducing the constant condenser flow rate from 2800 to 2200 gpm (10.9 to 8.6 L/sec) resulted in savings nearly identical to the optimally controlled condenser flow rate for July.

Storage Mode Regulation: The optimum start-time for mode 2 was found to be 8:00 a.m. for July. However, the annual simulation showed that this is not necessarily the optimum time for the other summer months. For the annual simulation, a start-time of 8:00 a.m. resulted in a reduction in peak demand of 161 kW and a savings of $5450.

Because of the different shapes of the actual and simulated load profiles, the start-times studied do not translate directly to the actual system. It is recommended that the long-term recorded data be used to develop the necessary input data for the plant simulation in order to study mode 2 operation. The variation of the best start-time from month to month should also be investigated with the simulated and actual loads.

Reset of the Chilled Water Set Point: Automatic reset of the chilled water set point saved $488 for July and $4350 on an annual basis. These savings are significant given the simplicity of this control strategy.

Storage Tank Subcooling: The estimated annual impact of storage tank subcooling showed that even a reduction in storage temperature of 1 F (0.6°C) caused an increase in the annual utility bill. The optimum temperature is the highest temperature acceptable. With the current utility rate schedule, the only way that subcooling might pay is if the peak days could be anticipated the night before and subcooling used only on these days.

Overall Impact: Though the savings reported here are a minor portion of the total power bill, most of the strategies can be implemented on the minicomputer system with only a minor effort.

An additional benefit of this project is the better understanding of the equipment and system operation by the plant personnel, both on a theoretical and practical basis. For one case in particular, this study pointed out that the actual outside air amounts were inconsistent with the supposed control strategies. An examination of the AHUs showed that many of the outside air dampers were out of adjustment, causing excessive amounts of outside air during hot weather. The reduced cooling loads resulting from the proper adjustment likely result in as great or greater savings than all of the computer strategies.

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TABLE 1
Optimized Strategies versus Base Case for July

		Consumption (MWh)	Peak (kW)	Total Bill ($)	Savings ($)
1.	Base case	5087.9	9766.7	186,433	-
2.	Optimum number of chillers	5086.0	9766.7	186,404	29
3.	Optimum cooling tower level	5086.2	9766.6	186,412	21
4.	Optimum condenser flow	5078.4	9758.0	186,147	286
5.	All 3 optimized strategies	5073.0	9747.2	185,938	495
6	a. Condenser flow constant @2800 gpm 2 optimized strategies	5085.4	9766.6	186,387	46
	b. 2500 gpm	5077.2	9750.4	186,087	346
	c. 2200 gpm	5072.9	9748.0	185,949	484

TABLE 2
The Effect of Time of Activation of Mode 2 in July

		Consumption (MWh)	Peak (kW)	Total Bill ($)	Savings ($)
1.	All 3 optimized strategies	5073.0	9747.2	185,938	-
2a.	Mode 2 on @9 a.m.	5071.5	9574.4	185,432	506
b.	8:30 a.m.	5072.2	9566.3	185,433	505
c.	8:00 a.m.	5072.2	9490.3	185,227	711
d.	7:30 a.m.	5071.6	9734.0	185,870	68

TABLE 3
Effects of Storage Tank Subcooling

Subcooling Temperature (F)	Electric Consumption (MWh)	Increase In Consumption Charge ($)	Peak Demand (kW)	Reduction In Demand Charge ($)	Increase In Consumption Charge ($)	Reduction In Demand Charge ($)	Net Increase ($)
44	5056.9	-	9485.5	-	-	-	-
43	5059.6	81	9476.3	23	324	276	48
42	5064.2	220	9465.1	51	880	612	268
41	5071.0	431	9463.9	54	1724	648	1076
40	5079.7	701	9451.0	85	2804	1020	1784

TABLE 4
Annual Simulation Results for Base Case

Month	Total Consumption (MWh)	Plant Peak (kW)	Total Bill ($)	Cooling Consumption (MWh)	Load (MBtu)
J	4435.19	8083.7	166,167	336.7	5,528
F	4002.26	8344.0	152,702	323.9	5,392
M	4477.70	8539.1	167,489	403.8	6,762
A	4500.32	9227.9	168,192	560.2	9,158
M	4923.25	9475.7	181,345	793.5	12,620
J	4848.07	9614.6	179,007	894.2	14,070
J	5087.91	9766.7	186,466	986.7	15,310
A	5123.26	9665.4	187,566	985.2	15,330
S	4705.06	9435.6	174,560	796.8	12,570
O	4681.57	9261.9	173,829	561.9	9,189
N	4357.42	8383.3	163,748	389.2	6,509
D	4370.56	8383.7	164,157	342.7	5,727
Total	55,512.8		$2,065,228	7374.8	118,200

TABLE 5
Annual Simulation Results for Mode 2 Activation at 8 AM

Month	Total Consumption (MWh)	Peak (kW)	Total Bill ($)
J	4428.17	8007.5	165,512
F	4000.63	8328.3	152,215
M	4479.17	8539.2	167,098
A	4499.48	9218.6	167,730
M	4922.73	9442.9	180.893
J	4850.08	9606.1	178,633
J	5086.81	9517.7	185,996
A	5124.71	9598.2	187,174
S	4703.44	9333.1	174,073
O	4680.52	9180.0	173,360
N	4359.11	8380.5	163,364
D	4370.92	8381.2	163,731
Total	55,505.8		$2,059,779

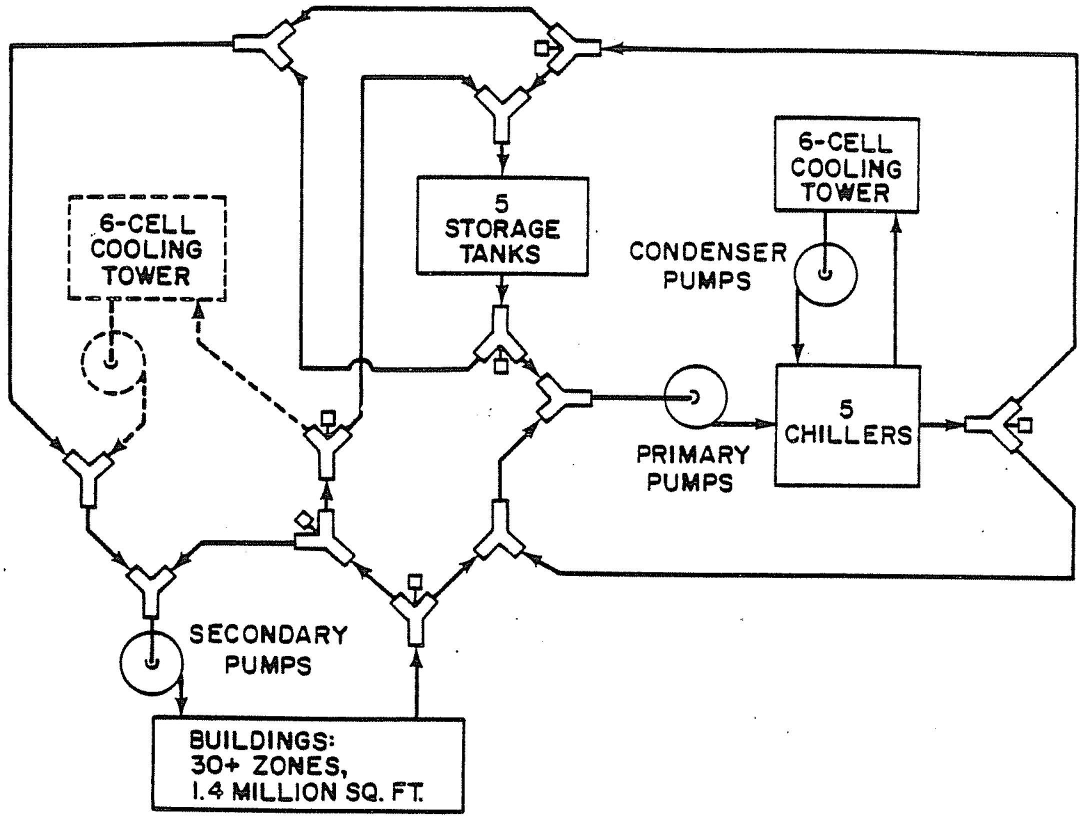

Figure 1. TRNSYS model schematic for the chilled water system of a large commercial building

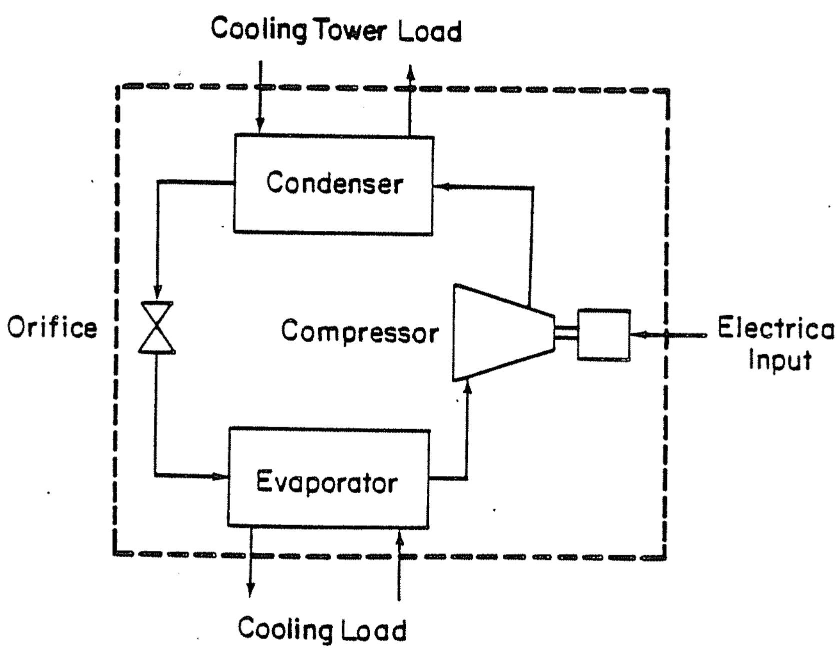

Figure 2. Chiller schematic showing system boundary and energy balance

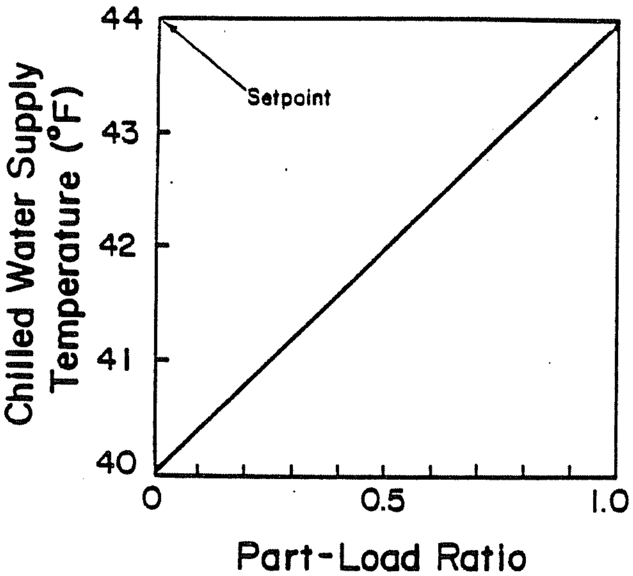

Figure 3. Effect of load on actual chilled water supply temperature

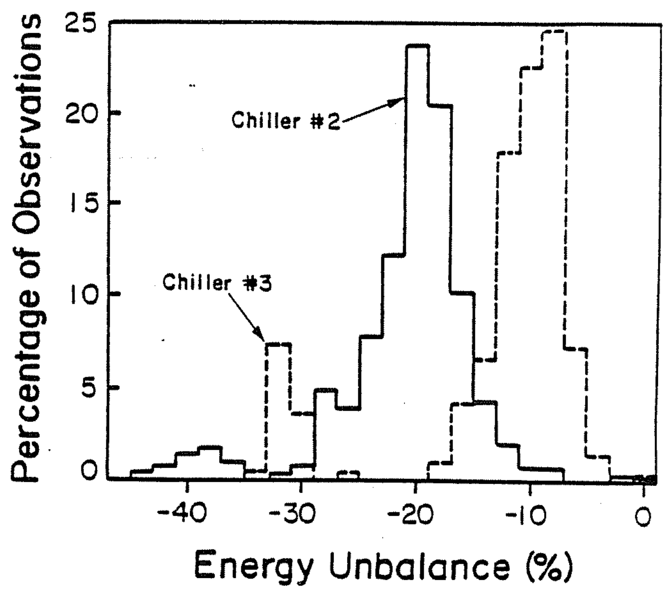

Figure 4. Distributions of chiller energy unbalances

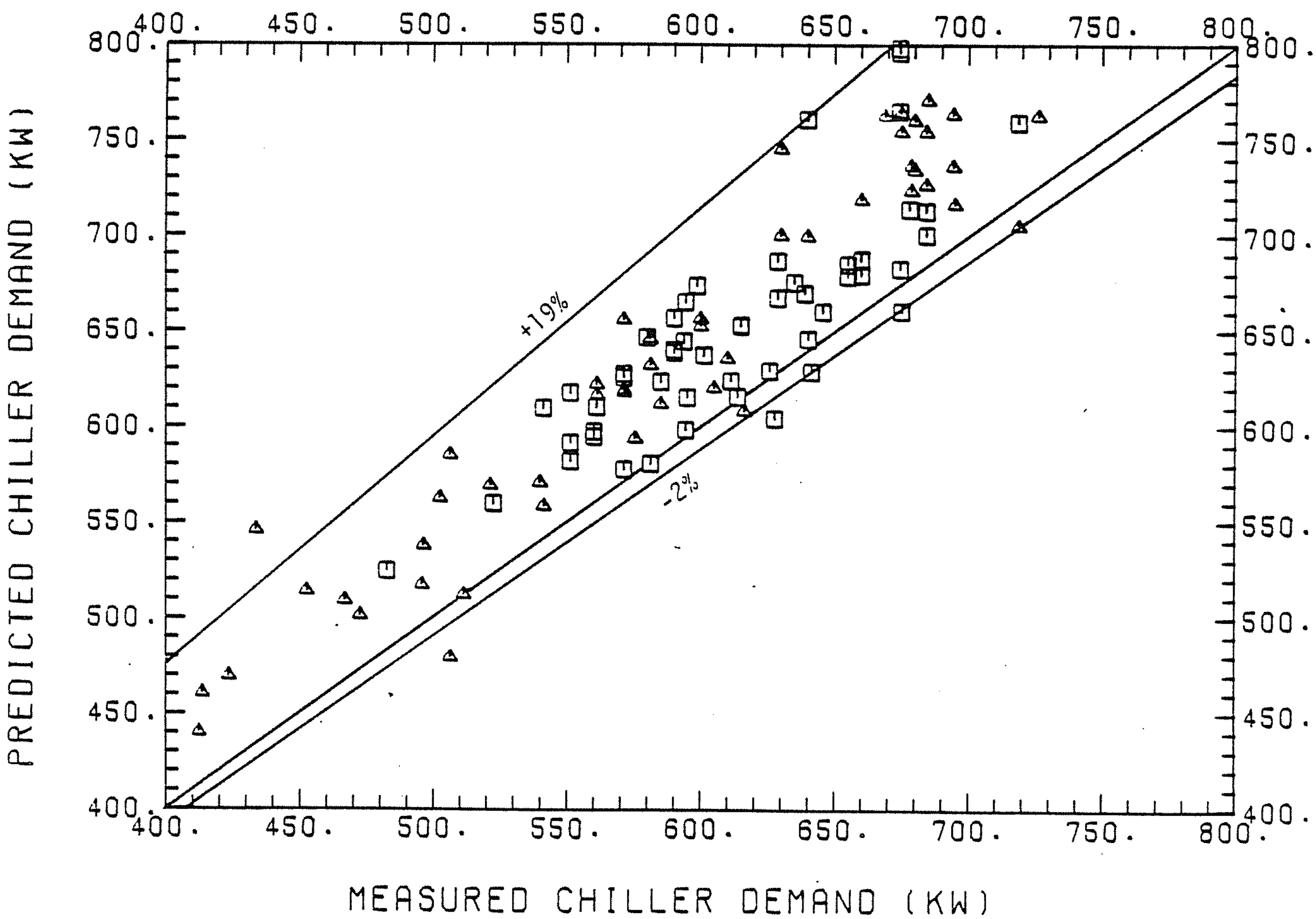

Figure 5. Comparison of predicted and measured demand for chiller 2 using one-hour average data

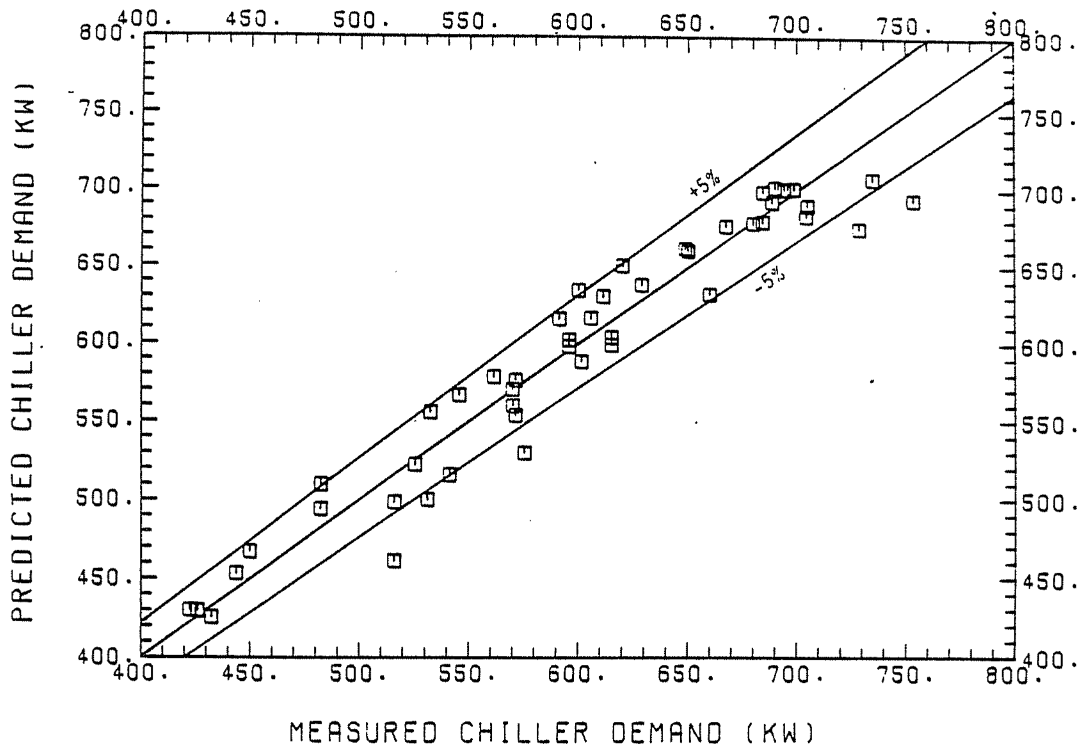

Figure 6. Comparison of predicted and measured demand for chiller 3 using one-hour average data

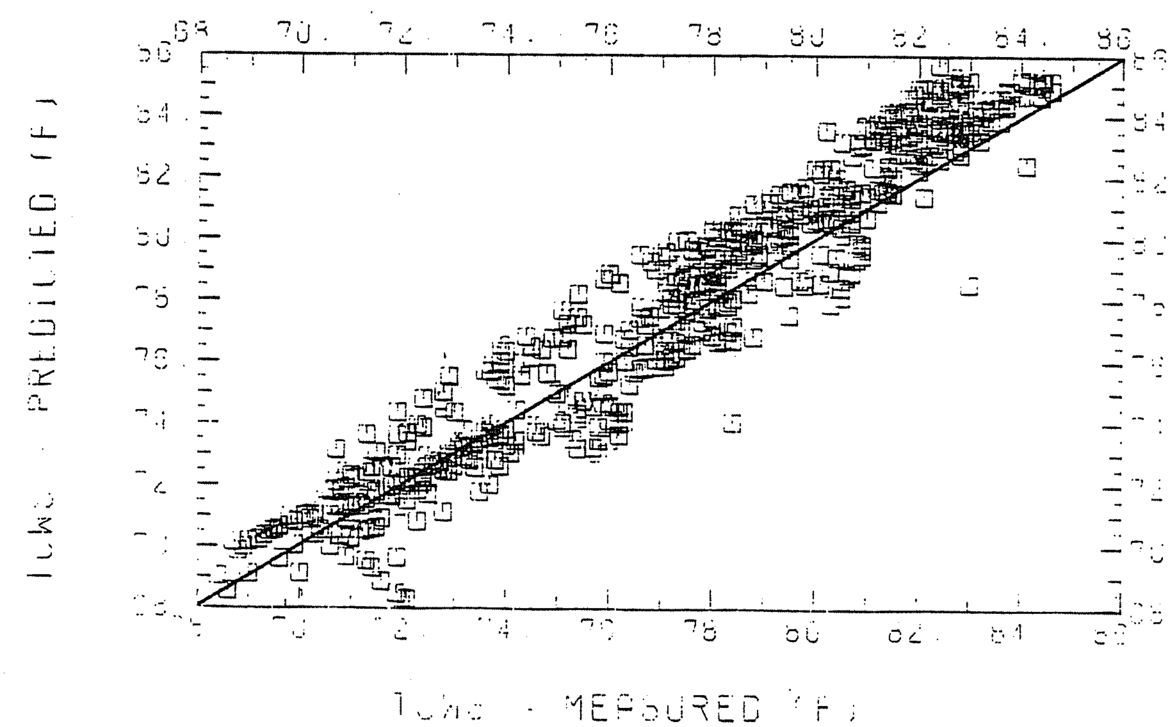

Figure 7. Predicted condenser water supply temperatures using four data tapes versus measured data

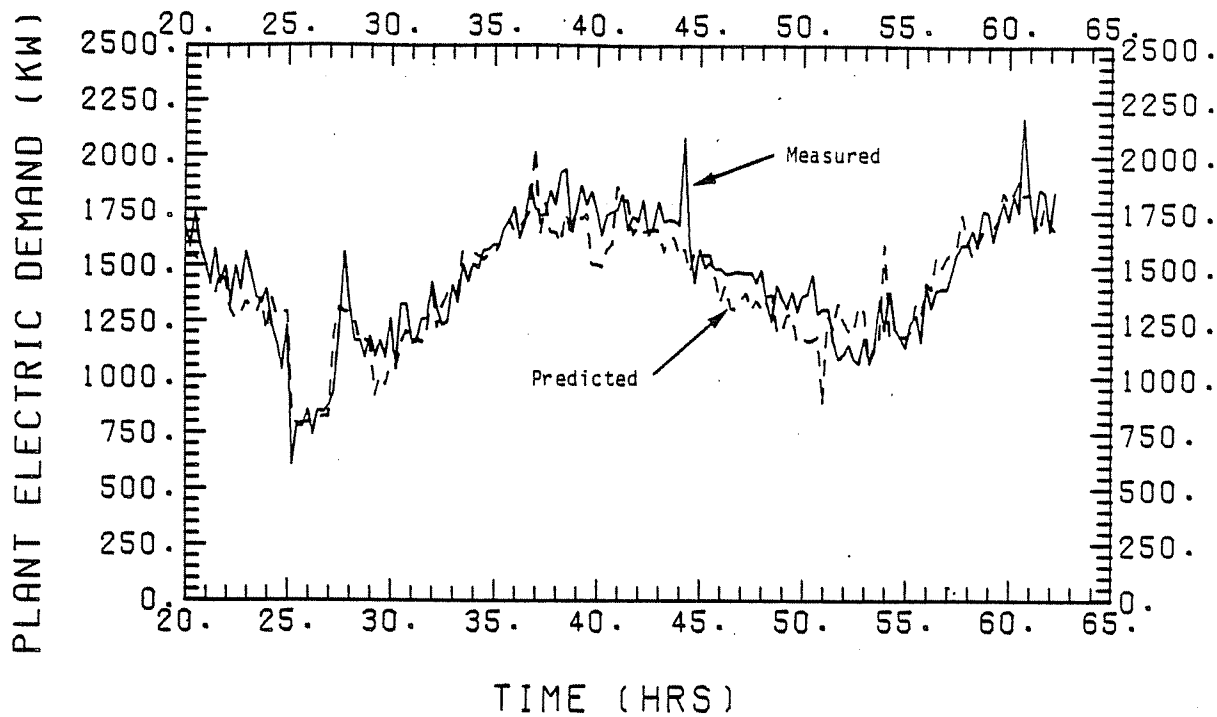

Figure 8. Simulated plant electric demand versus measured data

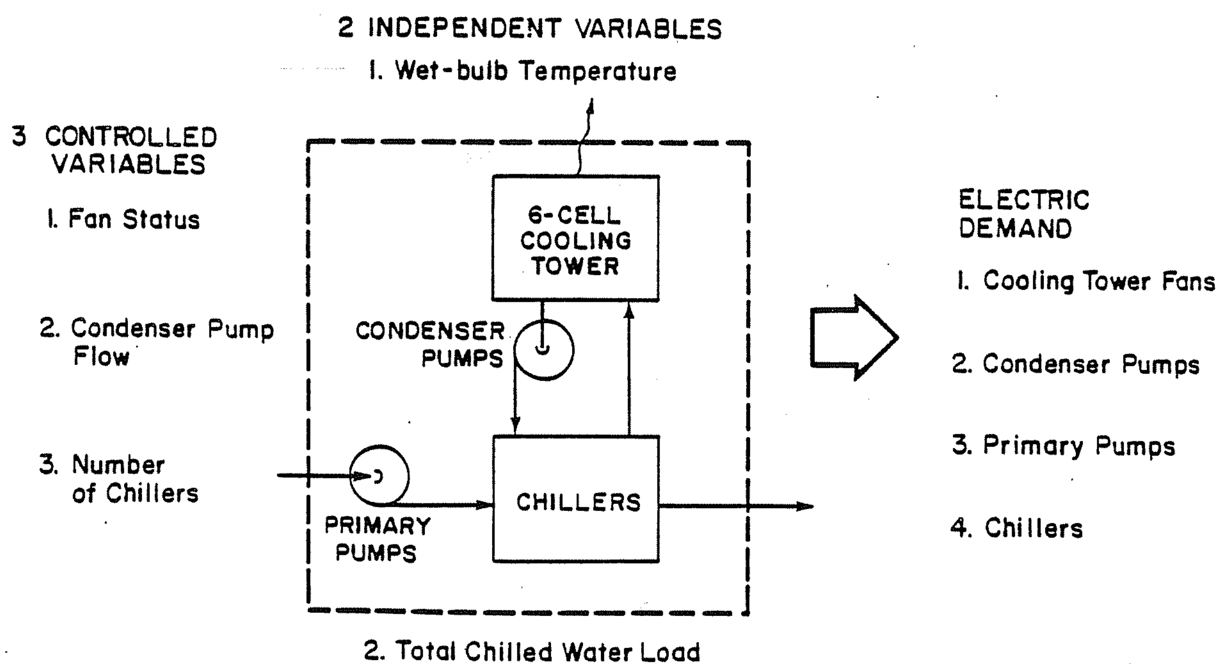

Figure 9. Subsystem for optimization studies