DEVELOPMENT OF OPTIMUM COMPUTER CONTROL FOR A LARGE STEAM TURBINE DRIVEN CHILLER PLANT

by

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ABSTRACT

This thesis presents a study of operational control optimization for steam turbine driven central chilled water plants. A computer simulation tool was constructed and used to investigate the performance of various control strategies. Each strategy was optimized on the basis of minimum overall operating costs. Optimal control "maps" were then generated. These maps were used to make the control decisions in seasonal simulations. The results of these seasonal simulations were then compared and conclusions and recommendations made.

The simulation program is based on TRNSYS, a modular transient simulation program developed at the University of Wisconsin-Madison. New component models were developed as necessary and all the components were verified using experimental data. The Walnut Street Chiller Plant, located on the University of Wisconsin campus, was used as the test facility. The plant was instrumented and data was collected for the three month period from May to July, 1986.

The plant performance simulator, composed of the verified models, was used to predict the cost of operating the plant under various control strategies, equipment configurations and varying fuel costs. An optimization method has been developed that allows for the fluctuations in fuel costs by using the unit costs as the independent variable. Assuming the present average as the fixed fuel costs, each scenario was optimized and the resulting equipment control logic was generated. This logic has been programmed into the seasonal simulator and full season performance and costs have been generated.

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A base case scenario has been devised which closely approximates the present, actual plant control. To assess the economic merit of each scenario, each have been compared relative to this base case.

The control and equipment modifications investigated include alternative fan speed and tower cell sequencing, different fan motor speeds, different condenser water flow rates and free cooling. Estimates of the potential reduction of electrical demand and steam consumption at the Walnut Street plant, as a result of utilizing the alternative control schemes, are presented

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NOMENCLATURE

Cp	specific heat
E	cooling tower effectiveness
FLCHW	mass flow rate of chilled water
FLCW	mass flow rate of condenser water
FLSTM	mass flow rate of steam
H	total enthalpy of pure substance
hp	pump head
HR	hour of the day
h_s	sigma energy of a moist air mixture per unit mass of dry air
M	mass flow rate
η	efficiency
P	pressure
PWR	power
Q	heat transfer rate
R	cooling tower capacity ratio
RPM	revolutions per minute
T	temperature
W	work
ω	humidity ratio
γ	specific weight
X	ratio of chilled water load to design load
Y	ratio of chiller leaving water temperature difference to design temperature difference

NOMENCLATURE (continued)

Subscripts

a	air stream
ch	chiller
chwr	chilled water return
chws	chilled water supply
cw	condenser water
cwr	condenser water return
cws	condenser water supply
des	design conditions
FC	free cooling mode
r	recirculation
stmr	steam flow return
stms	steam flow supply
t	turbine
twr	tower return
w	water stream
wb	wet bulb

Superscripts

ID	ideal conditions, isentropic

1.0 INTRODUCTION

The operation of a large central chiller plant is a complex task. It demands of the operator and the engineer a comprehensive understanding of the interactions of the equipment, the effects of control changes, varying loads and weather conditions. This knowledge is presently earned through years of on-site experience. When contemplating new equipment or alternative control strategies the interactions are not always intuitively obvious making it very difficult to predict the associated operational costs.

Diminishing resources and increasing fuel costs have added to the complexity of plant operations. The operator must not only provide the required cooling capacity to the end users but do so at a minimized cost. The use of computers for monitoring and controlling of mechanical equipment is growing. To utilize microprocessor technology the control variables must be identified and the decision making logic must be established.

Central chiller plants have been proven to be good applications of computerized energy management systems. Previous studies by Hackner (1983) and Lau (1984) have demonstrated significant savings potential for properly managed electrically driven central chilled water systems in commercial building applications. Braun (1987) has established operating guidelines for a much larger electrically driven chiller system at the DFW Airport. Each study derives an optimum control logic endemic to the application.

This study differs from those cited in that it is focused on a steam turbine driven chiller. Often in large central plant applications where heating and cooling are being generated the high grade electrical energy is replaced with the lower grade, readily available steam energy. Also, in this study, the refrigerant compressor is equipped with both variable speed and variable vane control.

The primary goal is to reduce the overall chiller plant operating costs. To study the many different options and to assess their costs and benefits a computer simulation tool has been developed based on TRNSYS (Klein 1983). Generic computer models have been developed for each of the major equipment components for use in the TRNSYS simulation. By simulating an entire cooling season, optimized control strategy guidelines are developed for various operational configurations.

The results of this research provide the necessary tools to investigate and quantitatively describe the operation of a central chilled water plant. The costs and benefits of alternative control strategies and equipment configurations may then be determined. This simulation may also be useful as an instructional tool to accelerate the learning curve for new operators.

2.0 TEST FACILITY, INSTRUMENTATION AND DATA COLLECTION

The Walnut Street Chiller Plant, at the University of Wisconsin-Madison, has been used as the test facility. This plant delivers chilled water to the west campus, primarily serving the University hospital. Chilled water is provided by two 3500 ton chillers, equipped with variable guide vane control and variable speed compressors. Each chiller is powered by a 3000 horse power mechanical drive, condensing steam turbine. The steam to the turbines is supplied by a separate coal-fired plant. Three, 2 cell, mechanical draft cooling towers provide the means of heat dissipation. The condenser and chilled water pumps are electrically driven. Figure 2.1 shows these major system components and the interconnections as modeled in the TRNSYS simulation.

The generic component models developed for use in TRNSYS have been specialized using equipment parameters from the Walnut Street Plant. The model testing, verification and resulting operational guidelines presented herein are therefore specific to this facility. The simulation models and presented analysis are however, applicable to other similar chiller plants provided that the appropriate parameter values are supplied to the simulation model.

An initial investigation of the plant and the potential modeling algorithms determined the necessary data collection points. The data consists of numerous temperature measurements throughout the system, system pressures, flow rates, steam conditions and operational; information. Figure 2.1 schematically illustrates the location of each data point.

Many of these points were previously being manually read and recorded in the plant operation logs at two hour intervals. A sample copy of the log for a single day is included in Appendix C. It was determined that to more closely model the transient performance of the components, one hour interval readings were necessary. The existing

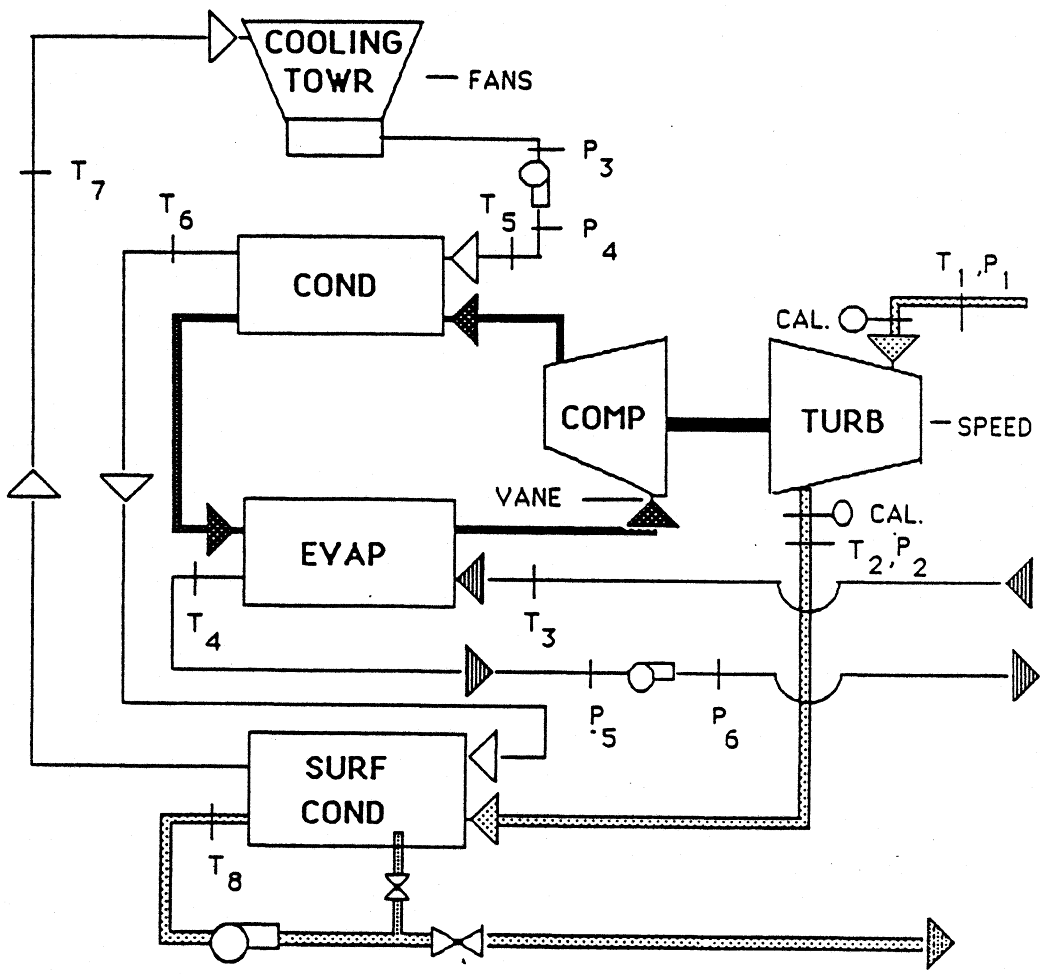


Figure 2.1 Schematic diagram of Walnut Street test facility with data reading locations

temperature sensors were standard mercury tube thermometers. Due to the small temperature drops across the condenser and evaporator (approximately 4 to 10 °F) misreading one degree could represent up to 25% error. For this reason copper-constantan thermocouples were installed in the thermometer wells at all the necessary points. These thermocouples were then connected to a centrally located Fluke Company, anolog to digital, ten channel thermocouple reader. All the temperatures could then be read simultaneously with better accuracy from the same location.

The pressure measurements were attained using the existing Bourdon-tube gages. These gages, as well as the flow measurement devices, were tested and calibrated by the plant personnel prior to this study. The turbine exhaust pressure was also verified with a manometer measurement.

The flow rates are measured in two different manners. The steam and chilled water flows are recorded continuously via pneumatic connections to in-line orifice meters. These measurements are registered on circular charts throughout the day. A sample chart is included in Appendix C. The condenser water flow is not directly measurable, but is calculated from the pressure drop across the pump. This method is discussed in more detail in section 3.4. An ultrasound flowrate measurement taken on the these flows by the physical plant personnel has added validity to these methods.

Steam properties are needed to perform an energy balance on the turbine to determine the work output. The temperature and pressure of the steam entering and exiting the turbine were directly measured. However, the steam was generally in the two phase region. The determination of the inlet steam conditions was accomplished by utilizing a steam throttling calorimeter. The calorimeter, assumed to be a constant enthalpy device, allows the enthalpy to be determined, thus defining the quality and all other properties in the two phase region. The turbine exhaust, however, is under a vacuum at 4 in. Hg which exceeded the physical throttling limit of the calorimeter. The determination of the turbine work, discussed in section 3.2, has been accomplished using the manufacturers data and the measured inlet steam conditions. The inlet steam quality had been determined to be 95 - 98 % vapor, which coincides with the findings at the steam turbine lab at the Mechanical Engineering department, which utilizes the same steam source.

Other necessary operational information, which is directly recorded by the operators are the turbine speed (RPM), chiller vane position, tower fan speed settings and the ambient wet and dry bulb temperatures. Each of the hourly readings have been logged into a computerized spread sheet, which was installed at the plant, and is used to verify the equipment models.

A summary of the hourly data collected and used for analysing the plant performance are listed in Table 2.1 and shown in Figure 2.1. The data was collected for a period of approximately three months from May thru July of 1986. Within this span there is one short period of erroneous data due to insufficient conductivity within the thermocouple wells. This problem was quickly rectified and the bad data discarded from the study.

TABLE 2.1

TEMPERATURES		
T_1	Steam Inlet	T_{stms}
T_2	Steam Exhaust	T_{stmr}
T_3	Chilled Water Return	T_{chwr}
T_4	Chilled Water Supply (Set Point)	T_{chws}
T_5	Condenser Water Return	T_{cwr}
T_6	Condenser Water Supply	T_{cws}
T_7	Condenser Water Tower Return	T_{twr}
T_8	Hot Well (Steam Condensate)	T_{hw}
T_9	Ambient Wet Bulb	T_{wb}
T_{10}	Ambient Dry Bulb	T_{db}
PRESSURES		
P_1	Steam Supply	P_{stms}
P_2	Steam Exhaust	P_{stmr}
P_3	Tower Pump Suction	P_{tws}
P_4	Tower Pump Discharge	P_{twr}
P_5	Chilled Water Pump Suction	P_{chws}
P_6	Chilled Water Pump Discharge	P_{chwr}
FLOW RATES		
	Chilled Water Flow Rate	FLCHW
	Condenser Water Flow Rate	FLCW
	Steam Flow Rate	FLSTM
OTHER		
RPM	Turbine/Chiller Speed	
VANE	Compressor Inlet Guide Vane Position	
FANS	Tower Fan Speed Setting	

Table 2.1 Summary of experimental data points

3.0 COMPONENT DEVELOPMENT

The TRNSYS 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. In simulation information flow follows the fluid flow as shown in Figure 2.1 with the output of one component being the input of the next component in the fluid flow stream.

The validation process for the TRNSYS models was to compare collected performance data with the simulation results. First, the general component models were specialized to the test site by a set of empirically-derived parameters. The component models were then individually validated using collected data as inputs, and comparing the simulated outputs to the recorded values. Once all the components were tested, the plant simulation was run using the chilled water load and ambient wet bulb temperature 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.

In the following sections the component models used in the plant simulation are individually described and validate. In section 3.6 all of the components are interconnected to simulated the entire plant as one entity. This overall plant simulation is verified with the collected data using the recorded chilled water load, ambient wet bulb and control settings as inputs. In section 3.7 the chilled water load generator is developed to be used in full season simulations with the automatic controllers which are discussed in Chapter 4.

3.1 COOLING TOWERS

The cooling tower provides the means by which the heat absorbed by the refrigerant and that due to steam condensation is dissipated from the refrigeration and steam turbine cycles. The tower utilizes ambient air as a heat sink to cool the condenser water stream. A schematic diagram of an induced-draft, cross flow tower is shown in Figure 3.1. Warm water from the refrigerant condenser is introduced to the top of the tower via distribution piping and sprayed over layers of baffles, known as fill. The fill breaks the water flow into a thin film creating a large surface area which is exposed to the cooler air stream. The cooling of the water is due to the combined effect of heat and mass transfer. Evaporation of the condenser water occurs due to the temperature and moisture content differences between the air and the water streams. This evaporative process removes the heat of vaporization from the water stream causing the water to leave at a cooler temperature and slightly reduced mass flow rate. Make up water must be introduced to compensate for this evaporative loss. Conversely, the air stream leaves at both a higher temperature and water content than the entering ambient condition.

Cooling towers are divided into cells. Each set of cells are physically separated from the others, each with its own set of fans and water distribution off the common manifold header. Towers can be coupled together or individually dedicated to separate chillers. The air flow circulation is accomplished with draft fans in each cell. These fans can be fixed speed, variable speed or some combination of fixed speed settings.

The analytical technique used to model the heat and mass transfer process occurring in the tower is that devised by Whillier (1967). The Whillier model is less complex and does not require the repetitive integration as in the more common Merkel method, which is the basis for the technique described in ASHRAE (1983). Recent

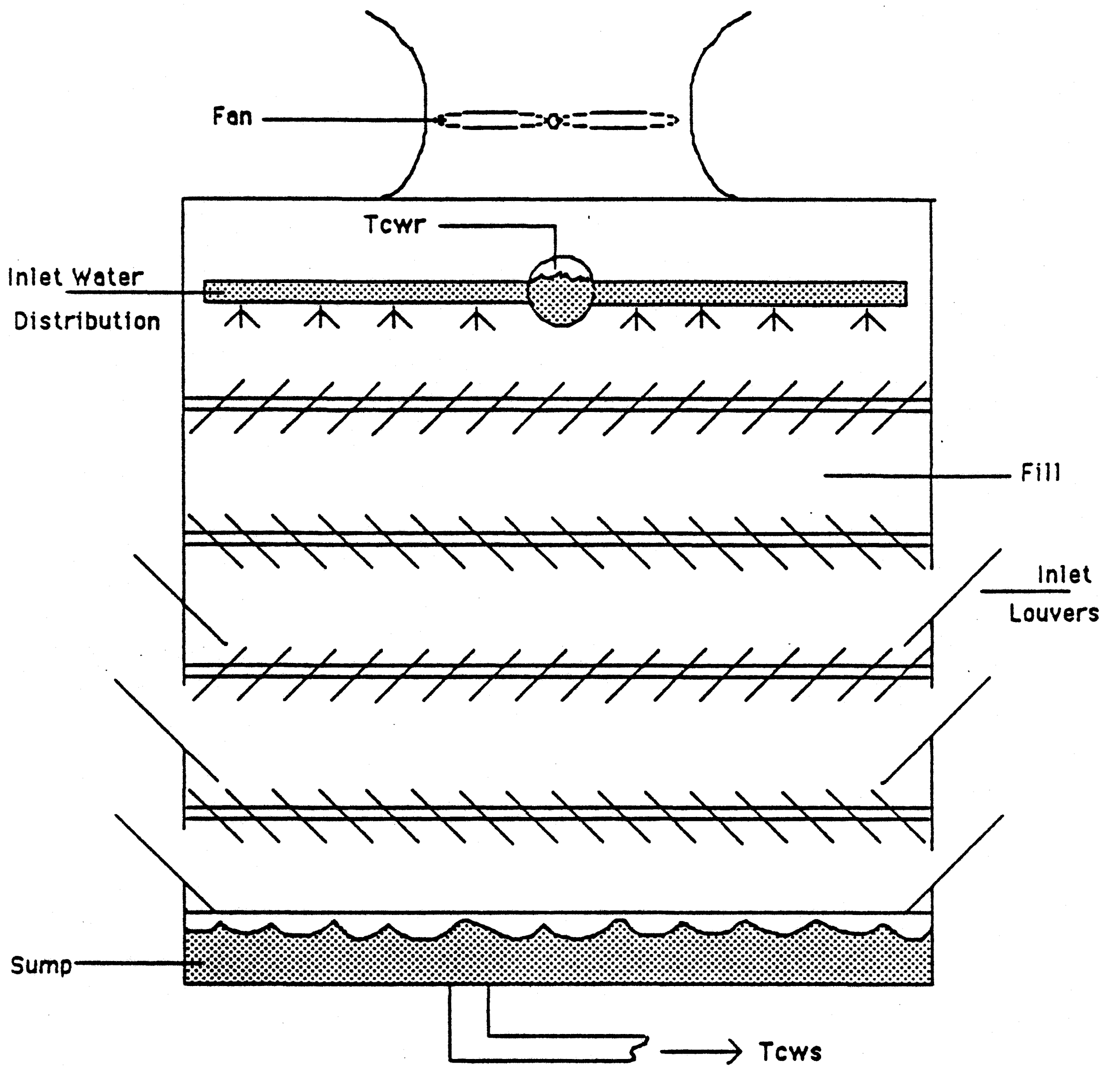


Figure 3.1 Schematic representation of induced draft, cross flow cooling tower

studies (Braun et al 1987) have indicated that there appears to be no advantage to the use of the Merkel model over the Whillier model. Both Lau (1983) and Hackner (1984) have utilized this model with very satisfactory results. This is substantiated when tested for the tower at hand.

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

Whillier defines a tower capacity factor, R, as the ratio of thermal capacities of the two fluid streams.

$$R = \frac{(Q_{a,max} \, , \, Q_{w,max})\ min}{(Q_{a,max} \, , \, Q_{w,max})\ max} \tag{3.1}$$

The maximum possible heat transfer for each stream are:

$$Q_{a,max} = M_a \, (h_{s,twr} - h_{s,i}) \tag{3.2}$$

$$Q_{w,max} = FLCW \; Cp_w \, (T_{twr} - T_{wb}) \tag{3.3}$$

where: M_a = Mass flow rate of air stream
FLCW = Mass flow rate of condenser water stream
Cp_w = Specific heat of condenser water
T_{twr} = Condenser water tower return temperature
T_{wb} = Ambient wet bulb temperature
h_s = Sigma energy term

Here $h_{s,twr}$ is the maximum possible sigma energy of the exiting air if it were to be at the temperature of the entering water, T_{twr}, and $h_{s,i}$ is the sigma energy of the inlet air. Sigma energy is a term first introduced by William Carrier in 1911. It is a unique grouping of terms in the energy balance which was found to be more convenient than mixture enthalpy in processes involving changes in moisture content.

$$h_s = h_m - \omega \, Cp \, T_m \tag{3.4}$$

where: h_m = enthalpy of moist air mixture

T_m = temperature of moist air mixture

The tower capacity ratio is analogous to C_{min}/C_{max} in conventional heat exchanger theory (Kays and London 1984). Similarly, the cooling tower effectiveness is defined, in the general sense, as the ratio of the actual to the maximum heat transfer between the tower streams.

$$E = \frac{Q_{twr}}{(Q_{a,max} \, , \, Q_{w,max})_{min}} \tag{3.5}$$

where :

$$Q_{twr} = FLCW \; Cp_w \; (T_{twr} - T_{cws})$$

Whillier correlates effectiveness to the tower capacity ratio with one empirical constant. A simpler method of determining exiting conditions was employed by Lau and Hackner using an alternative form of the effectiveness relationship as;

$$E = a \, R + b \tag{3.6}$$

The coefficients a and b are found empirically by linear regression applied to the actual tower performance data. Generally, b is approximately equal to 1 and $-1 < a < 0$.

Given the empirical coefficients a and b, the entering ambient air and condenser water conditions plus operational mode information (number of active fans and speed settings), from Equation (3.6) the effectiveness is determined and the exiting temperature is found from rearranging Equation (3.5) as;

$$T_{cws} = T_{cwr} - \frac{E\ (Q_{a,max}\ ,\ Q_{w,max})_{min}}{M_w\ Cp_w} \tag{3.7}$$

The effects of makeup water and the mass of the sump are neglected in this relationship.

In addition the tower model calculates the fan power necessary to run the fans at the desired settings. This is done using either the fan power laws as a function of flow rate (Tuve 1966) or measured values which can be input as parameters.

3.1.1 TOWER MODEL VERIFICATION

The Walnut Street plant uses three 2 cell towers. Two speed fans, with low speed being half the velocity of the high speed, are installed in the first four cells and one speed, high velocity, fans in the remaining two cells. The volumetric flow and power measurements made by the plant personnel in 1985 are used throughout this study. A copy of this document is included in Appendix C. The fan laws are used to determine flows and power at speeds other than those measured.

The coefficients a and b in Equation (3.6) were determined using linear regression applied to the recorded data of an approximately three week period in May.

These data include a wide range of wet bulb temperatures and tower operating conditions. The coefficients were found to be; a = -0.9617 and b = 1.153, with an RMS error of 0.0126.

The tower model was tested individually for accuracy in predicting the leaving water temperature, T_{cws}. Figure (3.2) compares the actual leaving water temperature to the model output for a wide range of wet bulb temperatures, time of day and condenser loads during the period from mid May to mid June. The majority of the points plotted are data other than those used to fit the effectiveness coefficients. The scatter in this figure can be associated both with limitations of the model and with the experimental error which enters into the measurements. The experimental errors result from estimates of flowrates and temperatures. The flow rates are the least accurately determined variable. The air flows are assumed to be constant for each fan setting as is the water flow. The latter is discussed in more detail in the pump component section, 3.4. It is also assumed that this water flow is evenly distributed among the cells. The error associated with the thermocouple readings is rated by the manufacturer to be within 1°F. The combined effect results in a root mean square error in Figure (3.2) of 1.53 °F.

3.2 STEAM TURBINE MODEL

A steam turbine converts the energy in high pressure steam to useful work by allowing the steam to expand through a set of rotary wheels which turn a shaft. Pressurized steam enters the turbine through the "steam ring" where it is throttled through nozzles and enters the first, high pressure stage, rotor compartment. The kinetic energy of the steam impinges upon the blades or "buckets" of the rotor wheel causing the rotor and shaft to turn. To more effectively utilize all the energy in the steam, multiple

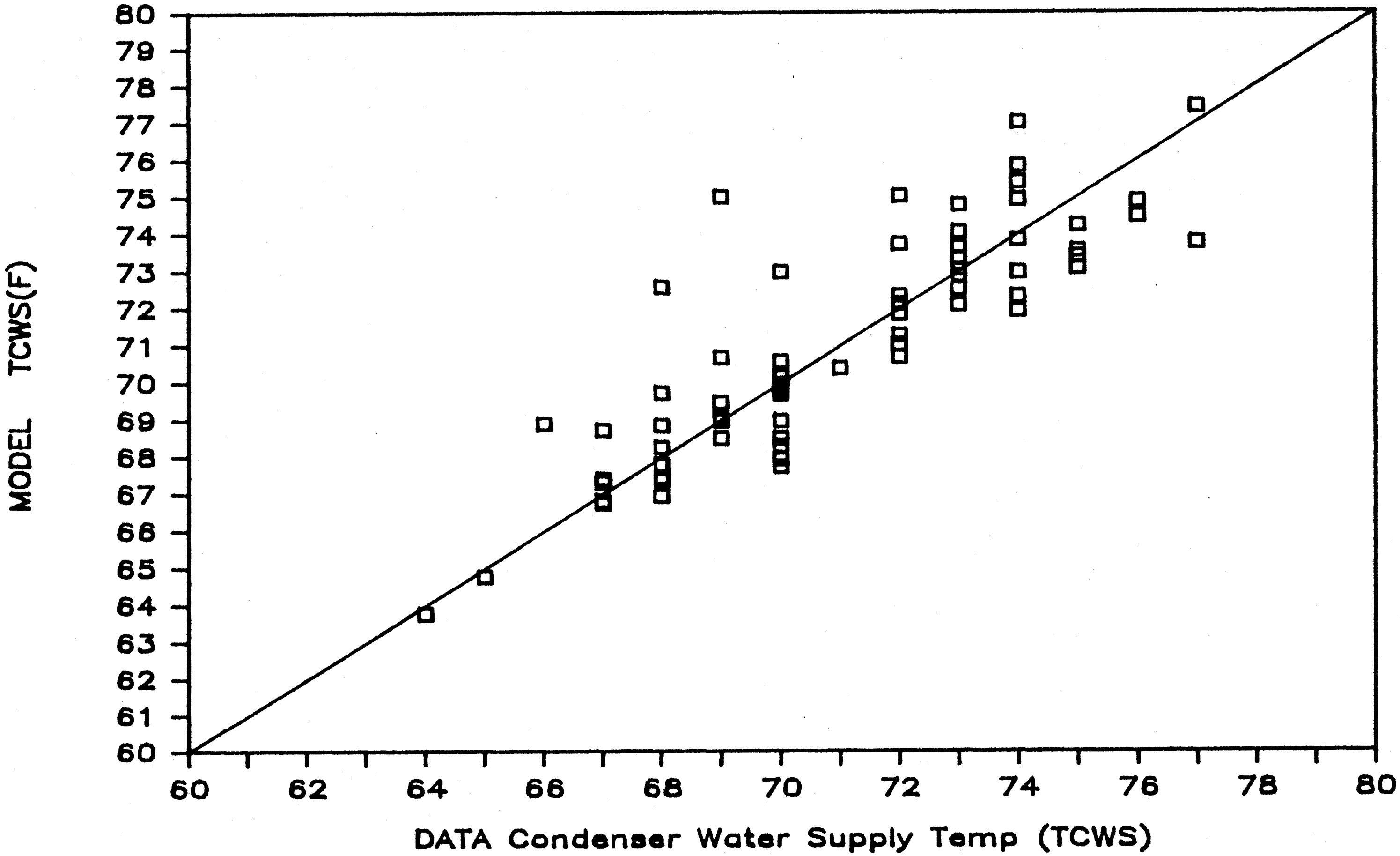


Figure 3.2 Comparison of predicted condenser water supply temperatures versus the measured data

stages of rotors and nozzles are employed. The energy is extracted from the successively decreasing pressure, but increasing volume of steam by the series of rotors until the steam is exhausted at sub-atmospheric pressure. It is then condensed and returned to the boiler. A schematic diagram of a multistage, mechanical drive, condensing, impulse turbine is shown in Figure (3.3).

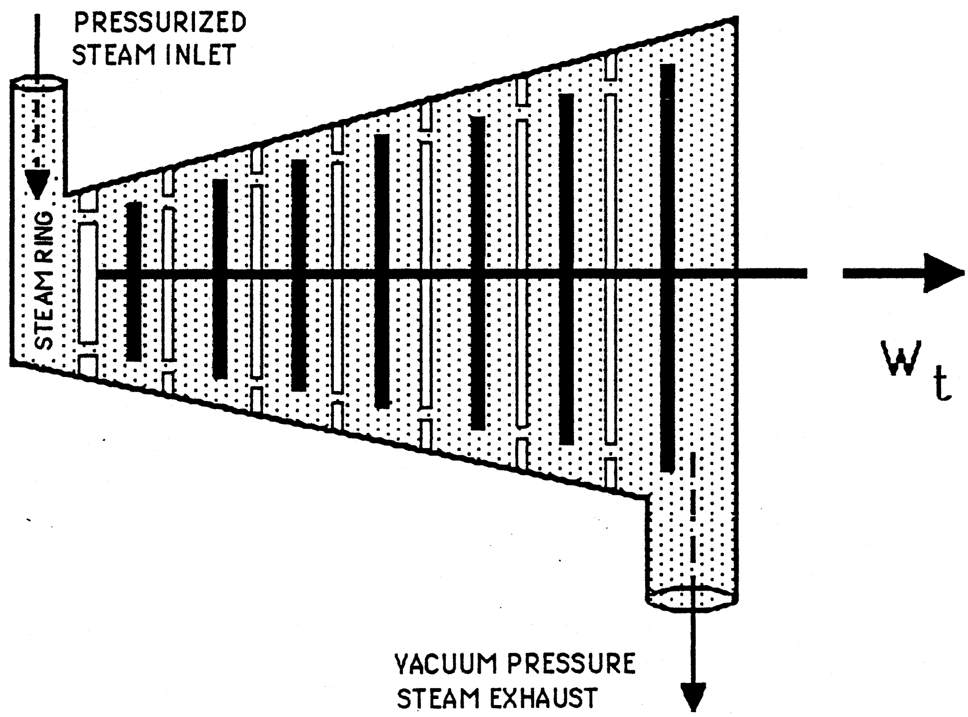


FIGURE 3.3 SCHEMATIC REPRESENTATION OF MULTISTAGE, CONDENSING, IMPULSE STEAM TURBINE

The object of the steam turbine model is to predict the quantity of steam consumed to produce a given amount of work. Due to the experimental difficulties associated with measuring two phase steam properties in the vacuum at the turbine exhaust the model has been based upon the turbine manufacturers performance curves rather than on the basic thermodynamic principles. The curves shown in Figure (3.4) have been generated, at this authors request, specifically for the turbine at hand operating under the measured inlet steam conditions and exhaust vacuum pressure. The discontinuances in these curves are at the points where additional steam inlet hand valves are opened to increase the supply of steam. For this case, the hand valves are not altered, therefore only the data to the left of the first discontinuance is considered. Similar curves may be attained for other turbine models and manufacturers.

The information presented in Figure(3.4) is fit to the following quadratic form:

$$FLSTM = A_1 + A_2RPM + A_3RPM^2 + A_4PWR + A_5(RPM\ PWR) \qquad (3.8)$$

where: FLSTM = Mass flowrate of steam (lbs/hr/1000)

RPM = Rotational speed of turbine (RPM/100)

PWR = Power output of turbine (KW/100)

A_1 - A_5 = Empirical coefficients

The empirical coefficients in Equation (3.8) are found by linear regression to the data of Figure (3.4).

The manufacturers information has been plotted in another useful form. Figure (3.5) shows the turbine efficiency as a function of RPM at constant steam flow rates.

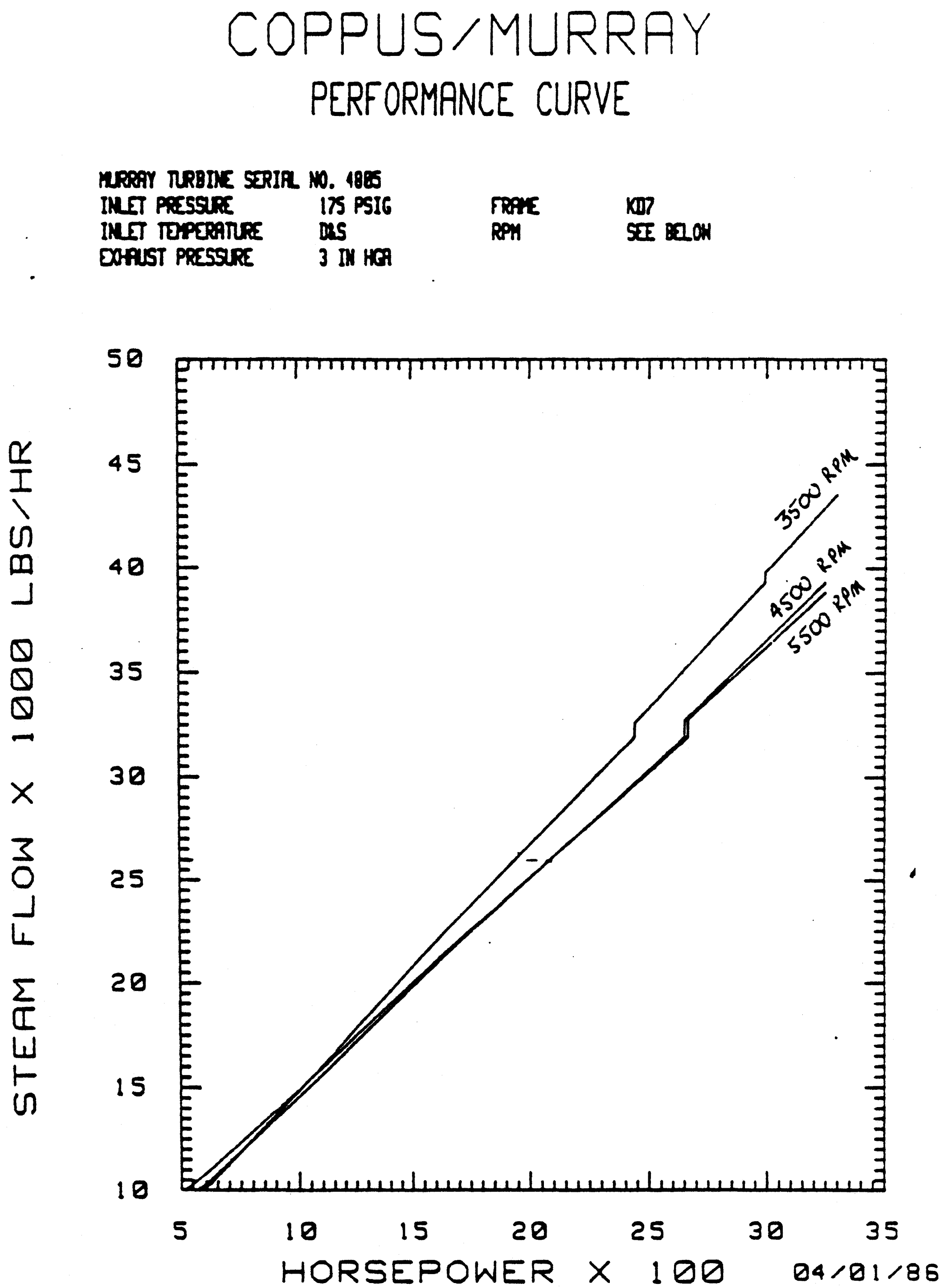


Figure 3.4 Turbine manufacturers performance curves

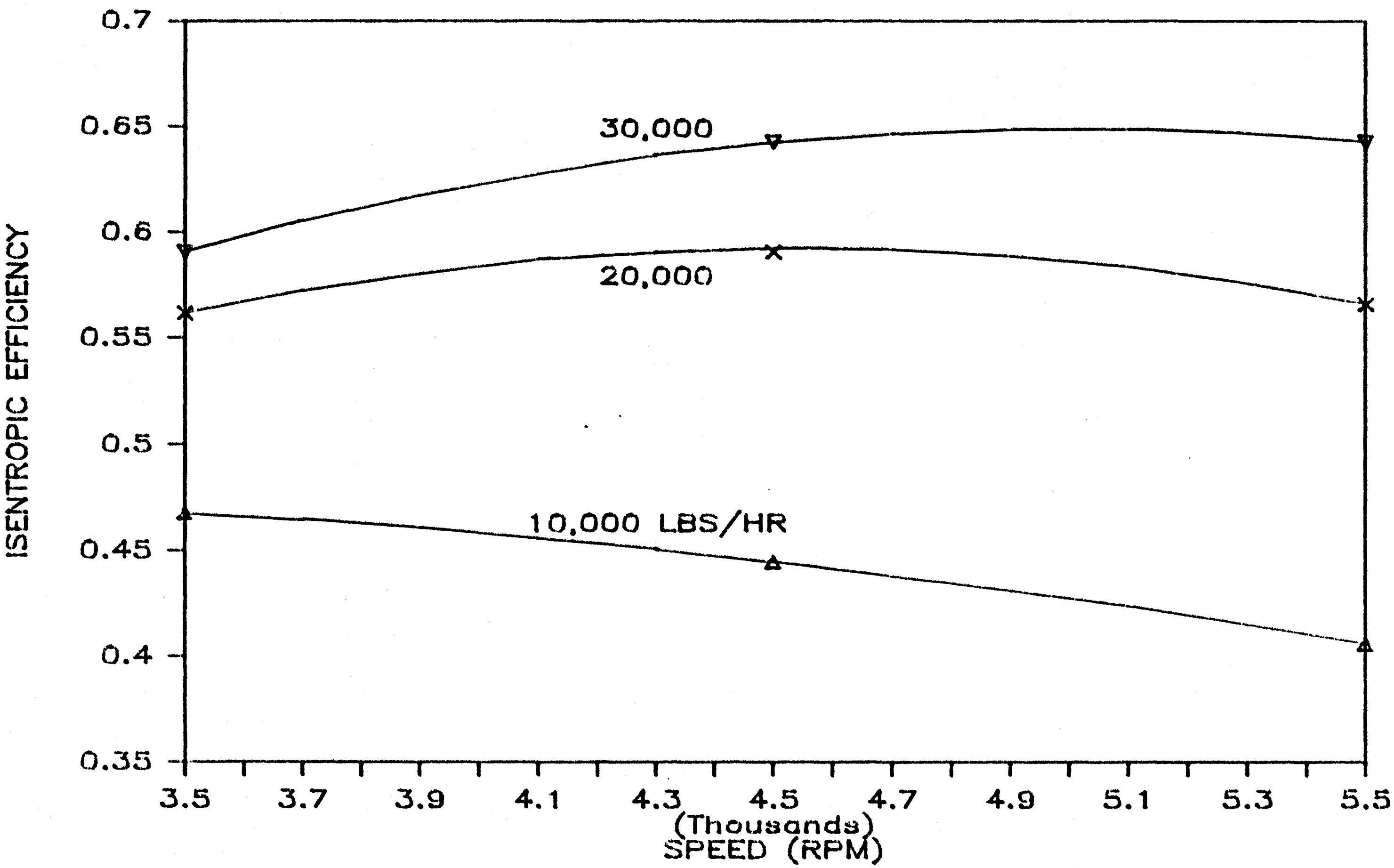


Figure 3.5 Steam turbine isentropic efficiency displayed as a function of RPM and steam flow rate

The efficiency is calculated as the ratio of actual work output to the ideal work output. Given the steam inlet conditions (T_{stms}, P_{stms}, H_{stms} and S_{stms}) from the calorimeter measurements, and knowing the exhaust pressure, P_{stmr}, the isentropic enthalpy at the exit, $H^{ID}{}_{stmr}$, is obtainable. For a given steam flow rate, the ideal work is found to be;

$$W^{ID} = FLSTM\,(H_{stms} - H^{ID}{}_{stmr}) \tag{3.9}$$

For the same steam flow, at the desired RPM, the actual work, W_{act}, is read from Figure (3.4). The ratio of these two yields the desired turbine efficiency;

$$\eta_t = \frac{W_{act}}{W^{ID}} \tag{3.10}$$

The efficiency curve is fit with the following quadratic form;

$$\eta_t = B_1 + B_2 RPM + B_3 RPM^2 + B_4 FLSTM + B_5 (RPM\ FLSTM) \tag{3.11}$$

Again, the empirical coefficients, B_1 - B_5 are determined by linear regression to the manufacturers data.

It is necessary to know the exiting steam properties to be used as an input to the surface condenser. Given a required work output from the turbine and knowing the inlet steam conditions, the exiting enthalpy can be determined from an energy balance on the turbine, assuming adiabatic operation;

$$H_{stmr} = H_{stms} - W_{act} \tag{3.12}$$

From the measurements, the steam exhaust pressure is known. With these two properties, H_{stmr} and P_{stmr}, the exiting steam state is fixed.

3.2.1 STEAM TURBINE MODEL VERIFICATION

The turbines located at the Walnut Street facility are 7-stage, 3000 horse power mechanical drive units coupled directly to the chiller compressors. The information presented in Figures (3.4) and (3.5) are specific to these machines. The curves plotted in these figures have been fit using linear regression on the data taken from the graphs. The coefficients that best describe the steam flow, Equation (3.8) and the efficiency, Equation (3.11), curves haven been determined and are listed below in Table 3.1.

The desired output of the turbine model is the steam consumption necessary to meet the chiller load. Since the actual work output of the turbine (input to the chiller) was not directly measurable, the steam flow curve fit, Equation (3.8), could not be verified independently. Therefore, the chiller model is interconnected directly to the turbine, as it is physically and the combined chiller-turbine model was verified. For a given measured chilled water load on the chiller, the compressor requires a specific power at a specific RPM to meet that load. This power and RPM are input to the turbine model, allowing for assumed windage and transmission losses, TLOSS, the steam consumption determined

TABLE 3.1

STEAM FLOW					
A_1	A_2	A_3	A_4	A_5	RMS
14.621	-0.5674	0.00714	1.9123	-0.0102	0.2003
EFFICIENCY					
B_1	B_2	B_3	B_4	B_5	RMS
0.0365	1.5179	-0.0225	1.5456	-0.0457	0.02

Table 3.1 Steam flow and efficiency equation coefficients for the Walnut Street turbine

directly from Equation (3.8). To verify the models, both the chiller and the turbine must be run together to eliminate the intermediate, unmeasured, power transfer. The presentation of this verification is reserved until the completion of the chiller model discussion in the next section.

3.3 CHILLER MODEL

A chiller is a refrigeration machine that produces chilled water for air conditioning needs. There are generally two modes of chiller operation, mechanical cooling and free cooling. The two modes require different modeling approaches; each is discussed separately.

3.3.1 MECHANICAL COOLING

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 is available to adjust refrigerant flow to maintain the desired leaving chilled water temperature (set point). The thermal load and work input absorbed by the refrigerant fluid is rejected at the condenser to the cooling towers. A schematic representation of the refrigerant cycle is shown in Figure (3.6).

A number of accepted methods of modeling the performance of chillers have been investigated. A mechanistic model of a centrifugal chiller has been developed by Braun et. al. (1987) which has been demonstrated to work well for both variable speed and variable vane control. This model is particularly useful when little or no performance data are available. It requires extensive knowledge of the machinery characteristics (i.e. compressor blade angles, staging, condenser/evaporator tube sizes and number of passes) and working fluids. With all the proper parameters and driving conditions this model is capable of determining both required compressor speed and power consumption as well as chiller capacity and the compressor surge limitations.

Another common method is to fit empirical relationships to manufacturers data. Stoecker (1971) has developed forms for these relationships which have been used successfully in previous studies by Lau (1983) and Hackner (1984). The initial limitation to this approach is that the model can only be trusted within the range of conditions to which it was fit. Also, it is not always possible to obtain comprehensive data from the manufacturer, as encountered by Lau.

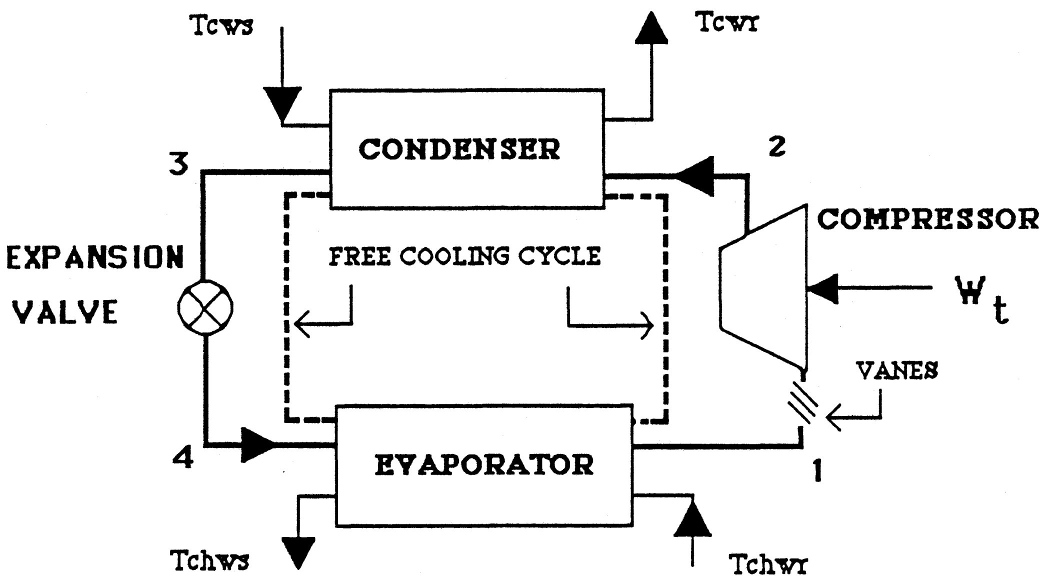


Figure 3.6 Schematic representation of vapor-compression refrigeration cycle and Free Cooling cycle

Braun (1987) has developed a simpler, empirical model particularly suited for cases in which experimental performance data are available over a wide range of conditions. This empirical model requires little computation resulting in a more suitable algorithm for seasonal performance simulations. To predict the power with Stoecker's model requires iteration between two sets of correlations to the manufacturers data each of which are defined by 9 empirically determined coefficients. Braun's model, however, is one closed form correlation requiring the determination of only 5 coefficients. Additionally, with the same set of variables, Braun's model is capable of predicting the chiller RPM as well as the power. Both models are empirical, but due to its' simplicity and the added RPM feature Braun's model has been used in this study. It is described here and its predictions compared with collected data.

Braun demonstrates that chiller power is primarily a function of two variables, the chilled water load and the temperature difference between the leaving chilled water, T_{chws}, and the leaving condenser water, T_{cwr}. This functional relationship is described by the following quadratic equation;

$$PWR = \alpha_0 + \alpha_1 X + \alpha_2 X^2 + \alpha_3 Y + \alpha_4 (XY) \tag{3.13}$$

where: $PWR = PWR_{ch} / PWR_{des}$; Chiller Power/Design Power consumption

$X = Q_{chw} / Q_{des}$; Chilled Water Load / Design Load

$Y = \Delta T / \Delta T_{des}$; $(T_{cwr} - T_{chws})$ / Design $(T_{cwr} - T_{chws})$

$\alpha_0 - \alpha_4$ = Empirical constants

To normalize the equation, the design values are chosen at the design capacity. The empirical coefficients are then determined by a linear least squares fit to the collected data.

The variable Y defines the pressure difference between which the saturated refrigerant fluid is allowed to vaporize and condense. The X variable, the thermal load to be met, essentially defines the resulting refrigerant flow rate which will be required to met that load within the limitations defined by Y.

Recall from the discussion in section 2.0, that this power, being delivered to the chiller compressor, was not a measured value. Therefore, to fit the coefficients of Equation (3.13) it was necessary to connect the compressor and turbine models.

Assuming windage and transmission losses, the power input to the chiller is equated to the output of the turbine. From section 3.2.1 Equation (3.8) is rearranged to calculate the turbine power output as a function of the measured steam consumption and RPM. This calculated power output, accounting for the transmission losses (multiplied by 1-TLOSS) is considered the power input to the chiller;

$$PWR_{ch} = \frac{FLSTM - \alpha_1 - \alpha_2\,RPM - \alpha_3 RPM^2}{\alpha_4 + \alpha_5\,RPM} \times (1\text{-}TLOSS) \tag{3.14}$$

This is the power used to fit the coefficients in Equation (313).

Braun has demonstrated that Equation (3.13) predicts both variable vane, constant speed and variable speed, constant vane operation quite well. However the data used to fit the coefficients to this equation must be for one or the other operational schemes. Equation (3.13) can be used only for the case in which either the vanes or the speed are held constant and the other is allowed to vary to keep up with the changing load. For situations in which both settings are varied, which includes the case at hand, additional correlations must be made, which are described below.

To model adjustable vane, variable speed operation or adjustable speed, variable vane operation the experimental data must be separated into bins of constant vane settings or speed settings, the setting which is manually adjusted. The coefficients of the power equation, are then fit to each set of bin data creating a individual power relationship for each bin.

For the case of adjustable vane, variable speed control, as an example, the data must be separated into bins of constant vane positions. A set of coefficients are then found, using linear regression, to describe the power relationship for each of the constant vane settings. For the same load and leaving water temperature difference each vane position will have a different power input.

The same variables used to predict the chiller power will also predict the chiller RPM.

$$RPM = \beta_0 + \beta_1 X + \beta_2 X^2 + \beta_3 Y + \beta_4 (XY) \quad (3.15)$$

Similar to the power equation, this RPM correlation is for fixed vane or fixed speed operation. Again, a separate set of coefficients must be empirically determined for each set of bin data.

In order to utilize this empirical model the required variables are the load and the leaving water temperature difference along with the associated design values used to normalize the equations, are needed. The load is determined directly from an energy balance on the evaporator using the experimentally measured flowrate and temperatures.

$$Q_{chw} = FLCHW \; Cp_{chw} \; (T_{chwr} - T_{chws}) \quad (3.16)$$

with;

$$X = Q_{chw} / Q_{des} \quad (3.17)$$

All these variables are known inputs to the chiller, whereas the leaving water temperature difference, Y, is not directly known. The leaving chilled water temperature is the desired set point temperature (T_{chws}), but the leaving condenser water temperature

(T_{cws}) is dependent upon the load, entering conditions and the power input. The temperature difference may be determined from the overall energy balance on the chiller;

$$Q_{chw} + P_{des}(\alpha_0 + \alpha_1 X + \alpha_2 X^2 + \alpha_3 Y + \alpha_5 XY) \\ = FLCW\ Cp_{cw}[\ Y\ (T_{cwr} - T_{chws})_{des} - (T_{cws} - T_{chws})] \tag{3.18}$$

Equation (3.18) may be rearranged to solve for Y explicitly, yielding;

$$Y = \frac{[(FLCW(T_{cws} - T_{chws}) + XQ_{des} - P_{des}(\alpha_0+\alpha_1 X+\alpha_2 X^2)]}{[FLCW\ Y_{des}\ -\ P_{des}\,(\alpha_3\ +\ \alpha_1 X)]} \tag{3.19}$$

To estimate the chiller power and RPM requirements with the above relationships the chiller must first be defined by the parameters that have been described here. These parameters include α_0 - α_4 the power equation coefficients, β_0 - β_4 the RPM coefficients and the design values. With this information and the inputs of the chilled water load and condenser water supply flow rate and temperature, Equations (3.17) and (3.19) directly determine X and Y, respectively. The power and RPM are then calculated from Equations (3.13) and (3.15), respectively.

The vane adjustments are primarily used to keep the turbine speed within its allowable operating range. The turbine possesses both a high and a low RPM limit recommended by the manufacturer for safe efficient operation. 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 within the turbine's upper RPM limit the vanes must be opened wider, releasing the flow restriction, thus slowing down the compressor.

Adjustable vane control is incorporated into the model by a programmed step logic. If for the given vane position the RPM is below the predetermined lower limit the vane setting is lowered to the next setting and the RPM is recalculated, with the appropriate coefficients, for this new vane position. If the RPM is still to low the vane setting is lowered again. This is continued until the RPM limitations are satisfied.

3.3.2 FREE COOLING

In the free cooling mode the compressor and the expansion valve are physically by-passed, 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. Thus, the refrigerant is cooled and condensed, returning to the evaporator by gravity to complete the cycle. The free cooling refrigerant flow is depicted by the dotted line in Figure (3.6).

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.

There is, essentially, a direct heat exchange occurring between the chilled water and the condenser (tower) water. 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 process is much like that of a counter flow heat exchanger.

The free cooling mode of the chiller model is programmed as a heat exchanger. The temperature difference between the condenser water supply and the chilled water set point is generally a predetermined constant, ΔT_{FC}. The necessary tower water supply temperature is then;

$$T_{cws} = T_{chws} - \Delta T_{FC} \tag{3.20}$$

Knowing the chilled water load to be met, an energy balance on the heat exchanger defines the condenser water return temperature.

$$Q_{chw} = Q_{twr}$$

$$Q_{twr} = FLCW\ Cp_{cw}\ (T_{cwr} - T_{cws}) \tag{3.21}$$

$$T_{cwr} = \frac{Q_{chw}}{FLCW\ Cp_{cw}} + T_{cws} \tag{3.22}$$

Because there is no steam being consumed, the condenser water is returned directly to the tower bypassing the surface condenser. With the tower inlet temperature for the next time step now known, the tower model and the corresponding climatic data are utilized to determine the required number of fans to meet the load and temperature delivery criteria, if possible.

3.3.3 CHILLER MODEL VERIFICATION

The chillers at the Walnut Street facility are each 3500 ton, single compression units, one of which has the capacity to do free cooling. The method of load capacity control is adjustable vanes, variable speed. 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.

A number of tests were run on the individual chiller model to test its' ability to predict both power and RPM for a range of conditions and operating configurations. To alleviate the intermediate, unmeasured power transfer, the turbine and the chiller models are incorporated together. The combined models are tested for their accuracy in predicting the steam consumption for the full range of conditions.

The initial set of data used to fit the model correlations was approximately a two week period in May/June of 1986 of hourly readings. The raw data were separated into bins of constant vane positions. It was noted that there existed only a few commonly used settings. For each of these vane positions, a linear least squares method was used to fit both the power and the RPM correlations, Equations (3.13) and (3.15). It was found that small changes in vane position did not significantly effect these variables. Therefore, only the three major vane positions were used in the model, the 100, 50 and 20 percent open positions. These empirically determined coefficients are listed in Table 3.2 along with their associated RMS error.

These coefficients and the associated design values were used as the defining parameters within the interactive model. From the collected data the load, condenser water supply flow rate and temperature have been used as inputs to the chiller model to test the power and RPM predictions. Utilizing the turbine manufacturers performance curves (Fig. (3.4)) the recorded steam and RPM were used to estimate the actual turbine power output. Including an assumed 5% transmission loss, this actual power is plotted against that predicted by the combined chiller-turbine models in Figure (3.7) for the 100% vane setting only.

TABLE 3.2

POWER						
VANE	$\alpha 0$	$\alpha 1$	$\alpha 2$	$\alpha 3$	$\alpha 4$	RMS
100	-0.1708	0.2838	-0.1065	0.00995	0.9891	0.0321
50	0.2076	-1.4704	1.7646	0.39003	0.2023	0.0386
20	0.2264	-0.9218	1.0120	-0.1174	1.0180	0.0326
RPM						
VANE	$\beta 0$	$\beta 1$	$\beta 2$	$\beta 3$	$\beta 4$	RMS
100	0.2691	0.6319	-0.1555	0.5720	-0.3123	0.0076
50	0.2815	0.1918	0.3241	0.8643	-0.6454	0.0106
20	0.6057	0.4718	-1.1159	0.0279	0.8538	0.0295

Table 3.2 Power and RPM equation coefficients for the Walnut Street Chiller

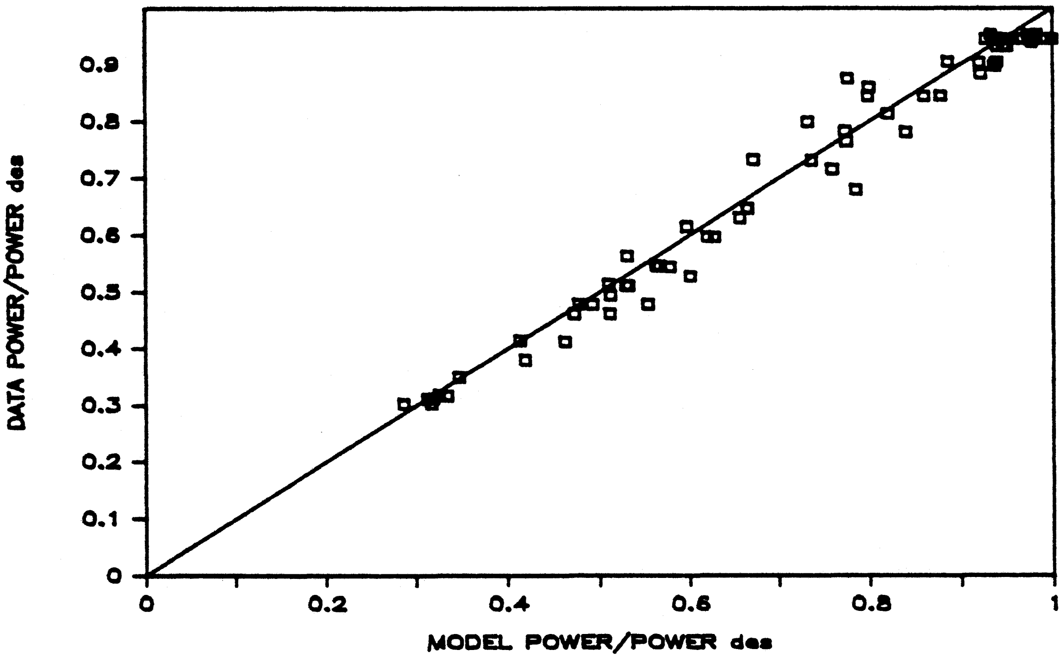


Figure 3.7 Predicted chiller power consumption for 100% vane position using measured condenser water supply temperature and chilled water load as inputs versus calculated power. Same data points used to fit empirical model

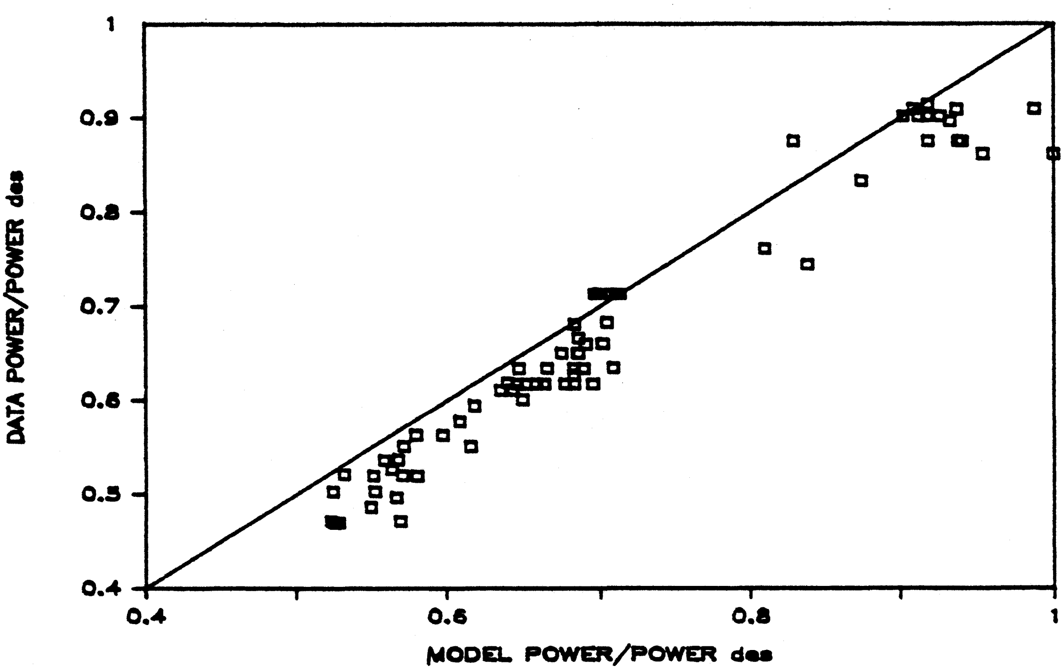


Figure 3.8 Predicted chiller power consumption for 100% vane position using measured condenser water supply temperature and chilled water load as inputs versus calculated power. Other data points than those used to fit model

A second, similar sized set of data, taken in July 1986, were used to check the versatility of this model. With the set of parameters determined from the May/June data and this new set of inputs Figure (3.8) presents the comparison of predicted and recorded power for the 100% vane position.

Similarly, the RPM output was tested and plotted for both sets of data. Figures (3.9) and (3.10) show these comparisons for the original and the other set of data, respectively

The effects of the vane position on RPM are important to the performance of the turbine. It is therefore necessary to verify the vane control logic programmed into the model.

If the vanes were to be left wide open at 100% the RPM would rise and fall with the chilled water load due to the relationship between the refrigerant flow rate and evaporator heat transfer. This is visually apparent in Figure (3.11) where the chiller is simulated for a three day period with the vane wide open. The chiller/turbine RPM varies with the load often falling below the manufacturers lower limit of 3500 RPM.

The iterative vane control described in the previous section is utilized here. Beginning with the 100% vane setting if, for the given load, the calculated RPM is below the lower limit the vanes are turned down to the 50 percent value and the RPM recalculated using the respective coefficients. If the RPM is still too low the vanes are set to 20 percent and left there.

By incorporating vane control the RPM can be regulated to within close approximation of the recorded values. The results of this test for the same three day period are presented in Figure (3.12).

The chiller and the steam turbine models have been merged together incorporating the intrinsic chiller vane control and speed control plus the turbine and

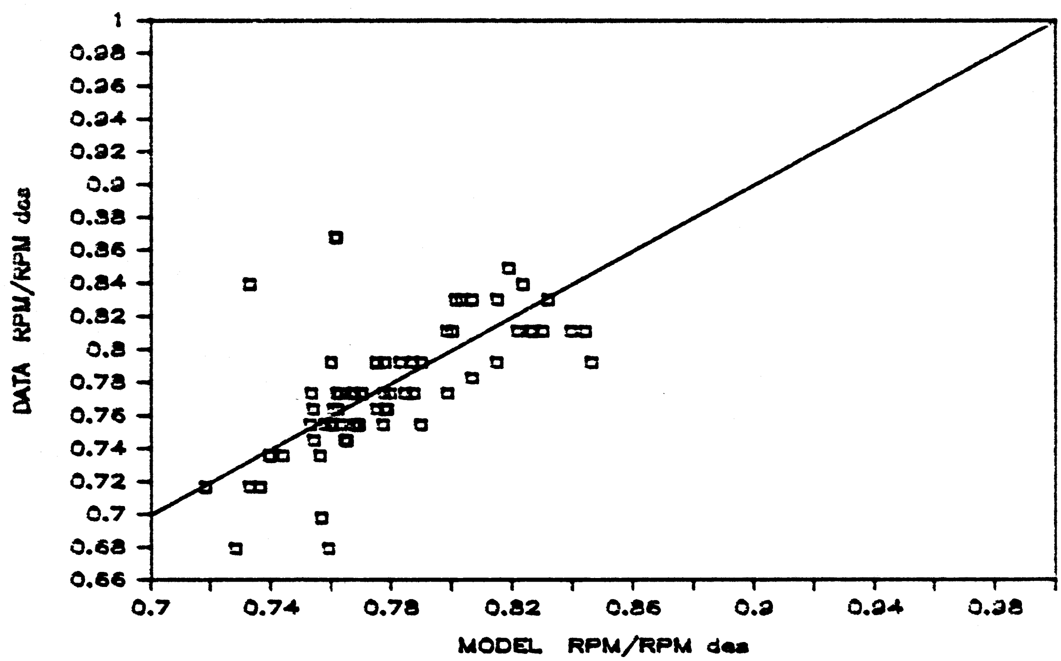


Figure 3.9 Predicted chiller RPM for 100% vane position using measured condenser water supply temperature and chilled water load as inputs versus measured RPM. Same data points used to fit empirical model

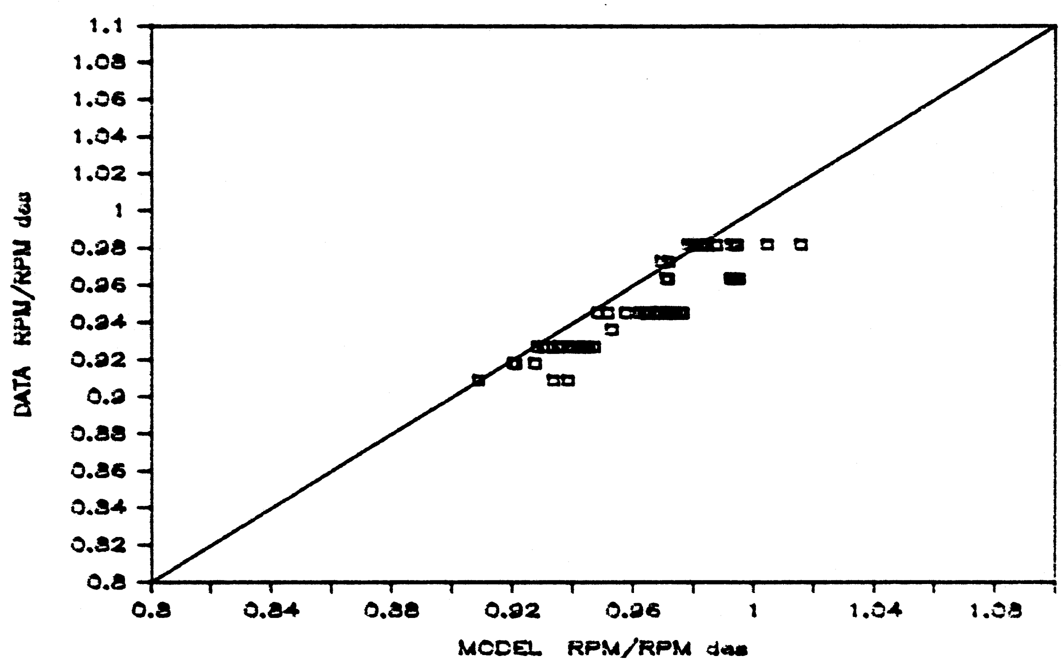


Figure 3.10 Predicted chiller RPM for 100% vane position using measured condenser water supply temperature and chilled water load as inputs versus measured RPM. Other data points than those used to fit the model

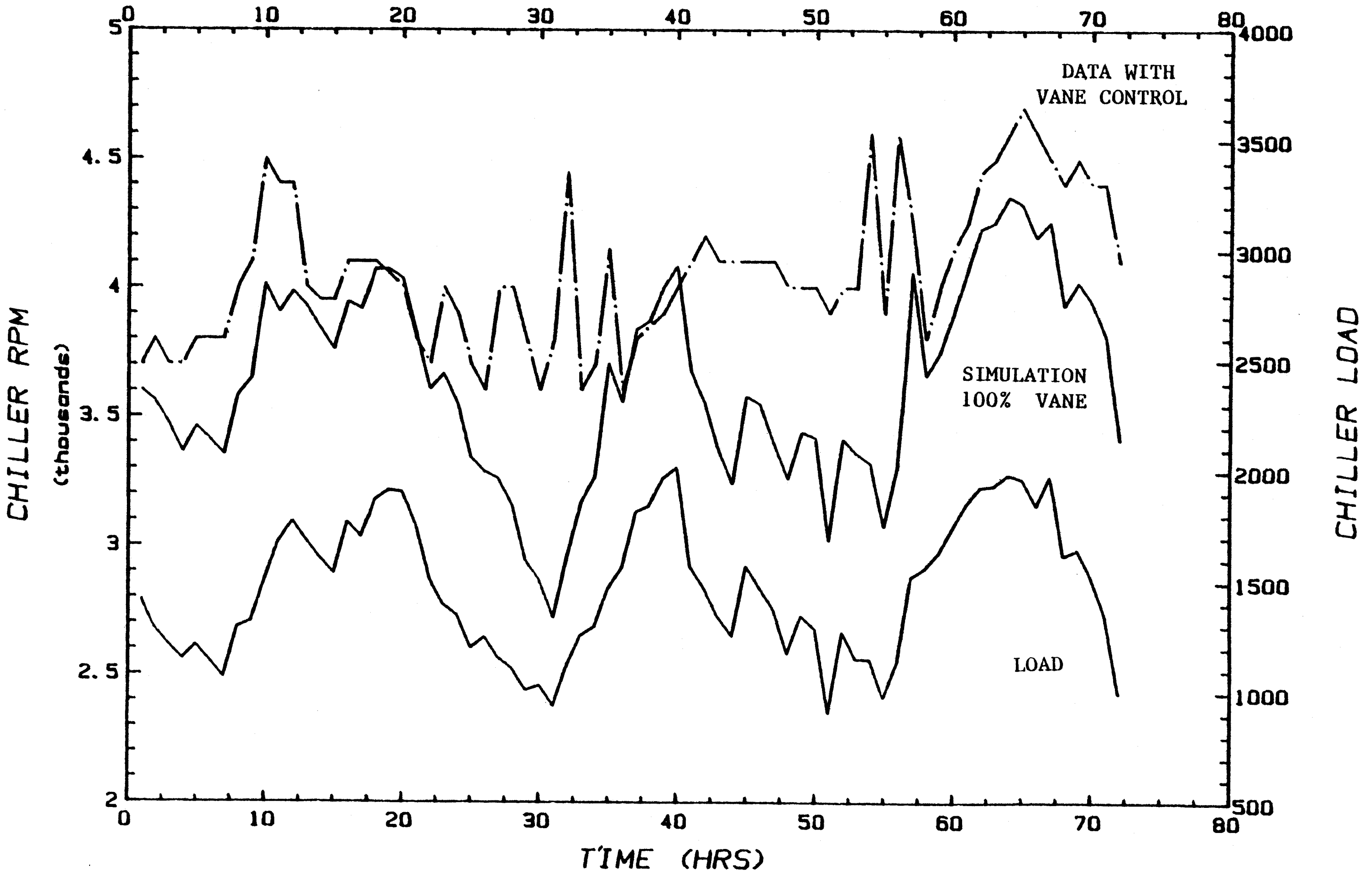


Figure 3.11 Simulated chiller RPM without vane control versus measured data with vane control indicating the relationship of RPM and chilled water load

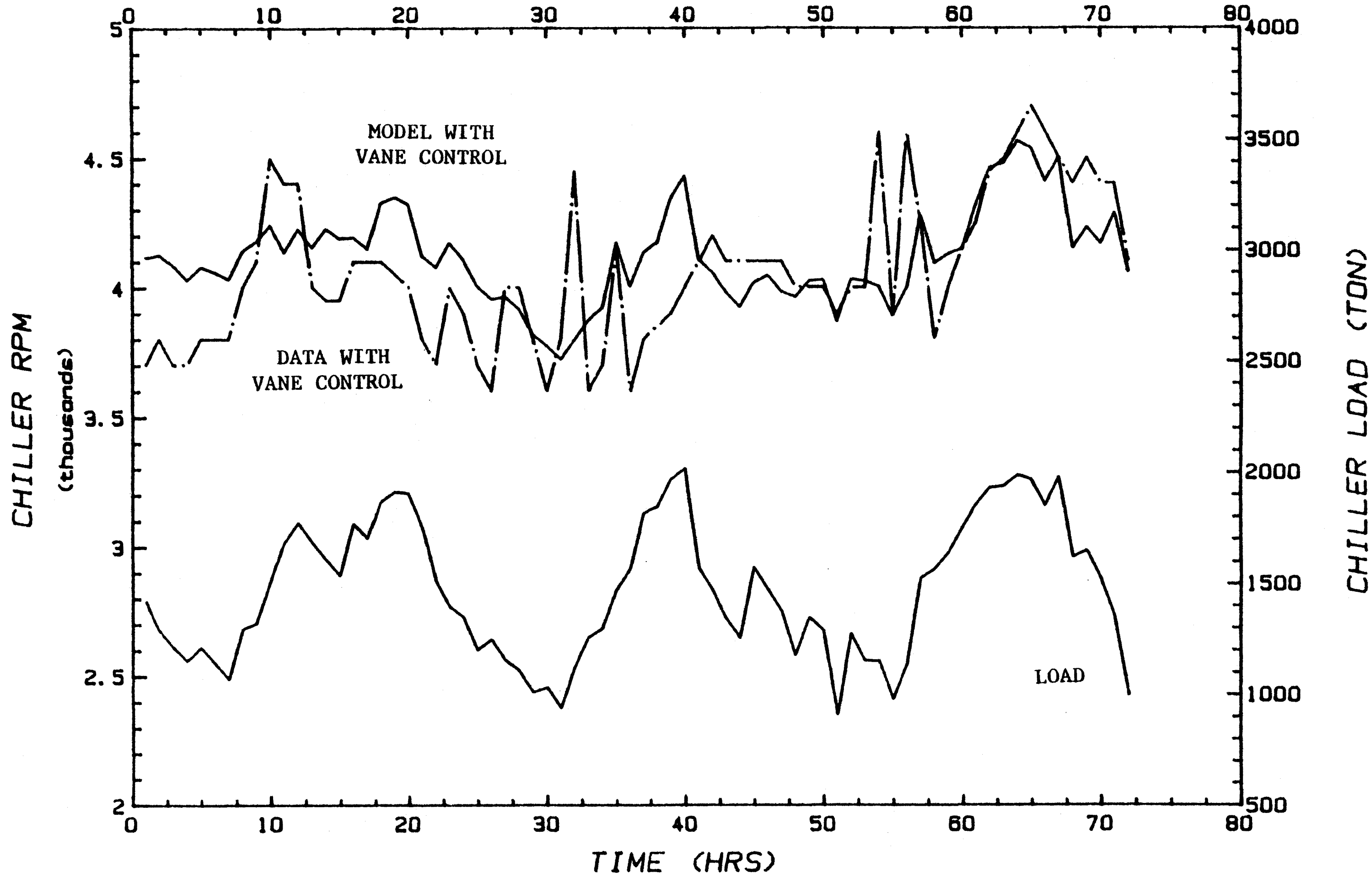


Figure 3.12 Predicted chiller RPM with vane control versus measured data plotted with chilled water load

transmission efficiencies. From the recorded data, 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 RPM requirement on the turbine. The turbine model thus provides an estimate of the necessary steam consumption to meet these requirements. Plotted in Figure (3.13) are the predicted and recorded steam consumption values for both the data used to fit the chiller model correlations and an equal number of other data. The predictions agree with the recorded values within an RMS error of 1113 lbs/hr, approximately 5%. It is noted that the the steam measurement readings are accurate only to within 500 lbs/hr due to the scale of the circular chart on which the recordings are plotted (see the sample chart in Appendix C). Also the condenser water flowrate is assumed to be constant in this test though it has been found to vary ±200 gpm.

3.4 PUMP MODEL

Hydronic pumps are a standard TRNSYS component, which outputs fluid temperature, mass flow rate and pumping power. The model assumes the fluid temperature and flow rate to remain constant in and out of the pump. The pumping power, which is of greatest importance to this study, is calculated as a user defined function of flow rate.

There are two sets of pumps that are included in this investigation, the chilled water pumps and the condenser water pumps. They are both capable of being electrically driven or, in case of a power failure, steam driven. Emergency operation is beyond the scope of this study and therefore only the electrical operation is considered. All the pumps are fixed speed, the flow rate is controlled by in-line restriction valves.

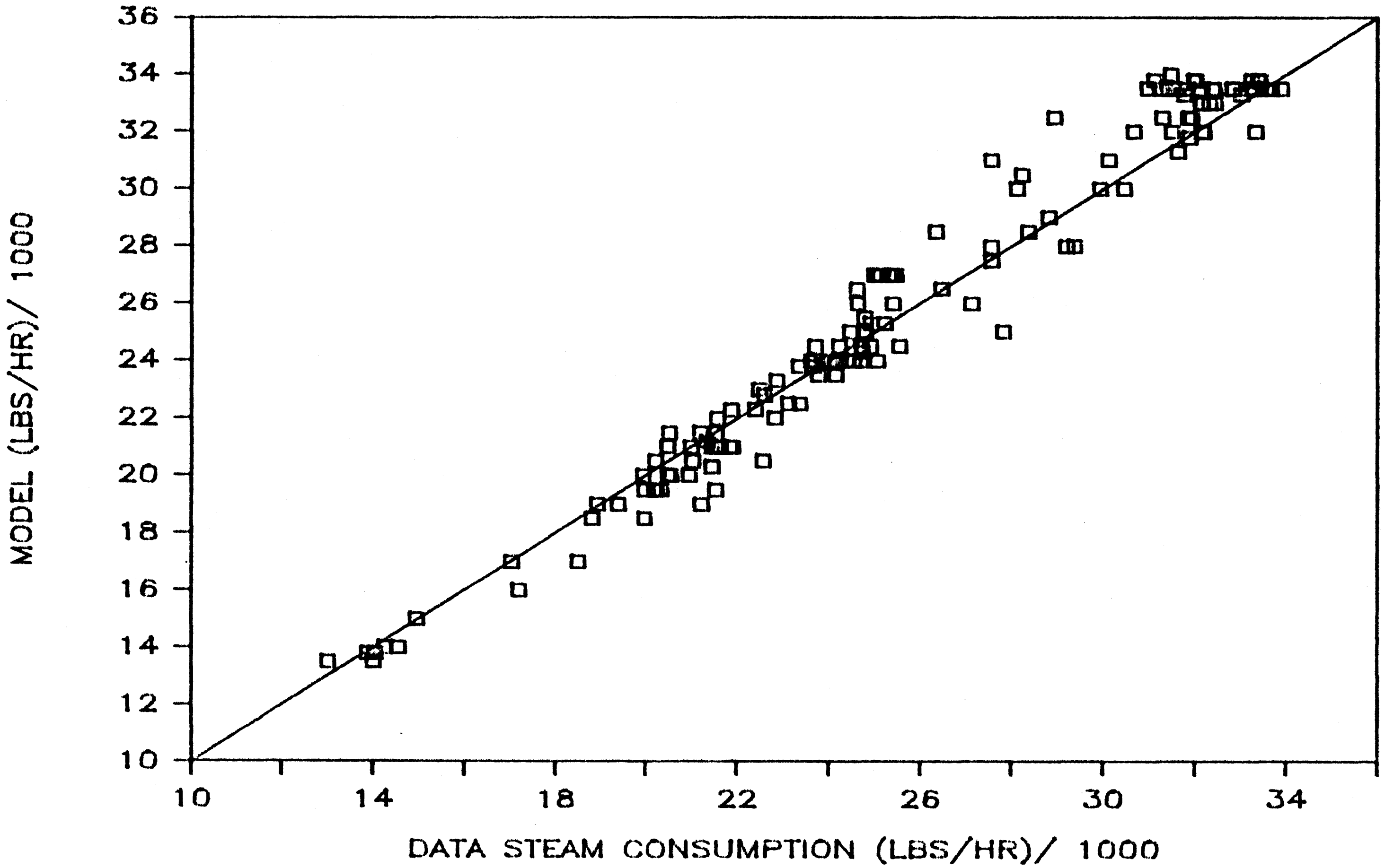


Figure 3.13 Predicted steam consumption for the combined chiller and turbine models using measured condenser water supply temperature and chilled water load as inputs versus the measured stream flow

The flow rate of the chilled water pump is measured and recorded by an in-line orifice device. The condenser water flow is not directly measured. Instead the flow has been calculated using the pump manufacturer's performance curves. By measuring the pressure increase across the pump and utilizing the Bernoulli equation, assuming constant velocity and elevation, the head produced by the pump is determined by;

$$h_p = \frac{P_d - P_s}{\gamma} \tag{3.23}$$

where: P_d = pump discharge pressure

P_s = pump suction pressure

γ = specific weight

Knowing the head and the pump size the flow is read from the manufacturers curve shown in Figure (3.14). The pressure increase measurements did not vary more than approximately 3 psi, which is within the accuracy with which the readings were taken. Therefore the flow has been considered to be a constant and calculated to be 11180 gpm. Ultra sound flow measurements were taken during the test period which reported an average condenser water flow of 11544 gpm, the three readings taken ranged from 11400 to 11620 gpm. A similar measurement was made on the chilled water flow reporting a flow within 48 gpm of the recorded value. Reading of the recorded values from the circular charts is limited to within 60 gpm.

For the fixed flow rate calculated from the pressure increase the manufacturers curve closely estimates the power previously measured by the plant personnel (the plant personnel's data is enclosed in Appendix C). The manufacturers curves are therefore used to correlate the power consumption for flows other than those that measured.

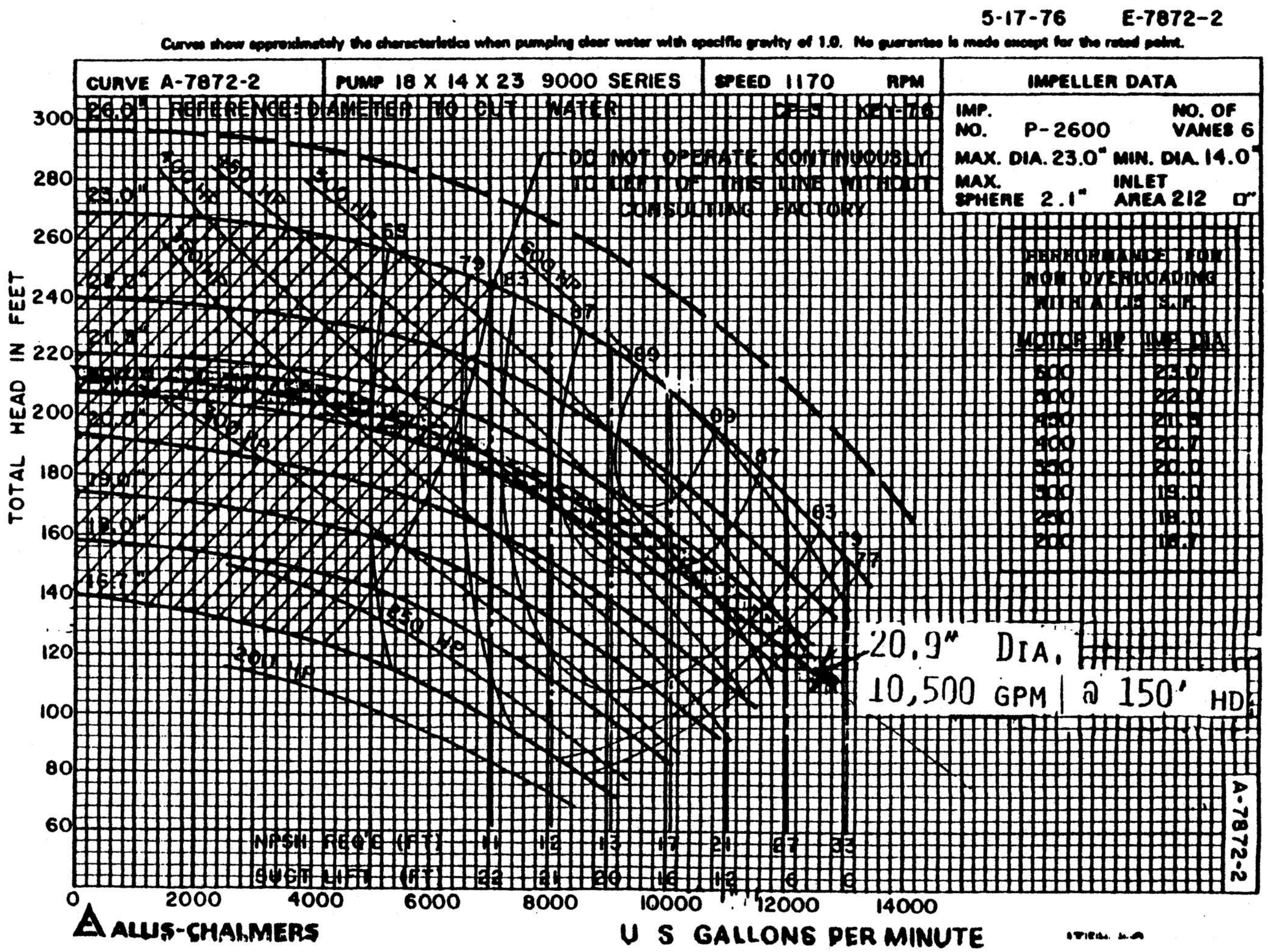


Figure 3.14 Condenser water pump manufacturers performance curves

3.5 SURFACE CONDENSER MODEL

The surface condenser is a shell-in-tube, cross flow heat exchanger which, as the name implies, condenses steam to be pumped back to the boiler. Heat exchangers are a standard TRNSYS component which uses conventional heat exchanger techniques (Kays and London 1984) to predict the leaving fluid steam temperatures.

The Mode 3 model, which is used here, requires the heat exchanger to be defined by its overall heat transfer coefficient, area product, (UA) and the fluid streams specific heats. The specific heat values are obtainable from thermodynamic property tables. The UA is found from an energy balance utilizing the log mean temperature difference, (ΔT_{lm}).

$$Q_{sc} = FLCW\ Cp_{cw}\ (T_{twr} - T_{cwr}) = UA\ \Delta T_{lm} \tag{3.24}$$

where: Q_{sc} = Surface condenser heat transfer rate

T_{twr} = Condenser water tower return, leaving cold fluid temperature

T_{cwr} = Condenser water return, entering cold fluid temperature

Rearranging Equation (3.24), UA can be solved for explicitly using the collected condenser water temperature data. The UA value for the Walnut Street condenser was calculated as the average of 10 readings taken over a three day period in May 1986, the values ranged from 380,000 to 410,000 the average being 395,028 BTU/hr F.

Figure (3.15) shows the results of the comparison of the leaving cold fluid temperatures, returning to the tower (T_{twr}), as predicted by the TRNSYS model, with a constant condenser water flowrate, to that of the measured data. The average difference for the test period is 1.65 °F with an RMS error for the hourly predictions of 2.16 °F.

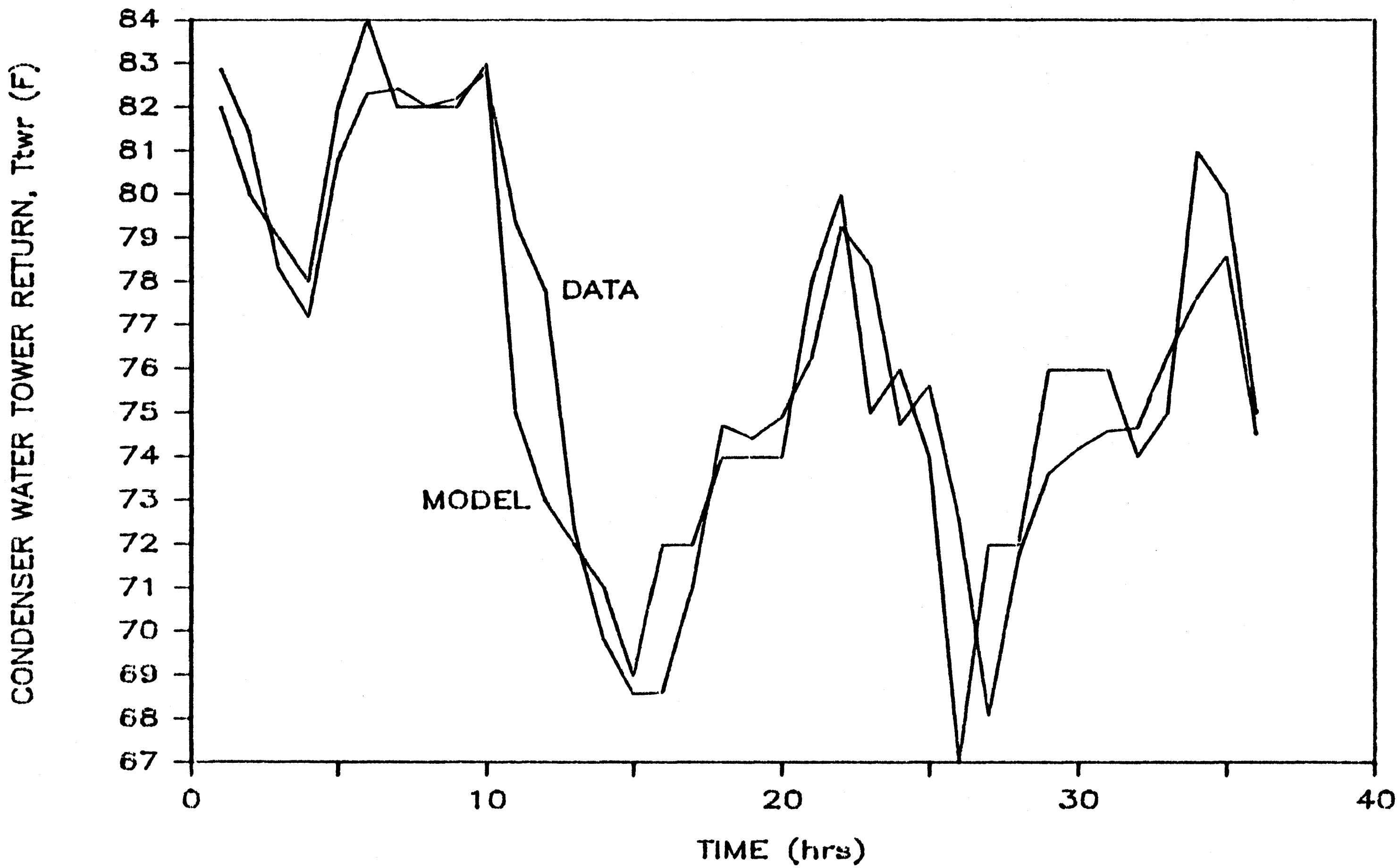


Figure 3.15 Predicted condenser water tower return temperature exiting surface condenser versus measured data

3.6 OVERALL PLANT SIMULATION

Each of the components discussed are interconnected to compose the overall plant simulation model. To verify the accuracy of this composite model, the complete model was used to predict the steam consumption for a 300 hour period in May and June for which steam flow measurements were recorded. Only the two external driving forces were used to propel the simulation, chilled water load and ambient wet bulb temperature. The actual cooling tower fan speed settings were used to control the operation, but the chiller intrinsically determines the vane position and consequent RPM.

The results are shown in Figure 3.16. The difference between the predicted and the recorded one hour interval steam readings for the entire period is 4.06%. The RMS error for the one hour points is 2687 lbs/hr. The distinct possibility that the manually recorded temperature readings to which the individual models were fit are not identically coordinated to the on the hour electronically recorded steam readings and the accuracy with which these steam charts can be read may account for the larger part of the discrepancies.

In addition to the prediction of steam flow the model also outputs the electric power consumption. The actual electric use for each individual component is not measured. Therefore a similar comparison can not be made for this fuel source. However, in future simulations the identical relationships of power and flow rate for all the pumps and fans are used enabling the electric power consumption in each simulation to be compared relative to each other.

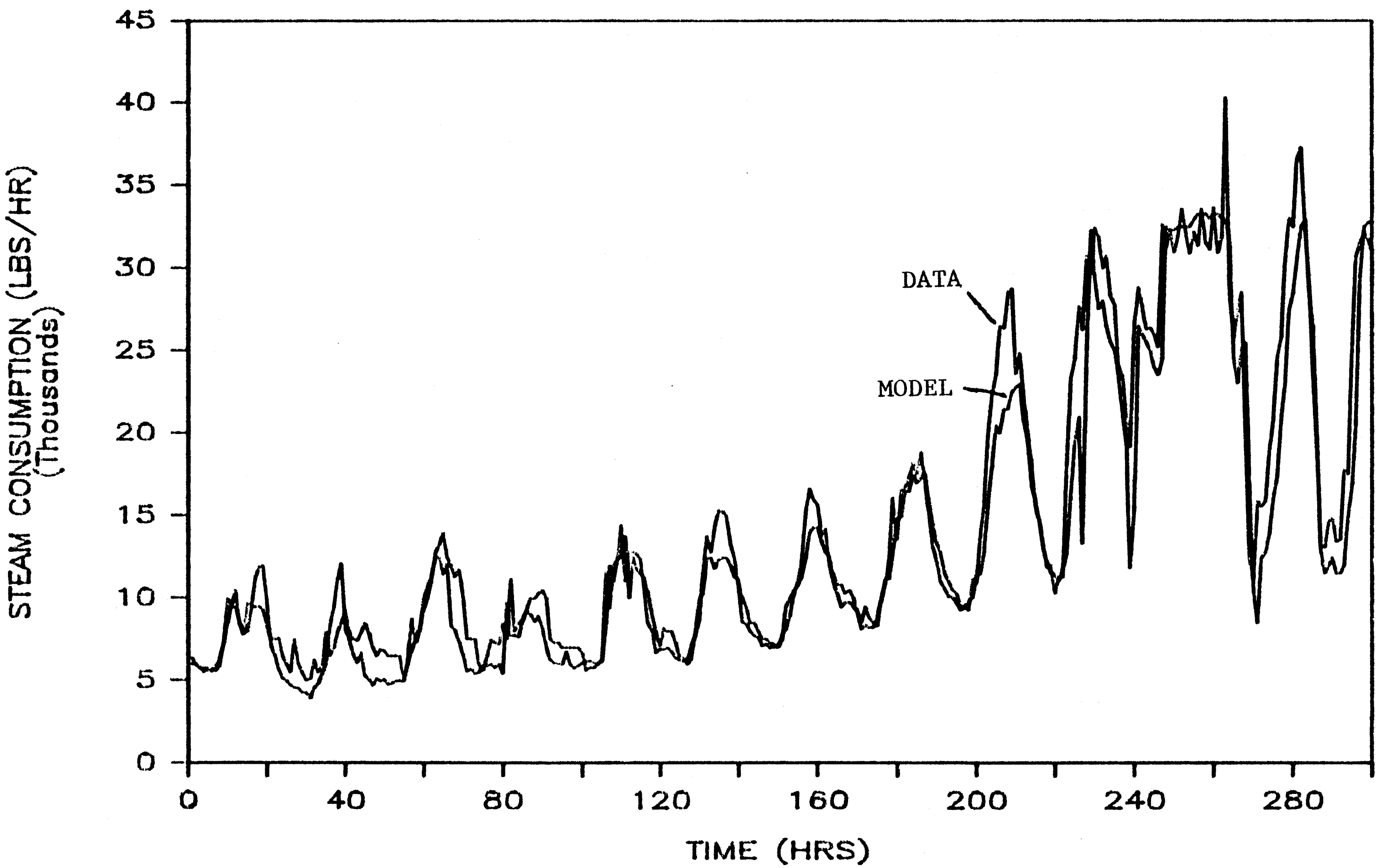


Figure 3.16 Predicted steam consumption for overall plant simulation using ambient wet bulb temperature and chilled water load as inputs versus measured data

3.7 CHILLED WATER LOAD GENERATOR

The two required inputs to perform yearly simulations are hourly chilled water load and ambient wet bulb temperature values. Since the collected data are limited to a four month period other sources of this information are needed. The wet bulb data is attainable from historical meteorological records. The source used in this study is the SOLMET Typical Meteorological Year (TMY) data. The historical chilled water load, however, has not been recorded in a readily accessible manner for computer simulations. For this reason a simulated yearly load has been used as an input.

It is important to note that though the load is a strong driving function it is not imperative that the generated load be identical to the actual load because firstly, all the yearly simulations will be run using the same load and will be compared relative to one another. Secondly, the load will vary from year to year, the important characteristics are the range of loads encountered and some approximation to the number of hours in each range.

To capture these salient features the recorded load has been correlated to the ambient driving force and the generally noted diurnal swing, being the wet bulb temperature and the hour of the day. Attempts to correlate the load with the relative humidity and day of the week were unsuccessful. The resulting correlation is ;

$$Q_{chw} = A(T_{wb} - 35) + B + C(HR) + D(HR)^2 \tag{3.25}$$

where: T_{wb} = wet bulb temperature

HR = hour of the day

A,B,C,D = empirically determined coefficients

Data for the months of May and June were used to fit this correlation with the assumption that this period is representative of the conditions encountered during the cooling season (May - Nov.). The empirical coefficients determined for this location and application are listed in Table 3.3.

TABLE 3.3

A	B	C	D	RMS
76.1579	451.079	-21.697	0.6235	399.09

Table 3.3 Load generator correlation coefficients

Figures (3.17) and (3.18) are plots of the generated load and the recorded load for the data used to fit the correlation and for another set of independent data, respectively. It is seen that the general trend is followed and that the peaks and valleys are of the same magnitude.

This correlation in coordination with the TMY data was used to generate the seven month long load and wet bulb input file to be used in the full season simulations presented in Chapter 5. The sum total of the hourly generated loads is within 10% of the average sum total load recorded at the plant for the previous 3 year period.

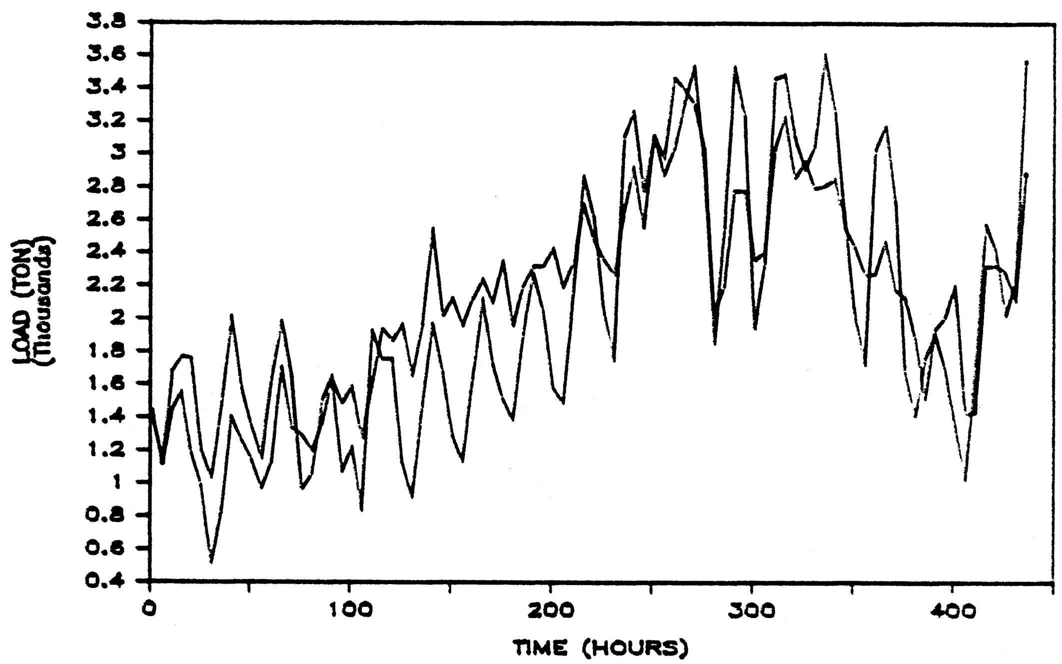


Figure 3.17 Simulated chilled water load as a function of wet bulb temperature and hour of the day versus actual load. Same data as used to fit the correlation

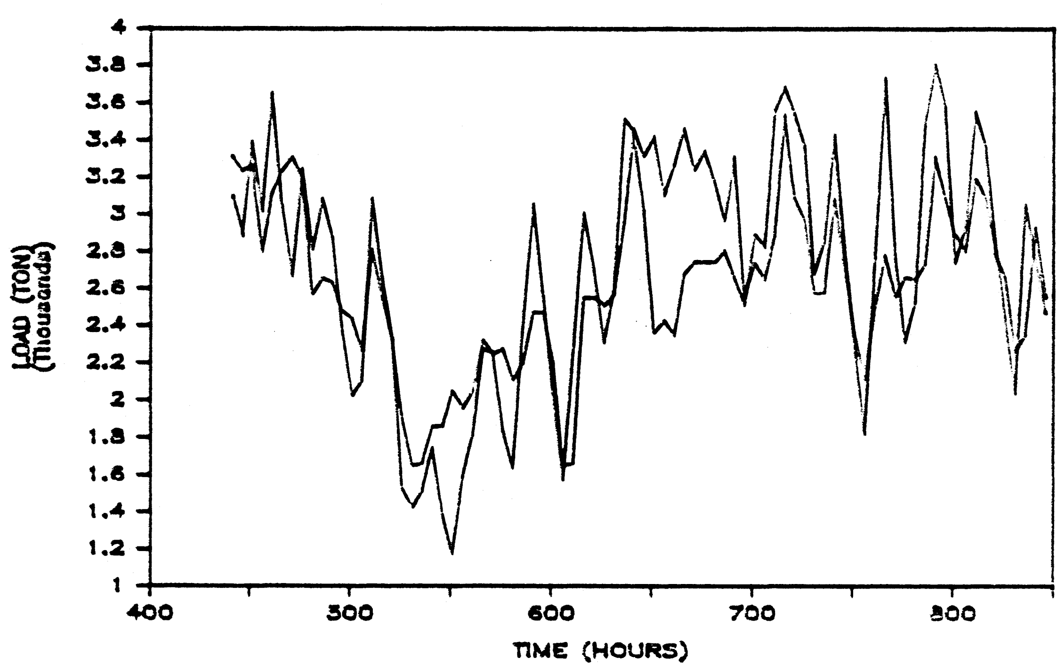


Figure 3.18 Simulated chilled water load as a function of wet bulb temperature and hour of the day versus actual load. Other data than used to fit the correlation

4.0 CONTROL LOGIC DEVELOPMENT

The goal in this study is to determine which combination of the available chiller plant control options and equipment configurations will minimize the overall operating costs. To achieve this, a comprehensive consideration of all interactions between the components is necessary. This is because the optimum operating point of one single component may adversely effect the operation of the others. Therefore, the entire system must be optimized as one entity.

The initial step is to identify and prioritize the control elements. They generally fall into three categories; those which are manually controlled, those automatically controlled and those which have fixed settings. The fixed setting elements such as the water flowrates, are not strong driving forces in the plant control interactions because they are constant. Therefore, these flowrates do not enter into the day to day control decisions, but the long term effects of different flowrates will be investigated. The automatic controls, such as the turbine speed, require little attention because they are intrinsic to the machinery and will spontaneously respond to the signals from the other controls. The remaining control elements which have strong effects on the others and on the operating costs are to be identified and the resulting effects studied with the simulation model. This methodology has been utilized at the Walnut Street facility.

4.1 CONTROL ELEMENTS

In the previous Chapter it was shown that the chiller power could be represented by a quadratic relationship in two variables. When considering the total plant power and steam consumption, there are six important variables 1) chilled water load, 2) chilled water set point, 3) ambient wet bulb temperature, 4) condenser water flow, 5) tower air flow and 6) compressor vane position with the associated RPM.

The end use of the chilled water dictates the load and set point temperature and in this study it is assumed as uncontrollable. The wet bulb is a given, uncontrollable function of the climate and the condenser water flow is a preset constant. The remaining tower air flow rate and chiller vane position will therefore be the control elements with the strongest effect on the system interactions. The tower air flow, which is regulated by the tower cell and fan speed settings, will determine the condenser water return temperature (T_{cwr}) which in turn dictates the chiller power, vane position and RPM for the given load. Fan control is then the first priority in the control logic investigation.

For a given set of external conditions, load and wet bulb, there exists many different combinations of fan settings which will provide adequate heat dissipation. The question that remains is "which of these combinations, in concert with the other components, will do the job for the least overall cost?".

To investigate the effect of the fan combinations, the total system power consumption has been simulated, at a constant load and wet bulb, for incremented fan speeds for different numbers of tower cells. The relative tower air flow is the ratio of the sum total of all the air flow to the maximum 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 4.1 indicates that the total power required to meet the load is substantially increased with decreased number of cells. This is due to the fact that fan power is a cubic function of air flow. Therefore the best fan strategy to employ is to operate as many cells in parallel as possible, all at part speed. Similar results were found for other load and wet bulb conditions. The limiting factor of this illustrative example is the availability of fan speed variety. If the fan speeds are not infinitely

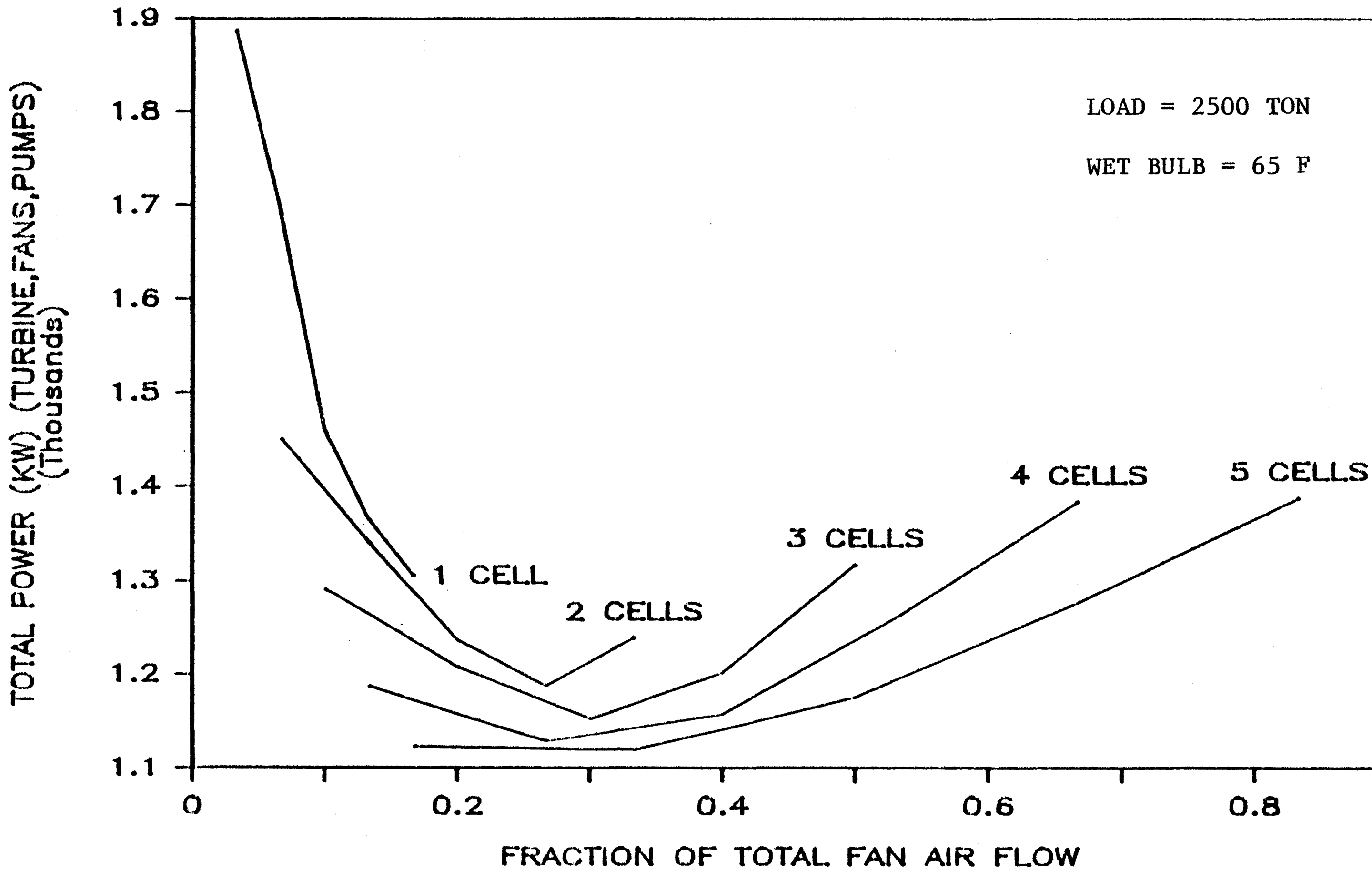


Figure 4.1 Simulated total power consumption for increasing tower air flow and different number of operating tower cells at moderate load and wet bulb

variable as shown the absolute optimum relative air flow may not be attainable, thus the minimum costs will be associated with fewer cells on at their available part speeds. Each possible combination for the given equipment must be investigated. A more detailed discussion of fan strategies is presented in Chapter 5.

The effects of control changes are not always intuitively obvious. The TRNSYS simulation program, composed of all the interconnected components, enables the investigation of the resulting interactions to be made easily. The effects of various fan settings are shown here.

The fans are varied by half speed increments one cell at a time. Setting number one being the minimum air flow and number twelve represents all cells on at full speed. For each fan setting, at a load and wet bulb chosen near the middle of their respective range, the steam and electrical consumption is simulated and plotted in Figure 4.2. There is an obvious trade off made between the two power sources. As the fan speeds are increased the condenser water return temperature is decreased resulting in a decrease in required chiller/turbine power and thus less steam is consumed. The objective is to determine which combination of these two quantities is the least expensive. This will depend upon the relative unit costs of each fuel source. This must be done for each conceivable set of load and wet bulb combinations, which uniquely describe the operating criteria.

4.2 OPERATING MAPS

To simplify the control decision making process, operating maps are developed which indicate the mode of cooling (mechanical or free cooling), the fan setting and the vane setting that will result in the minimum instantaneous total operating cost for any

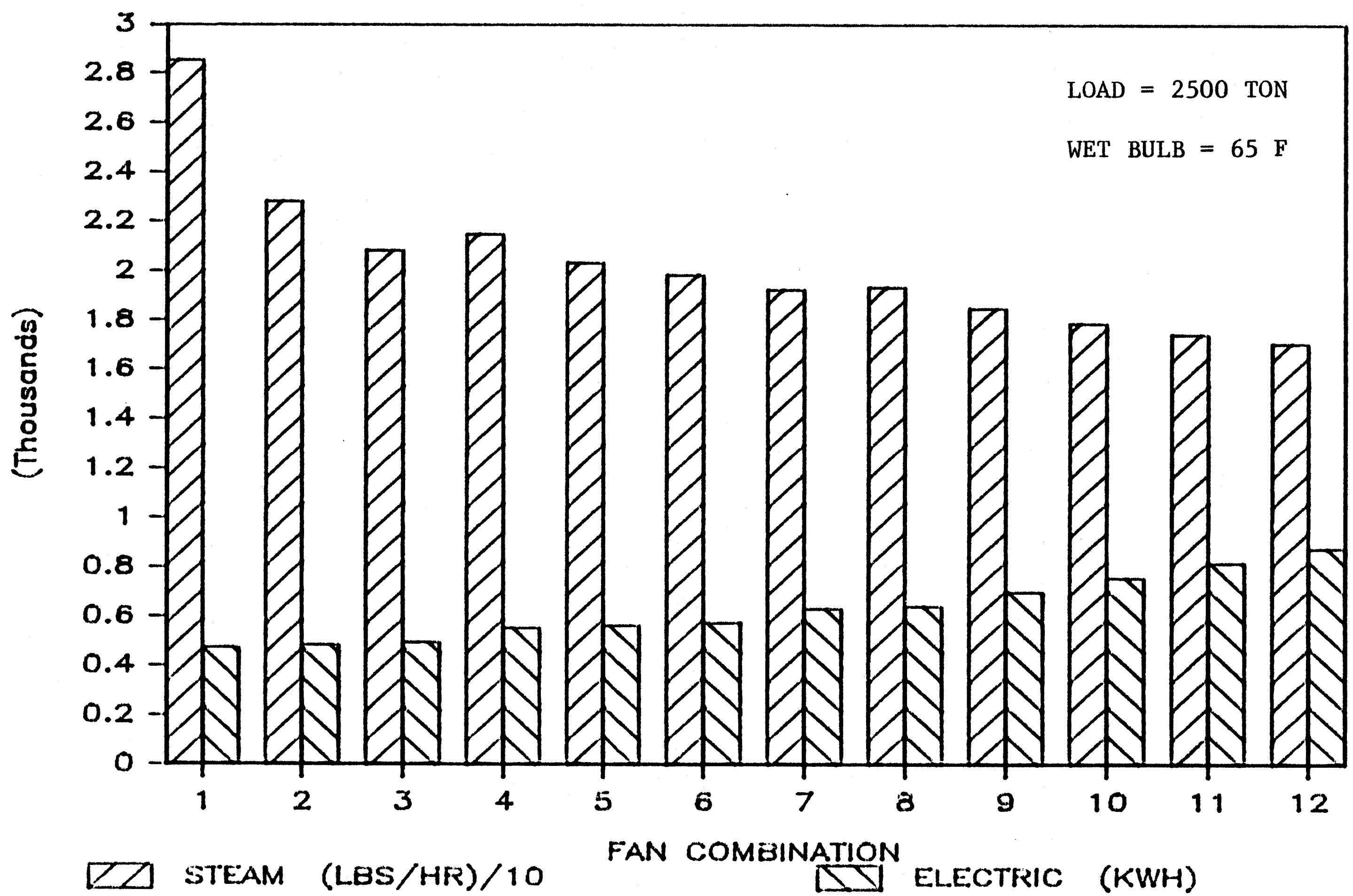


Figure 4.2 Relationship of electric and steam consumption for increasing speeds and number of tower fans for moderate load and wet bulb

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. These costs will vary throughout the season. 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. This procedure is described in the following two sections concluding with an example set of operating maps.

4.2.1 FUEL USE MATRIX DEVELOPMENT

The operating maps are optimized based on the total instantaneous operating cost. Though the unit fuel costs may vary, having 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. This section describes the process of determining the fuel use and the matrices in which these values and their associated fan and vane settings are stored.

The four important variables to be considered when simulating the plant performance are 1) chilled water load, 2) ambient wet bulb, 3) tower air flow and 4) compressor vane position. As discussed in Chapter 3, the load and wet bulb are the independent driving forces and for a chosen tower air flowrate (fan setting) there exists a unique vane setting that properly regulates the RPM. To develop the matrices each possible combination of these four variables is used in the simulation to determine the rate of steam and electric use. A fan operation strategy to be used must be defined by choosing the order in which the tower cells will be activated and at what speed the fans will be set, at specified conditions. Each of these fan setting combinations is assigned a numerical value, increasing values being assigned to increasing cells and air flow rates.

At a particular fan setting, the instantaneous performance for every combination of load and wet bulb is simulated. The resulting electric use, steam use and refrigerant compressor vane position are recorded in three separate matrices. These simulations are repeated for each of the numbered fan setting combinations. The result is 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 fan setting value defines the third dimension as depicted in Figure 4.3. With these three matrices the required vane position and the resulting instantaneous fuel use can be determined for all the desired fan settings at any given set of external conditions.

This same methodology is used to define the free cooling mode. Since the compressor is bypassed and the turbine eliminated in the free cooling mode only the electrical consumption matrix is 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.

The generation of these matrices are specific to the plant being simulated and the fan setting scenario chosen. However, they are independent of the fuel cost. For the Walnut Street plant the range of chilled water load and wet bulb temperatures used in the investigation were 100 to 4000 tons and 10 to 85 °F, respectively. The fan setting scenarios are discussed in more detail in Chapter 5, but in general fan setting number one is for one cell on at half speed and number 12 representing all cells on at full speed, unless other wise indicated.

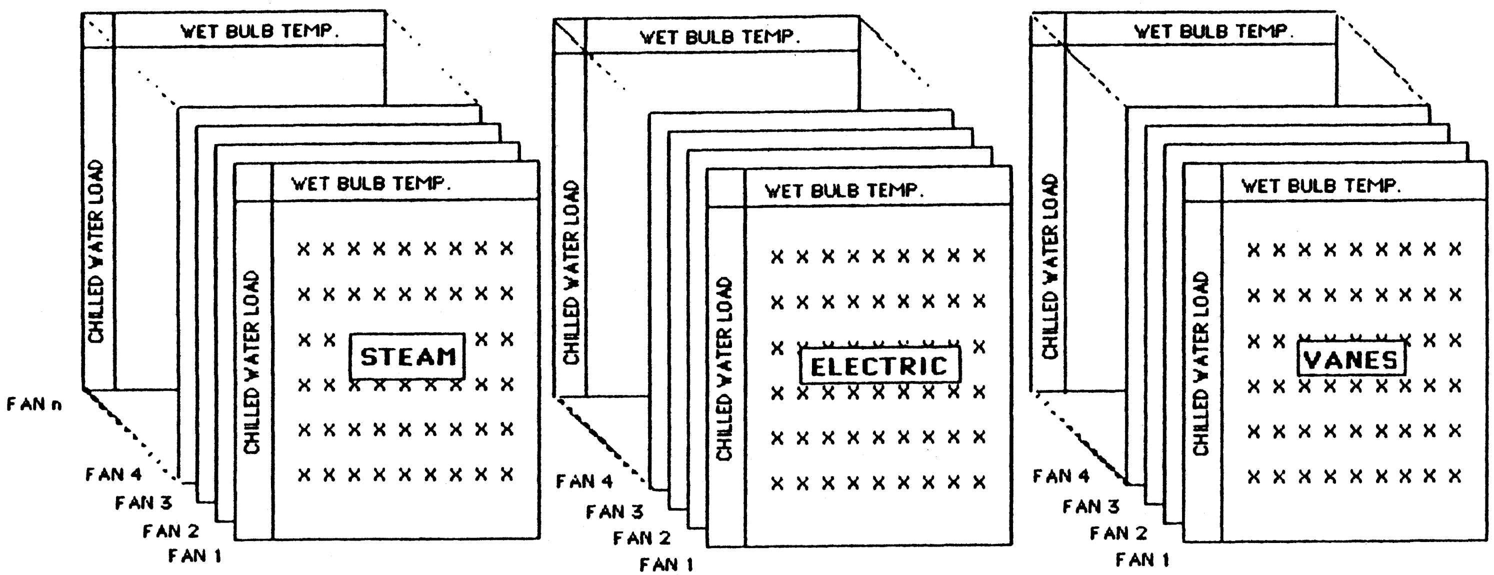


Figure 4.3 Illustrative representation of 3 dimensional fuel use and vane position matrices

4.2.2 OPERATING MAP GENERATION

The relative costs of electricity and steam will dictate the optimum trade off point between the two fuel consumption rates. A means of determining this trade off point and the fan setting and vane setting combination which will attain it is required.

Utilizing the previously generated steam and electric matrices the three independent variables, load, wet bulb and fuel cost are input and the sum total fuel cost is calculated for each fan setting. Figure 4.4 is a plot of this total cost for a variety of fan settings over a range of loads all at a constant wet bulb temperature and unit fuel costs. For visual clarity only three of the twelve possible combinations have been plotted. At each load 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 unique vane position 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 it is noted that the optimum operating cost is not very sensitive to the fan setting choice. This is evident in Figure 4.4 where the fan lines lie very close together.

At each combination of load and wet bulb the optimum fan setting and associated vane position are recorded. What emerges from this are unique operating maps indicating the fan and vane settings that will minimize the overall costs for the entire operating range at the particular unit fuel costs for which they are generated. Figures 4.5 and 4.6 are examples of such maps generated for the mechanical cooling mode. A similar map for the free cooling mode, shown in Figure 4.7, was developed in the same fashion.

Utilizing these maps, plant control decisions are reduced to the two independent variables; load and wet bulb. Referring to Figure 4.7, if free cooling is attainable the

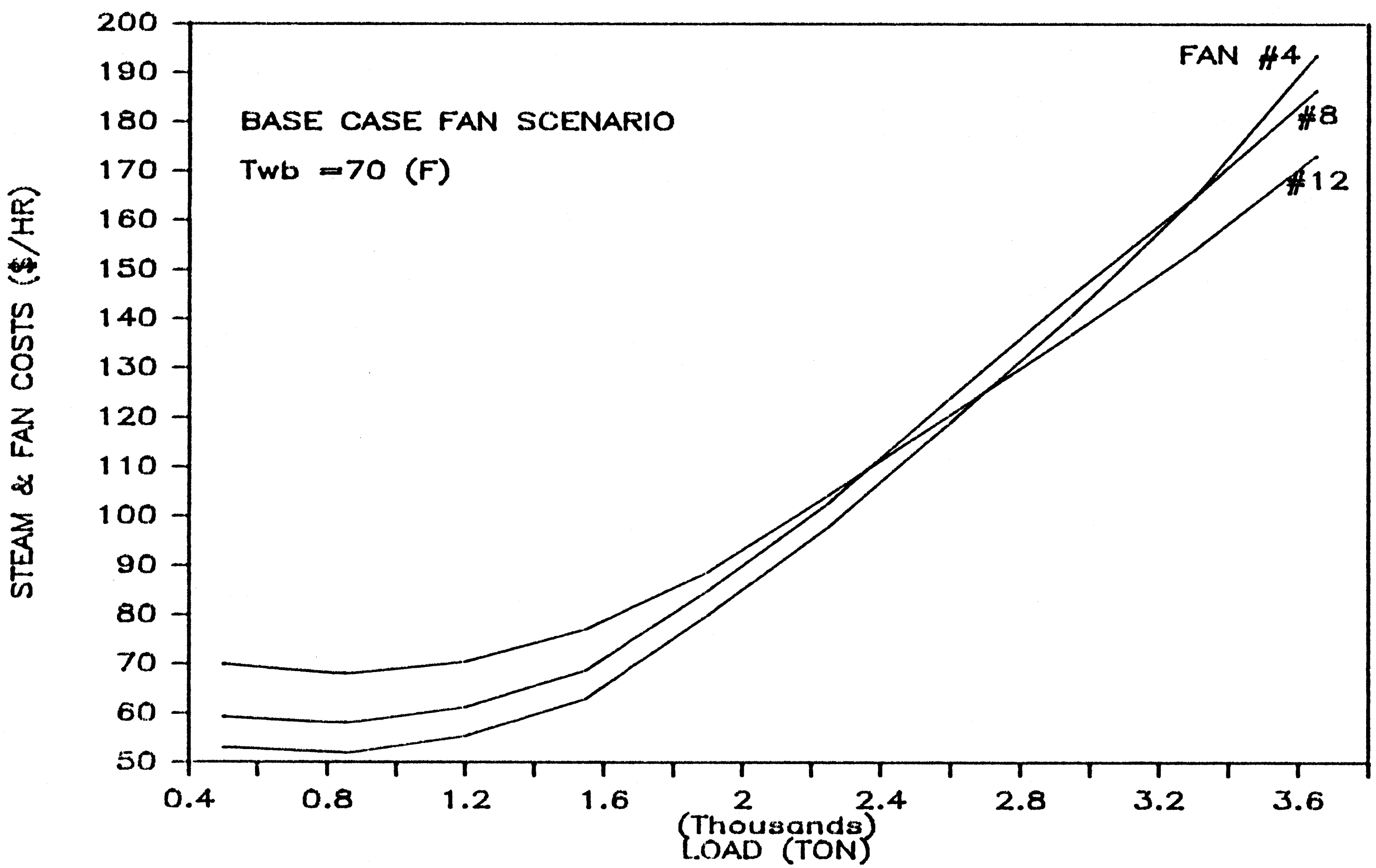


Figure 4.4 Total operating costs, over range of chilled water loads, for a variety of constant fan combination settings

fan settings for instantaneous optimization are indicated. If free cooling is insufficient the maps in Figures 4.5 and 4.6 are referenced to determine the proper fan and vane settings that will minimize the operating costs in the mechanical cooling mode.These maps are used to make the operating decisions for the seasonal simulations presented in Chapter 5 and can be used for day to day operating decisions as well.

Operating costs can be easily obtained for any set of fuel costs by making use of a separate computer program. This program utilizes the three dimensional electrical, steam and vane matrices particular to the plant and fan operation scenario for which they were generated. By inputting the desired fuel costs and external driving forces, load and wet bulb, the sum total instantaneous operating cost is calculated for each fan setting. These sum totals are output, in order of increasing costs, additionally the individual fan speeds and the vane position are indicated.

An interactive program such as this has been installed at the Walnut Street plant. The fan operating scenario used is that which has been determined to be a cost effective utilization of the existing equipment. This is the Base Case scenario discussed in detail in the next chapter. With the outputs from this program the operators can make their control decisions based on a judgement of the estimated fuel cost savings and other external implementation costs, such as added man hours. A listing of this program and a sample input / output statement are included in Appendix D.

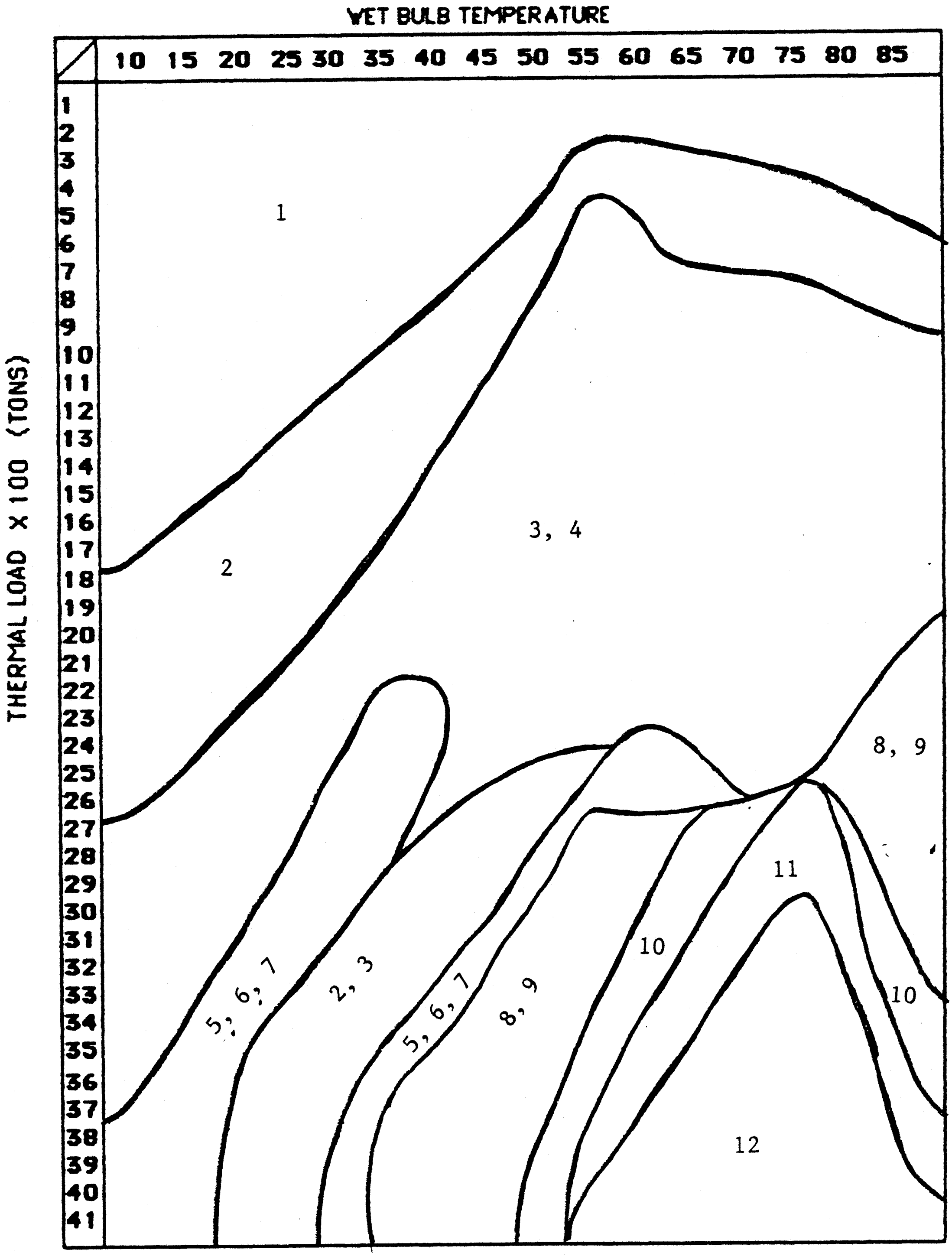


Figure 4.5 Sample mechanical cooling fan control map for base case control scenario

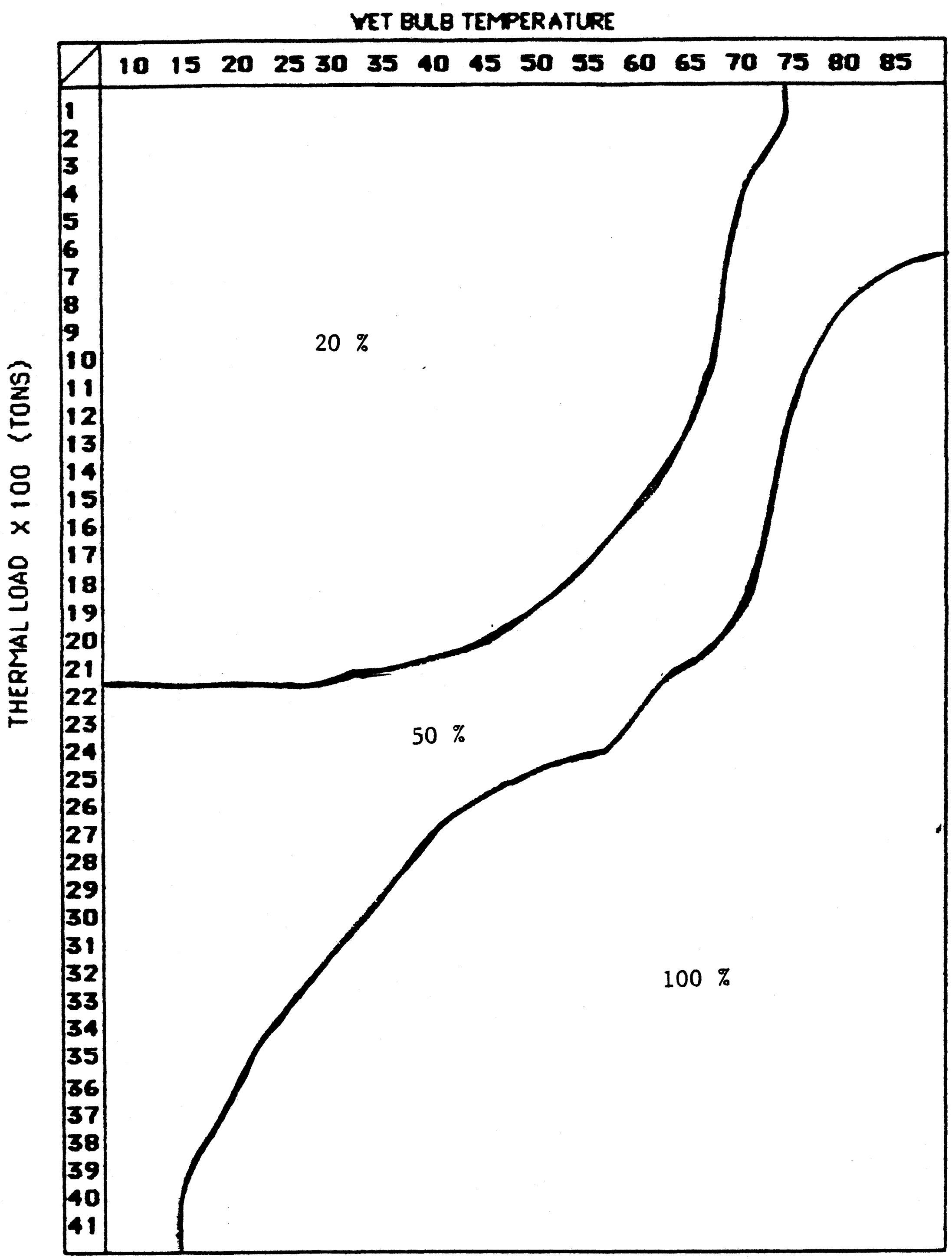


Figure 4.6 Sample mechanical cooling chiller vane control map for base case scenario

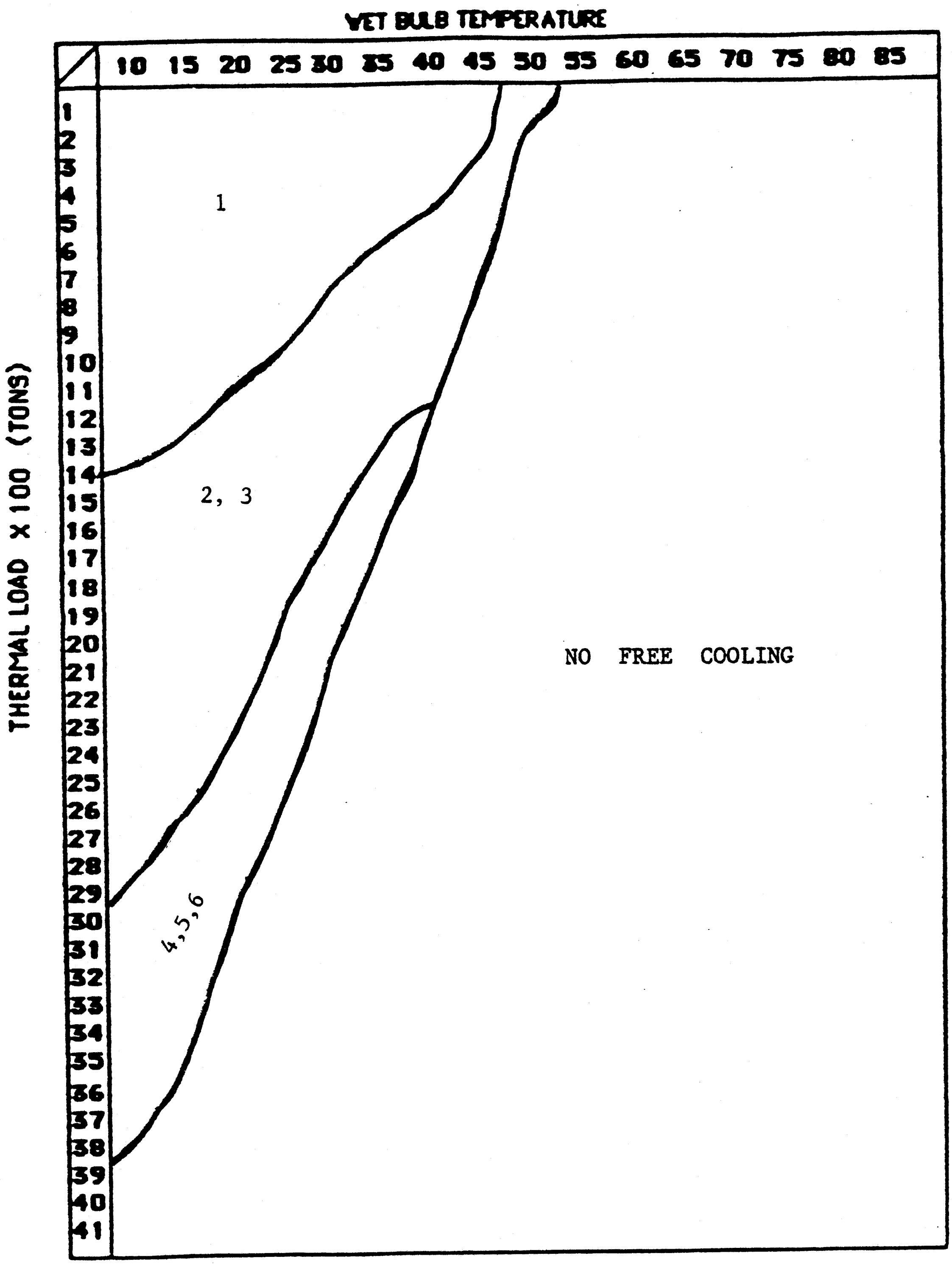


Figure 4.7 Sample free cooling fan control map for base case scenario

5.0 SEASONAL SIMULATIONS

In Chapter 3 the individual equipment models were developed and verified. Additionally, the simulated chilled water load used to drive TRNSYS seasonal simulations was introduced. In Chapter 4, a methodology for generating operating maps was developed. In this section, several different control strategies are investigated. The results of yearly simulations using the TRNSYS model driven by the simulated load are presented for the various control alternatives.

To this point the development of models and the discussion of methodologies have been in general terms, followed by the specifications for the Walnut Street facility. The following discussion and quantitative results are particular to the Walnut Street plant.

The simulation results presented provide a quantitative means by which contemplated control and equipment retrofit decisions may be based. In addition, the economic merit of previous retrofits are quantified. These findings lead to some general conclusions and recommendations which can be made for the operation of this and other similar chilled water plants that are summarized in the next chapter.

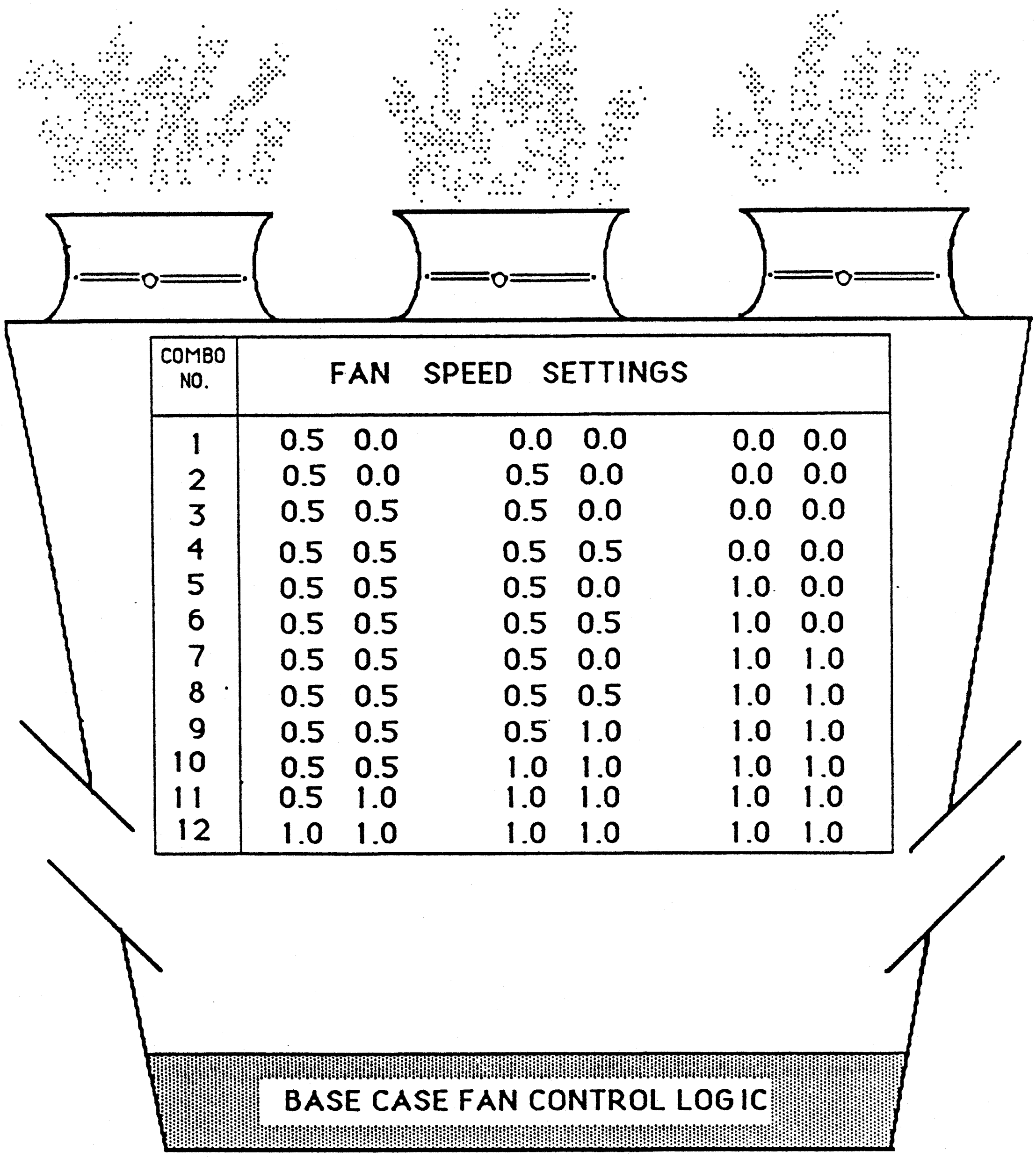
For comparison purposes, the seasonal simulations utilize unique control maps developed for each scenario. These optimized maps were generated for the present average unit fuel costs of $4.08/1000 lbm. of steam and $0.05/KWhr of electricity using the methodology developed in Chapter 4. Each simulation was executed using identical load and climatic profiles and a condenser water flow rate of 11180 gpm. 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 the scenarios, unless otherwise stated, the free cooling map is accessed first. Whenever possible free cooling is utilized, otherwise control is transferred to the mechanical cooling map. The individual fan speeds associated with the fan combination number are input to the tower model with the wet bulb and the chiller is driven by the load. The simulation calculates the predicted instantaneous fan and pumping power and the turbine steam consumption each hour and integrates these quantities over the cooling season. A sample TRNSYS simulation deck is listed in Appendix B and each fan control strategy is illustrated in Appendix E.

5.1 BASE CASE

Presently the plant control is based on learned manual 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 which spills out the sides of the tower due to lack of air intake suction). This scenario is depicted in Figure 5.1. The assigned fan setting combination number is listed at the left and the fan speeds for each of the two cells of the three towers indicated to the right.

The control maps using this scenario have been developed using the methodology described in Chapter 4. In fact, as stated, the maps shown in Figures 4.5 to 4.7 are for this base case logic.



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 5.1 Base case fan control scenario, combination number and associated individual fan speeds

The basis by which comparisons of alternative strategies are made is the overall operating costs for the entire season. Though the actual electric demand is not measured, the same chilled water and condenser water pump flowrates, power correlations and fuel costs used in this base case are used throughout. The electric costs are therefore relative to each other. Differences in electrical costs are due to the variations in fan operations. The base case seasonal operating costs have been generated using the Madison TMY wet bulb data and the simulated chilled water load developed in section 3.7. For the 5640 hours of operation, the total accumulated load is 9.502×10^6 ton hours requiring 3.118×10^6 kW hrs of electrical power and 6.205×10^7 lbs_m of steam resulting in a total annual operating cost of $409,100. The alternative strategies are compared relative to this annual cost.

5.2 ALTERNATIVE TOWER AND FAN CONTROL

The control scenarios described in this section are those which consider the separate tower interconnections and/or the fan speeds and sequencing. For each of the strategies described, the respective control maps have been developed by the methodology introduced in Chapter 4. To assess the relative merit of each scenario, the seasonal simulation utilizes these individual maps and the identical load and wet bulb inputs as the base case.

5.2.1 TWO TOWER CELL OPERATION

The original design of this chiller plant allotted one tower, 2 cells, to each individual chiller. The reason for the extra set of cells is that the initial plant specifications called for three chillers to be installed, but due to financial constraints only two were purchased. 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 the tower reconfiguration, a representative tower fan sequencing scenario for the one tower per chiller operation was devised. There are only two 2-speed fans to consider, producing four possible fan combinations; one cell at half speed, two cells half speed, one at half, one at full and both on full. The seasonal simulation results operating under this scenario, relative to the base case, are presented in Figure 5.2 and 5.3 along with the other configurations presented in this section. The negative value associated with this scenario, labled 2CELL in Figure 5.2, indicates that operating the plant in this manner would cost substantially more per season than operating under the base case control scenario (a negative savings). In Figure 5.3, the relationship between the steam and electrical consumption is presented. Due to the limited surface area in the two cells, a large air flow rate is required to provide adequate evaporative cooling. The result is large electrical costs to drive the fans. Because the condenser return water temperature can not be driven down due to limited air flow the chiller operates less efficiently and results in higher steam consumption as well. It was found that by implementing the tower interconnections an approximate savings of \$17000 a season is realized.

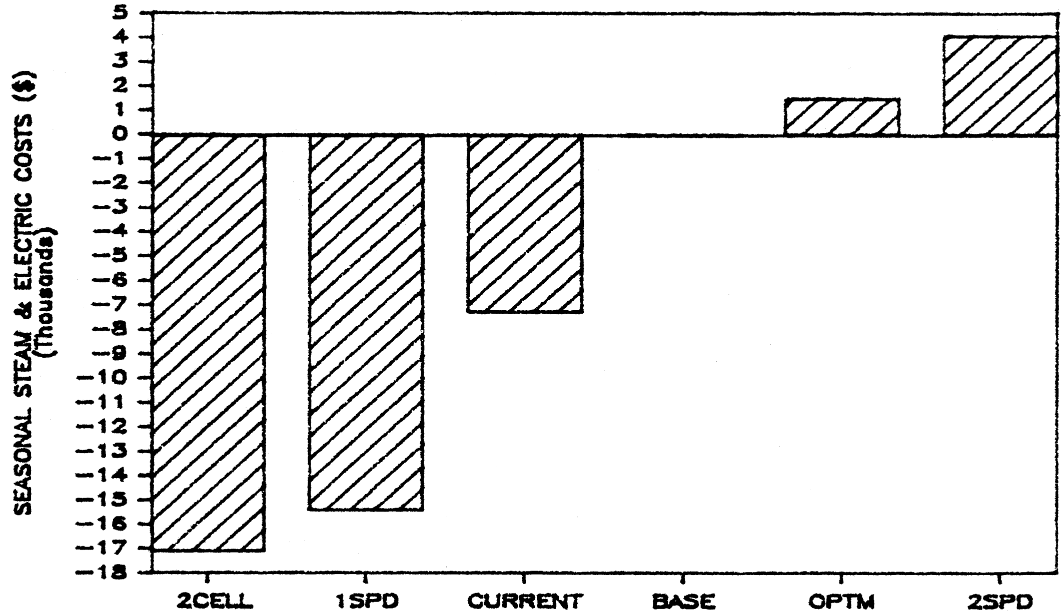


Figure 5.2 Seasonal savings of alternative fan and tower strategies relative to the base case

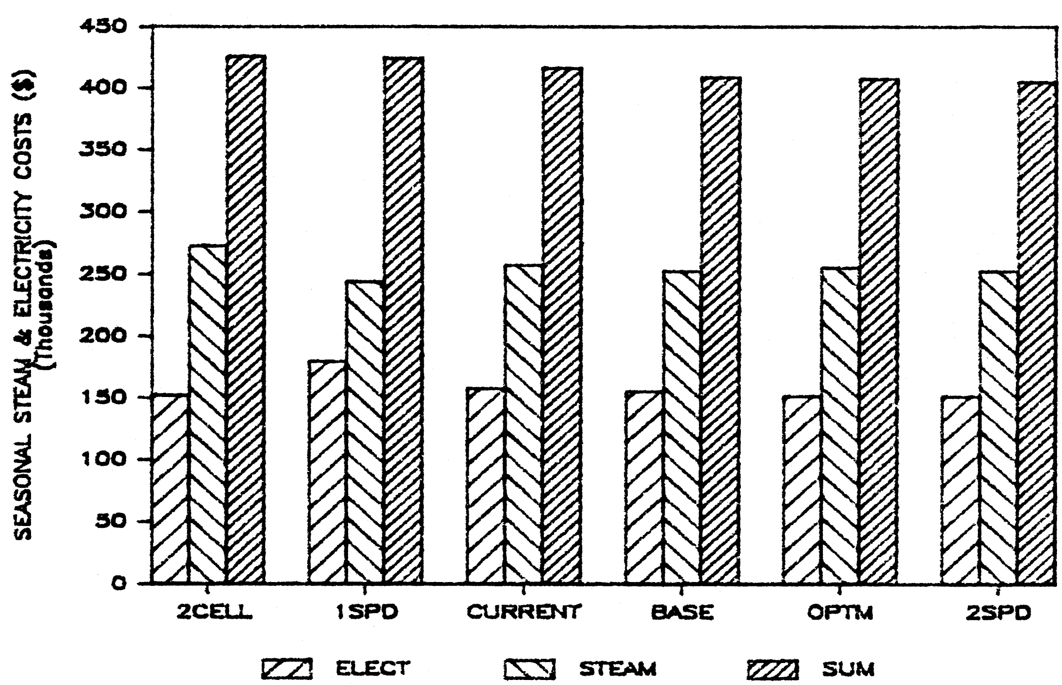


Figure 5.3 Relative seasonal electric and steam costs of alternative fan and tower strategies

5.2.2 ONE SPEED FANS

It was desired to quantify the seasonal cost advantage of using two speed fans, as is presently configured, compared to one speed fans. This strategy 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 (the difference between the leaving condenser water temperature and the limiting ambient wet bulb temperature). This substantially decreases the chiller power required and subsequently the steam consumption, but this is at the cost of additional electrical power to drive the high speed fans, as shown in Figure 5.3. The seasonal simulation indicates that the six one speed fans provide a slight savings over the two cell operation, but that the two speed fans in the base case account for an overall savings of approximately $15400 annually.

5.2.3 REFINED BASE CASE

The operation of the existing equipment defined by the base case has the potential of being slightly improved. This comes at a cost of additional fan speed and tower interconnection manipulation. These costs, which come in the form of 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 all operating. The half speed increments are continued by incorporating the one speed, high velocity fans one at a time and counter balancing the large step in speed by turning off one half speed fan. This is continued until all six cells are on at full speed. This refined scenario results in a slight savings over the base case, indicated by the positive value labeled OPTM in Figure 5.2. It is estimated to be $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.

5.2.4 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 one speed fans in the third tower with two speed fans. This scenario was simulated with 12 possible fan combinations, each of the six cells being activated one at a time at half speed increments again, requiring that at least on cell of each active tower is in use.

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.

5.2.5 APPROXIMATION OF ACTUAL OPERATION

Though the current operation was not possible to emulate with a computerized control logic the resulting operation 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, as described in section 5.1, 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 15.75% 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. This resulted in an annual electric cost of $158,624 and a steam cost of $257,522 totaling $416,146. The base case, therefore, saves $7046 over the current control due to more careful control.

5.3 EQUIPMENT MODIFICATIONS

In addition to the daily equipment control decisions, four modifications intrinsic to the construction of the plant and its installed equipment were investigated. A similar approach has been taken here as presented in the previous sections. Each scenario was simulated for the entire cooling season and the results were evaluated relative to the base case. In this section, only equipment modifications were considered, therefore the fan operations were not effected. For all the scenarios described in this section the base case fan control maps, described in section 5.1, were used.

5.3.1 FREE COOLING

To utilize free cooling, the chiller must be equipped with the compressor and expansion valve bypass piping. The existing equipment is so equipped. All the seasonal simulations thus far have utilized free cooling whenever possible. To determine the cost 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 indicate that, by operating in the mechanical cooling mode all season there is a $22,100 increase in operating costs. This is shown in Figure 5.7. The added cost is associated with the increase in steam consumption necessary to keep the turbine running all season.

5.3.2 IDEAL TRANSMISSION

The ideal transmission is a novel idea investigated as an upper limit of performance. It is noted that there are technical limitations and inefficiencies imposed on the physical application of this idea that have not been considered in this study. An ideal transmission is one which would allow both the turbine and the chiller to turn at their own respective optimum RPM independent of one another with no mechanical losses or inefficiencies.

The chiller operates most efficiently when there are no refrigerant flow restrictions, the 100% vane position. This is apparent in Figure 5.4 which shows the relative chiller power consumption for each vane position. The cross hatched areas in this figure are extrapolated results outside the range of normal operation for that vane position. As discussed in section 3.3.1, to compensate for the varying load on the evaporator, the refrigerant mass flow rate must be regulated. When holding the vane setting constant at the wide open position to minimize the power input the compressor speed must be altered to compensate for the load changes.

The steam turbine operates at its highest efficiency at very high RPM's, as seen in Figure 3.5. It would be beneficial if the turbine could always be run at this high rate of speed. Unfortunately, this is not the same speed at which the chiller would optimally

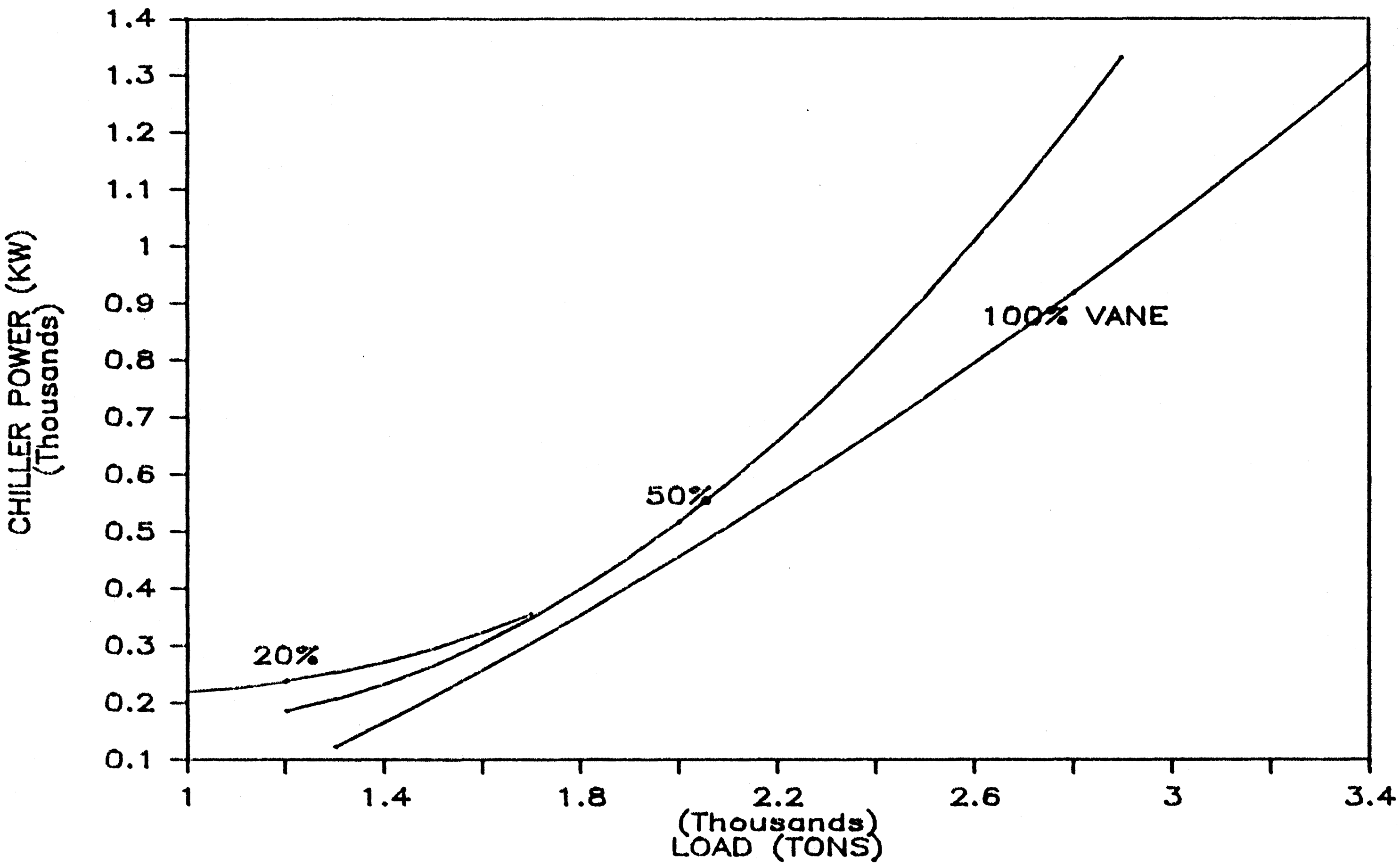


Figure 5.4 Chiller power requirement, for different vane settings, over a range of chilled water loads

operate. Ideally, a transmission which will compensate for these differences in RPM and allow each component to operate at its respective optimum for all load conditions would be used.

This ideal scenario was simulated by utilizing only the 100 percent vane setting chiller coefficients to determine the chiller power and RPM requirements. The resulting power requirement as determined from Equation (3.12) was input to the turbine model. The optimum turbine RPM, however, is that which minimizes the steam flow for this given power output requirement. To determine this value, the first derivative of the turbine steam consumption equation, Equation (3.7), is set to zero and rearranged yielding;

$$RPM = \frac{-A_2 - A_5\,PWR}{2\,A_3} \tag{5.1}$$

The steam consumption was determined utilizing this RPM and the required chiller power as inputs to the turbine model.

The seasonal simulation for this scenario estimates an annual savings of \$64,900 relative to the original base case.

5.3.3 TOWER AIR RECIRCULATION

At the Walnut Street 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 recirculation of the approximately saturated tower exhaust air. This recirculated moist air impairs the evaporative heat exchange occurring in the tower. It was desired to determine to what extent recirculation exists and what effect it has on the overall performance.

To determine the amount of recirculation, experimental measurements of the air stream humidity ratios were necessary. 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) the ambient conditions well removed from the tower (see Figure 5.5).

The recirculation rate was determined from a mass balance on the moist air flows;

$$m_a \, \omega_a + m_r \, \omega_o = (m_a + m_r) \, \omega_{mix} \tag{5.2}$$

where: m_a = mass flow rate of ambient air

m_r = mass flow rate of recirculated air

ω_a = humidity ratio of ambient air

ω_o = humidity ratio of tower exhaust air

ω_{mix} = humidity ratio of ambient and recirculated air mixture

By rearranging Equation (5.2) the ratio of recirculated air to fresh intake air is determined as;

$$\frac{m_r}{m_a} = \frac{\omega_{mix} - \omega_a}{\omega_o - \omega_{mix}} \tag{5.3}$$

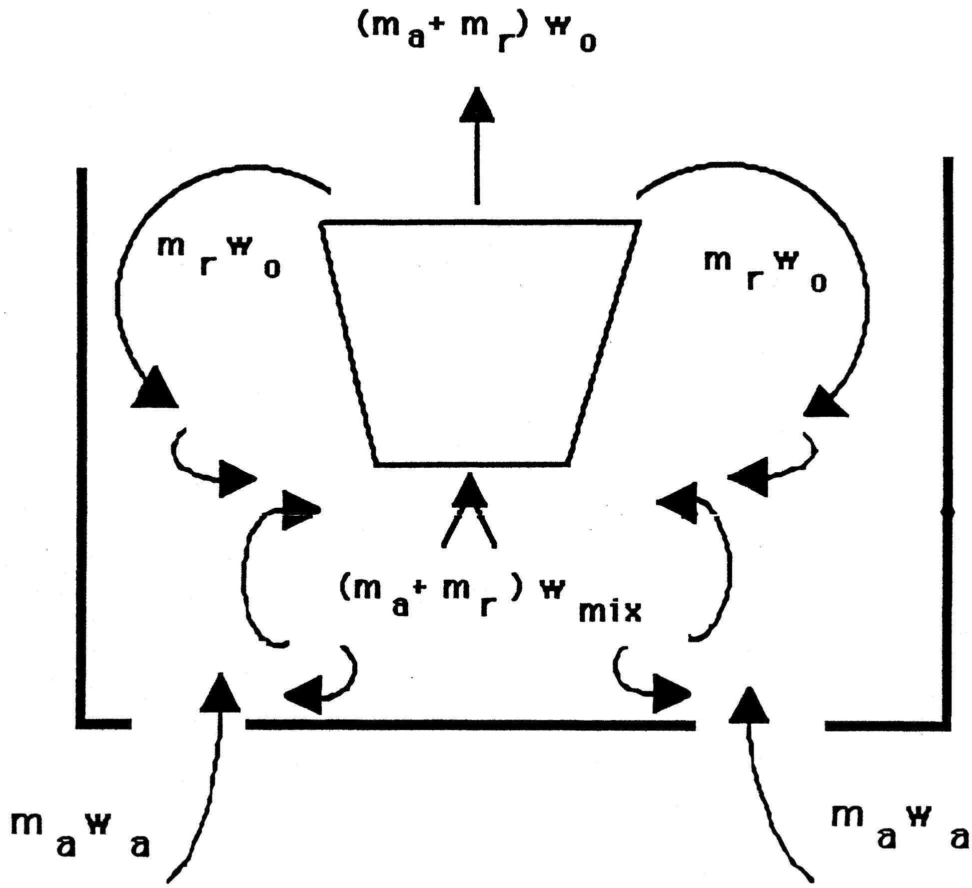


Figure 5.5 Schematic representation of tower exhaust air recirculation within the building facade

The psychometric readings taken at the Walnut Street plant on three separate calm, dry days in September indicated a range of recirculation rates from 8.9 to 19.4 percent with an average rate of 13.5%.

$$\frac{\bar{m}_r}{m_a} = 13.5\ \% \tag{5.4}$$

To simulate the effect of recirculation on the plant performance, the wet bulb and dry bulb temperatures of the tower inlet air mixture of fresh and recirculated air are needed. For a given tower fan setting, the total air flow is known and the relative ambient air and recirculation air flows are easily determined from Equation (5.4) for an assumed constant recirculation rate. With these flows and ambient air conditions known and assuming saturated tower exhaust air, an adiabatic mixing analysis defines the mixture temperatures to be input to the tower model.

The literature indicates that generally a recirculation rate of 5 - 8% can be assumed for a free standing cooling tower. This scenario has been simulated over the entire season for a number of assumed constant recirculation rates ranging from 5 to 15%. The full season operational costs associated with each of the simulated recirculation rates are plotted in Figure 5.6. The savings realized by removing the wall and assuming a 5 percent recirculation rate is shown in Figure 5.7, relative to the base case.

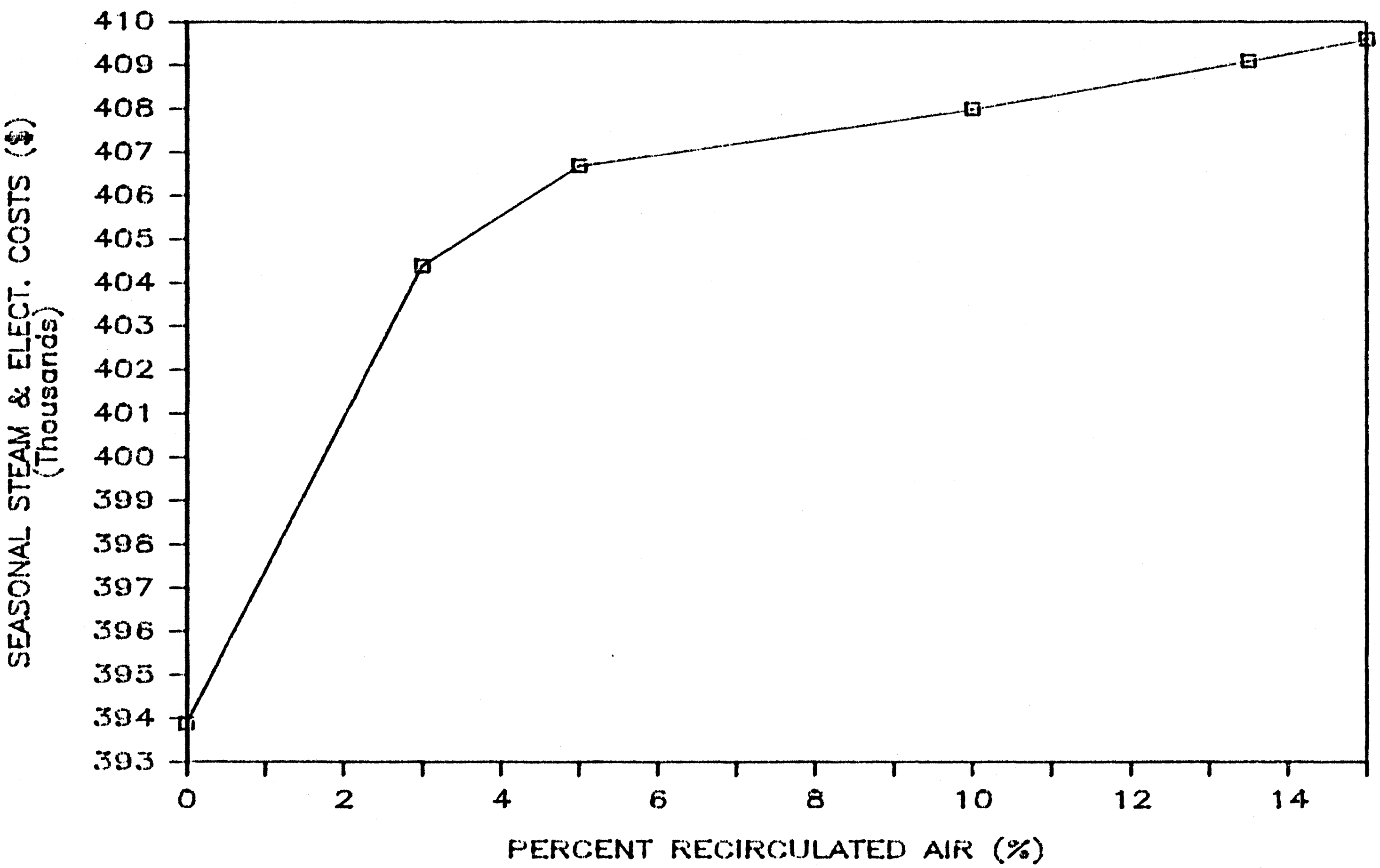


Figure 5.6 Seasonal operation costs presented as a function of recirculation rate

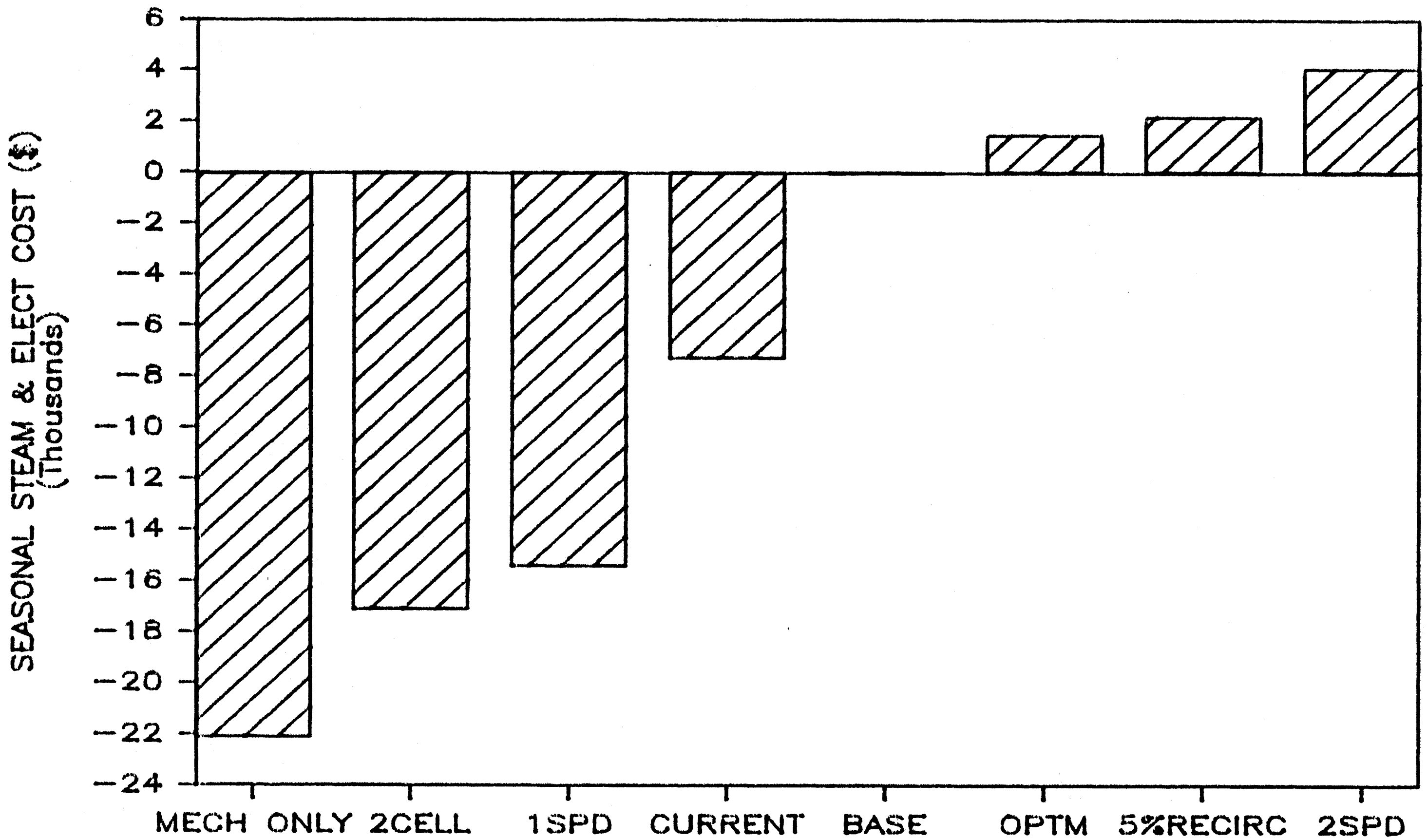


Figure 5.7 Seasonal savings of alternative equipment modifications relative to the base case

5.3.4 CONDENSER WATER FLOW RATE

The condenser water pumps at the Walnut Street plant are an example of a fixed setting control element. They are one speed pumps with flows preset by in line restriction valves. This flow rate effects 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.

Seven different constant condenser water flow rates were simulated for the full season. The results of these simulations are presented in Figure 5.8. This figure indicates a minimum overall operating cost to exist at a flow of 11500 gpm, which is approximately the flow at which the pumps are now set to operate.

5.4 TWO CHILLER OPERATION

The test facility is capable of operating both chillers simultaneously, in parallel. Both the chilled and the condenser water flow rates are doubled when the second chiller is brought on line. The chilled water load is divided evenly between the two separate evaporators, but the condenser water load must still be met by the same six cell cooling tower. The individual chillers will therefore require less power due to the divided load, but to determine the most cost effective coordination of the two chillers the total overall costs must be considered.

To determine the point at which it is most economical to operate two chillers at part load, as opposed to one chiller at near full load, simulations for both scenarios were performed. The total load was incremented from 2500 to 4500 tons and the wet bulb range considered was 75 to 95 °F. The control setting that resulted in the minimum overall cost at each set of conditions was utilized. The total operational cost over the

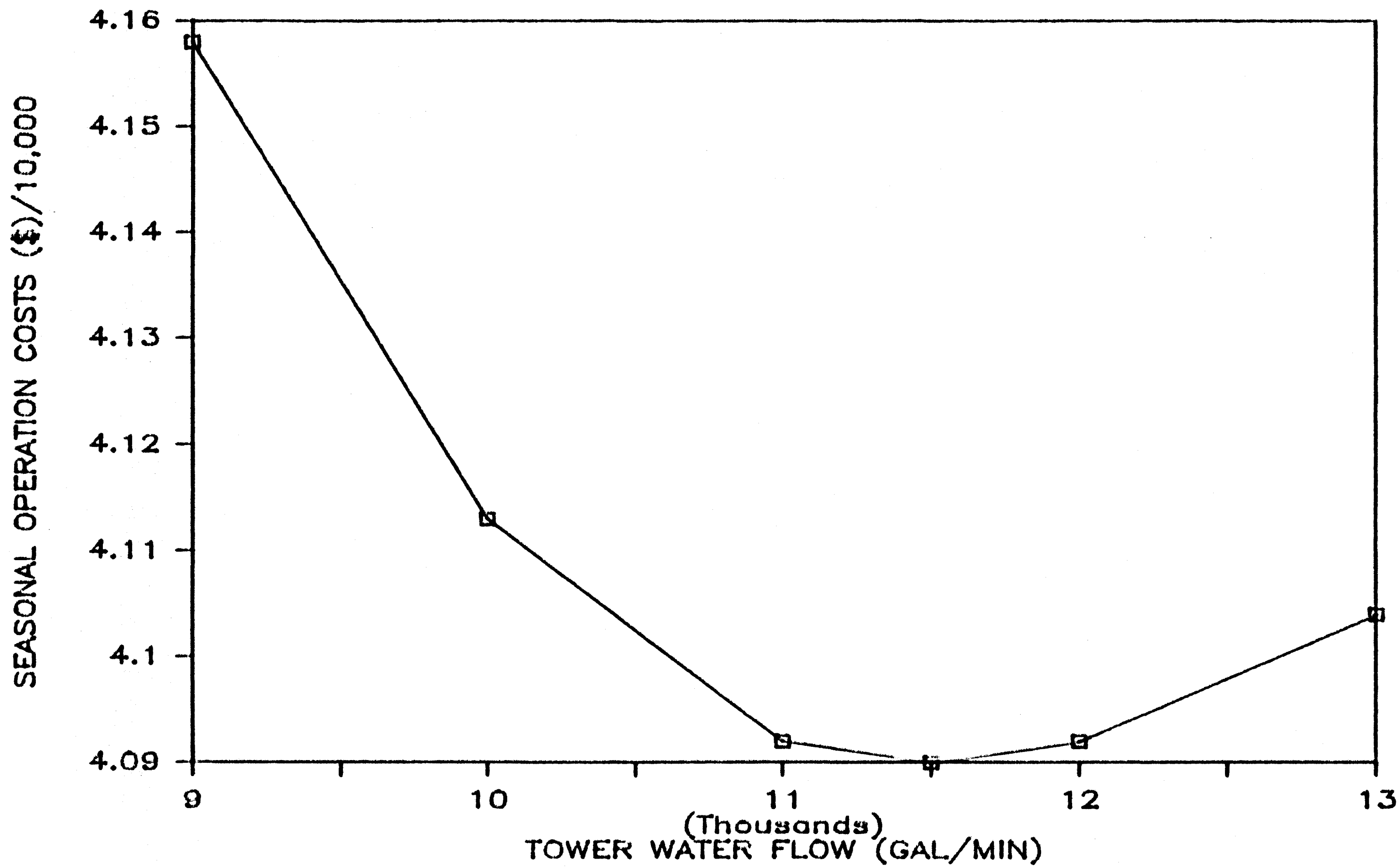


Figure 5.8 Seasonal operating costs for different constant condenser water flow rates

range of loads considered, at a constant wet bulb, for both one and two chiller operation are plotted in Figures 5.9 to 5.11. It is shown that for wet bulb conditions greater than 75°F, it is more economical to operate a single chiller until it reaches its maximum capacity before bringing the second chiller on line. The discontinuities exhibited in the curves of Figures 5.9 to 5.11 occur at the points where the vane positions are changed.

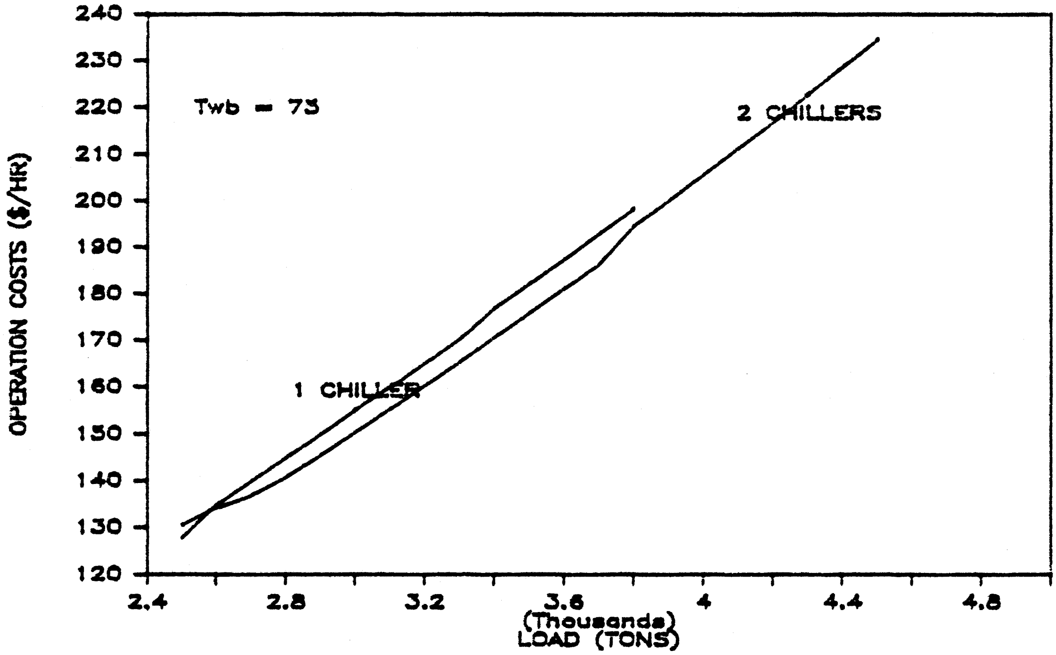


Figure 5.9 Seasonal operating costs of one chiller and two chiller operation (wet bulb temperature = 75°F)

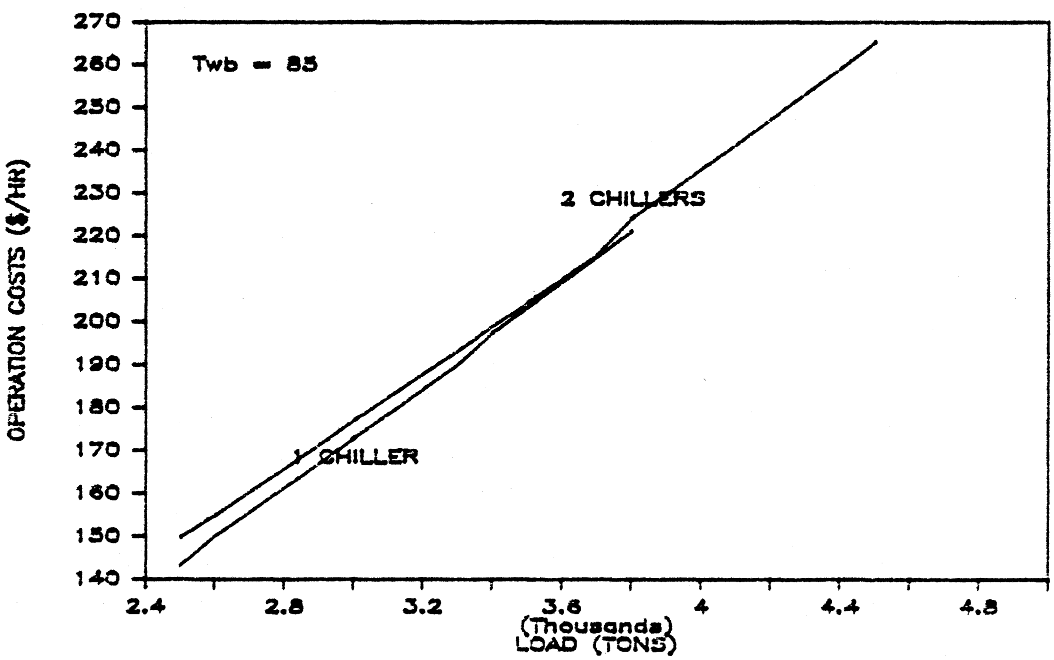


Figure 5.10 Seasonal operating costs of one chiller and two chiller operation (wet bulb temperature = 85°F)

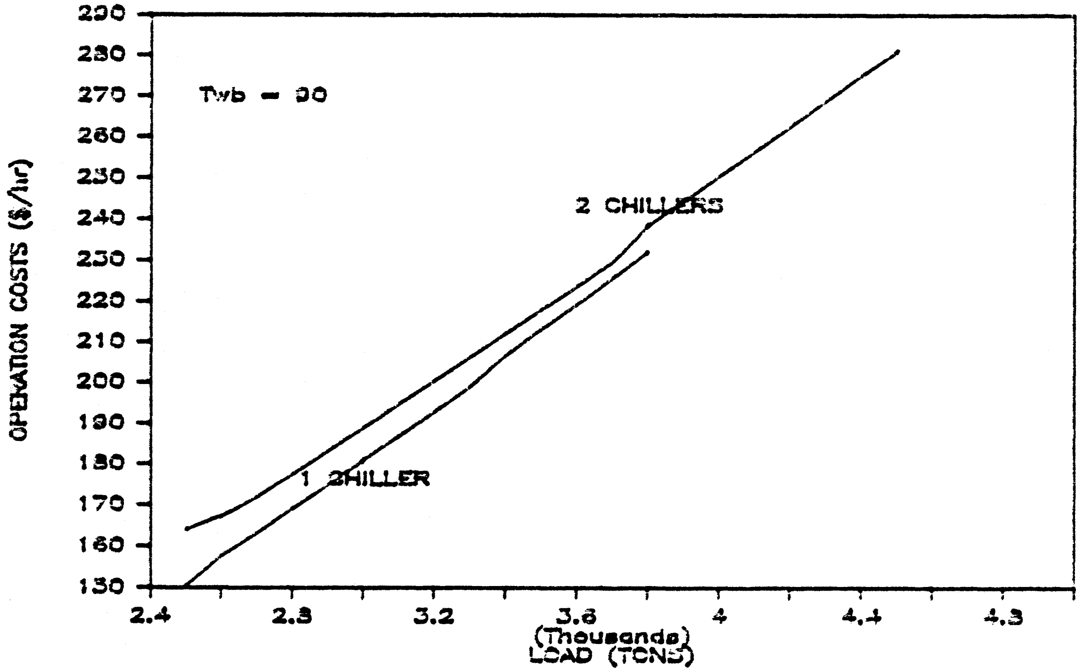


Figure 5.11 Seasonal operating costs of one chiller and two chiller operation (wet bulb temperature = 90°F)

6.0 CONCLUSIONS AND RECOMMENDATIONS

This thesis has presented an analysis of optimized control of a steam turbine driven central chilled water plant. This analysis has provided the means by which to simulate such a facility and a methodology to determine the cost effective operational control. The objective of developing these capabilities was to evaluate alternative control strategies for the Walnut Street test facility and provide a model for future investigations on similar facilities. From the results of this study the following conclusions are drawn and recommendations made.

6.1 CONTROLS AND EQUIPMENT MODIFICATIONS

1). Interconnecting the three towers provided a savings of $17100 on an annual basis. The savings at this plant are significant. It was a wise decision to implement this modification. It is recommended that this modification be investigated in all future plant studies.

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.

2). The use of two speed fans has been shown to be a definite advantage over one speed fans. 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. The savings resulting from converting the two one-speed fans to two speed fans was found to be \$4100. The payback time for this conversion has not been investigated.

3). The utilization of free cooling was found to save \$22,100 a year. This is a very cost effective means of providing cooling. It is recommended that free cooling continue to be used whenever possible. It is also recommended in future studies that for chillers without the free cooling option the cost and benefit on converting be immediately investigated.

4). The ideal transmission is a novel idea which requires more thorough investigation. It was presented here only as an example of the upper limit of savings. The inefficiencies and control of such an device have not been taken into consideration in this study. The manufacturers data and specifications or experimental tests would be required to adequately simulate the true performance of a variable speed transmission.

5). There is a substantial amount of recirculation occurring within the tower facade at the Walnut Street plant. The exact amount for all conditions was not determinable due to the many variables involved. Assuming a constant free standing recirculation rate of 5% the penalty associated with the facade was estimated to be \$2210 annually. These savings must be weighed against the aesthetic value of the facade and the cost of alternative rectifications.

6). The overall plant performance was found to be relatively insensitive to the choice of condenser water flowrate near the optimum flow. At the Walnut Street plant the optimal flow is presently in use. This flowrate has been preset by the chiller manufacturer. Flowrates within approximately 15% of the optimum will not greatly effect the seasonal performance. Outside this range resetting the flow should be considered.

6.2 COMPUTER MODEL USE AND BENEFITS

The major portion of this project was the development of the computer models used in the simulations. Substantial effort was invested in modifying existing models to included the added control variables and developing new models. All the models were designed to be general in nature which are then made specific with experimental parameters. The same components can be used in future studies which will now require less effort to implement. The components need only be characterized by the design parameters.

The other major part of the investigative procedure was the development of the control maps and the program which uses these maps to determine the optimum fan settings and vane position for a specified load and wet bulb. The determination of the independent variables and their relationships with the control decisions were discovered by implementing numerous simulations and observing the system behavior. The resulting control maps, presented as a function of two variables, enabled the investigation of alternative strategies to be greatly simplified and accelerated.

Similar maps may be generated for each unique strategy and plant configuration using these developed components and methodology. This is the ultimate advantage of a modular computer simulation tool like TRNSYS. Long term, quantitative results can be generated for any variety of scenarios easily, quickly and accurately. The alternative would be a trial and error experimental approach with the actual equipment. This would

The three dimensional matrices, developed with the TRNSYS model, greatly facilitate the determination of varying fuel cost effects on operation costs. They enable the user to investigate the instantaneous consequences of fuel cost changes for any set of operating conditions. In addition, they provide a quick and easy means of comparing the costs associated with operating with the different fan speed and vane settings.

6.3 RECOMMENDATIONS FOR FUTURE SIMULATIONS

As with any first time experimental endeavor lessons are learned and improvements can be made. In future studies on similar plants the following suggestions are made that may make life a little easier and the investigation more encompassing.

1). If the budget allows, all experimental readings should be automatically collected at prescribed intervals by one central logging device. This will alleviate human error and the problem of readings being recorded separately becoming uncoordinated and erroneous. This will also save considerable data input time

2). With the cooperation of the plant operators controlled tests on the chiller to determine the surge conditions would be beneficial. With these data surge control can be built into the chiller model.

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APPENDIX A

TRNSYS SUBROUTINE DOCUMENTATION

— Centrifugal Chiller Component

— Steam Turbine Component

— Fan Controller Component

— Individual Fan Setting Routine

```
*****************************************************************
*  BRAUN CHILLER MODEL WITH THE ADDITION OF RPM AND ADJUSTABLE  *
*  VANES. THIS ROUTINE DETERMINES THE VANE SETTING NECESSARY TO *
*  MAINTAIN THE ACCEPTABLE RPM AND CALCULATES THE CHILLER POWER *
*  AND RPM AT THAT VANE SETTING. IT ALSO INCLUDES THE FREE      *
*  COOLING OPTION. WHEN THE CONTROLLER SIGNALS FOR FREE COOLING *
*  THE COMPRESSOR IS TURNED OFF AND THE PWR & RPM ARE SET TO    *
*  ZERO.                                                        *
*****************************************************************
      SUBROUTINE  TYPE49(TIME,XIN,OUT,T,DTDT,PAR,INFO)
      DIMENSION PAR(43),XIN(7),OUT(10),INFO(10)
      DATA IMAX/50/,TOL/0.0001/,CPW/1./,NSTK/4./

      INFO(6)=10
      EXEC1 = 0.
      EXEC2 = 0.
      STICK = 0.

***  FIRST ITERATION IN TIME STEP SET OSCILATION COUNTER TO 0 **

      IF(INFO(7) .EQ. 0) OUT(10) = 0.0
         VLAST = OUT(9)
         IOSC =  INT(OUT(10) + 0.1)

****  CAPACITY AND SURGE COEFFICIENTS   ****
        B0= PAR(12)
        B1= PAR(13)
        C0= PAR(14)
        C1= PAR(15)
        C2= PAR(16)

****  RPM LIMITS, 100% & 50% VANE  ****
      VLIM100 = PAR(42)
      VLIM50  = PAR(43)

****  100% VANE PARAMETERS     *****
100     VANE =100.
        A0= PAR(7)
        A1= PAR(8)
        A2= PAR(9)
        A3= PAR(10)
        A4= PAR(11)

        D0= PAR(17)
        D1= PAR(18)
        D2= PAR(19)
        D3= PAR(20)
        D4= PAR(21)
```

```
**  CALL THE CHILLER SUBROUTINE TO CALCULATE  100% VANE OPERATION
      CALL CHILL(PAR,XIN,TIME,A0,A1,A2,A3,A4,B0,B1,C0,C1,C2,
     .           D0,D1,D2,D3,D4,TCHWS,QEVAP,POWER,SPEED,TCWR,VANE)

      IF(STICK .EQ. 1) GO TO 200

**  CHECK RPM TO ADJUST THE VANES, IF RPM TOO LOW SET VANES TO 50%
      IF(SPEED .LT. VLIM100  .AND. EXEC1 .NE. 1) THEN
50        VANE = 50.
          A0=PAR(22)
          A1=PAR(23)
          A2=PAR(24)
          A3=PAR(25)
          A4=PAR(26)
          D0=PAR(27)
          D1=PAR(28)
          D2=PAR(29)
          D3=PAR(30)
          D4=PAR(31)
      CALL CHILL(PAR,XIN,TIME,A0,A1,A2,A3,A4,B0,B1,C0,C1,C2,
     .           D0,D1,D2,D3,D4,TCHWS,QEVAP,POWER,SPEED,TCWR,VANE)
          IF(STICK .EQ. 1) GO TO 200
      ENDIF

***  CHECK RPM AGAIN TO ADJUST VANES,
***  IF RPM STILL TOO LOW SET VANES TO 20%   ***
      IF(SPEED .LT. VLIM50  .AND.  EXEC2 .NE. 1) THEN
20        VANE = 20.
          A0=PAR(32)
          A1=PAR(33)
          A2=PAR(34)
          A3=PAR(35)
          A4=PAR(36)
          D0=PAR(37)
          D1=PAR(38)
          D2=PAR(39)
          D3=PAR(40)
          D4=PAR(41)
      CALL CHILL(PAR,XIN,TIME,A0,A1,A2,A3,A4,B0,B1,C0,C1,C2,
     .           D0,D1,D2,D3,D4,TCHWS,QEVAP,POWER,SPEED,TCWR,VANE)
          IF(STICK .EQ. 1) GO TO 200
      ENDIF

*  CHECK OSCILATIONS. IF VANE SETTING HAS CHANGED SINCE LAST ITERATION
*  ADD TO COUNT, IF COUNT IS MORE THAN NSTK USE GREATER VANE SETTING AN
D
*  CALL CHILLR SUBROUTINE AGAIN THEN TO OUTPUTS.

      IF(ABS(VLAST-VANE) .LT. 1.E-06  .AND.  IOSC .NE. NSTK) GO TO 5
```

```
         OUT(10) = OUT(10) + 1.0
5     IF(IOSC .GE. NSTK) THEN
         VANE = AMIN1(VANE,VLAST)
         IF(VANE .EQ. 100) THEN
            EXEC1 = 1
            EXEC2 = 1
            STICK = 1
            GO TO 100
         ENDIF
         IF(VANE .EQ. 50) THEN
            EXEC2 = 1
            STICK = 1
            GO TO 50
         ENDIF
         IF(VANE .EQ. 20) THEN
            STICK = 1
            GO TO 20
         ENDIF
      ENDIF

****   OUTPUTS   *********

200   OUT(1)=TCHWS
      OUT(2)=XIN(3)
      OUT(3)=TCWR
      OUT(4)=XIN(5)
      OUT(5)=QEVAP
      OUT(6)=POWER
      OUT(7)=SPEED
      OUT(8)=TCWR-TCHWS
      OUT(9)=VANE
      END

*****************************************************************
*                 CHILLER SUBROUTINE                            *
*****************************************************************
      SUBROUTINE CHILL(PAR,XIN,TIME,A0,A1,A2,A3,A4,B0,B1,C0,C1,C2,
     .           D0,D1,D2,D3,D4,TCHWS,QEVAP,POWER,SPEED,TCWR,VANE)
      DIMENSION PAR(41),XIN(7),OUT(10),INFO(10)
      DATA IMAX/50/,TOL/0.0001/,CPW/1./,NSTK/4./

      QDES=PAR(1)
      PDES=PAR(2)/3.515
      DTDES=PAR(3)
      RPMDES=PAR(4)
      EFFMOT=PAR(5)
      PDES=PDES*EFFMOT
      COPDES=QDES/PDES
      GLOSS=PAR(6)/3.515
```

```
      TCHWS=XIN(1)
      TCHWR=XIN(2)
      FLCHW=8.3333*60.*XIN(3)/12000.*CPW
      TCWS=XIN(4)
      FLCW=8.3333*60.*XIN(5)/12000.*CPW
      NCH=XIN(6)
      ONOFF = XIN(7)


      IF(FLCW.GT.0. .AND. FLCHW.GT.0.) THEN

*** CHECK STATUS OF CHILLER ON/OFF **************

      IF(ONOFF .LT. 0.7)THEN
        QEVAP = FLCHW*(TCHWR-TCHWS)
        POWER = 0
        SPEED = 0
        TCWR = TCWS + QEVAP/FLCW
        VANE = 0
        GOTO 200
      ENDIF

* DETERMINE MAXIMUM ALLOWABLE TEMPERATURE RISE BETWEEN LEAVING
* CONDENSER AND CHILLED WATER TEMPERATURES IN ORDER TO AVOID
* SURGE AND MAXIMUM CAPACITY.
        A=C2
        B=C1-B1
        C=C0-B0
        YMAX=QUAD(A,B,C,1.,10.)

*** DETERMINE TEMPERATURE RISE ASSUMING NORMAL OPERATION

        X=FLCHW*(TCHWR-TCHWS)/QDES
        Y=(FLCW*(TCWS-TCHWS)+X*QDES-GLOSS+PDES*(A0+A1*X+A2*X*X))/
     .     (FLCW*DTDES-PDES*(A3+A4*X))
        IF(Y.LT.YMAX) THEN

          XCAP=B0+B1*Y
          IF(X.GT.XCAP) THEN
           CAP = 1

**** LOAD EXCEEDS CHILLER CAPACITY ******

            A=A2*B1*B1+A4*B1
            B=A1*B1+2.*A2*B0*B1+A3+A4*B0+B1*COPDES*(1.+FLCW/FLCHW)
     .        -FLCW*DTDES/PDES
            C=A0+A1*B0+A2*B0*B0+(FLCW*(TCWS-TCHWR)-GLOSS)/PDES+
```

```
     .        B0*COPDES*(1.+FLCW/FLCHW)
            Y=QUAD(A,B,C,0.,YMAX)
            X=B0+B1*Y
          ELSE
            XSURGE=C0+C1*Y+C2*Y*Y
            IF(X.LT.XSURGE) THEN
         SURGE = 1

***  LOAD LESS THAN MINIMUM SAFE VALUE  ****

              ITER=0
10            ITER=ITER+1
                X=C0+C1*Y+C2*Y*Y
                DXDY=C1+2.*C2*Y
                F=PDES*(A0+A1*X+A2*X*X+A3*Y+A4*X*Y)+
     .            (1.+FLCW/FLCHW)*X*QDES-GLOSS+FLCW*(TCWS-TCHWR-Y*DTDES
)
                DFDY=PDES*((A1+2.*A2*X)*DXDY+A3+A4*(X+Y*DXDY))+
     .                (1.+FLCW/FLCHW)*QDES*DXDY-FLCW*DTDES
                YLAST=Y
                Y=Y-F/DFDY
              IF(ABS(Y-YLAST).GT.TOL .AND. ABS(F).GT.TOL  .AND.
     .                   ITER.LT.IMAX) GO TO 10
              IF(ITER.EQ.IMAX) THEN
                WRITE(*,101) TIME,X,Y,F
101             FORMAT(//2X,'LACK OF CONVERGENCE IN CHILLER MODEL'/
     .            'TIME, X, Y, F = ',4(1X,1PE11.3))
              ENDIF
            ENDIF
          ENDIF

****  BOTH LIMITS EXCEEDED   *****

        ELSE
          Y=YMAX
          X=B0+B1*Y
        ENDIF

****  DETERMINE POWER AND LEAVING CONDENSER WATER TEMPERATURE **

        TCHWS=TCHWR-X*QDES/FLCHW
        QEVAP=X*QDES
        PLF=A0+A1*X+A2*X*X+A3*Y+A4*X*Y
        RPM=D0+D1*X+D2*X*X+D3*Y+D4*X*Y
        POWER=3.515*PDES*PLF/EFFMOT
        SPEED=RPM*RPMDES
        TCWR=TCHWS+Y*DTDES

      ELSE
```

```
****  NO WATER FLOW  *****

        QEVAP=0.
        TCHWS=TCHWR
        TCWR=TCWS
        POWER=0.

      ENDIF

200   CONTINUE

      END

      REAL FUNCTION QUAD(A,B,C,XMIN,XMAX)

      IF(A .EQ. 0) GO TO 106
      ROOT=SQRT(B*B-4.*A*C)
      X=(-B-ROOT)/2./A
      IF(X.LT.XMIN) X=(-B+ROOT)/2./A
      QUAD=AMIN1(XMAX,AMAX1(XMIN,X))
106   IF(B .EQ. 0) THEN
        QUAD = -C
      ELSE
         QUAD = -C/B
      ENDIF
      RETURN
      END
```

```
*******************************************************************
*                    TURBINE MODEL                                *
*  THIS ROUTINE CALCULATES THE STEAM FLOW AND TURBINE EFFIC-     *
*  IENCY DIRECTLY FROM THE CURVE FITS TO THE MANUFACTURERS        *
*  DATA. IT INCLUDES THE FREE COOLING OPTION, WHEN THE CONTROL-   *
*  LER SIGNALS FOR FREE COOLING THE TURBINE TURNS OFF AND THE     *
*  STEAM FLOW IS SET TO ZERO.                                     *
*******************************************************************
      SUBROUTINE TYPE 50 (TIME,XIN,OUT,T,DTDT,PAR,INFO)
      DIMENSION XIN(3),OUT(4),PAR(14),INFO(10)

**** STEAM FLOW COEFFICIENTS  ******
      A0=PAR(1)
      A1=PAR(2)
      A2=PAR(3)
      A3=PAR(4)
      A4=PAR(5)

**** EFFICIENCY  COEFFICIENTS  ******
      B0=PAR(6)
      B1=PAR(7)
      B2=PAR(8)
      B3=PAR(9)
      B4=PAR(10)

**** SUPPLY STEAM ENTHALPY & IDEAL ENTHALPY DROP  ******
      H1=PAR(11)
      DHIDEAL=PAR(12)

**** EXHAUST PRESSURE   *******
      P2=PAR(13)

**** TRANSMISSION LOSSES, TLOSS  ******
      TLOSS = PAR(14)

**** INPUTS    ********
      PWR=XIN(1)
      RPM=XIN(2)
      ONOFF = XIN(3)

      INFO(6)=4

**** CHECK STATUS OF TURBINE CONTROL *****
      IF(ONOFF .LT. 0.5)THEN
         STM = 0
         T2 =0
         WACT = 0
         GO TO 50
      ENDIF
```

```
**** CURVE FIT TO TURBINE PERFORMANCE CURVE  *****
      PWR = (PWR/100)*(1+TLOSS)
      RPM = RPM/100
      STM = A0 + A1*RPM + A2*RPM*RPM + A3*PWR + A4*PWR*RPM
      EFF = B0 + B1*RPM + B2*RPM*RPM + B3*STM + B4*STM*RPM

      STM = STM*1000
      EFF = EFF/100

**** WORK TERMS (BTU/HR)  ******
      WIDEAL = STM * DHIDEAL
      WACT = EFF * WIDEAL

**** EXIT CONDITIONS  *****
      H2 = H1-WACT
      CALL STEAM('US',T2,P2,H2,S2,X2,V2,U2,23)

**** OUTPUTS  ********
50    OUT(1) = STM
      OUT(2) = T2
      OUT(3) = WACT

      RETURN
      END
```

```
*****************************************************************
*                     CONTROL ROUTINE                            *
*   THIS ROUTINE DETERMINES THE PROPER FAN SETTING FOR THE       *
*   GIVEN LOAD AND TWB. THIS COMPONENT ALSO MAKES THE DECISIION *
*   AS TO WHICH MODE TO OPERATE, FREE COOLING OR MECHANICAL      *
*   COOLING. THE OPERATION MAPS MUST BE MADE ACCESSBLE TO THIS  *
*   ROUTINE. ADDITIONALLY, THE SUBROUTINE WHICH DESIGNATES THE  *
*   INDIVIDUAL FAN SPEED SETTINGS FOR THE ASSOCIATED COMBINA-   *
*   TION NUMBER MUST BE LINKED TO THIS PROGRAM. THE PARAMETERS  *
*   ARE THE MAXIMUM LOAD AND WET BULB  UNDER WHICH FREE COOLING *
*   CAN BE ACHIEVED.                                             *
*****************************************************************
      SUBROUTINE TYPE22(TIME,XIN,OUT,T,DTDT,PAR,INFO)
      DIMENSION FAN(42,17),OUT(9),XIN(2),INFO(10),PAR(2),FREE(42,17)
      REAL  MECH(42,17)
      LOGICAL LOADFR,LOADCH
      DATA LOADFR/.TRUE./ LOADCH/.TRUE./

      Q=XIN(1)
      TWB= XIN(2)
      QFRMAX= PAR(1)
      TWBFRMAX=PAR(2)
      INFO(6)=9

**** CHECK IF IN FREECOOLING RANGE   ****

      IF(Q .LE. QFRMAX  .AND.  TWB .LT. TWBFRMAX) THEN
          ONOFF = 0

**** READ IN FREECOOLING MATRIX  ********

      IF(LOADFR) THEN
         DO 100 N=1,42
           READ(20,*)FREE(N,1),FREE(N,2),FREE(N,3),FREE(N,4),FREE(N,5),
     .           FREE(N,6),FREE(N,7),FREE(N,8),FREE(N,9),FREE(N,10),
     .           FREE(N,11),FREE(N,12),FREE(N,13),FREE(N,14),
     .           FREE(N,15),FREE(N,16),FREE(N,17)
100      CONTINUE
           LOADFR = .FALSE.
      ENDIF

**** LOCATE LOAD IN THE MATRIX  ********

      CALL LOADFIND(Q,FREE,K)

*** LOCATE TWB IN THE MATRIX, IF Q,TWB IN THE FREECOOL RANGE THEN
*** USE FREECOOL, OTHERWISE SEND TO THE MECHANICAL  MODE
```

```
      DO 300 J=2,16
        IF(TWB .GT. FREE(1,J)  .AND.  TWB .LE. FREE(1,J+1)) THEN
          IF(FREE(K,J) .EQ. 0  .AND.  FREE(K,J+1) .EQ. 0) THEN
             ONOFF = 1
             GOTO 350
          ENDIF
          IF((TWB-FREE(1,J)) .GT. 2.5  .AND.  FREE(K,J+1) .EQ. 0) THEN
             ONOFF = 1
             GOTO 350
          ENDIF
          IF(FREE(K,J+1) .EQ. 0) THEN
             FANS = FREE(K,J)
             IFANS = INT(FANS)
             ONOFF=0
             GOTO 500
          ELSE
             CALL INTERP(FREE,TWB,K,J,IFANS)
               ONOFF = 0
               GOTO 500
          ENDIF
        ENDIF
300   CONTINUE
      ELSE

**** NOT IN FREECOOL MODE GO TO CHILLER  *****

        GOTO 350
      ENDIF

*********  CHILLER MODE   ************

350   ONOFF = 1

**** READ IN CHILLER MATRIX *******

      IF(LOADCH) THEN
        DO 380 N= 1,42
           READ(10,*)MECH(N,1),MECH(N,2),MECH(N,3),MECH(N,4),MECH(N,5),
     .            MECH(N,6),MECH(N,7),MECH(N,8),MECH(N,9),MECH(N,10),
     .            MECH(N,11),MECH(N,12),MECH(N,13),MECH(N,14),
     .            MECH(N,15),MECH(N,16),MECH(N,17)
380     CONTINUE
          LOADCH = .FALSE.
      ENDIF

**** LOCATE LOAD IN CHILLER MATRIX  *********

      CALL LOADFIND(Q,MECH,K)
```

```
**** LOCATE TWB IN CHILLER MATRIX  *******

      DO 400 J=2,16
         IF(TWB .GT. MECH(1,J)  .AND.  TWB .LE. MECH(1,J+1)) THEN
            CALL INTERP(MECH,TWB,K,J,IFANS)
               GOTO 500
         ENDIF
400   CONTINUE

**** DETERMINE INDIVIDUAL FAN SETTINGS   *******

500   CALL FANS1TWR(IFANS,FRAC1,FRAC2,FRAC3,FRAC4,FRAC5,FRAC6,TIME)

**** OUTPUTS  ******

      OUT(1) = Q
      OUT(2) = TWB
      OUT(3) = ONOFF
      OUT(4) = FRAC1
      OUT(5) = FRAC2
      OUT(6) = FRAC3
      OUT(7) = FRAC4
      OUT(8) = FRAC5
      OUT(9) = FRAC6

      END

**** SUBROUTINES  *************

      SUBROUTINE LOADFIND(Q,FAN,K)
      DIMENSION FAN(42,17)


      DO 200 I=1,42
        IF(Q .GT. FAN(I,1)  .AND.  Q .LE. FAN(I+1,1)) THEN
           IF(Q .GT. (FAN(I,1)+50))THEN
              K=I+1
              QTEMP = FAN(I+1,1)
           ELSE
              K=I
              QTEMP = FAN(I,1)
           ENDIF
         ENDIF
200   CONTINUE

      RETURN
      END

*
```

```
SUBROUTINE INTERP(FAN,TWB,K,J,IFANS)
INTEGER IFANS
DIMENSION FAN(42,17)

DIF1 = FAN(1,J+1) - FAN(1,J)
DIF2 = TWB - FAN(1,J)
WT = DIF2/DIF1
FDIF = FAN(K,J+1) - FAN(K,J)
FANS = (WT*FDIF)+FAN(K,J)
IFANS = INT(FANS)

IF((FANS - IFANS) .GE. 0.5) THEN
   IFANS = IFANS +1
ENDIF

RETURN
END
```

```
*****************************************************************
*                  FAN SETTING ROUTINE                          *
*    THIS SUBROUTINE SET UP TO CONVERT THE FAN SET COMBINATIO   *
*    NUMBER TO THE ACTUAL FAN SETTINGS, FRAC1 ......FRAC6.      *
*    RETURNS THOSE FRACTIONS, 0.5=HALF SPEED, 1=FULL SPEED.     *
*****************************************************************
      SUBROUTINE FANS1TWR(IFANS,FRAC1,FRAC2,FRAC3,FRAC4,FRAC5,
     .           FRAC6,TIME)
      INTEGER IFANS
      GOTO(10,20,30,40,50,60,70,80,90,100,110,120) IFANS
        WRITE(*,*) 'FANSET IS NOT AN INTERGER BETWEEN 1 AND 12,
     .              FANSET = ',IFANS
        GO TO 130

**** CASE #1

10    FRAC1 = 0.5
      FRAC2 = 0.
      FRAC3 = 0.
      FRAC4 = 0.
      FRAC5 = 0.
      FRAC6 =0.
      GO TO 130

**** CASE #2

20    FRAC1 = 0.5
      FRAC2 = 0.
      FRAC3 = 0.5
      FRAC4 = 0.
      FRAC5 = 0.
      FRAC6 = 0.
      GO TO 130

**** CASE #3

30    FRAC1 = 0.5
      FRAC2 = 0.
      FRAC3 = 0.5
      FRAC4 = 0.5
      FRAC5 = 0.
      FRAC6 = 0.
      GO TO 130

**** CASE #4

40    FRAC1 = 0.5
      FRAC2 = 0.
      FRAC3 = 0.5
```

```
      FRAC4 = 0.
      FRAC5 = 1.0
      FRAC6 = 0.
      GO TO 130

**** CASAE #5

50    FRAC1 = 0.5
      FRAC2 = 0.5
      FRAC3 = 0.5
      FRAC4 = 0.
      FRAC5 = 1.0
      FRAC6 = 0.
      GO TO 130

**** CASE #6

60    FRAC1 = 0.5
      FRAC2 = 0.5
      FRAC3 = 0.5
      FRAC4 = 0.5
      FRAC5 = 1.0
      FRAC6 = 0.
      GO TO 130

**** CASE #7

70    FRAC1 = 1.0
      FRAC2 = 0.5
      FRAC3 = 0.5
      FRAC4 = 0.5
      FRAC5 = 1.0
      FRAC6 = 0.
      GO TO 130

**** CASE #8

80    FRAC1 = 0.5
      FRAC2 = 0.5
      FRAC3 = 0.5
      FRAC4 = 0.5
      FRAC5 = 1.0
      FRAC6 = 1.0
      GO TO 130

**** CASE #9

90    FRAC1 = 0.5
      FRAC2 = 0.5
```

```
      FRAC3 = 0.5
      FRAC4 = 1.0
      FRAC5 = 1.0
      FRAC6 = 1.0
      GO TO 130

**** CASE #10

100     FRAC1 = 0.5
      FRAC2 = 0.5
      FRAC3 = 1.0
      FRAC4 = 1.0
      FRAC5 = 1.0
      FRAC6 = 1.0
      GO TO 130

**** CASE #11

110     FRAC1 = 0.5
      FRAC2 = 1.0
      FRAC3 = 1.0
      FRAC4 = 1.0
      FRAC5 = 1.0
      FRAC6 = 1.0
      GO TO 130

**** CASE #12

120     FRAC1 = 1.0
      FRAC2 = 1.0
      FRAC3 = 1.0
      FRAC4 = 1.0
      FRAC5 = 1.0
      FRAC6 = 1.0
      GO TO 130

130   CONTINUE

      RETURN
      END
```

APPENDIX B

TRNSYS SIMULATION DECK

Walnut Street Chiller Plant Simulation Deck

```
*****************************************************************
*                 SAMPLE SIMULATION DECK                        *
*   THIS DECK SIMULATES THE TOTAL PLANT BEHAVIOR. IT PREDICTS   *
*   THE PERFORMANCE OF THE CHILLER AND TURBINE, TOWER, SURFACE  *
*   CONDENSER AND WATER PUMPS. THE SIMULATION IS DRIVEN BY THE  *
*   BY THE CHILLED WATER LOAD AND WET BULB TEMP. THE EQUIPMENT  *
*   IS CONTROLLED BY THE FANCONTROLLER. OUTPUTS ARE THE  FLSTM, *
*   FAN AND PUMP POWER AND ASSOCIATED COSTS.                    *
*****************************************************************
SIMULATION 0  5640  1
LIMITS 50 10 47
TOLERANCES  -0.01  -0.01
WIDTH 132

UNIT 1  TYPE 9   WEATHER DATA READER  LOADMARCHNOV.DAT
PARAMETERS 16
4  1  -1 1 0  -2 1 0  -3 1 0  -4 1 0  30 0

UNIT 2  TYPE 19  TCHWS CALC
PARAMETER 1
1
INPUTS 3
0,0   0,0   1,4
5800   55   300

UNIT 3  TYPE 22  FANCONTROLLER
PAR 2
2800 47.5
INPUTS 2
1,4   1,3
300   25

UNIT 4  TYPE 46   COOLING TOWER
PAR 6
2.557E07  68.2  9.97  6  1.1502  -0.9617
INPUTS 10
9,1   0,0  1,2  1,3    3,4  3,5  3,6  3,7  3,8  3,9
40   11180  30   25     0    0    0    0    0    0

UNIT 5  TYPE 49  CHILLER
PAR 43
3815  1973.51  42.3  5300  1 0 -0.1708059  0.283802
-0.106513  0.00995258  0.989071  1.0  0.0  0.0  0.2  0.08
0.26911  0.6319153  -0.1554947  0.5720074  -0.3123009
0.10799  -1.15204  2.12516  0.342745  -0.032505
0.404795  -0.39464  1.85781  1.082052  -1.60985
0.22635  -0.92188  1.01201  -0.117368  1.018014
0.60569  0.47178  -1.1159  0.027966  0.8537876
4600  4100
```

```
INPUTS 7
2,2  2,3  2,1  4,1  4,2   0,0  3,3
40   55   5800  38  11500  1    0

UNIT 6  TYPE 15   COND. FLOW CONVERTER(GPM-LBS/HR)
PAR 1
1
INPUTS 2
5,4      0,0
11500  499.98

UNIT 7  TYPE 50  STEAM TURBINE
PAR 14
14.6212  -0.5674  0.0071375  1.9123  -0.0102029
0.03648  1.51787  -0.02246  1.54563   0.026542
1197.68  322.9  1.44  0.05
INPUTS 3
5,6  5,7    3,3
300  3000    0

UNIT 8  TYPE 5   SURFACE CONDENSER
PAR 4
3  395028.6  0.449  1
INPUTS 4
7,2  7,1    5,3    6,1
114  5000    40    5749770

UNIT 9 TYPE 3  COND. WATER PUMP
PAR 4
5999760  358.68  0.5821  0.4081
INPUTS 3
8,3  8,4      0,0
45  5749770   0..958

UNIT 10  TYPE 15  CHILLED WATER CONVERT. (GPM-LBS/HR)
PARAMETERS 7
0  0  1  -3  0  2  -4
INPUTS 3
5,2   0,0      0,0
5600  499.98   2999880

UNIT 11  TYPE 3  CHILLED WATER PUMP
PARAMETERS 4
2999880  121.55  0.8104  0.20603
INPUTS 3
5,1   10,1    10,2
40  2799888  0.933

UNIT 12  TYPE 28    OUTPUTS
```

```
PAR 25
10 0 8000  70  2  -11 -3  -1 0.00408 1 -3  -12 -13 3 -14 3 -3
-1 0.05 1 -3  3  -4  -15 -4
INPUTS 7
7,1   11,3   4,4  9,3  1,4  1,3  3,3
LABELS 6
FLSTM  FLSTM-$  KW KWSUM-$   TOTALSUM-$  LOAD

END
```

APPENDIX C

- SAMPLE DATA LOGS
- SAMPLE CIRCULAR CHARTS
- TOWER AIR FLOW MEASUREMENT
- DOCUMENTATION
- FAN AND PUMP POWER MEASUREMENTS

WEST CAMPUS HEATING & CHILLING

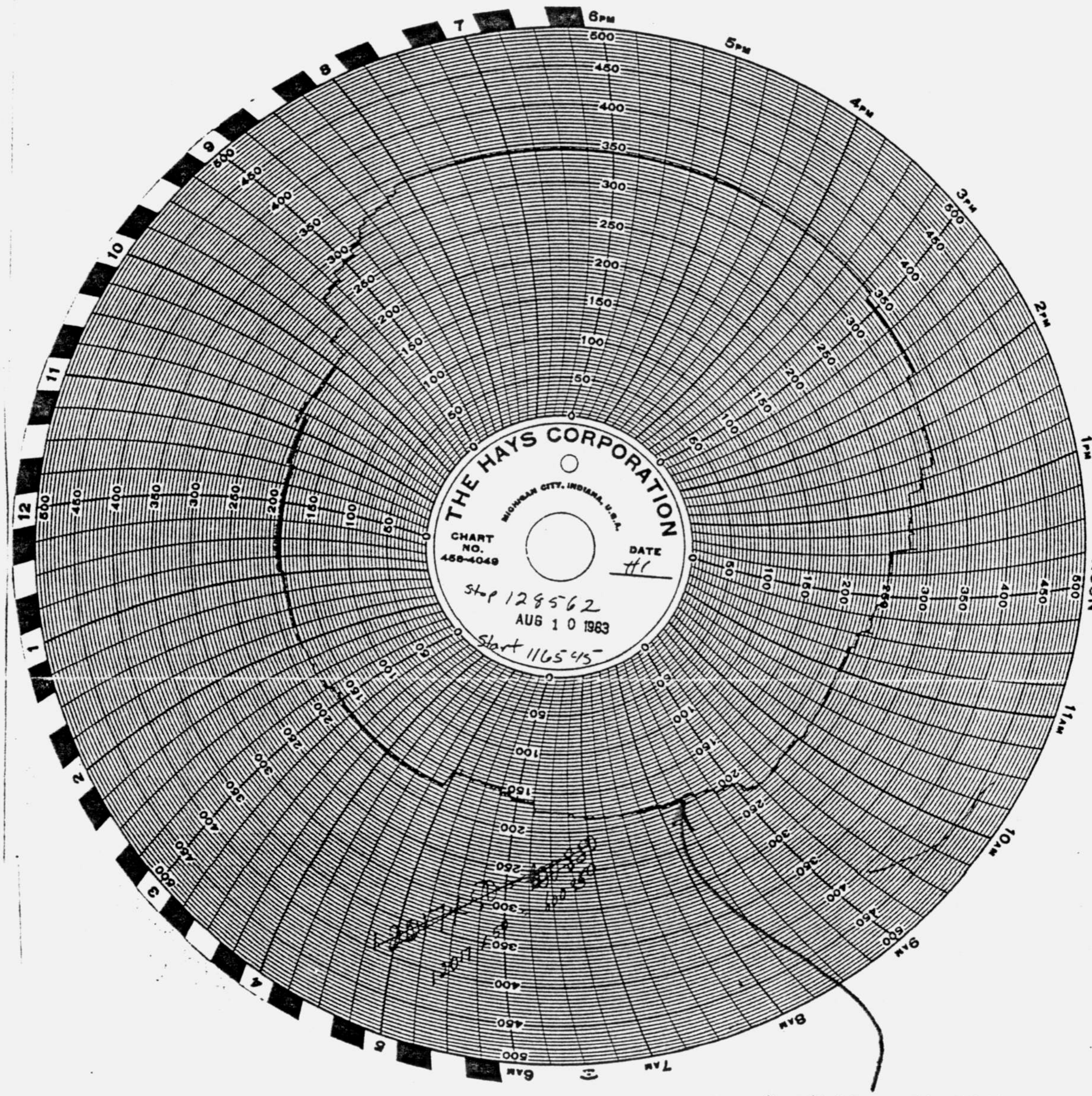
CHILLER NO. II

DATE

	MID	2	4	6	8	10	NOON	2	4	6	8	10
COMPRESSOR												
Cooler Refg Temp	35	33	35	36	35	34	34	34	34	34	34.5	35
Cond Refg Temp	85	81	80	80	84	86	84	84	84	84	76	75
Comp Brg Oil PSI	20.5	20.5	20.5	20.5	20.5	20.5	20.5	20.5	20.5	20.5	20.5	20.5
Cooler Oil Temp	107	102	102	100	106	106	[illegible]	106	105	105	100	96
Comp Brg Temp	125	125	135	135	130	130	[illegible]	130	130	130	130	[illegible]
Brg Oil Temp Inboard	138	134	132	132	136	138	[illegible]	138	[illegible]	137	129	[illegible]
Brg Oil Temp Outboard	136	132	130	130	134	136	[illegible]	136	[illegible]	135	128	[illegible]
Oil Filter PSID	4.5	4.5	4.5	4.5	4.5	4.5	[illegible]	4.5	[illegible]	4.5	4.5	[illegible]
Resv Oil Level	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
Vane Position	100%	100	[illegible]	106	100	100	100	100	100%	100%	100%	100%
Comp Discharge Temp	112	106	104	[illegible]	110	114	110	110	110	110	100	96
TURBINE												
Gland Seal Steam PSI	2	2	2	2	2	2	2	2	2	2	1	1
Oil Cooler Outlet Temp	118	116	[illegible]	119	122	124	122	122	123	123	119	114
Oil Cooler Inlet Temp	120	118	118	120	124	126	126	124	124	124	118	113
Oil Cooler PSID	5	5	5	5	5	6	5	5	5	5	2	2
Outboard Brg Oil Temp	121	118	118	120	124	126	126	126	125	125	119	115
Inboard Brg Oil Temp	119	116	[illegible]	118	120	122	122	122	121	121	117	114
Oil Resv Level	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
Turbine Exhaust	27	27.5	27.5	27.5	27	27	27	27	27	27	27	27
Brg Oil PSI	15.6	15.5	15.4	15.2	15.6	15.6	15.6	15.6	15.5	15.5	15.2	15.2
Relay Oil PSI	68	67	67	67	68	68	68	69	68	68	66	65
Turbine Speed	5200	5000	4900	4900	5200	5300	5200	5200	5200	5200	4900	4600
Steam Inlet PSI	170	175	175	178	170	165	170	170	170	168	175	177
Steam Ring PSI	125	120	95	88	140	150	135	140	139	135	85	67
COOLER/CONDENSOR												
Chilled Water PSID	16	16	16	16	16	16	16	16	16	16	16	16
Chilled Water Inlet Temp	49	47.5	47	47	50	50	50	50	50	50	47	45
Chilled Water Outlet Temp	38	38	38	38	37	37	37	37	37.5	37.5	37.5	37
Tower Water PSID	12	12	12	12	12	12	12	12	12	12	12	12
Tower Water Inlet Temp	76	73	72	73	73	75	74	74	73	74	69	68
Tower Water Outlet Temp	84	80	79	79	82	84	82	82	82	82	74	73
TOWER PUMP												
Strainer Inlet PSI	18	18	18	18	18	18	19	18	19	19	19	19
Pump Suction PSI	16	16	16	16	15	15	15	15	15	15	15	15
Pump Discharge PSI	74	75	75	75	74	75	75	75	74	74	74	74
CHILLED WATER PUMP												
Strainer Inlet PSI	95	83	81	96	90	90	87	83	99	97	91	90
Pump Suction PSI	84	82	80	89	89	89	86	82	98	95	90	89
Pump Discharge PSI	130	126	126	134	134	134	132	127	193	141	135	134
SURFACE CONDENSOR												
Inlet Temp	80	78	77	76	80	82	80	80	80	81	73	72
Discharge Temp	82	80	79	78	82	84	82	82	92	83	75	73
Hotwell Temp	98	[illegible]	[illegible]	93	96	100	99	99	99	97	90	89
Tower Water PSID	6	6	6	6	6	6	6	6	6	6	6	6
Pump Discharge PSI	32	28	27	27	31	27	33	33	27	27	26	26
TOWER WATER												
TDS Actual	2000	1950	2000	1900	2100	2000	2100	2100	2100	2100	2000	2000
Tower Pac	2050	2000	2025	2000	2000	2200	2100	2100	2050	2050	2000	2050
PH Actual	8.6	8.6	8.6	8.5	8.5	8.5	8.5	8.5	8.5	8.5	8.5	8.5
Tower Pac	8.0	8.0	8.0	7.9	8.0	8.0	7.9	7.8	7.9	7.9	8.0	7.9
PH adj Up Or Down									↑	↑		
"M" Alkalinity	348	348	340	360	360	364	384	380	[illegible]	408	380	364
	1 3	1 3	1 3	1-3	1-3	1-3	1-3	1-3	1 3	1 3	1 3	1 3
Odd # Fan Speed Hi-Lo	HH	HH	LH	H-H	HH	HH	HH	HH	HH	HH	HH	LL
	2 4	2 4	2 4	2-4	2-4	2 4	2-4	2-4	2 4	2 4	2 4	2 4
Even # Fan Speed Hi-Lo	HH	LL	HH	LL	LH	HH	LL	HH	HH	HH	HH	LL
Security Check In	✓	✓	✓	✓						✓	✓	✓
D	76	75	[illegible]	74	74	80	82	80	83	81	65	62
W	72	70	65	70	69	64	64	68	70	70	75	73
RH	83	77	83	83	74	58	52	54	53	59	59	38

Operational Notes

Sample of Daily Log Chart

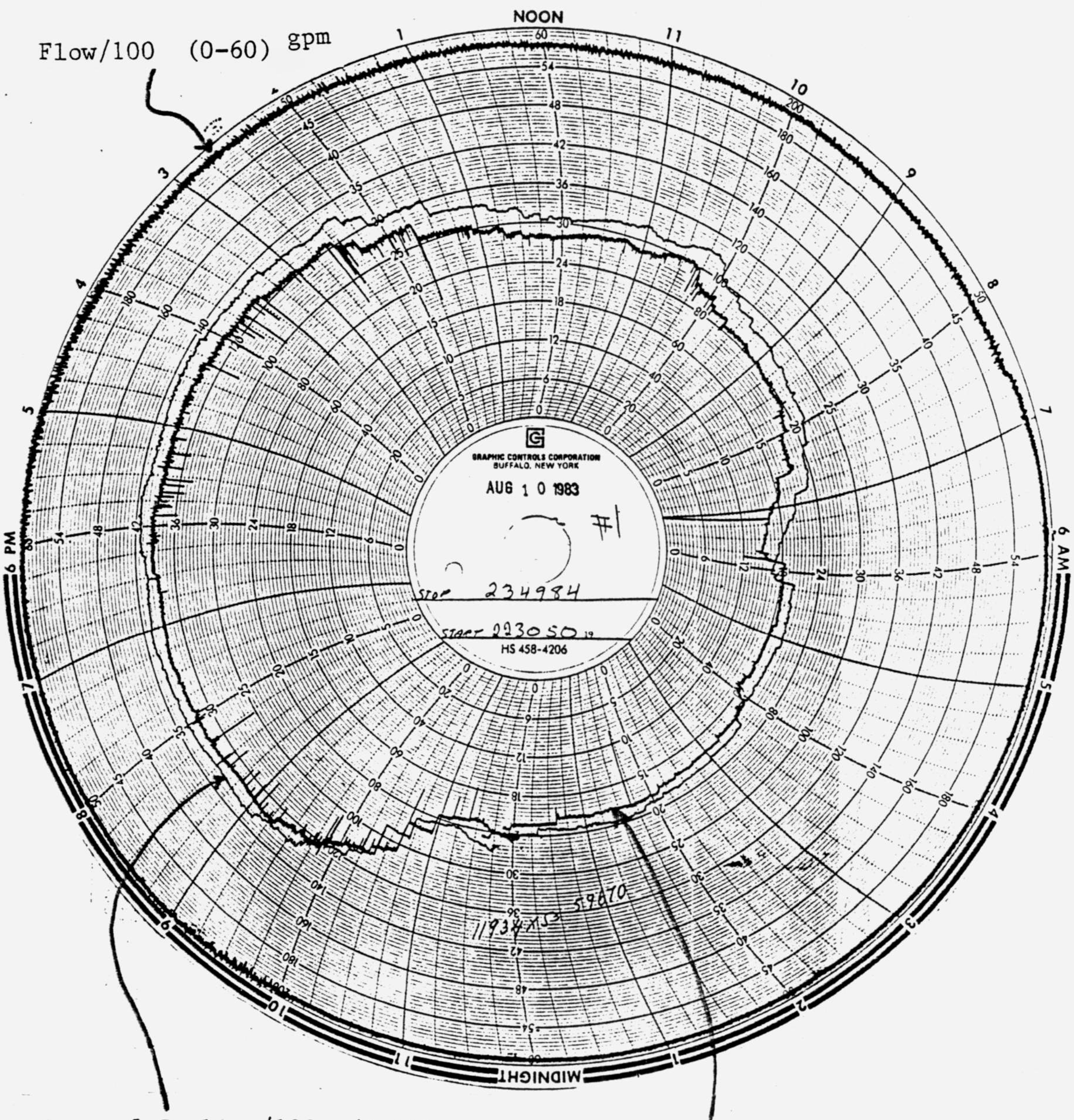


Lbsm/hr/100 (0-500)

Steam Flow Circular Chart

Chilled Water Circular Chart

Flow/100 (0-60) gpm



Tons of Cooling/100 (0-50)

Temp. Drop Across Evaporator/0.1 (0-200) F

FAN CONDITIONS: 1-H, 2-H, 3-L, 4-L, 5&6-"ON"

#3 TOWER FAN

(LOW SPEED)

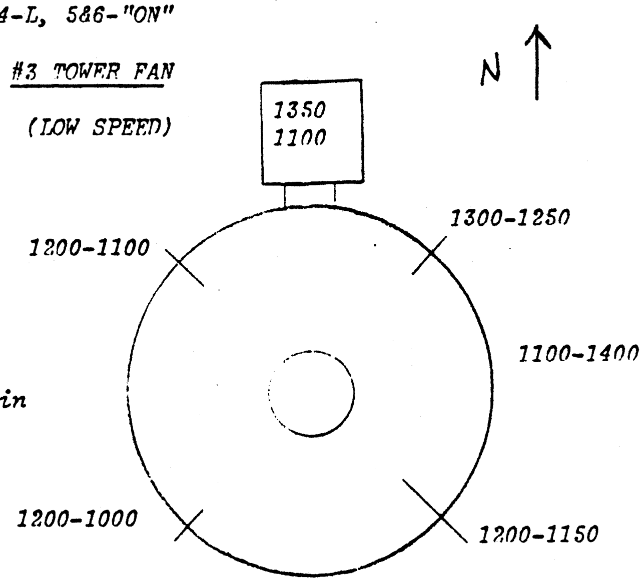
INNER TOTAL = 7900

OUTER TOTAL = 8650

=16550

AVERAGE =1182.14

VOLUME OF AIR = 268322.14 cft/min



FAN CONDITIONS: 1-H, 2-H, 3-L, 4-L, 5&6 - "ON"

#4 TOWER FAN

(LOW SPEED)

INNER TOTAL = 8150

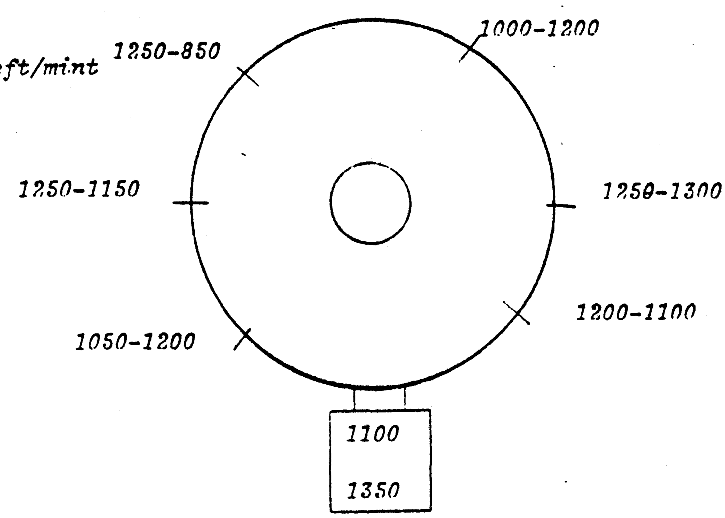
INNER TOTAL = 8150

OUTER TOTAL = 7900

=16050

AVERAGE = 1146.43

VOLUME OF AIR = 260216.68 cft/mint



FAN CONDITIONS: 1-H, 2-H, 3-L, 4-L, 5&6 - "ON"

#5 TOWER FAN

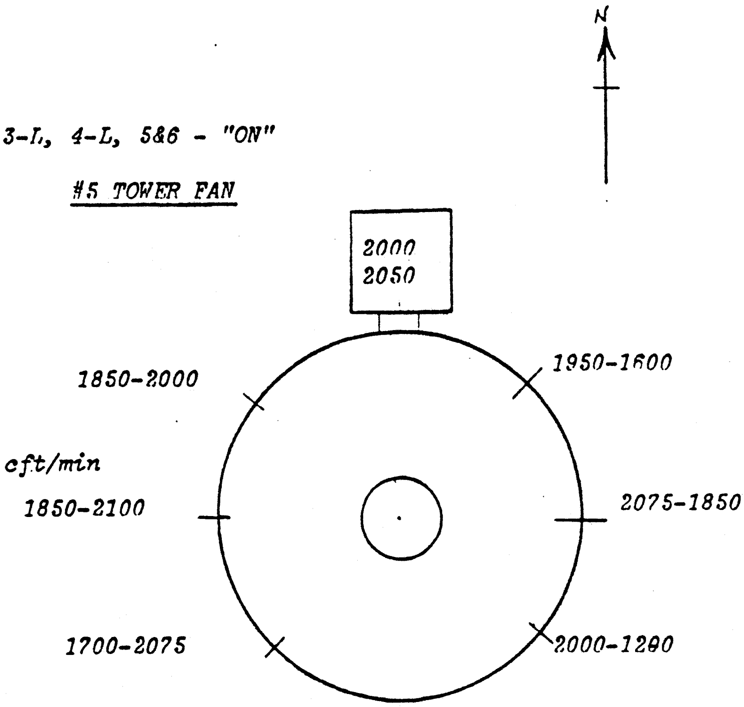
INNER TOTAL = 16400

OUTER TOTAL = 13800

= 30200

AVERAGE = 1887.5

VOLUME OF AIR = 428424.75 cft/min



FAN CONDITIONS: 1-H, 2-H, 3-L, 4-L, 5&6 - "ON"

#6 TOWER FAN

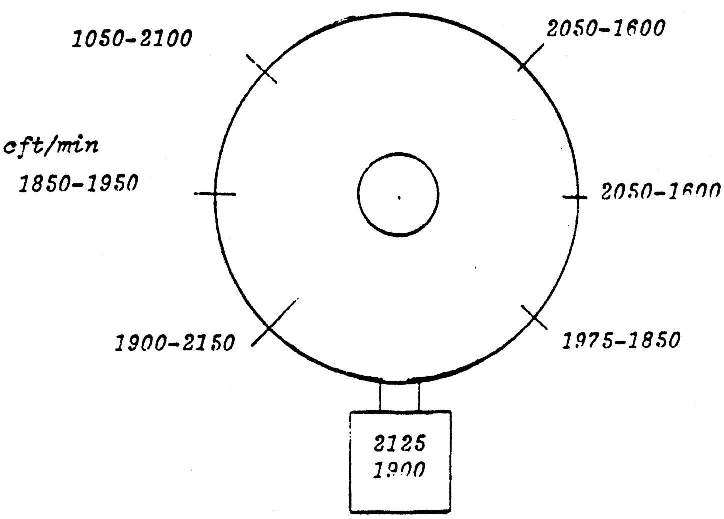
INNER TOTAL = 14400

OUTER TOTAL = 11750

= 26150

AVERAGE = 1867.85

VOLUME OF AIR = 423964.59 cft/min



WALNUT STREET HEATING PLANT

CALCULATION FOR MOTOR H.P & POWER FACTOR

H.P = VOLTS X AMPS X P.F X 1.73 / 746

P.F = H.P X 746 / VOLTS X AMPS X 1.73

#1 TOWER PUMP MOTOR DESIN:

VOLTS = 460

AMPS = 580

H.P = 500

P.F = 500 X 746 / 460 X 580 X 1.73 = .81 Or 81%

TOWER PUMP MOTOR ACTUAL:

VOLTS = 480

AMPS = 567

PF = .81

H.P = 480 X 567 X .81 X 1.73 / 746 = 511

SEPT 10, 1985 AMP readings = 572, 570, 561 & averages to 567

VOLTS = 480

#1 CHILLED WATER PUMP DESIGN:

VOLTS = 460

AMPS = 234

H.P = 200

P.F = .80

SEPT 10 1985 AMP readings = 203, 198, 202 & the average is 201 AMPS

VOLT = 480

#1 CHILLED WATER PUMP ACTUAL:

H.P = 480 X 201 X .80 X 1.73 / 746 = 179

#1 TOWER FAN MOTOR ON HIGH SPEED-DESIGN:

VOLTS = 460

AMPS = 119

HP = 100

P.F = .79

FAN MOTOR ON HIGH SPEED ACTUAL:

H.P = 480 X 104 X .79 X 1.73 / 746 = 91.5

SEPT 10, 1985 AMP readings = 104, 104, 104 & the average is 104 AMPS.
VOLT = 480

 # 1 TOWER FAN MOTOR ON LOW SPEED-DESIGN:

VOLTS = 460

AMPS = 45

H.P = 25

P.F = .52

AMP readings on SEPT 10, 1985 = 36,36, 36 & the average is 36 AMPS

VOLT = 480

#1 TOWER FAN MOTOR ACTUAL:

H.P = 480 X 36 X .52 X 1.73 / 746 = 20.8

#2 TOWER FAN MOTOR ON HIGH SPEED-DESIGN:

VOLTS = 460

AMPS = 119

H.P = 100

P.F = .79

SEPT 10, 1985 AMP readings = 108,106, 106 & the average is 106.6 AMPS

#2 TOWER FAN MOTOR ACTUAL:

H.P = 480 X 106.6 X .79 X 1.73 / 746 = 93.7

#2 TOWER FAN MOTOR ON LOW SPEED-DESIGN:

VOLTS = 460

AMPS = 45

H.P = 25

P.F = .52

SEPT 10, 1985 AMP readings = 36, 36, 36 & the average is 36 AMPS.

VOLT = 480

#2 TOWER FAN MOTOR ACTUAL:

H.P = 480 X 36 X .52 X 1.73 / 746 = 20.8

#3 TOWER FAN MOTOR ON HIGH SPEED-DESIGN

VOLTS = 460

AMPS = 119

H.P = 100

P.F = .79

SEPT 10, 1985 AMP readings = 105,104,104 & the average is 104.3 AMPS

#3 TOWER FAN MOTOR ACTUAL:

H.P = 480 X 104.3 X .79 X1.73 ÷ 746 = 91.7

#3 TOWER FAN MOTOR LOW SPEED-DESIGN:

SEPT 10, 1985 AMP readings = 37.1, 36.5, 36.2 & the average is 36.6 AMPS.

#3 TOWER FAN MOTOR ACTUAL:

H.P = 480 X 36.6 X .52 X 1.73 ÷ 746 = 21.2

#4 TOWER FAN MOTOR ON HIGH SPEED-DESIGN:

VOLTS = 460

AMPS = 119

H.P = 100

P.F = .79

SEPT 10, 1985 AMP readings = 108, 108, 108 & the average is 108 AMPS

<u>#4 TOWER FAN MOTOR ACTUAL:</u>

H.P = 480 X 108 X .79 X 1.73 ÷ 746 = 95

<u>#5 & 6 TOWER FAN MOTOR DESIGN;</u>

VOLTS = 460

AMPS = 120

H.P = 100

P.F = .79

SEPT 10, 1985 AMP readings = 68, 68, 68 & the average is 68 AMPS.

<u>#5 & 6 TOWER FAN MOTOR ACTUAL:</u>

H.P = 480 X 68 X .79 X 1.73 ÷ 746 = 59.8

Using 25 H.P power factor of .52

H.P = 480 X 68 X .52 X 1.79 ÷ 746 = 39.4

APPENDIX D

INTERACTIVE OPTIMAL CONTROL PROGRAM

— Program Listing

— Sample Session Output

```
*******************************************************************
*   THIS IS AN OVERALL PLANT OPERATION PROGRAM. IT INCLUDES        *
*   OPTIONS FOR INVESTIGATING DIFFERENT FUEL COSTS AND  DIFFER-   *
*   ENT FAN CONTROLS (OTHER THAN THE OPTIMUM FOR THE EXISTING      *
*   FAN MOTORS, WHICH IS PRE-PROGRAMED IN). THE PROGRAM CAN BE     *
*   USED AS A DAY TO DAY PLANT OPERATION DECISION MAKER OR         *
*   GENERAL INVESTIGATIVE TOOL. THE THREE DIMENSIONAL MATRICES     *
*   MUST BE MADE ACCESSABLE TO THIS PROGRAM                        *
*******************************************************************

      REAL ELECT(42,17,12),STEAM(42,17,12),VANE(42,17,12)
      REAL FANS(42,17,12),FAN(12),KWD,STMD
      REAL PRICE(12),HRS(42,17),FANSET(6,12)
      INTEGER PROG,ANS

      OPEN(10,FILE='[NUGENT.MATRIX]ELECT.DAT',STATUS='OLD')
      OPEN(20,FILE='[NUGENT.MATRIX]STEAM.DAT',STATUS='OLD')
      OPEN(30,FILE='[NUGENT.MATRIX]VANE.DAT',STATUS='OLD')
      OPEN(40,FILE='[NUGENT.MATRIX]FAN.DAT',STATUS='OLD')
      OPEN(50,FILE='[NUGENT.MATRIX]HRS.DAT',STATUS='OLD')
      OPEN(60,FILE='[NUGENT.MATRIX]FANSET.DAT',STATUS='OLD')

*****  READ IN MATRICIES  **********

      DO 200 IFS = 1,12
        DO 100 L=1,41
          READ(10,*)(ELECT(L,IWB,IFS),IWB=1,16)
          READ(20,*)(STEAM(L,IWB,IFS),IWB=1,16)
          READ(30,*)(VANE(L,IWB,IFS),IWB=1,16)
          READ(40,*)(FANS(L,IWB,IFS),IWB=1,16)
100     CONTINUE
200   CONTINUE

      DO 300 L=1,41
        READ(50,*) (HRS(L,IWB),IWB=1,16)
300   CONTINUE

      DO 310 J=1,6
        READ(60,*) (FANSET(J,K),K=1,12)
310   CONTINUE

*****     OPERATION PROGRAM OR INVESTIGATION TOOL    *******************
*

5     WRITE(*,*)'DO YOU WISH TO RUN THE OPERATION PROGRAM OR THE
     . INVESTIGATION PROGARM ?'
```

```
10    WRITE(*,*) '1= OPERATION       0= INVESTIGATION'
      READ(6,*) PROG
      IF(PROG .NE. 1  .AND. PROG .NE. 0)THEN
        WRITE(*,*) 'INPUT NOT IN RANGE, PLEASE MAKE SELECTION AGAIN'
         GO TO 10
      ENDIF

      IF(PROG .EQ. 1)GO TO 400
      IF(PROG .EQ. 0)GO TO 600

400   WRITE(*,*)'*****************************************************'
      WRITE(*,*)'*                OPERATION PROGRAM                  *'
      WRITE(*,*)'*  THIS PORTION OF THE PROGRAM IS FOR DAILY USE.    *'

      WRITE(*,*)'*  IT IS DESIGNED TO TAKE INPUTS OF FUEL COSTS,     *'
      WRITE(*,*)'*  CHILLER LOAD, AND WET BULB TEMPERATURE AND       *'
      WRITE(*,*)'*  OUTPUT THE APPROXIMATE ASSOCIATED COSTS OF EACH *'
      WRITE(*,*)'*  FAN SETTING POSSIBILITY. THE OPERATOR MAY THEN  *'
      WRITE(*,*)'*  MAKE THE CONTROL DECISION BASED ON THIS INFOR-  *'
      WRITE(*,*)'*  MATION. THERE IS AN OPTION FOR YEARLY COSTS AND *'
      WRITE(*,*)'*  HOURLY COSTS. FUEL COSTS, LOAD AND WET BULB MAY *'
      WRITE(*,*)'*  BE ALTERED WITHIN THE PROGRAM AND COMPARISONS   *'
      WRITE(*,*)'*  MAY BE MADE.                                     *'
      WRITE(*,*)'*****************************************************'

410   CALL FUELCOST(KWD,STMD)
      CALL QTWB(Q,TWB,L,IWB)
      CALL COST(ELECT,STEAM,KWD,STMD,L,IWB,PRICE,FAN)
      CALL HROUTPUT(FANSET,PRICE,FAN,VANE,L,IWB)

      WRITE(*,*)'DO YOU WISH TO SEE ESTIMATES OF YEARLY COSTS FOR
     . THESE SCENARIOS ?'
415   WRITE(*,*) '1= YES     0=NO'
      READ(6,*)ANS
      IF(ANS .NE. 1  .AND.  ANS .NE. 0) THEN
        WRITE(*,*)'RESPONCE NOT WITHIN RANGE, PLEASE REENTER'
         GO TO 415
      ENDIF
      IF(ANS .EQ. 1) GO TO 450
      IF(ANS .EQ. 0) GO TO 500

**********   YEARLY  COST ESTIMATES FOR EACH FAN SETTING *******

450   CALL  ACCUM(ELECT,STEAM,HRS,L,IWB,KWD,STMD,VANE,FAN)

*********   CHANGE INPUTS  **************************************

500   WRITE(*,*)'DO YOU WISH TO CHANGE INPUTS? (LOAD,Twb,FUEL COSTS)'
515   WRITE(*,*)' 1=YES      0=NO'
```

```
      READ(6,*)ANS
      IF(ANS .NE. 1  .AND.  ANS .NE. 0) THEN
        WRITE(*,*)'RESPONCE NOT WITHIN RANGE, PLEASE REENTER'
         GO TO 515
      ENDIF
      IF(ANS .EQ. 1) GOTO 410
      IF(ANS .EQ. 0) GOTO 520

**********   EXIT OR RE SET FOR NEXT HOUR  ******************

520   WRITE(*,*)' QUIT PROGRAM OR RE-SET FOR NEXT HOUR?'
525   WRITE(*,*)'  1= RE-SET          0= QUIT'
      READ(6,*)ANS
      IF(ANS .NE. 1  .AND.  ANS .NE. 0) THEN
        WRITE(*,*)'RESPONCE NOT WITHIN RANGE, PLEASE REENTER'
         GO TO 525
      ENDIF

      IF(ANS .EQ. 1)GOTO 410
      IF(ANS .EQ. 0)GOTO 550

550   WRITE(*,*)'QUIT PROGRAM ??'
      WRITE(*,*)' 1= YES   0= NO'
      READ(6,*)ANS
      IF(ANS .NE. 1  .AND.  ANS .NE. 0) THEN
        WRITE(*,*)'RESPONCE NOT WITHIN RANGE, PLEASE REENTER'
         GO TO 550
      ENDIF
      IF(ANS .EQ. 1)GOTO 1000
      IF(ANS .EQ. 0)GOTO 520

*****************************************************************
*                     INVESTIGATION TOOL                        *
*                                                               *
*    THIS PORTION OF THE  PROGRAM IS USED TO  INVESTIGATE THE   *
*    EFFECTS OF VARYING FUEL COSTS ON THE OVERALL COST OF       *
*    OPERATIONS.                                                *
*****************************************************************

600   CALL FUELCOST(KWD,STMD)
      CALL COOLSEASON(ELECT,STEAM,HRS,KWD,STMD)

      WRITE(*,*)'DO YOU WISH TO CHANGE THE FUEL COST INPUTS?'
610   WRITE(*,*)' 1 = YES     0 = NO'
      READ(6,*) ANS
      IF(ANS .NE. 1  .AND.  ANS .NE. 0)THEN
        WRITE(*,*)'RESPONCE OUT OF RANGE, PLEASE RE-ENTER'
          GO TO 610
      ENDIF
```

```
      IF(ANS .EQ. 1)GOTO 600

      WRITE(*,*)'QUIT PROGRAM OR GO TO OPERATIONAL PROGRAM?'
620   WRITE(*,*)' 1 = OPERATIONAL PROGRAM    0 = QUIT'
      READ(6,*) ANS
      IF(ANS .NE. 1  .AND.  ANS .NE. 0)THEN
        WRITE(*,*)'RESPONCE OUT OF RANGE, PLEASE RE-ENTER'
         GO TO 620
      ENDIF

      IF(ANS .EQ. 1)GO TO 410

      WRITE(*,*)'QUIT PROGRAM ??'
      WRITE(*,*)'  1 = YES'
      READ(6,*) ANS

      IF(ANS .EQ. 1)THEN
        GO TO 1000
      ELSE
        GO TO 5
      ENDIF
*****************************************************************


1000  STOP
      END

*****************************************************************
*   FUEL COST SUBROUTINE                                          *
*****************************************************************
      SUBROUTINE FUELCOST(KWD,STMD)
      REAL KWD,STMD

10    WRITE(*,*)'INPUT ELECTRIC COST, $/KWH'
      READ(6,*) KWD
      IF(KWD .LT. 0.03  .OR.  KWD .GT. 0.25) THEN
        WRITE(*,*)'ELECTRIC COST OUT OF RANGE($ 0.03 - 0.25),
     . PLEASE RE-ENTER '
         GO TO 10
      ENDIF

20    WRITE(*,*)'INPUT STEAM COST, $/1000 LB'
      READ(6,*)STMD
      IF(STMD .LT. 2  .OR.  STMD .GT. 20) THEN
        WRITE(*,*)'STEAM COST OUT OF RANGE($ 2 - 20),
     . PLEASE RE-ENTER '
         GO TO 20
      ENDIF
```

```
      RETURN
      END

************************************************************************

**** LOAD AND WET BULB  INPUTS  ********

      SUBROUTINE QTWB(Q,TWB,L,IWB)
      INTEGER L,IWB
      REAL Q,TWB

10    WRITE(*,*)'INPUT WET BULB TEMPERATURE, F'
      READ(6,*) TWB
      IF(TWB .LT. 0  .OR.  TWB .GT. 95) THEN
        WRITE(*,*)'WET BULB TEMP OUT OF RANGE(0 - 95), PLEASE REENTER'
         GO TO  10
      ENDIF
      ITWB=INT(TWB)
      IF((TWB - ITWB) .GT. 0.5)THEN
        TWB=ITWB+1
      ELSE
        TWB=ITWB
      ENDIF

      IWB = (TWB/5)-1


20    WRITE(*,*)'INPUT CHILLED WATER FLOW RATE, GAL/MIN'
      READ(6,*) FLCHW
      IF(FLCHW .LT. 5000  .OR. FLCHW .GT. 6500) THEN
        WRITE(*,*)'CHILLED WATER FLOW RATE OUT OF RANGE,PLEASE REENTER'
          GO TO 20
      ENDIF

30    WRITE(*,*)'INPUT CHILLED WATER SET TEMPERATURE, F'
      READ(6,*) TCHWS
      IF(TCHWS .LT. 40  .OR. TCHWS .GT. 55) THEN
        WRITE(*,*)'CHILLED WATER SET TEMP OUT OF RANGE,PLEASE REENTER'
          GO TO 30
      ENDIF

40    WRITE(*,*)'INPUT CHILLED WATER RETURN TEMPERATURE, F'
      READ(6,*) TCHWR
      IF(TCHWR .LT. 40  .OR. TCHWR .GT. 60) THEN
        WRITE(*,*)'CHILLED WATER RETURN TEMP OUT OF RANGE,
     .  PLEASE REENTER'
          GO TO 40
      ENDIF
```

```
      Q = (FLCHW*500.4*(TCHWR - TCHWS))/12000
      Q = Q/100
      L = INT(Q)
      IF((Q-L) .GT. 0.5)THEN
        L = L+1
      ELSE
        L = L
      ENDIF

      RETURN
      END
******************************************************************
*        COST AND SORT SUBROUTINE                                *
******************************************************************
      SUBROUTINE COST(ELECT,STEAM,KWD,STMD,L,IWB,PRICE,FAN)
      REAL ELECT(42,17,12),STEAM(42,17,12),PRICE(12),FAN(12),KWD,STMD
      INTEGER L,IWB

      DO 10 IFS =1,12
        PRICE(IFS) = KWD*ELECT(L,IWB,IFS)+STMD*(STEAM(L,IWB,IFS)/1000)
        FAN(IFS) = IFS
10    CONTINUE

      SWITCH = 1
200   IF(SWITCH .EQ. 1) THEN
        SWITCH = 0
        DO 20 I=1,11
          IF(PRICE(I+1) .LT. PRICE(I)) THEN
            SWITCH = 1
            TEMP1 = PRICE(I)
            TEMP2 = FAN(I)

            PRICE(I) = PRICE(I+1)
            FAN(I) = FAN(I+1)

            PRICE(I+1) = TEMP1
            FAN(I+1) = TEMP2
          ENDIF
20      CONTINUE
          GO TO 200
      ENDIF

      RETURN
      END
******************************************************************
*               HOURLY OUTPUTS SUBPROGRAM                        *
******************************************************************
      SUBROUTINE HROUTPUT(FANSET,PRICE,FAN,VANE,L,IWB)
```

```
      REAL FANSET(6,12),PRICE(12),FAN(12),VANE(42,17,12)
      INTEGER L,IWB

      WRITE(*,15) (FAN(N),N=1,12)
      WRITE(*,18) (VANE(L,IWB,FAN(N)),N=1,12)
      WRITE(*,21) (FANSET(1,FAN(N)),N=1,12)
      WRITE(*,22) (FANSET(2,FAN(N)),N=1,12)
      WRITE(*,23) (FANSET(3,FAN(N)),N=1,12)
      WRITE(*,24) (FANSET(4,FAN(N)),N=1,12)
      WRITE(*,25) (FANSET(5,FAN(N)),N=1,12)
      WRITE(*,26) (FANSET(6,FAN(N)),N=1,12)
      WRITE(*,*) '_____________________________________________________
     ._________________________'
      WRITE(*,27)(PRICE(N),N=1,12)

15    FORMAT(1X,'COMBO # ',12(3X,F3.0))
18    FORMAT(1X,'VANE    ',12(2X,F4.0))
21    FORMAT(1X,'FAN 1   ',12(3X,F3.1))
22    FORMAT(1X,'FAN 2   ',12(3X,F3.1))
23    FORMAT(1X,'FAN 3   ',12(3X,F3.1))
24    FORMAT(1X,'FAN 4   ',12(3X,F3.1))
25    FORMAT(1X,'FAN 5   ',12(3X,F3.1))
26    FORMAT(1X,'FAN 6   ',12(3X,F3.1))
27    FORMAT(1X,'COST$/HR',12(2X,F4.0))

      RETURN
      END
*****************************************************************
*                   ACCUMMULATED COST FOR EACH FAN SETTING      *
*****************************************************************
      SUBROUTINE ACCUM(ELECT,STEAM,HRS,L,IWB,KWD,STMD,VANE,FAN)
      REAL ELECT(42,17,12),STEAM(42,17,12),HRS(42,17),COST(12)
      REAL VANE(42,17,12),KWD,STMD,FAN(12),YRCOST(12)
      INTEGER L,IWB

      IF(HRS(L,IWB)  .LT. 1) THEN
        WRITE(*,*)'ZERO HOURS ESTIMATED AT THESE CONDITIONS'
        GO TO 300
      ENDIF

      DO 100 I=1,12
        COST(I) = KWD*ELECT(L,IWB,FAN(I)) +
     .            STMD*(STEAM(L,IWB,FAN(I))/1000)
        YRCOST(I) = COST(I)*HRS(L,IWB)
100   CONTINUE

      WRITE(*,*)'COMBO #     VANE     HRS/YR     COST $/YR'
      DO 150 N=1,12
        IFAN = FAN(N)
```

```
      WRITE(*,10)FAN(N),VANE(L,IWB,IFAN),HRS(L,IWB),YRCOST(N)
150   CONTINUE

10    FORMAT(3X,F3.0,11X,F4.0,8X,F4.0,7X,F8.0)

300   RETURN
      END
*****************************************************************
*                  TOTAL YEAR OPERATIONAL COSTS                 *
*****************************************************************
      SUBROUTINE COOLSEASON(ELECT,STEAM,HRS,KWD,STMD)
      REAL ELECT(42,17,12),STEAM(42,17,12),HRS(42,17),KWD,STMD
      REAL LCOST,LSTM,LKW,KWCOST

      SUM = 0
      SUMKW = 0
      SUMSTM = 0
      DO 100 L=1,41
        DO 110  IWB = 1,16
          IF(HRS(L,IWB) .LT. 1) GOTO 100
            LCOST = 1.0E10
            DO 120 IFS=1,12
              STMCOST = STMD*(STEAM(L,IWB,IFS)/1000)
              KWCOST = KWD*ELECT(L,IWB,IFS)
              CCOST = KWCOST + STMCOST
              IF(CCOST .LT. LCOST) GO TO 120
                 LIFS = IFS
                 LCOST = CCOST
                 LSTM = STMCOST
                 LKW = KWCOST
120         CONTINUE
            SUM = SUM+HRS(L,IWB)*LCOST
            SUMKW = SUMKW+HRS(L,IWB)*LKW
            SUMSTM = SUMSTM+HRS(L,IWB)*LSTM
110     CONTINUE
100   CONTINUE

      WRITE(*,*)'TOTAL COOLING SEASON ESTIMATED OPERATIONAL COST'
      WRITE(*,*)'-----------------------------------------------'
      WRITE(*,*) ' '
      WRITE(*,10) KWD
      WRITE(*,20) STMD
      WRITE(*,*)' '
      WRITE(*,*)'APPROX. SEASONAL ELECTRIC COST=  $',SUMKW
      WRITE(*,*)'APPROX. SEASONAL STEAM    COST=  $',SUMSTM
      WRITE(*,*)'APPROX. SEASONAL TOTAL    COST=  $',SUM

10    FORMAT(1X,'ELECTRIC UNIT COST = ',F4.2,'  $/KWH')
20    FORMAT(1X,'STEAM UNIT COST    = ',F5.2,'  $/1000 LB')
```

```
30

      RETURN
      END
************************************************************************
```

```
DO YOU WISH TO RUN THE OPERATION PROGRAM OR THE INVESTIGATION PROGARM ?
1= OPERATION        0= INVESTIGATION
1

*****************************************************
*                 OPERATION PROGRAM                 *
*   THIS PORTION OF THE PROGRAM IS FOR DAILY USE.   *
*   IT IS DESIGNED TO TAKE INPUTS OF FUEL COSTS,    *
*   CHILLER LOAD, AND WET BULB TEMPERATURE AND      *
*   OUTPUT THE APPROXIMATE ASSOCIATED COSTS OF EACH *
*   FAN SETTING POSSIBILITY. THE OPERATOR MAY THEN  *
*   MAKE THE CONTROL DECISION BASED ON THIS INFOR-  *
*   MATION. THERE IS AN OPTION FOR YEARLY COSTS AND *
*   HOURLY COSTS. FUEL COSTS, LOAD AND WET BULB MAY *
*   BE ALTERED WITHIN THE PROGRAM AND COMPARISONS   *
*   MAY BE MADE.                                    *
*****************************************************
INPUT ELECTRIC COST, $/KWH
0.05

INPUT STEAM COST, $/1000 LB
4.08

INPUT WET BULB TEMPERATURE, F
65

INPUT CHILLED WATER FLOW RATE, GAL/MIN
5800

INPUT CHILLED WATER SET TEMPERATURE, F
45

INPUT CHILLED WATER RETURN TEMPERATURE, F
55
```

COMBO #	6.	3.	7.	9.	8.	5.	10.	11.	12.	4.
VANE	100.	100.	100.	100.	100.	100.	100.	100.	100.	100.
FAN 1	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	1.0	0.5
FAN 2	0.5	0.5	0.5	0.5	0.5	0.5	0.5	1.0	1.0	0.5
FAN 3	0.5	0.5	0.5	0.5	0.5	0.5	1.0	1.0	1.0	0.5
FAN 4	0.5	0.0	0.0	1.0	0.5	0.0	1.0	1.0	1.0	0.5
FAN 5	1.0	0.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	0.0
FAN 6	0.0	0.0	1.0	1.0	1.0	0.0	1.0	1.0	1.0	0.0
COST$/HR	103.	103.	103.	106.	106.	106.	107.	108.	109.	110.

```
DO YOU WISH TO SEE ESTIMATES OF YEARLY COSTS FOR THESE SCENARIOS ?
1= YES      0=NO
1

COMBO #        VANE        HRS/YR        COST $/YR
   6.           100.         113.          11825.
   3.           100.         113.          11825.
   7.           100.         113.          11914.
   9.           100.         113.          11967.
   8.           100.         113.          11998.
   5.           100.         113.          11999.
  10.           100.         113.          12042.
  11.           100.         113.          12178.
  12.           100.         113.          12350.
   4.           100.         113.          12403.
   2.           100.         113.          12631.
   1.           100.         113.          15088.

DO YOU WISH TO CHANGE INPUTS? (LOAD,TWB,FUEL COSTS)
 1=YES       0=NO
0

QUIT PROGRAM OR RE-SET FOR NEXT HOUR?
  1= RE-SET          0= QUIT
0

QUIT PROGRAM ??
 1= YES   0= NO
1
```

APPENDIX E

ALTERNATIVE FAN CONTROL STRATEGIES

FAN SPEED AND SEQUENCING SCENARIOS

COMB #	TOWER 1		TOWER 2		TOWER 3	
1	0.5	0.0	0.0	0.0	0.0	0.0
2	0.5	0.5	0.0	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

REFINED BASE CASE

COMB #	TOWER 1		TOWER 2		TOWER 3	
1	0.5	0.0	0.0	0.0	0.0	0.0
2	0.5	0.5	0.0	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	0.5	0.5	0.0
6	0.5	0.5	0.5	0.5	0.5	0.5
7	0.5	0.5	0.5	0.5	0.5	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

2 SPEED FANS

COMB #	TOWER 1		TOWER 2		TOWER 3	
1	0.5	0.0	0.0	0.0	0.0	0.0
2	0.5	0.5	0.0	0.0	0.0	0.0
3	0.5	1.0	0.0	0.0	0.0	0.0
4	1.0	1.0	0.0	0.0	0.0	0.0

2 CELL OPERATION

COMB #	TOWER 1		TOWER 2		TOWER 3	
1	1.0	0.0	0.0	0.0	0.0	0.0
2	1.0	1.0	0.0	0.0	0.0	0.0
3	1.0	1.0	1.0	0.0	0.0	0.0
4	1.0	1.0	1.0	1.0	0.0	0.0
5	1.0	1.0	1.0	1.0	1.0	0.0
6	1.0	1.0	1.0	1.0	1.0	1.0

1 SPEED FANS