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INVESTIGATION OF CONTROL ALTERNATIVES FOR A STEAM TURBINE DRIVEN CHILLER

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ABSTRACT

The subject of this paper is a large central plant with a steam turbine driven chiller whose refrigerant compressor is equipped with both variable speed and variable vane control. The cooling towers are multi-cell with both one- and two-speed fans. The primary goal of the project was to find ways to reduce the overall operating costs of the chiller plant.

To study the many different options and to assess their costs and benefits, a computer simulation of the chiller plant was developed for use with TRNSYS (Klein et al. 1983). Generic computer models were developed for each of the major equipment components, with the specific parameters determined through measurements on the system.

The control and equipment modifications investigated include alternative fan speed and tower cell sequencing, different fan motor speeds, different condenser water flow rates, free cooling, and the effect of tower air outlet recirculation. Each control strategy was optimized on the basis of minimizing overall operating costs. Optimal control maps were generated to indicate the optimal setting of tower cells, fans, and chiller vane position. By simulating an entire cooling season, optimized control strategy guidelines were developed for various equipment configurations. Energy consumption under optimal strategies is compared to that under current operation.

INTRODUCTION

The operation of a large central chiller plant demands of the operator and the engineer a comprehensive understanding of the interactions of the equipment and the effects of control changes, varying loads, and weather conditions. This knowledge is usually earned through years of on-site experience. When contemplating new equipment or alternative control strategies, the interactions between equipment and the environment are not always obvious, which makes it difficult to predict the associated operational costs. Computer simulation of plant operation and control provides a means by which these decisions can be quantified and day-to-day operations can be optimized.

The subject of this paper is a central plant with a steam turbine driven chiller whose refrigerant compressor is equipped with both variable-speed and variable-vane control. The cooling towers are multi-cell with both one- and two- speed fans. The primary goal of the project was to find ways to reduce the operating costs of the plant. To study the many different options and to assess their costs and benefits, a computer simulation of the chiller plant was developed for use with TRNSYS (Klein et al. 1983). Generic computer models have been developed for each of the major equipment components, with the specific parameters determined through measurements on the system. By simulating an entire cooling season, optimized control strategy guidelines were developed for various equipment configurations. Energy consumption under optimal strategies is compared to that under current operation.

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COOLING FACILITY

A university's chiller plant has been the focus of this study. Chilled water is provided by two 3500-ton chillers, each equipped with free cooling capability, variable guide-vane control, and variable speed compressors. Each chiller is powered by a 3000 hp mechanical drive, condensing steam turbine. The steam for the turbines is supplied by a separate coal-fired plant. Three two-cell, mechanical draft cooling towers with two-speed fans provide the means of heat dissipation. The condenser and chilled water pumps are electrically driven. Figure 1 shows the major system components, the interconnections, and the relevant measured quantities.

MODEL DEVELOPMENT

TRNSYS is a modular simulation program that allows system component models to be formulated independently. These models can then be connected together, analogous to the manner in which pipes and wires connect the actual equipment. The chiller plant simulation is composed of interconnected individual component models for the chillers, turbines, pumps, cooling towers, surface condensers, and controls, each with several input and output variables.

Collected performance data were compared with the simulation results. First, the general component models were made specific to the test site by determination of a set of empirically derived parameters. The component models were then individually validated where possible using additional collected data. Once all the components were tested, the plant simulation was run using the chilled water load and ambient wet-bulb temperature as driving forces along with the associated vane and fan settings as the inputs. The simulated steam consumption was compared to the measured flow as a test of the overall validity of the simulation.

The measured data were collected hourly over the three-month period from May to July 1986. The component parameters and model verification were established using data from an approximately two-week period in May and June. The models were also compared to another, independent set of data in July. The three major components and the models are described below.

Cooling Towers

The heat and mass transfer processes occurring in the cooling tower are simulated using the model devised by Whillier (1967). The Whillier model is less complex and does not require an iterative solution as in the more common Merkel method, which is the basis for the technique described in ASHRAE Fundamentals (1985). Recent studies (Braun 1987) have indicated that there appears to be no advantage to the use of the Merkel model over the Whillier model.

The Whillier model correlates a tower temperature effectiveness term to the ratio of thermal capacities of the air and water streams and an overall coefficient. This effectiveness is then used directly to determine the exit conditions of the air and water streams, in the same manner as for a conventional heat exchanger analysis.

The tower model was tested for its accuracy in predicting the leaving water temperature, T_{cws}. The RMS error for the test data points is 1.53 F (0.85°C). The experimental errors result largely from uncertainties in the estimates of flow rates and temperatures. The flow rates are the least accurately determined variable. The air flow and water flow are assumed to be constant for each fan setting. It is also assumed that the water flow is evenly distributed among the cells. The error associated with the thermocouple readings is approximately 1 F (0.56°C). The agreement between simulation and measurement is within experimental uncertainty. Further details are given in (Nugent 1987).

Steam Turbine

The steam turbine model used in this study is based upon the turbine manufacturer's performance curves. The curves shown in Figure 2 have been generated by the manufacturer specifically for the turbine at hand operating under the measured inlet steam conditions and exhaust vacuum pressure. The discontinuities in these curves are at the points where additional steam inlet valves are opened by hand to increase the supply of steam. These hand valves are not opened during normal operation at the chiller plant.

The steam flow rate shown in Figure 2 has been fit as a quadratic function of power and rpm for each set of valve openings. The actual turbine power output was not measured directly, so this component could not be verified individually. The turbine model was connected to the chiller, and the combined models were tested as described below.

Chiller

There are two modes of chiller operation, mechanical cooling and free cooling, and each is modeled separately. Mechanical cooling is accomplished with the familiar vapor-compression refrigeration cycle. To meet the changing load imposed on the water side of the evaporator, variable inlet guide vanes (dampers) and/or compressor speed control are available to adjust refrigerant flow to maintain the desired leaving chilled water temperature (set point). As presently configured, the guide vanes are manually set and the speed is automatically controlled by a feedback loop between the leaving chilled water temperature and the desired set point temperature.

Braun (1987) has developed an empirical chiller model capable of correlating experimental performance data over a wide range of conditions. This empirical model requires little computation and results in a suitable algorithm for seasonal performance simulations. Braun's model is capable of predicting both the chiller power requirement and the rpm with the same set of variables.

Braun demonstrated that chiller power and speed are primarily a function of two variables, the chilled water load and the temperature difference between the leaving chilled water, T_{chws} (set point), and the leaving condenser water, T_{cwr}. These functional relationships are described by the following quadratic equations;

$$PWR/PWR_{des} = \alpha_0 + \alpha_1 X + \alpha_2 X^2 + \alpha_3 Y + \alpha_4 (XY) \tag{1}$$

$$RPM/RPM_{des} = \beta_0 + \beta_1 X + \beta_2 X^2 + \beta_3 Y + \beta_4 (XY) \tag{2}$$

where:

PWR = chiller power

RPM = chiller speed

X = Q_{chw}/Q_{des}

Y = $(T_{cwr}-T_{chws})/(T_{cwr}-T_{chws})_{des}$

α_i, β_i = empirical constants

des = designates design conditions

The variable Y is related to the pressure (or temperature) difference between which the saturated refrigerant fluid is allowed to evaporate and condense. The X variable, a non-dimensional form of the evaporator load, defines the resulting refrigerant flow rate, which will be required to meet that load within the limitations defined by Y. The design values that normalize the equations are chosen at the maximum or design capacity. The empirical coefficients are then determined by a linear least squares fit to the collected data.

Braun has demonstrated that Equations 1 and 2 predict both variable vane, constant speed and variable speed, constant vane operation quite well. However, the data used to fit the coefficients to these equations must be for one or the other operational scheme.

To model adjustable vane, variable speed operation, the experimental data have been separated into bins of constant vane settings. The coefficients of the power and rpm equations are then fit to each set of bin data, creating an individual relationship for each vane setting. For the same load and leaving water temperature difference, each vane position will require a different power input and resulting compressor speed.

The vane adjustments are primarily used to keep the turbine speed within its allowable operating range. The turbine has both a high- and a low-speed operating limit. At low chiller loads, the required refrigerant mass flow rate is small, causing the compressor, and thus the turbine, to turn slowly. By restricting the refrigerant inlet flow passage, the compressor is forced to turn faster to provide the same mass flow rate. Conversely, at higher loads, a larger mass flow rate is demanded. To keep the speed below the turbine upper speed limit, the vanes must be opened wider, releasing the flow restriction, thus allowing a lower compressor speed.

Adjustable vane control is incorporated into the model by a programmed series of steps. The vanes are first assumed to be 100% open and the turbine speed is calculated with Equation 2. If the speed is below the predetermined lower limit (approximately 3500 rpm) the vane setting is lowered to the next setting and the speed is recalculated with the appropriate coefficients for this new vane position. If the speed is still too low, the vane setting is lowered again. This logic is continued until the speed limitations are satisfied. The collected data show only a few commonly used settings. It was found that small changes in the vane position did not significantly affect the power and speed requirements. The three major settings used in the model are at 100%, 50%, and 20% open. By incorporating this vane control, the speed has been regulated to within close approximation of the recorded values.

In the free cooling mode, the compressor and the expansion valve are physically bypassed, causing the refrigerant to cycle without compression. The warm refrigerant in the evaporator rises by natural convection to the condenser, where the energy of the chilled water load is exchanged directly with the tower water. The refrigerant is then cooled and condensed and returns to the evaporator by gravity to complete the cycle. The free cooling refrigerant flow is shown by the dotted line in Figure 1. Free cooling is almost, but not entirely, "free." This mode alleviates the use of steam to run the turbine, but there is a cost associated with the fan power required to expel the heat absorbed in the condenser.

Free cooling may be utilized only when the ambient wet-bulb temperature is low enough to enable the cooling tower to dissipate the entire chilled water load and, at the same time, return the condenser water at a temperature lower than the chilled water set point. The free cooling mode model treats the process as a conventional heat exchanger with an overall conductance determined from the data.

The chiller and the steam turbine models have been merged together incorporating the intrinsic chiller vane control and speed control plus the turbine and transmission efficiencies. The chilled water load and the condenser water supply variables are input to the combined model. The chiller determines the proper vane position and, in turn, imposes a power and speed requirement on the turbine. The turbine model thus provides an estimate of the necessary steam consumption to meet these requirements. The results from the model for the three month period agree with the hourly recorded values within an RMS error of 1113 lbs/h (506 kg/h), which is approximately 5% of the maximum steam flow. The steam measurement readings are accurate only to within 500 lbs/h (227 kg/h) due to the scale of the circular chart on which the readings are recorded. The condenser water flow rate has been assumed to be constant in this test, although it has been found to vary ±200 gpm (12.6 L/s). As a result, the model predictions are within experimental accuracy of measurements.

OVERALL PLANT SIMULATION

As a test of the overall plant simulation model, all of the components were interconnected to calculate the steam consumption. The two external driving forces input to the simulation were the chilled water load and ambient wet-bulb temperature. The actual cooling tower fan speed settings were used to control the operation, with the chiller model determining the vane position and consequent speed.

The comparison between prediction and measurement is shown in Figure 3. The difference between the calculated and the recorded one hour interval steam readings for the entire period is 4.06%. The RMS error for all the one hour points is 2687 lbs/h (1221 kg/h). The discrepancies are believed to be largely due to the errors associated with reading the steam charts and the time gap (as much as one hour) between the readings of the temperatures and the steam use.

Since the collected data are limited to a three month-period, other sources of wet-bulb and chilled water load have been used for yearly simulations. The load has been correlated as a function of the wet-bulb temperature and the hour of the day. The wet-bulb values used are from the Typical Meterological Year (TMY) weather data (SOLMET).

CONTROL LOGIC DEVELOPMENT

A major goal was to determine which combination of the available chiller plant control options and equipment configurations will minimize the overall operating costs. To achieve this, a consideration of the interactions between the system components is necessary requiring that simulations of the entire system performance for the cooling season be run for each alternative.

Control Elements

The end use of the chilled water dictates the load and set point temperature that, in this study, has been taken as inputs. The tower airflow, which is regulated by the tower cells and fan speeds, determines the condenser water return temperature, which in turn dictates the chiller power, vane position, and speed for the given load. Tower fan control is, therefore, a very important control element.

To investigate the effect of the fan combinations, the total system power consumption has been simulated, at a constant load and wet-bulb, for all combinations of fan speeds and number of tower cells. The results are shown in Figure 4 for one load and wet-bulb. The fraction of total fan air flow is the ratio of the sum total of all the tower cell airflows to the maximum flow rate for all cells operating at full speed. The turbine work has been converted to equivalent kilowatt hours to be consistent with pumping and fan power units. Figure 4 indicates that the total power required to meet the load is substantially increased with a decreased number of cells. This is due to the fact that fan power is a cubic function of airflow. Similar results were found for other load and wet- bulb conditions. The best fan strategy is to operate as many cells in parallel as possible at a total airflow rate that is optimal for the load and wet-bulb temperature. The limiting factor to this fan control strategy is the number of fan speeds and cells that can be independently controlled. Minimum operating costs will result with variable fan speed control. However, two- or three-speed fans with multiple cell towers provide nearly optimum control.

Performance Maps

To simplify the control decision-making process, operating maps were developed that indicate the mode of cooling (mechanical or free cooling), the fan settings, and the vane setting that results in the minimum instantaneous total operating cost for any given set of operating conditions. The maps reduce the decisions to two measurable variables: chilled water load and ambient wet-bulb temperature.

The development of these maps is dependent upon the unit fuel costs being considered. Therefore, the fuel costs are an input to the map generation process. Producing the operating maps is a two-step process involving the production of general fuel use matrices followed by optimization for the specific unit fuel costs.

The operating maps are optimized based on the minimum total instantaneous operating cost. Although the unit fuel costs may vary and can have a large effect on the optimization, the quantity of fuel consumed is constant for each given set of operating criteria. Knowing the fuel use, the total cost can be determined for any unit fuel cost by simply multiplying the quantity of fuel by the respective unit cost.

For each particular fan setting, the instantaneous performance for every combination of load and wet-bulb was simulated. The resulting electric use, steam use, and refrigerant compressor vane position were recorded in three separate matrices. These simulations were repeated for each of the fan setting combinations. The result was a set of three-dimensional matrices, one for each output. The rows and columns are defined by the load and wet-bulb, respectively, and the discrete fan setting value defines the third dimension, as depicted in Figure 5. With these three matrices, the required vane position and the resulting instantaneous fuel use can be determined for all combinations of fan settings at any given set of external conditions.

The fan setting combination number increases with increasing tower airflow rate, as depicted in Figure 6. The numbers listed to the right of the fan setting combination number are the relative fan speeds for each of the two cells of the three towers. The actual system limited fan speed changes to one-half speed increments for four of the cells and full speed increments for two of the cells. As the combination number increases, the flow rate increases due to steps in the fan speeds. As shown in Figure 6, the progression is to increase the total flow rate in steps of one-half speed, although, as seen for the change from combination 4 to 5, as one fan is turned on to full speed, a compensation is made by turning off a one-half speed fan.

This same methodology was used to define the free cooling mode. Since the compressor is bypassed and the turbine is inoperative in the free cooling mode, only the electrical consumption matrix was generated. If, at a given set of load and wet-bulb, the tower is unable to provide adequate heat dissipation or deliver the condenser water at the required temperature, free cooling is not attainable. This is indicated by a zero in that matrix element.

In order to generate an operating map, the previously determined steam and electric matrices are used together with the three independent variables, load, wet-bulb and fuel costs. The sum total fuel cost is then calculated for each fan setting. For this plant, the fuel costs used are \$4.08/1000 lb_m (\$8.97/1000 kg) of steam and \$0.05/kWh of electricity. Figure 7 is a plot of the total cost for several fan settings over a range of loads, all at a constant wet-bulb temperature and fixed fuel costs. (For visual clarity, only three of the twelve possible fan speed combinations have been plotted.) At each load and wet-bulb, there exists a minimum cost. The fan setting associated with this minimum is the desired optimum setting for that particular load and wet-bulb. The vane position that is associated with this load, wet-bulb, and choice of fan setting is retrievable from the vane matrix. Similar plots result for the full range of wet-bulb temperatures. At moderate loads and wet-bulb temperatures, the optimum operating cost is not very sensitive to the fan setting choice.

At each combination of load and wet-bulb, the optimum fan setting and associated chiller vane position are recorded. What emerges from this are unique operating mode maps that indicate the fan and vane settings that will minimize the overall costs for the entire operating range at the particular unit fuel costs.

The resulting maps for the fan and vane control for the mechanical cooling mode are shown in Figure 8 and Figure 9 for electric and steam costs of \$0.05/kWh and \$4.08/1000 lbm (\$8.97/1000 kg), respectively. These maps are useful for making plant control decisions in terms of the two independent variables, load and wet-bulb. For example, if the load is 2200 tons (7735 kW) and the wet-bulb is 50 F (10°C), Figure 8 indicates that either fan combination 3 or 4 will result in minimum cost operation. From Figure 6, this is seen to be either three or four fans at half-speed. From Figure 9, the optimum vane setting is seen to be 50% of wide open. As either the load or wet-bulb temperature changes, these maps prescribe the changes necessary to achieve optimal operation. These maps have been programmed into a microcomputer and are currently used by the plant operators to make the control decisions.

Figure 10 shows the fan control during the free cooling mode. As indicated, there is a limit in the load and wet-bulb conditions for which free cooling is attainable. If free cooling is possible, the desired fan settings for instantaneous optimization are as indicated.

SEASONAL SIMULATIONS

The seasonal simulations utilize unique control maps developed for each control alternative. These optimized maps were generated for the present average unit fuel costs using the methodology described above. Each simulation was executed using identical load and climatic profiles and a constant condenser water flow rate. The control decisions are made each time step by finding the optimum fan setting combination in the respective control map for the load and wet-bulb inputs. For all of the alternatives, unless otherwise stated, the free cooling mode is utilized if possible.

Base Case

Presently the plant control is based on manual operator responses to changing loads and climatic conditions. The complexity of these decisions made it difficult to develop a simple, logical, step-wise computerized control scheme capable of emulating the present plant operation. Instead, from discussions with the operators and examination of the data logs, a close approximation has been established. This logic is used as the base case to which the alternatives are compared.

The investigation revealed that, as the load builds, the two-speed fans are generally turned on, one at a time, in half-speed increments. Towers 1 and 2, equipped with the two-speed fans, are virtually always operating. As the need arises, the additional two full-speed tower cells are added. At all times, at least one fan in each open tower is operating to reduce blow-down water losses (i.e., the water that spills out the sides of the tower due to lack of air intake suction). As the load increases, the fan settings proceed downward in the table in Figure 6 but not including combination numbers 5 and 7.

The base case seasonal operating costs have been generated using the Madison TMY wet-bulb data and the simulated chilled water load. For the 5640 hours of cooling season operation, the total accumulated load is 9.5×10^6 ton-hours (120 GJ) requiring 3.12×10^6 kWh of electrical power and 6.2×10^7 lb_m (2.82×10^7 kg) of steam resulting in a total annual operating cost of $409,100. The alternative strategies are compared relative to this annual cost.

Two Tower Cell Operation

The original design of this chiller plant allotted one tower with two cells to each individual chiller. In 1982, the three towers were plumbed together to allow one chiller to utilize the full capacity of all six cells. To quantify the benefit of this tower reconfiguration, a representative tower fan sequencing scenario for the one tower per chiller operation was devised. There are only two two-speed fans to consider, producing four possible fan combinations. In Figure 11, the seasonal operating cost for this mode, labeled 2-Cell, relative to the base case, is shown. The results indicate that operating the plant in this manner would cost substantially more per season (approximately $17,100) than operating under the base case control scenario. The tower interconnections that were made produce significant savings. The tower interconnection has an added advantage supplementing the financial savings. Should some mechanical difficulty arise in one of the towers, it can be isolated and repaired while the others carry the load. This alleviates down time and costly purchased capacity.

Single-Speed Fans

The seasonal cost advantage of using two-speed fans, as is presently configured, compared to one-speed fans was simulated. Single-speed fans for each tower cell results in six possible combinations. Each of the six cells are sequentially activated, at full speed, until all six are operating simultaneously. Because all six cells are utilized, the water to air heat exchange area is increased, thus improving the tower performance and decreasing the tower approach temperature. This substantially decreases the chiller power required and subsequently the steam consumption but at the cost of additional electrical power to drive the high speed fans. As shown in Figure 11 labeled 1-Spd, the seasonal simulation indicates that the six one speed fans provide a slight savings over the two-cell operation but still cost about $15,400 annually more than the base case with the two-speed fans.

Refined Base Case

The operation of the existing equipment defined by the base case has the potential of being slightly improved. The improvement comes at a cost of additional fan speed and tower interconnection manipulation. These costs, which represent added man hours and possible increased blow-down losses, are very difficult to predict and are not included in the simulation.

The concept of operating as many cells as possible at part speed is more closely simulated by bringing the two-speed fans on line one at a time at half-speed until all four cells are operating. The half-speed increments are continued by incorporating the one-speed, high velocity fans one at a time and counterbalancing the large increase in speed by turning off a one-half-speed fan. This is continued until all six cells are on at full speed. This refined scenario is labeled Optm in Figure 11. It results in a slight reduction in cost over the base case ($1800 annually). The savings are due to slight decreases in both steam and electrical demand due to the closer adherence to the optimal fan sequencing.

Two-Speed Fans

The savings incurred by the use of two-speed fans in two of the towers provide the motivation to investigate replacing the single-speed fans in the third tower with two-speed fans. This scenario was simulated with 12 possible fan combinations with each of the six cells being activated one at a time at half-speed increments. At least one cell of each active tower was in use at all conditions. This fan control scenario provided the most savings above the base case. Both the electric and steam costs are decreased, resulting in a total savings of $4600 a season over the base case, as shown in Figure 11 by the bar labelled 2-Spd. The additional speed provides another degree of freedom when devising control strategies. The half-speed increments should be used at all times in an effort to continually operate as many cells at part speed whenever possible.

Approximation Of Actual Operation

Although it was not possible to emulate the current operation with a computerized control logic, the resulting operational costs can be approximated. For the test period, the actual control scheme was simulated by using the recorded load, wet-bulb, and actual fan speed settings as inputs. For the same period, the base case has also been simulated using the same recorded load and wet-bulb inputs, but the respective control maps were used to determine the fan settings. The results of these simulations indicate that the actual steam flow is 3.9% less than that predicted by the base case, but the electrical demand is 15.8% more. Due to the relative costs of the fuels, the base case is ultimately 1.65% less costly than the current actual control scheme. It was assumed that these proportional differences remained constant throughout the year, resulting in an annual cost reduction of $7046 over the current operation, labeled Current in Figure 11, due to more careful control.

Free Cooling

All the seasonal simulations thus far have utilized free cooling whenever possible. To determine the economic benefit associated with free cooling, the base case seasonal simulation was executed without free cooling using the mechanical cooling mode only. The results of this simulation are labeled Mech in Figure 11. They indicate that, by operating in the mechanical cooling mode all season, there is a $22,100 increase in operating costs. The added cost is associated with the increase in steam consumption necessary to keep the turbine running all season.

Tower Air Recirculation

At the central chilled water facility, a decorative facade surrounding the cooling towers has been built in an effort to improve the aesthetics of the plant. This facade restricts the flow of fresh ambient intake air to some extent and causes some recirculation of the approximately saturated tower exhaust air. This recirculated moist air impairs the evaporative heat exchange occurring in the tower.

To determine the amount of recirculation, experimental measurements of the airstream humidity ratios were recorded. A psychrometer was used to measure the wet-bulb and dry-bulb temperatures at three sites: (1) the tower air inlet, (2) the tower exhaust, and (3) ambient conditions well removed from the tower. The recirculation rate was determined from a mass balance on the moist airflows. The psychometric readings taken on three separate calm, dry days in September indicated a range of recirculation rates from 8.9% to 19.4% with an average rate of 13.5%. This compares to the literature, which indicates that generally a recirculation rate of 5-8% can be assumed for a freestanding cooling tower.

To simulate the effect of recirculation of moist air on the plant performance, the wet- and dry-bulb temperatures of the tower inlet air, which is a mixture of fresh and recirculated air, are determined assuming adiabatic mixing. For a given tower fan setting, the total airflow is known and the relative ambient air and recirculation airflows are determined. This scenario has been simulated over the entire season for a number of assumed constant recirculation rates ranging from 5% to 15%. The savings realized by removing the wall and assuming a 5% recirculation rate is labeled 5% Recirc in Figure 11. The reduction in cost due to reducing recirculation from 13.5% to 5% is approximately $2210.

Condenser Water Flow Rate

The condenser water pumps at the plant are an example of a fixed-setting control element. They are single-speed pumps with flow rates preset by in-line restriction valves. This flow rate affects the individual performance of all the major components, the cooling towers, chillers, turbines, and surface condensers. The flow rate is, therefore, optimized by its overall effect on the plant performance. Several different constant condenser water flow rates were simulated for the full season. The optimum was found to be relatively flat within 15% of the flow rate that produced minimum operating costs. The plant performance is, therefore, relatively insensitive to the condenser water flowrate near the optimum.

CONCLUSIONS

This study has demonstrated the advantages of computer simulation and control of energy systems. The proper operation of cooling towers and chillers can provide substantial savings of operating costs. In addition, plant simulation provides an instructional tool for new operators. The major fuel cost savings over a base case operation, in order of impact, were determined to be:

1. The use of free cooling at low wet-bulb temperatures.
2. The interconnection of towers to allow as many cells as possible to be utilized.
3. The use of multiple-speed fans in the cooling tower.
4. The continual updating of the set points to ensure optimal control at all times.
5. Changing single-speed fans to two-speed units.

The changes that were investigated and that would not result in substantial savings are:

1. Changing fan motor control from two-speed operation to continuously varying speed units.
2. Removing the decorative cooling tower facade to reduce air recirculation.
3. Changing the condenser water flowrate.

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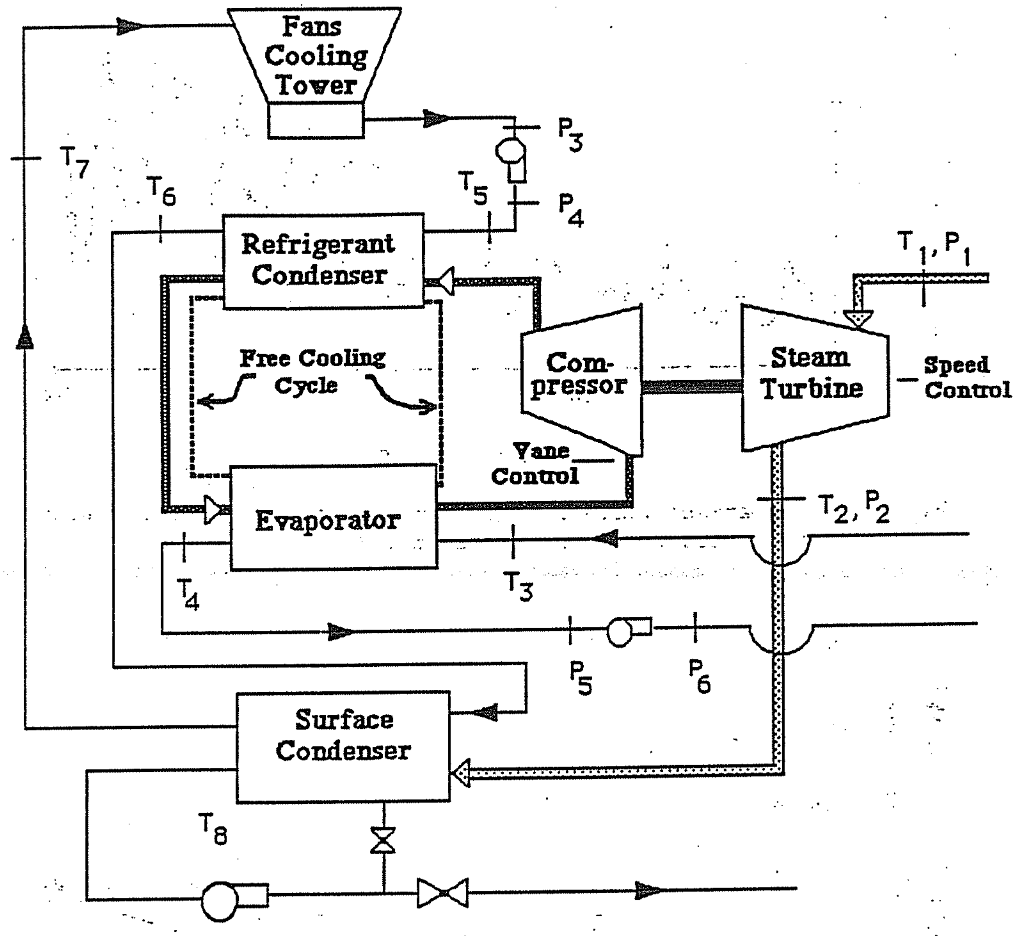


Figure 1 Schematic of the test facility showing data reading locations

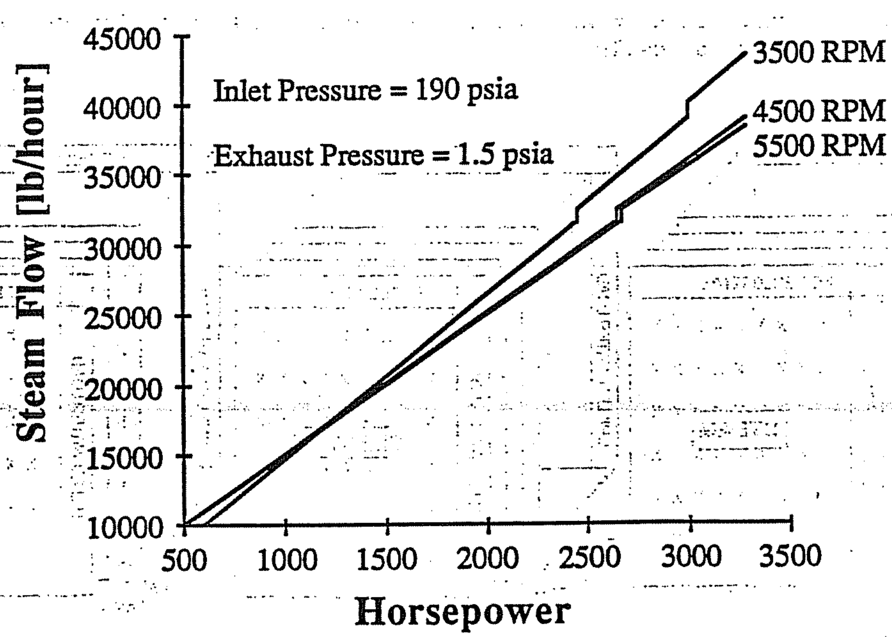


Figure 2 Turbine manufacturer's performance curves

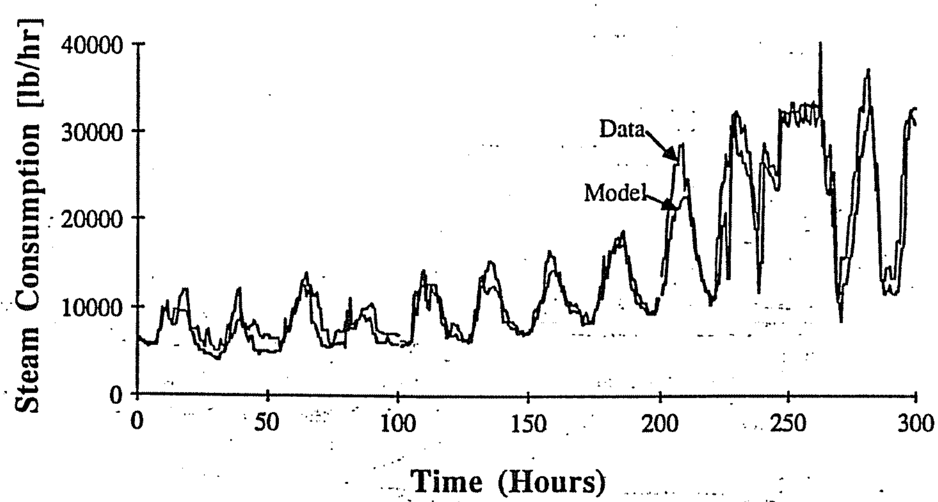


Figure 3 Comparison between steam consumption as measured and as predicted using the system model

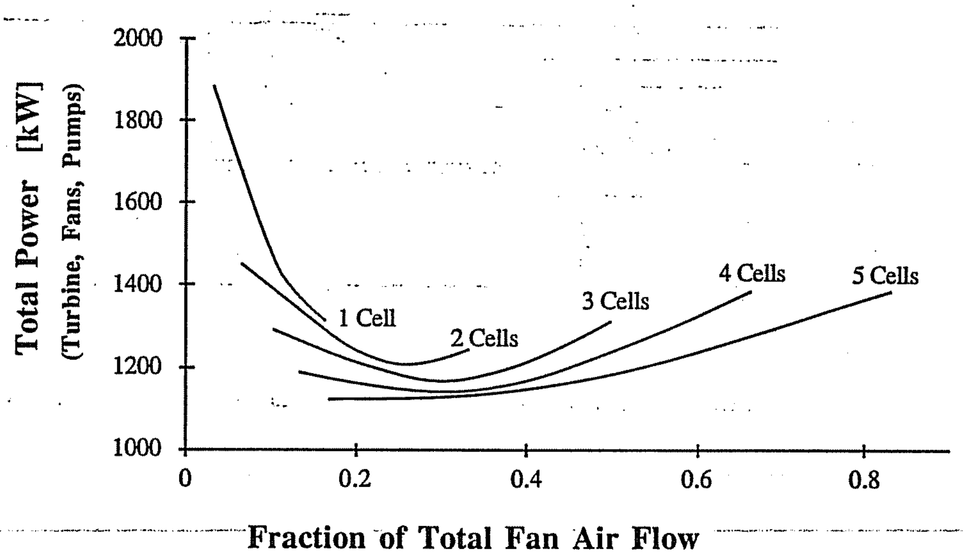


Figure 4 Variation of plant total power as a function of fan speed and number of cells

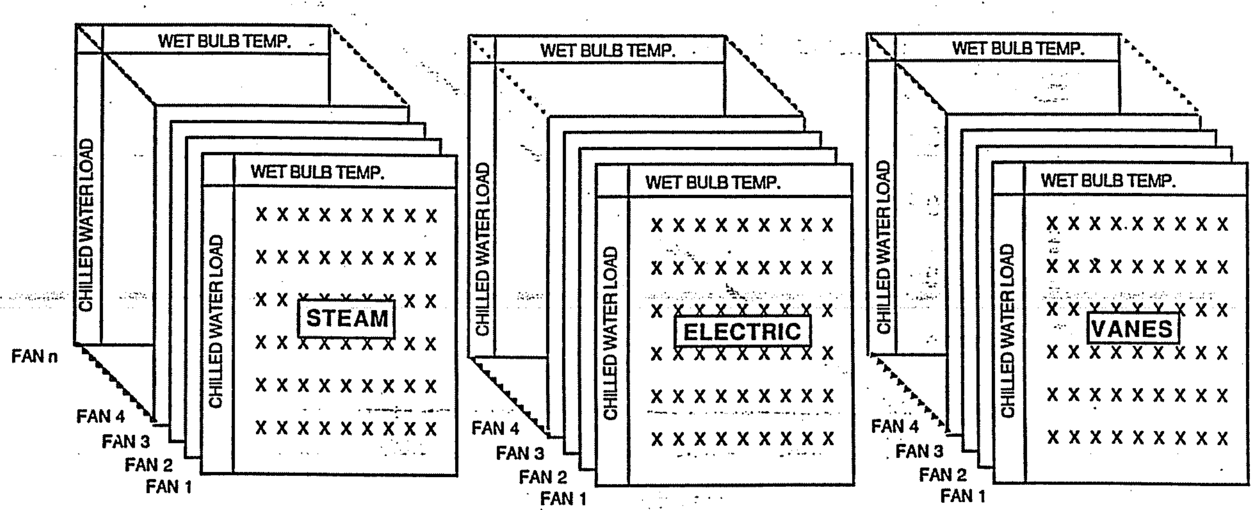
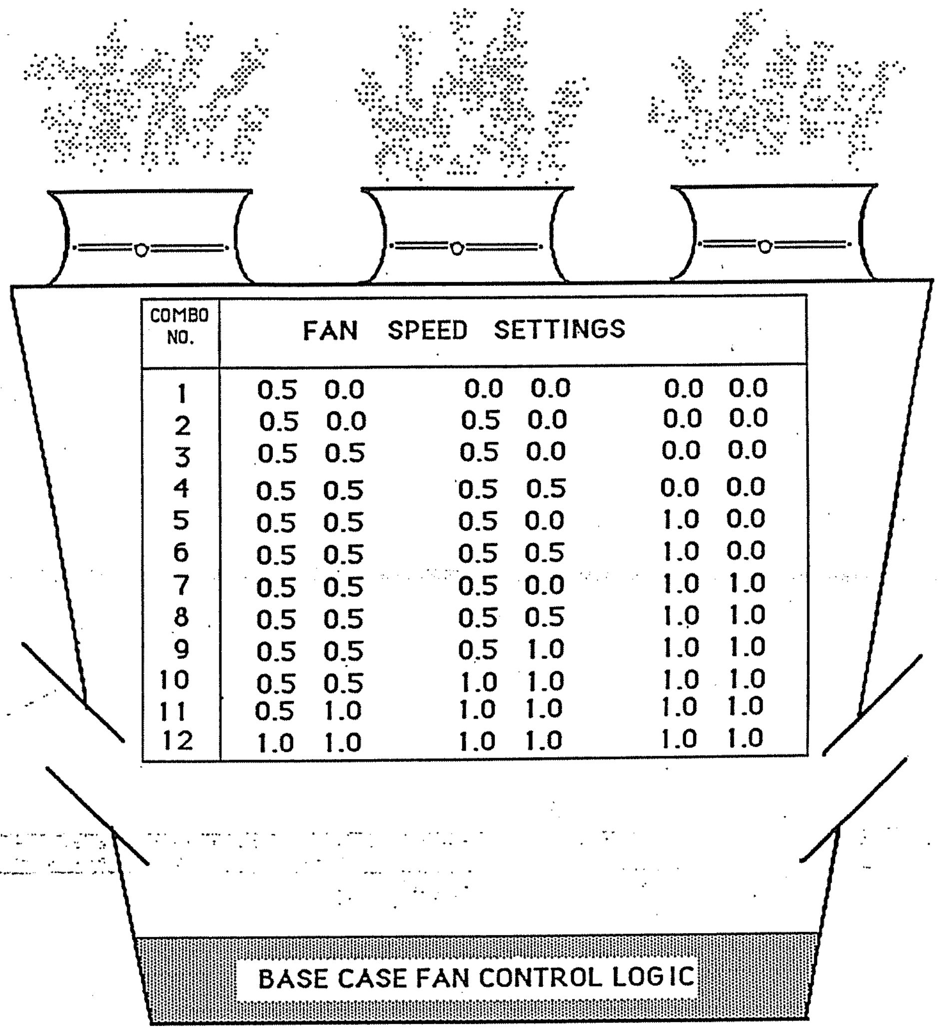


Figure 5 Schematic of the development of the matrices for optimal control



COMBO NO.	FAN SPEED SETTINGS					
1	0.5	0.0	0.0	0.0	0.0	0.0
2	0.5	0.0	0.5	0.0	0.0	0.0
3	0.5	0.5	0.5	0.0	0.0	0.0
4	0.5	0.5	0.5	0.5	0.0	0.0
5	0.5	0.5	0.5	0.0	1.0	0.0
6	0.5	0.5	0.5	0.5	1.0	0.0
7	0.5	0.5	0.5	0.0	1.0	1.0
8	0.5	0.5	0.5	0.5	1.0	1.0
9	0.5	0.5	0.5	1.0	1.0	1.0
10	0.5	0.5	1.0	1.0	1.0	1.0
11	0.5	1.0	1.0	1.0	1.0	1.0
12	1.0	1.0	1.0	1.0	1.0	1.0

Figure 6 Fan control combination numbers and associated relative fan speeds

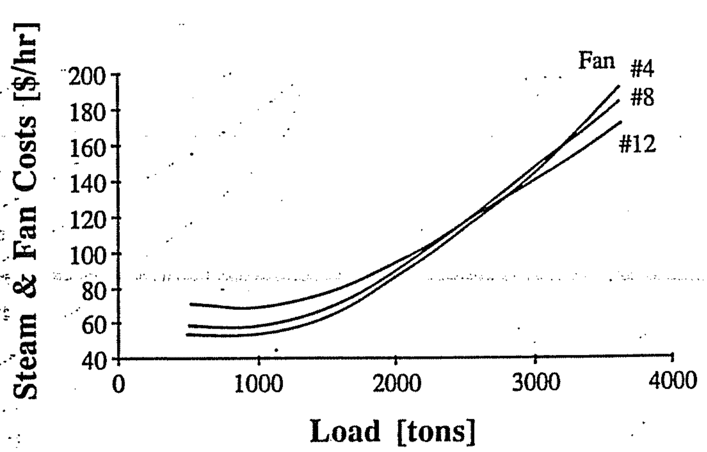


Figure 7 Total steam and fan power costs as a function of load for different fan speed combinations

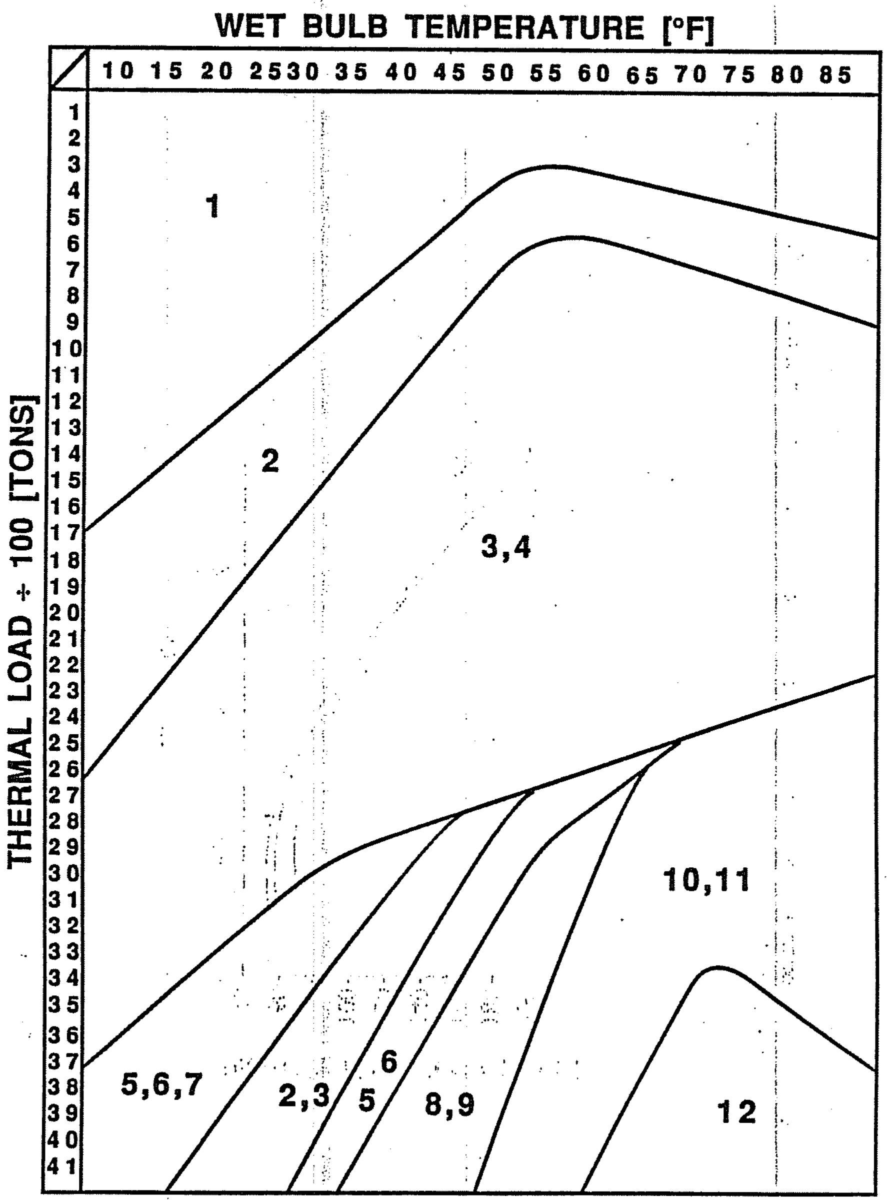


Figure 8 Fan control matrix as a function of load and wet bulb temperature for mechanical cooling

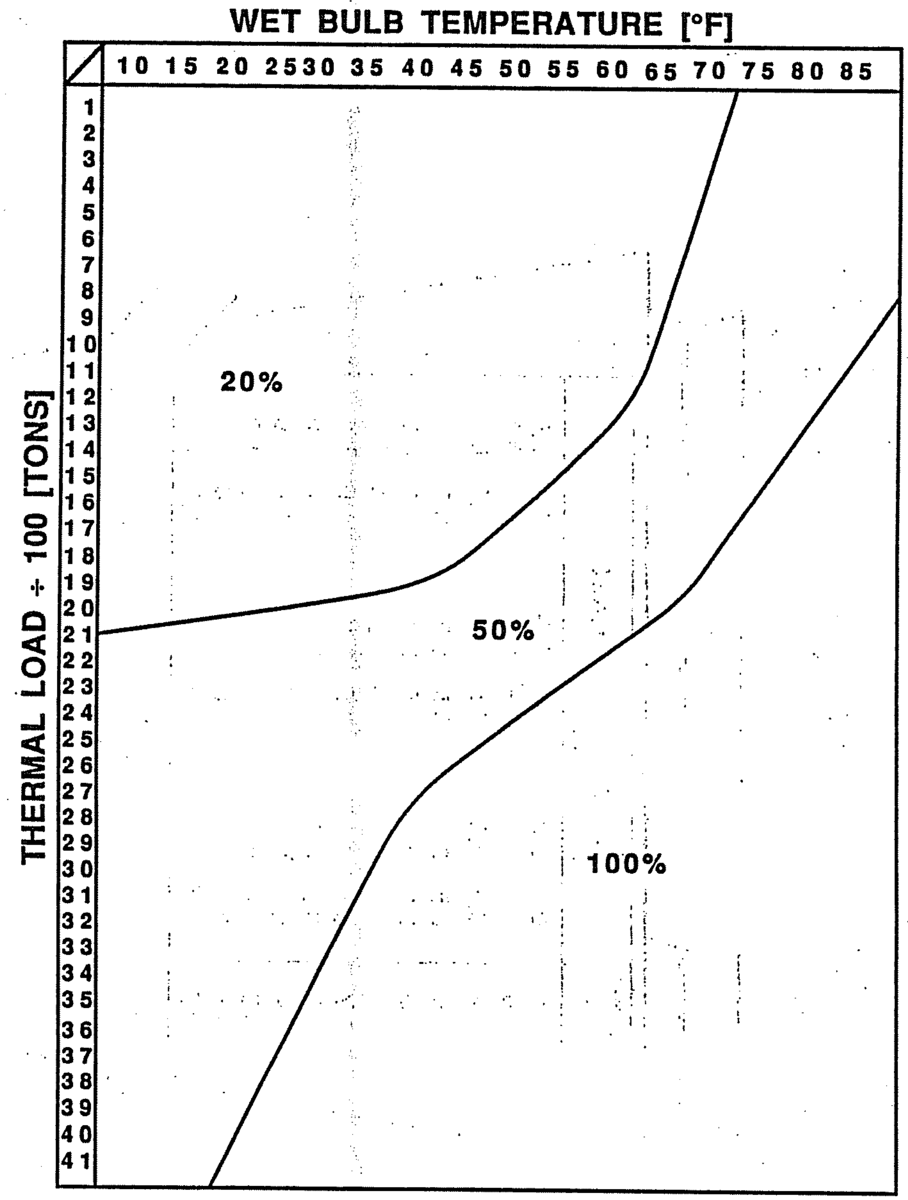


Figure 9 Chiller vane control matrix as a function of load and wet temperature for mechanical cooling

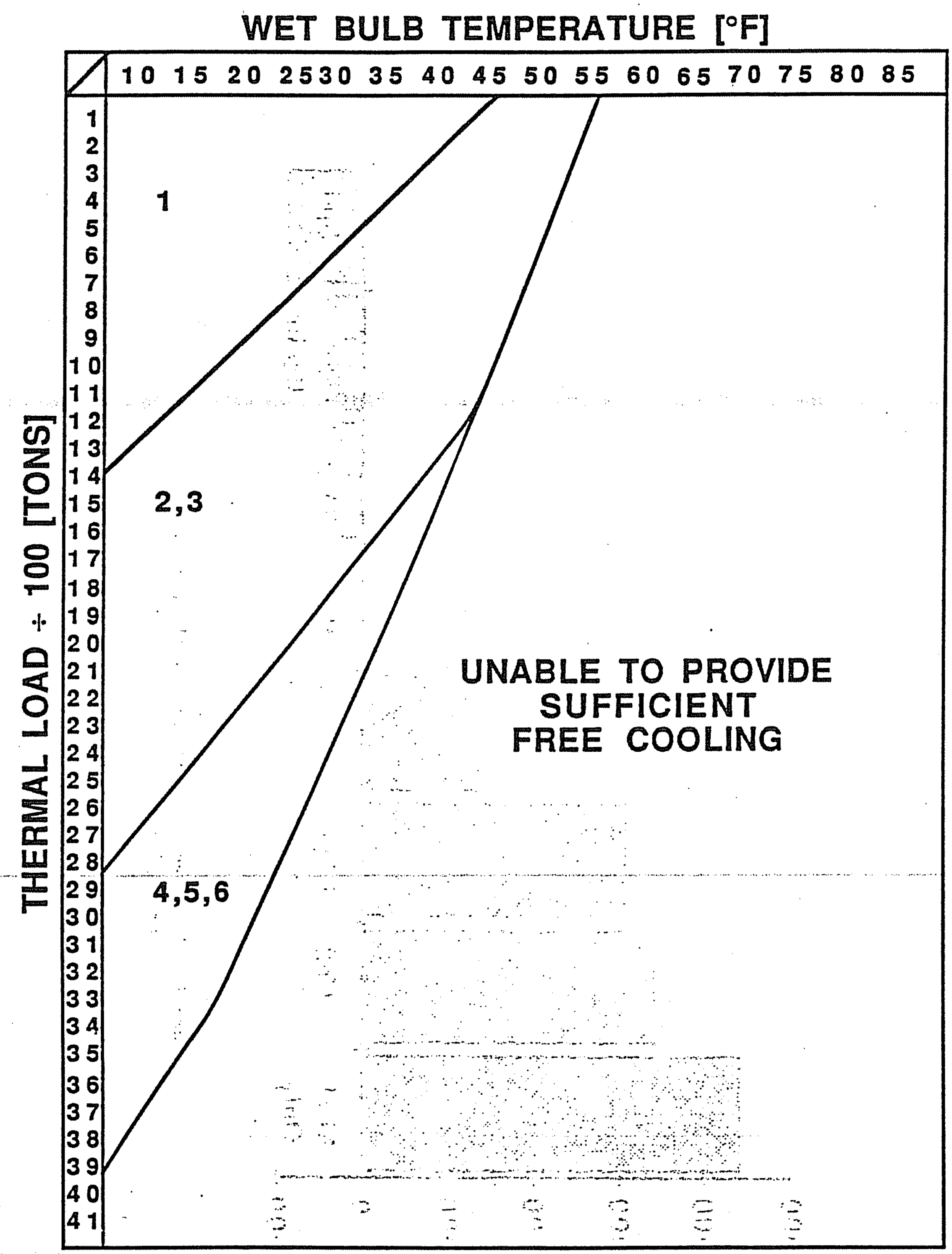


Figure 10 Fan control matrix as a function of load and wet bulb temperature for free cooling

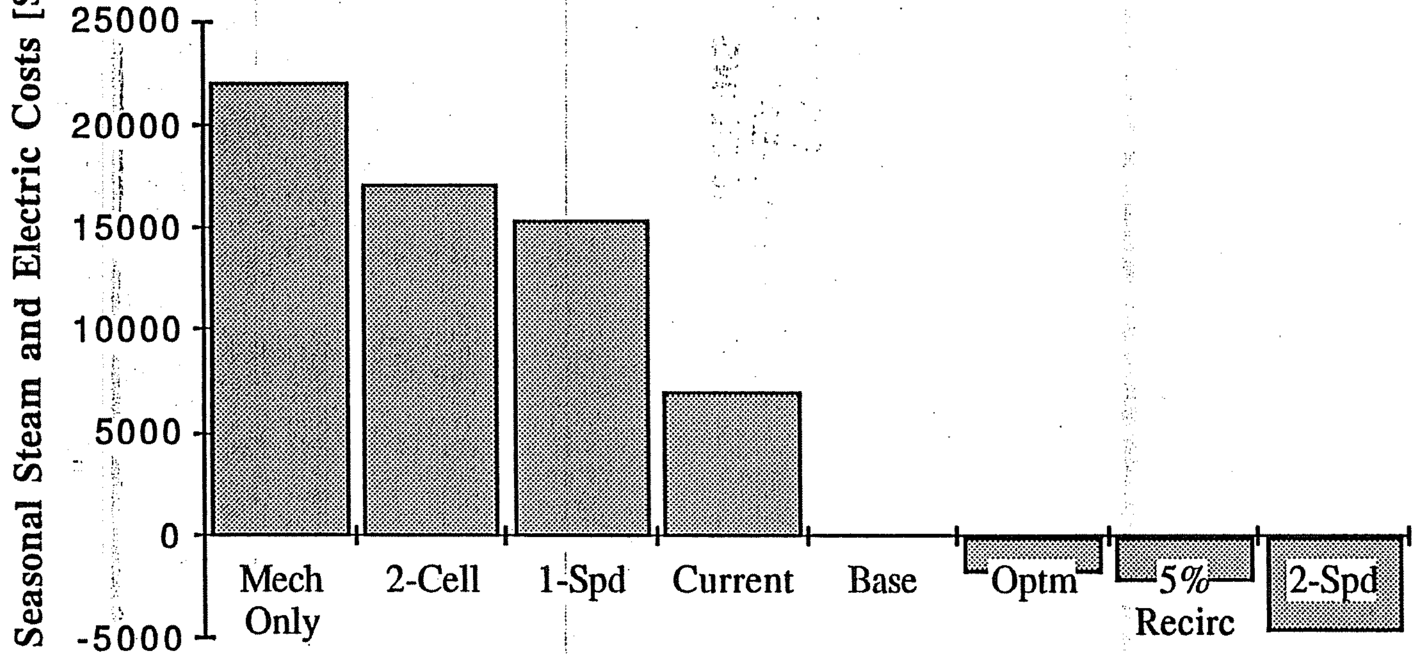


Figure 11 Total seasonal steam and electrical costs for different plant modifications

Discussion

G.J. WILLIAMS, McClure Engineering Associates, St. Louis, MO: Although I am aware that some of the energy retrofits were completed prior to your study, were any estimates of the installed cost of these options made to determine which, if any, of the efforts would be viable as energy retrofit projects based on a return on investment through energy savings?

S.A. KLEIN: The retrofit projects at this facility were (1) addition of free-cooling capability on one of the two chillers; (2) tying together the cooling towers so that all six cells could operate with either one or both chillers in operation; and (3) variable-speed control of the steam turbine and chiller (between speeds of 3500 and 5500 rpm). The addition of free-cooling capability had the largest effect on reducing operating costs. Although I do not have the cost of the retrofit at hand, I have been assured by the facility operators that it was a cost-effective retrofit. The cost of replumbing the towers was small. Our estimates showed a savings of $16,000 per year. The facility operators estimated a reduction of $30,000 in the two-year period after the retrofit was made. The saving of the variable-speed retrofit was not specifically examined.