OPTIMAL SEQUENCING OF CENTRAL REFRIGERATION EQUIPMENT IN AN INDUSTRIAL PLANT

Donald P. Fiorino, Facilities Engineering Manager
Texas Instruments, Inc.
Dallas, Texas 75265

John W. Priest, Ph.D., P.E.
The University of Texas at Arlington
Arlington, Texas 76019

ABSTRACT

A model was developed to find a viable solution to the problem of selecting the optimal sequence of refrigeration equipment (chillers, cooling towers, pumps) to operate in a Central Utility Plant. The optimal equipment sequence is that sequence which has the lowest energy cost to operate at a given plant cooling load and outside air wet bulb temperature. and satisfies all the constraints associated with the refrigeration system. Selection of the optimal equipment sequence is very difficult given the complexity of the refrigeration system and the dynamic nature of the plant cooling load. As a solution a computer program was developed to generate optimal equipment sequences to operate for combinations of a wide range of plant cooling loads and outside air wet bulb temperatures.

Analysis of the solution identified the need for a retrofit project to remove "vital" constraints in order to improve the refrigeration system's performance. The solution to the problem was then incorporated in the operating procedures for the Central Utility Plant.

INTRODUCTION

The Texas Instruments plant on Forest Lane in Dallas, Texas, is a mixed-use facility of approximately 1.3 million square feet. the plant experiences a very large and dynamic cooling load composed of comfort cooling, i.e. 75°F and 50% RH in areas of the plant occupied by personnel, and process cooling, i.e. removal of heat generated by manufacturing processes and equipment. The plant cooling load fluctuates according to a variety of factors including work shifts, production levels, and outside air conditions. The plant was constructed in four phases between 1979 and 1985 and is shown in Figure 1. Administration, engineering, support, and manufacturing activities are conducted at the plant. Approximately 4400 personnel worked at the plant as of May 1985 with an additional 1200 personnel expected by December 1986. The plant was designed and constructed in a distributed manner with centralized support areas, i.e. utility plant, warehouse, cafeteria, etc., and modular work areas, i.e. modules, linked by major corridors.

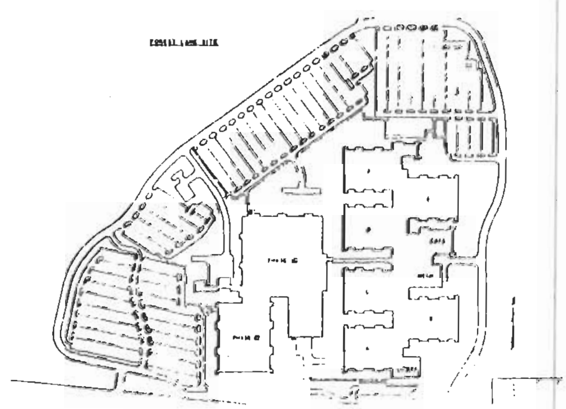

Fig. 1 Site Plan

The Forest Lane plant is equipped with a variety of large capacity refrigeration equipment. The primary cooling medium used throughout the plant is 42°F chilled water. It is produced by four chillers totaling 4200 tons of refrigeration. The chilled water is pumped through the chillers and then through the plant's chilled water supply and return lines by five chilled water primary pumps totaling 625 horsepower and two chilled

Proceedings of the Third Symposium on Improving Building Systems in Hot and Humid Climates, Arlington, TX, November 18-19, 1986

water booster pumps totaling 200 horsepower. Heat liberated by the chillers' vapor compression cycles is rejected to the atmosphere by five cooling towers totaling 4335 tons of refrigeration. Condenser water is pumped between the chillers and the cooling towers by four condenser water pumps totaling 400 horsepower.

EXISTING PROCEDURE

The Central Utility Plant Operator is responsible for selecting and operating the necessary sequences or combinations of refrigeration equipment to satisfy any particular plant cooling load. Equipment selection is based upon a bi-hourly calculation of the plant's instantaneous cooling load, measurement of the outside air wet bulb temperature, and instrument readings of current equipment parameters i.e. flow rates, temperatures, pressures, etc. Knowledge, experience, and judgment is then applied to select the optimal sequence of refrigeration equipment which will satisfy the plant's cooling load with the lowest possible energy cost. The size and complexity of the plant's refrigeration system, the dynamic plant cooling load and outside air wet bulb temperature, and the judgment of the operators reduce the possibility of selecting an optimal sequence of refrigeration equipment. Schematics of the chilling and condensing systems are shown in Figures 2 and 3.

OPTIMIZATION PROJECT

A project was undertaken to find a viable solution to the common problem of selecting an optimal sequence of refrigeration equipment. This sequence was defined as the sequence of refrigeration equipment which satisfies the plant cooling load with the lowest possible energy cost and without violating any constraints associated with the refrigeration system. The project was an optimization problem to be solved by constructing a mathematical model of the refrigeration system and deriving the solution to the problem. The model incorporated those features of the refrigeration system which have a significant effect on the system's performance and ignored those features which have only negligible effect. The optimal equipment sequence must be selected from a set of approximately 10,000 alternative sequences of refrigeration equipment.

The first step was to determine the relevant parameters to be included in the model. The input/output diagram shown in Figure 4 identifies the plant cooling load and outside air wet bulb temperature as the input variables for the model. The output variables for the model are the number and size of chillers to operate, the number and type of chilled water pumps to operate, the number and size of cooling towers to operate, and the number of condenser water pumps to operate. It should be noted that although the output variables are discrete, partial loading conditions may exist. Partial loading is an important concept because some types of refrigeration equipment, particularly chillers, operate most economically when partially loaded. The objective function is the sum of several single-variable objective

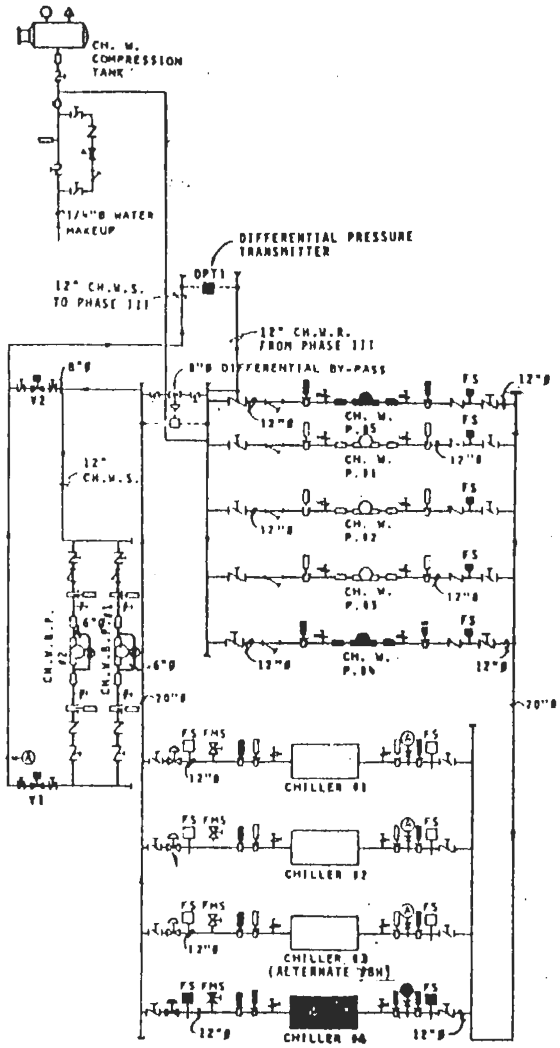

Fig. 2 Schematic of Chilled Water

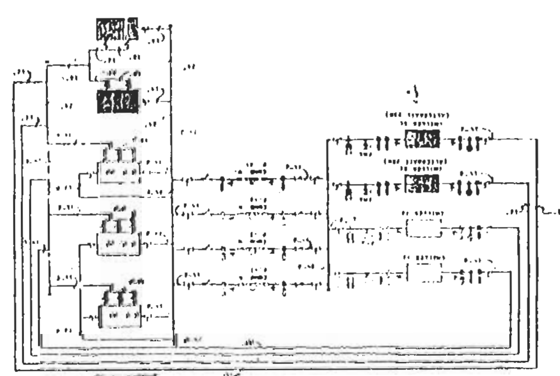

Fig. 3 Schematic of Chillers

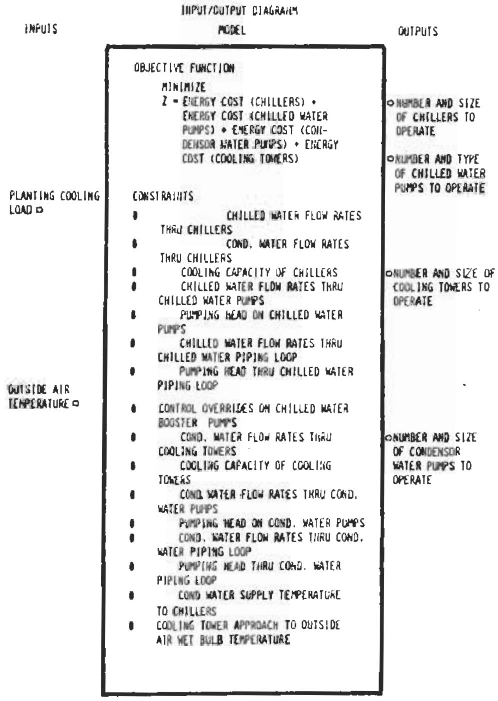

Fig. 4 Input/Output Model

functions that express the hourly energy cost to operate the refrigeration system. The constraints included in the model consist of particular limitations, i.e. maximum and minimum flow rates, temperatures, pressures, etc., associated with the refrigeration equipment and system.

DYNAMIC PROGRAMMING APPROACH

The second step was to select an optimization technique capable of solving the model. Given the Markovian nature of the refrigeration system, i.e. the sequence of interrelated decision stages and a finite number of states associated with each decision stage, a dynamic programming technique was selected. This technique translates this complex problem with seven decision variables into seven sub-problems each with one variable. These sub-problems can then be solved one at a time.

The key elements of dynamic programming are stages, states, decisions, policies, and sub-policies. The concept of a stage allows a sequence of interrelated decisions to be ordered. An index variable n is associated with each stage of the refrigeration system. The stages are listed in Table 1. The state variables then describe the possible conditions in which the system might be in a given stage of the problem; these are also listed in Table 1. An index variable s is associated with each state of the refrigeration system. A decision $x_n(s_n)$ is a choice made in state s at stage n that transforms the system to another state s' in stage n + 1.

TABLE 1

STATES

s	n	Description
1	1	0x100 HP C W Booster Pumps
2	1	1x100 HP C W Booster Pumps
3	1	2x100 HP C W Booster Pumps
4	2	0x125 HP C W Primary Pumps
5	2	1x125 HP C W Primary Pumps
6	2	2x125 HP C W Primary Pumps
7	2	3x125 HP C W Primary Pumps
8	2	4x125 HP C W Primary Pumps
9	2	5x125 HP C W Primary Pumps
10	3	0x900 Ton Chillers
11	3	1x900 Ton Chillers
12	3	2x900 Ton Chillers
13	4	0x1200 Ton Chillers
14	4	1x1200 Ton Chillers
15	4	2x1200 Ton Chillers
16	5	0x100 HP Cond W Pumps
17	5	1x100 HP Cond W Pumps
18	5	2x100 HP Cond W Pumps
19	5	3x100 HP Cond W Pumps
20	5	4x100 HP Cond W Pumps
21	6	0x750 Ton Cooling Towers
22	6	1x750 Ton Cooling Towers
23	6	2x750 Ton Cooling Towers
24	7	0x945 Ton Cooling Towers
25	7	1x945 Ton Cooling Towers
26	7	2x945 Ton Cooling Towers
27	7	3x945 Ton Cooling Towers

Multiple choices are available to proceed from any given state at any given stage in the refrigeration system. A policy is then an ordered set of decisions,

$$X = (x_1, x_2, x_3 \ldots x_7)$$

there being one decision for each state as the system progresses from stage to stage. Figure 5 shows the network of decisions with the system. The optimal policy is that policy which minimizes the objective function $f_n(s_n, x_n)$ while satisfying all constraints. Figure 6 shows this relationship. The principle of optimality is central to dynamic programming and requires that a optimal policy consist only of optimal sub-policies.

The basis of the solution procedure for a dynamic programming problem is that, given the current state, the optimal policy for the remaining stages of the system is independent of the policy adopted in the previous stages. The solution procedure begins by finding the optimal

Proceedings of the Third Symposium on Improving Building Systems in Hot and Humid Climates, Arlington, TX, November 18-19, 1986

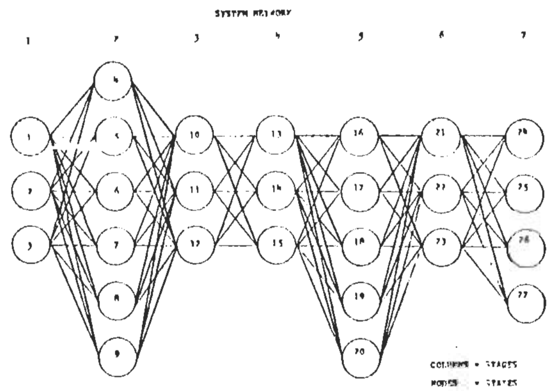

Fig. 5 System Network

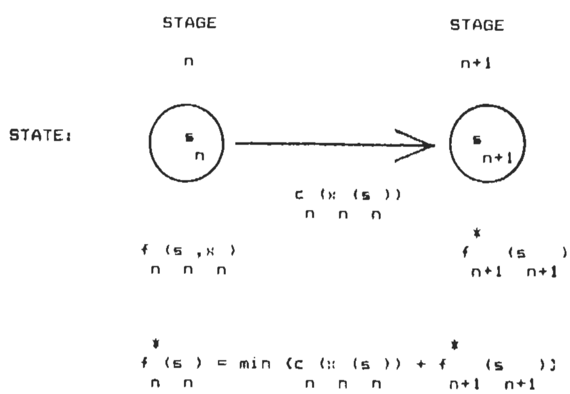

Fig. 6 Recursive Equation

sub-policy for each state of the last stage. The solution procedure then moves backward stage by stage n = (N,N-1,...1) determining the optimal sub-policy for each state until it finds the optimal policy when starting at the initial stage of the system. This is accomplished by utilizing a recursive relationship to identify the optimal sub-policy for each state at stage n given the optimal sub-policy for each state at stage n + 1.

The computer program was written to iterate the input variables over their prescribed ranges; calculate the optimal equipment sequence for each iteration of the input variables according to the recursive relationship and dynamic programming algorithm; calculate the operating parameters, i.e. flow rates, pressures, temperatures, of the refrigeration system and verify that the constraints are satisfied for each equipment sub-sequence at each iteration of the input variables; and generate output data for each iteration of the input variables. The central feature of the computer program is the close correspondence between its architecture, i.e. sub-routines, and the structure of the refrigeration system as represented by the dynamic programming model, i.e. stages and states.

MODEL TESTING AND VALIDATION

The next step was to determine that the model reasonably represented the refrigeration system. Validation testing was accomplished by comparing refrigeration system performance as represented by the model to actual system performance as represented by system operating data and equipment performance data. Comparisons were performed for input parameters, equipment performance, and output parameters.

The computer program iterated plant cooling load from 500 to 4000 tons of refrigeration. Actual cooling loads experienced at the Forest Lane plant were recorded on a bi-hourly basis during 1985 and ranged from 511 tons in November to 2861 tons in June. These values were adjusted by a factor of 1.31 to account for additional cooling loads in Phase IV, i.e. 800 tons, and the warehouse, i.e. 135 tons, which the Forest Lane plant expected to realize during 1986. The range of imputed plant cooling loads, i.e. 500 to 4000 tons, compared closely to the range of actual plant cooling loads (adjusted), i.e. 671 to 3756 tons.

Actual operating data for the refrigeration system also provided the basis for testing the model's representation of key output parameters. The model's representation of the system's condenser water flow rate, i.e. line 1010, yielded a value identical to the actual figure, and its representation of the system's condenser water head, i.e. line 5300, yielded a value within 3 percent of the actual figure. The model's representation of condenser water supply temperature, i.e. state 26 of stage 7, yielded a value identical to the actual figure. Its representation of the system's chilled water flow rate, i.e. line 7290, yielded a value 17 percent below the actual figure; however, the variance was attributed to the fact that chiller #2 was operating below design conditions. The model's representation of the system's chilled water head, i.e. line 13460, yielded a value within 8 percent of the actual figure, and its representation of the system's chilled water supply temperature, i.e. state 14 of stage 4, yielded a value identical to the actual figure.

For comparison purposes, computer output data for the combination of plant cooling load and outside air wet bulb temperature under actual conditions was displayed opposite the actual operating data. This comparison of model output data and actual operating data at the system level provided the basis for testing the model is representation of the optimal equipment sequence and operating cost. The model's representation operated less and on smaller refrigeration equipment than that actually in operation at the time of comparison.

MODEL SENSITIVITY

Sensitivity analysis was used to investigate the effect on the optimal solution when the parameters changed values. Sensitive parameters were particularly serious in this project when a change in their value caused an inferior or infeasible value of the objective function to result.

Data of optimal operating cost versus plant cooling load with outside air wet bulb temperature fixed was organized in graphical form. Optimal operating cost varied moderately with changes in plant cooling load in the range of 700 to 2700 tons but became infeasible when the plant cooling load ranged below 700 tons or above 2700 tons. Validation testing revealed that plant cooling loads below 700 tons and above 3600 tons are rarely experienced. However, the sensitivity of optimal operating cost to plant cooling load in the range of 2800 to 3600 tons was of significance to the system and solution and warranted further investigation.

Data of optimal operating cost versus outside air wet bulb temperature with plant cooling load fixed was organized in graphical form. Optimal operating cost varied moderately with changes in outside air wet bulb temperature in the range of 20°F to 75°F but became infeasible when the outside air wet bulb temperature reached 80°F. Fortunately, outside air temperatures of 80°F are rarely experienced.

PARETO ANALYSIS

A Pareto analysis was used to identify which specific system constraints and sub-systems were causing the above problems. Once identified, a retrofit project could be undertaken to reduce or remove the constraint at fault by modifying the sub-system. This improved the refrigeration system and solution performance in the sensitive region.

The computer model simulated system performance for 36 iterations of the plant cooling load with 13 iterations of the outside air wet bulb temperature nested inside each iteration of the plant cooling load. Thus, the model simulated system performance for 36 x 13 = 468 unique combinations of the input variables.

Each of the simulations, in which an infeasible solution resulted, was analyzed to determine the system constraint(s) which was violated and the associated sub-system(s), i.e. stage(s), of the dynamic programming network at which the violation occurred. The results of the analysis revealed that violations were concentrated among only 8 of 22 constraints and among 3 of 4 sub-systems.

A Pareto analysis of the relative and cumulative frequency of violations by constraint was prepared in a graphical plot. This Pareto analysis revealed that nearly 50 percent of the violations were attributable to only 2 "vital few" constraints, i.e. maximum chilled water flow rate to Phases III/IV and minimum unsatisfied chilled water head to operate a chilled water booster pump. The output data showed the violations of these two constraints to have occurred largely within the range of plant cooling load that was identified by sensitivity analysis as being of significance to the system and solution. This Pareto analysis also revealed that the remaining 50 percent of the violations were attributable to 20 "trivial many" constraints.

RETROFIT MEASURES

Maximum chilled water flow rate to Phases III/IV and minimum unsatisfied chilled water head to operate a chilled water booster pump, in one sub-system, i.e. the chilled water pumps and piping sub-system, were identified as the most significant causes of infeasible solutions. The infeasible solutions caused by the above constraints and sub-system were concentrated in the range of plant cooling load, i.e. 2800-3600 tons, which was identified by sensitivity analysis as a sensitive range of that parameter and was identified by validation testing as a reasonable range of that parameter.

This was corroborated by chilled water flow/pressure problems actually experienced at the Forest Lane plant. A retrofit project to reduce or remove the two "vital" constraints in the chilled water pumps and piping sub-system became evident as a means of improving the refrigeration system's performance. Increasing the chilled water piping capacity between the Central Utility Plant and Phases III/IV was defined first, and chilled water booster pump modifications were defined thereafter based on the final definition of chilled water piping capacity.

A field survey was performed to estimate the peak building and process chilled water demands in Phases III/IV. An overlay of combined peak chilled water demand for Phases III/IV on existing chilled water piping capacity showed that the 4844 gallons per minute peak demand exceeded the 2500 gallons per minute pipe capacity between the Central Utility Plant and Phases III/IV by 2344 gallons per minute or 93.8%. Alternatives proposed to remove this piping capacity shortfall consisted of a "four pipe" system (i.e. installing a second set of twelve inch diameter chilled water supply and return lines in parallel with the existing set of twelve inch diameter chilled water supply and return lines between the Central Utility Plant and Phases III/IV) and a three pipe system (i.e. installing an eighteen inch diameter chilled water supply line between the Central Utility Plant and Phases III/IV and converting the existing twelve inch diameter chilled water supply line between the Central Utility Plant and Phases III/IV into a second return line). Both alternatives would supply 5000 gallons per minute to Phases III/IV at a reasonable velocity of 7.17 feet per second and a moderate pressure loss of 0.515 pounds per square inch per 100 feet, thereby satisfying Phase III/IV peak chilled water demand. The advantages and disadvantages of each piping system were determined and compared, as were those for two pipe routing alternatives and two pipe anchoring alternatives. The combination of alternatives selected for installation were the "three pipe" system with the eighteen inch diameter chilled water supply line routed through the Module A corridor and buried.

The constraint of minimum unsatisfied chilled water head to operate a chilled water booster pump related to both the chilled water primary and booster pumps as the chilled water booster pumps are piped in parallel in the chilled water supply line to Phases III/IV and are piped in series with the chilled water primary pumps in the chilled water primary piping loop. Hence, the chilled water flow rate delivered by the booster pumps is common to that delivered by the primary pumps and the chilled water head delivered by the booster pumps is in addition to that delivered by the primary pumps. Thus, if system chilled water head is 185 feet, a likely value of that parameter as established by validation testing, and the primary pumps are delivering their maximum head of 150 feet, then the unsatisfied head to be delivered by the booster pumps would be 185 - 150 = 35 feet. Yet, the minimum head which a booster pump can deliver at 1150 revolutions per minute is 44 feet as shown by the booster pump performance curve; therefore, a booster pump cannot be energized. The result is an infeasible model solution and poor system performance. Yet, the performance curve of the same booster pump operating at 860 revolutions per minute rather than 1150 revolutions per minute was noted in Figure 7. At 860 revolutions per minute, the booster pump can deliver the 35 feet of head unsatisfied by the primary pumps. In this instance, a feasible model solution and satisfactory system performance would result. Alternatives proposed to modify the booster pumps consisted of replacing the two existing booster pumps with multi-speed pumps (i.e. identical pumps each coupled to both an 1150 revolution per minute motor and an 860 revolution per minute motor) and installing adjustable frequency speed controllers (i.e. very advanced electronic devices which vary input power frequency to adjust motor speed) on the two existing booster pumps. Results of adjustable frequency speed controller testing conducted earlier were reviewed and the advantages and disadvantages of each alternative were determined and compared. Installation of adjustable frequency speed controllers was selected.

The original model representing the existing refrigeration system was updated to represent the retrofitted refrigeration system. Updating the original model was simplified by the fact that all of the additions and modifications are concentrated in the chilled water piping and pumps sub-system of the refrigeration system. An updated input/output diagram highlighted five system constraints and one component of the objective function which had to be updated. A solution was derived from the updated model, validated, and analyzed. The performance of the retrofitted refrigeration system was then compared to that of the existing refrigeration system by comparing the solutions from their respective models.

Optimal operating cost varied moderately with changes in plant cooling load in the range of 700 to 3600 tons. Optimal operating cost became infeasible when the plant cooling load ranged below 700 tons or above 3600 tons. Validation testing revealed that plant cooling loads in those ranges are rarely experienced. Optimal operating cost also become infeasible when the outside air wet bulb temperature reached 80°F. Fortunately outside air wet bulb temperatures of 80°F are rarely experienced.

An updated Pareto analysis of the relative and cumulative frequency of violations by constraint was then evaluated. This analysis

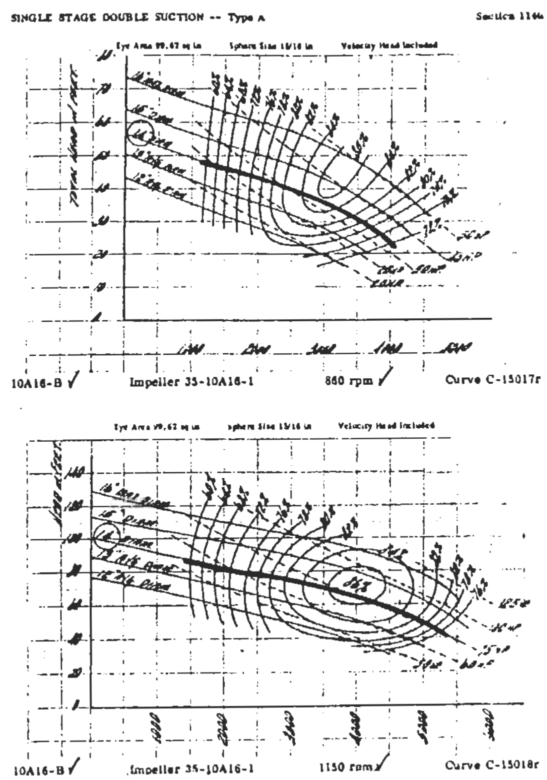

Fig. 7 Chilled Water Booster Pump Curves

revealed that nearly 45% of the violations were attributable to one "vital" constraint, i.e. maximum system condensor water flow rate. A scan of the updated output data showed the violations of this constraint occurred within the range of plant cooling load that was identified by validation testing as rarely occurring. This Pareto analysis also revealed that the remaining 55% of the violations were attributable to 20 "trivial" constraints. Performance statistics resulting from establishing control over the original model representing the existing refrigeration system and the updated model representing the retrofitted refrigeration system were compared. The percentage by which refrigeration system performance improved as a result of the retrofit project was calculated for key statistics measuring system performance. The updated model showed that the retrofitted refrigeration system's performance was greatly improved over that of the existing refrigeration system as represented by the original model.

IMPLEMENTING THE SOLUTION

The first step in implementing the solution was undertaking and completing the chilled water piping and pumps retrofit measures. A formal project proposal, i.e. capital request, was submitted through a series of capital approval committees to secure funding for the project.

Proceedings of the Third Symposium on Improving Building Systems in Hot and Humid Climates, Arlington, TX, November 18-19, 1986

Facilities cost analysis personnel performed a formal return on investment analysis and a risk assessment analysis for the project.

Detailed engineering of the chilled water piping project was performed in a two-stage design process. Drawings and specifications for installation of the retrofit measures were prepared at the Forest Lane facilities engineering office. These were then forwarded to the architect's office to be reviewed and modified and/or supplemented as necessary. In particular, the architect issued appendix items for structure interface matters, i.e. pipe supports and anchors, wall/floor/roof penetrations, thrust blocks and expansion measures, and reviewed equipment submittal data. This two-stage design process identified protection of the buried eighteen inch diameter black iron pipe against corrosion as a latent, yet critical, aspect of the design. Corrosion protection involves coating, electrically isolating, and cathodically protecting structures to minimize corrosion and maximize service life. A corrosion protection consultant was retained to write an amendment to the drawings and specifications addressing corrosion protection for the buried eighteen inch diameter black iron pipe.

Installation and start-up of the chilled water piping project was initiated following a bid solicitation process. The construction contract was awarded to the winning bidder - a large piping/mechanical contractor with an extensive record of similar projects completed satisfactorily at the Forest Lane plant and other Texas Instruments facilities.

A "critical path method" network and schedule was developed to monitor equipment/material delivery, installation sequencing and scheduling, system start-up, related work by crews other than the contractor, and contingency planning. Daily worksite inspections were performed to monitor the above as well as assure quality workmanship, safety/environmental and security measures, and avoidance of interference with plant operations. Weekly project review meetings were conducted to review progress to date and coordinate critical project elements for the following week. The "critical path" network and schedule was updated following each project review meeting to reflect actual progress and changes/contingencies. Start-up and testing of the retrofitted refrigeration system was accomplished by a start-up team and the contractor during a scheduled shutdown of plant chilled water service. Minor mechanical and electrical problems which surfaced during start-up and testing were quickly corrected by troubleshooting and debugging. Extending beyond start-up and testing, the retrofitted refrigeration system's operation was monitored closely for a period of six months during which a wide range of plant cooling loads and outside air wet bulb temperatures were experienced. During this period of extended operation, mechanical reliability of the retrofitted refrigeration was proven, its performance capabilities were demonstrated, and operator training was strengthened.

The second step in implementing the solution was testing and proving the updated solution during actual operation of the retrofitted refrigeration system. This was accomplished during the period of operation of the retrofitted refrigeration system extending beyond start-up. The improved procedure for selecting the optimal sequence of refrigeration equipment to operate in the Central Utility Plant now consists of the Central Utility Plant Operator calculating the plant's instantaneous cooling load and measuring the outside air wet bulb temperature as before, but now referring to the solution (i.e. the updated computer output listing optimal equipment sequences for combinations of plant cooling load ranging from 500-4000 tons and outside air wet bulb temperature ranging from 20-80^{o}F) to select the optimal equipment sequence to operate. This improved procedure is now standard and is incorporated in the written operating instructions for the Central Utility Plant.