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Aquifer Thermal Energy Storage at Mid-Island Postal Facility: Phase I Final Report

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May 1993

Prepared for
New York State Energy Research
and Development Authority
Albany, New York
under Contract Number 18576

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AQUIFER THERMAL ENERGY STORAGE AT MID-ISLAND POSTAL FACILITY: PHASE I FINAL REPORT

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SUMMARY

The successful widespread commercialization of aquifer thermal energy storage (ATES) in the United States will depend on how the experiences gained from early full-scale projects are used as guides in the design, installation, and operation of future projects. One early system, built in the mid-1980s, is the U.S. Postal Service (USPS) Mid-Island Mail Processing Facility (MPF), in Melville, New York. The heating, ventilation, and air conditioning (HVAC) of the MPF's workroom is provided by an ATES system, which is operated year-round to provide a source for both heating and cooling, in combination with a triethylene glycol (TEG) liquid-desiccant system for humidity control. Because the facility affords a unique opportunity to study this innovative system, the U.S. Department of Energy's (DOE) Pacific Northwest Laboratory (PNL) entered into agreements with the USPS, the U.S. Geological Survey (USGS), and the New York State Energy Research and Development Authority (the Energy Authority) to assess the operation and performance of the system.

Two essentially independent questions were to be addressed by the project. The first question is "How does the MPF ATES/TEG technology compare to conventional technologies?" The second is "What can be done to make operation of the USPS MPF more economical?" However, modelling of the MPF ATES/TEG HVAC system and its loads helped to address both of these questions by showing how much energy is used by the different system components. Note that the question of how an optimally designed and operated ATES/desiccant system compares to conventional technologies is not addressed in this study. Also note that USPS concerns for control of dust, which the MPF designers addressed by providing high air circulation rates and air washing, made the ATES/TEG system, with its inherently high circulation rates and air washing, particularly attractive. However the high air circulation rate was not used in the conventional HVAC technology cases reported here because it is relatively uncommon.

Comparison of unique systems with conventional technology inevitably requires removal of the systems from significant aspects of their design and implementation context. This study is no exception. The MPF ATES/TEG/HVAC system is the only one of its kind in the world. It is a highly integrated

design that sought to provide a number of secondary social and environmental benefits whose value is difficult, if not impossible, to estimate. No assessment was made in this study of either these benefits or their value. Unique designs almost always suffer from overdesign, to ensure functionality, impacting the costs and operation of the systems. Invariably, considerations that drive design decisions (such as future energy costs, availability of fuels, future functional requirements) are rarely accurate, resulting in suboptimal designs when viewed a decade later. Additionally, functional requirements evolve with time, and systems must operate to match these requirements. All of the foregoing factors should be kept in mind by the reader because all apply to the ATES/TEG/HVAC system at the Mid-Island MPF.

As the first task of this assessment, the annual energy performance of the ATES/TEG HVAC system was estimated assuming the system is operated "per design" according to existing operating schedules in effect at the MPF. Computer simulation models were developed for the MPF and individual components and subsystems that make up the ATES/TEG HVAC system. These models were based on original design documents, monitored data, anecdotal reports, and equipment and system logs kept by the USPS. Two baseline HVAC system models were also developed for comparison to the ATES/TEG system. The first baseline system consisted of a two-pipe heating and cooling plant with an electrically driven water chiller and propane boilers, and standard air-handling units to replace the liquid-desiccant system's conditioners. The second or "alternate baseline" system was an extension of the first with the addition of HEPA filters and desiccant-wheel dehumidification to provide air quality and humidity control comparable to that provided by the liquid-desiccant system's conditioners.

From the standpoint of space temperature control, the first baseline system fully meets the usual requirements for this building application. However, the liquid-desiccant system can provide some additional capabilities that, because their impacts have not been quantitatively assessed, are not credited in this evaluation. These include precise control of space humidity levels in summer, and the ability to act as an air scrubber to remove gaseous pollutants (Chung 1993) and particulates. At high enough concentrations, the liquid desiccant has also been found to kill some airborne organisms

(Amsterdam 1988). The alternative baseline system provides air quality and humidity control comparable to that provided by the liquid-desiccant system.

Peak cooling loads calculated as part of this task were considerably lower than the original design (191 tons versus 329 tons), but latent loads were more than twice as high (742 lb/hr versus 354 lb/hr moisture removal rates). These higher latent loads are attributable to ventilation equipment that was added after design and construction. Both the simulation and field test results indicate that the nominal 500 lb/hr capacity of the existing concentrator would be adequate if these ventilation loads had not been added.

Because the building envelope of the MPF is well insulated and internal heat gains caused by equipment are high relative to other commercial buildings, the estimated average and peak heating loads are quite low relative to the cooling loads.

Total annual energy consumption for the baseline HVAC system was predicted to be 1159.4 MWh with an additional 6850 gal of propane for heating. Total annual energy consumption for the alternate baseline system is predicted to be 1375 MWh in electrical energy, 6850 gal of propane for heating, and 20,249 gal of propane for desiccant-wheel regeneration. By comparison, the predicted annual energy consumption for the ATES/TEG system is 1545 MWh, with 51,918 gal of propane used for liquid-desiccant regeneration. While the cost to operate the ATES/TEG system is higher than the baseline systems, it is not excessive relative to other buildings of its vintage. The HVAC energy consumption per square foot of conditioned floor area was predicted to be 17,700 Btu/ft^2 for the baseline, 30,400 Btu/ft^2 for the alternate baseline, and 45,000 Btu/ft^2 for the ATES/TEG system. The USPS design goal for the facility in 1982 was to reduce energy consumption below 88,000 Btu/ft^2.

The higher electrical energy consumption of the ATES/TEG system relative to the baseline systems is primarily caused by higher air flow rates for this system relative to the baseline. Higher air flow rates are required to produce the same net cooling effect because the ATES/TEG system cannot produce the low supply air temperatures provided by the baseline system. Mechanical refrigeration requirements of the ATES/TEG system were found to be reduced by more than two-thirds compared to the baseline system. Parasitic pumping

energy required to operate the ATES and liquid-desiccant systems is the second highest contributor to the annual electrical load. Additional performance penalties associated with operating the liquid-desiccant system include TEG losses caused by evaporation and the increase in annual plant cooling load resulting from the heat of solution of the water absorption process and from the heat produced in TEG circulation, spray and pumps.

Note that the performance of the baseline systems was based on the performance of 1982 technologies but that the life-cycle costs were based on current equipment, labor and energy costs. The reason for using 1982 technology performance numbers is this: while the past decade has seen no appreciable progress in the performance of commercially available liquid-desiccant systems, there have been continuing research efforts that, if commercialized today, would result in significantly better performance. Progress in cost-effective conventional chiller technologies, on the other hand, has been continuously transferred to commercially available equipment. To prevent these artificial technology transfer discontinuities from unfairly penalizing the liquid-desiccant technology we chose to use the 1982 performance numbers. Current cost numbers were used primarily because the 10-year old cost numbers are very hard to document. Current energy cost numbers were used because we were primarily interested in the current and future potential of the ATES/TEG concept, not how it might have looked a decade ago. The MPF is not an ideal case study because construction costs are generally higher for government projects. This could bias the life-cycle cost results to favor the lowest first cost alternative.

A life-cycle cost analysis was performed to compare the costs of the ATES/TEG HVAC system to those of the baseline HVAC systems. The analysis translated predicted energy consumption of the three systems into energy costs. Capital costs for the liquid-desiccant system were obtained from the manufacturer, and well drilling costs were obtained from the Melville well contractor. Other capital equipment costs were estimated using conventional engineering methodology. Unfortunately, the results of the economic assessment indicated that both the initial capital and the annual operation and maintenance costs are approximately double those of the alternate baseline system and four times those of the baseline system. The existing ATES/TEG

system design results in little reduction in size and cost of the components common to the three systems. Adding ATES and liquid-desiccant system components results in a cost increase of about 300% over the baseline system and about 140% over the alternate baseline system.

The second goal of the technical assessment was to explore plant modifications and operational and control issues relevant to performance and ability of the system to meet the needs of the facility in its current use. These issues can be largely resolved by the following actions:

- Control infiltration loads caused by shrink-wrap machines and other exhaust fans. Dedicated air supplies for the machines could reduce latent loads caused by induced infiltration air flow of over 17,000 cfm when all three shrink-wrap machines are operating.

- Improve efficiency of propane boilers from around 70% to 90% through either replacement with a new high-efficiency model or through reclaim of flue gas heat.

- Operate the liquid-desiccant system main circulating and conditioner spray pumps only as needed rather than continuously, or replace pump motors with variable- or two-speed pump models.

- Use variable-speed pumping on the well-water system to better match cooling loads (thus more effectively using the limited resource of stored chill) and to increase the amount of chill produced by the closed-circuit cooler in winter.

Because workroom cooling loads were predicted to be quite different from those estimated in the original design, conditioner performance calculations were made. The results of these calculations suggest the following operational modifications could be made to improve system performance:

- Reduce design supply and return fan air flow rates on the workroom conditioners to better match actual cooling loads. Preliminary estimates suggest this could reduce fan horsepower on the order of 25%. Additional savings can be achieved by efficient motor and airfoil impeller retrofits.

- Change the design supply chilled water temperature from 50°F to 54°F. These higher chilled water temperatures should still provide sufficient sensible cooling of the space. Based on the aquifer model, this could reduce mechanical refrigeration requirements an additional two-thirds.

An operational assessment of the liquid-desiccant system was performed to characterize the performance of the system, to identify operating

anomalies, and to provide recommendations (where possible) for improving and maintaining the system. This task included short- and long-term monitoring of the liquid-desiccant system, analysis of log data, and development of conditioner performance algorithms and a moisture load model. Results from the long- and short-term monitoring of the liquid-desiccant system concentrator suggest there are a number of opportunities for increasing capacity and energy efficiency. In addition to the opportunities already identified above, these include:

- Increase concentrator air flow by increasing horsepower and/or mitigating air flow restrictions to increase the moisture removal capacity of the concentrator. A variable-speed drive or two-speed motor retrofit should be provided so that the concentrator can operate at low fan power most of the time but also operate at a high power when necessitated by humid conditions. An efficient airfoil blower retrofit should also be considered.

- Improve concentrator thermal coefficient of performance by recovering heat and avoiding mixing desiccant streams of different temperature and concentration, which occurs in the current system. The addition of a heat pipe heat exchanger to extract heat upstream of the reflux coil and add it to the intake air upstream of the hot water coil is the first priority.

- Enlarge the existing desiccant plate heat exchanger and possibly use return reflux water to further cool the desiccant returned to the system from the concentrator.

The results of the technical and economic assessments show that the ATES/TEG system, as implemented at the MPF, can[a] provide the HVAC service and high air quality for which it was designed but at relatively high cost. It is likely that the first cost would have been much less if the conservative design criteria were relaxed to levels consistent with the criteria applied to an accepted, mature technology. Some relatively low-cost retrofits and operational changes have been identified that should reduce energy costs to operate the system by about 40%. Further reductions in energy cost are possible but probably not cost effective to implement in an existing system. In terms of parasitic energy use, the ATES/TEG system is more sensitive to

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(a) if measures to increase concentrator capacity to design capacity and reduce dehumidification loads by eliminating the shrink-wrap ventilation loads, which were added after design and construction, are implemented.

control strategy than the conventional system. It appears that continuous simultaneous optimization of pump and fan speeds, supply air conditioners, and chilled water temperature is needed, as well as carefully optimized pipe and duct system design, to obtain reasonable parasitic energy use. However, investigation of advanced control was beyond the scope of work to be addressed in Phase I of the project.

In evaluating the broader implications suggested by the results of this assessment, it should be remembered that this is a unique and innovative integration of unconventional technologies. As such, an important objective has been simply to improve our understanding of the operating characteristics of these integrated technologies. This objective has been met both by the work reported here and through the USPS's continued operation of the system. The operational experience and analysis of the ATES/TEG system at the MPF provide valuable groundwork upon which system and component designs using similar technologies may be considered. In addition, the results of this study provide targets for improving economic and technical performance of the ATES/TEG system, and help determine where improvement will be most beneficial.

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1.0 INTRODUCTION

The U.S. Postal Service's (USPS) Mid-Island Mail Processing Facility (MPF) is located in Melville, New York. The facility serves as the central mail collection and distribution point for a population of 2.5 million.

Heating, ventilation, and air conditioning (HVAC) is provided to most of the MPF by the unique combination of an aquifer (ground water) heating/cooling system and a liquid-desiccant dehumidification system as shown in Figure 1.1. The seasonal aquifer thermal energy storage (ATES) system is designed to provide "cold" water to handle the cooling load during summer and "warm" water as the heat source for the chiller, operated as a heat pump, in winter. Backup cooling is provided by the chiller; emergency heating capability is provided by modular boilers. Triethylene glycol (TEG) is the liquid desiccant used to remove moisture from the workroom supply air. Modular boilers provide heat to regenerate the desiccant. The liquid-desiccant system, shown in Figure 1.2, can also add water to the air during periods of low humidity. We will refer to the MPF HVAC system and its loads (actual and modelled) as the ATES/TEG system and to the HVAC system alone as the ATES/TEG HVAC system.

Conceptually, ATES/TEG HVAC systems of the type implemented at the MPF may offer significant operational advantages over more conventional systems. Mechanical refrigeration for cooling is downsized from peak cooling load levels, and its operation schedule can be reduced. Thus, a sizable reduction in peak summer electric loads could occur relative to conventional systems, along with a decrease in total summer electrical use. Downsizing of the air-conditioning equipment also reduces the amount of ozone-depleting compounds that is required, such as chlorofluorocarbons (CFCs). Because the desiccant system provides dehumidification, cooling water temperatures are not as low as those needed in other ATES cooling designs; this reduces thermal losses in the ATES cycle. Finally, the TEG liquid-desiccant system provides a more precise control of conditioned space humidity than conventional commercial HVAC systems. TEG-based dehumidification systems may also reduce certain gaseous pollutants (Chung 1993) and bacteriological pollutants (Amsterdam 1988), and can help in the removal of airborne particulates.

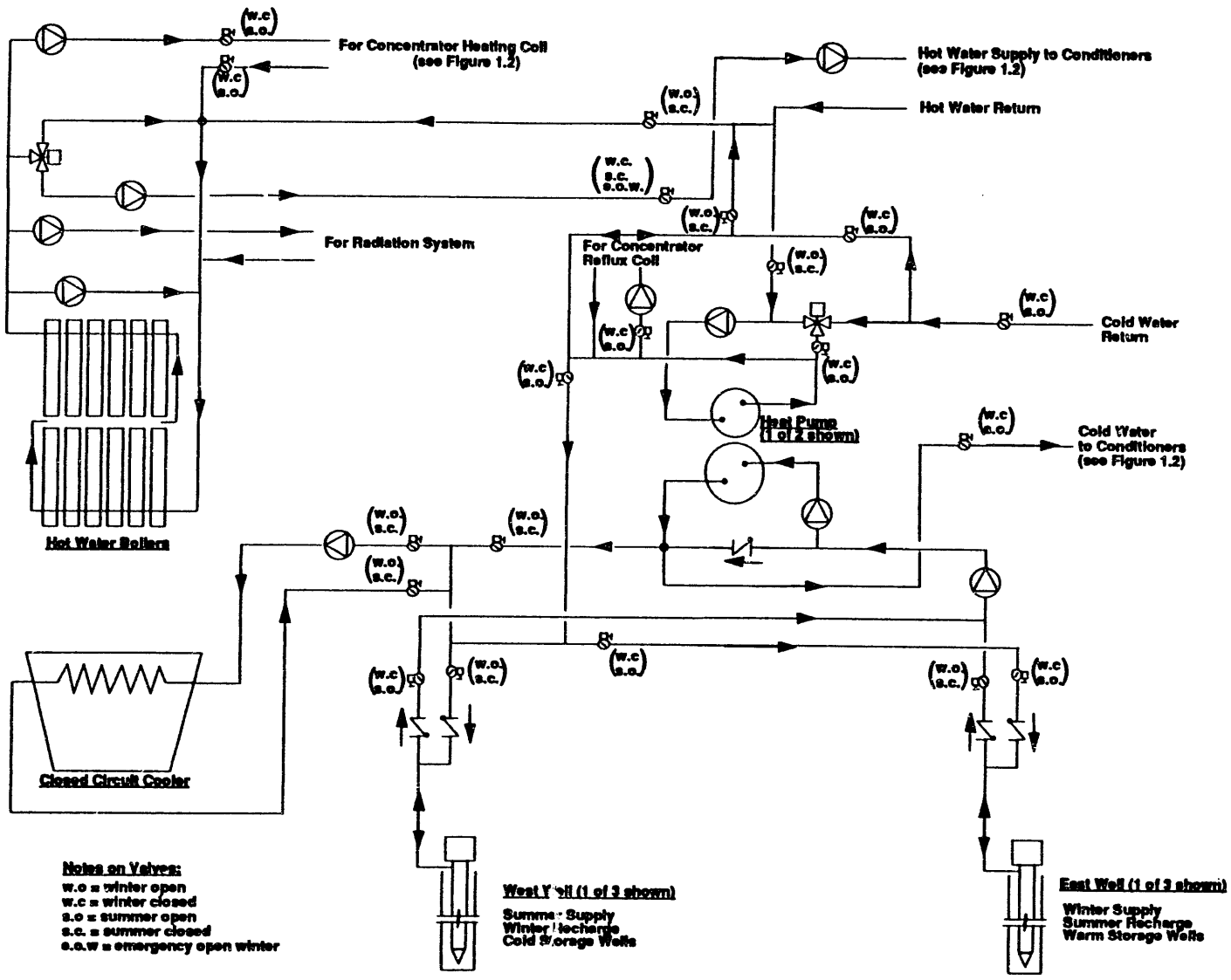

FIGURE 1.1. Existing ATES/TEG HVAC System Schematic (Backup Cooling Mode Not Shown)

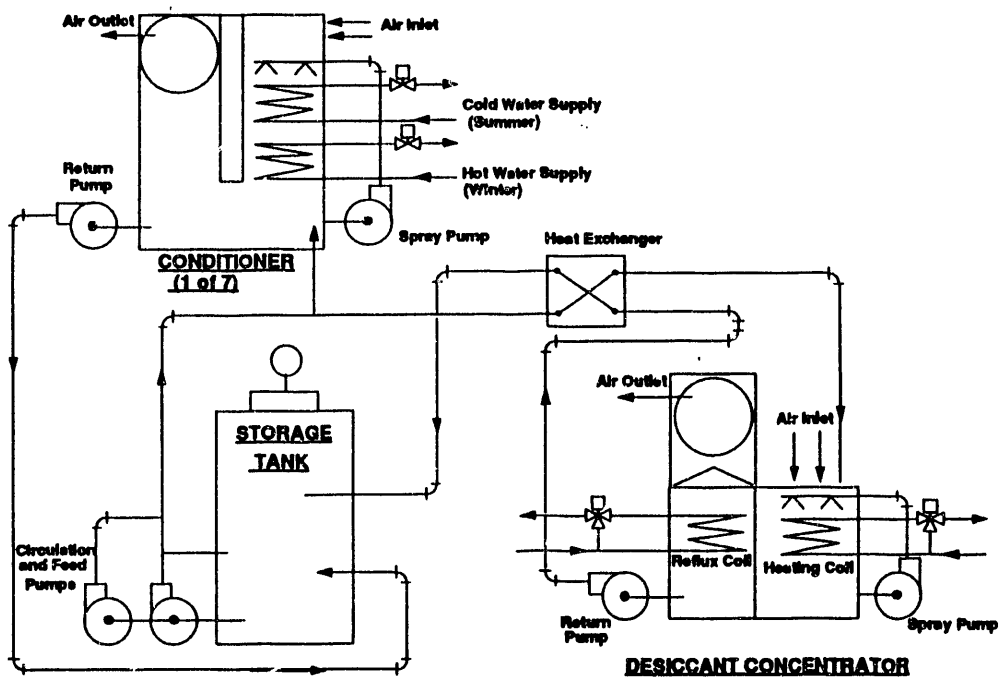

FIGURE 1.2. Liquid-Desiccant System Schematic

Because of these perceived advantages and the unique opportunity for study this innovative existing application affords, the U.S. Department of Energy's (DOE) Pacific Northwest Laboratory (PNL)[a] entered into agreements with the USPS, the U.S. Geological Survey (USGS), and the NYS Energy Research and Development Authority to evaluate the system. Elements of this evaluation included:

- A technical assessment to 1) estimate the annual electrical and propane use associated with operating the MPF HVAC system as originally designed, and 2) compare this to a more conventional "baseline" HVAC system. These comparisons provide a gauge for the relative energy merit of using an ATES/liquid-desiccant system at the MPF, as well as general design and operation insights that might prove valuable when considering such a system at another site.

- A life-cycle cost analysis to compare the costs of owning and operating the ATES/TEG system relative to two baseline HVAC systems representing conventional HVAC technologies.

(a) Operated for the U.S. Department of Energy by Battelle Memorial Institute under Contract DE-AC06-76RLO 1830.

- An operational assessment of the liquid-desiccant system to characterize the performance of the system, identify operating anomalies, and provide recommendations (where possible) for improving and maintaining the system. This task included on-site monitoring and analysis of other collected operational data (both logs and anecdotal data) on the liquid-desiccant system.

An additional investigation, not reported here, was conducted by the USGS to better understand the geotechnical performance of the ATES wells at the MPF. The objectives of this investigation included defining the local geohydrologic setting, evaluating the potential causes of well clogging, and providing general recommendations for improving overall system operation. Because of funding restrictions, only initial investigations were made in these areas.

The balance of this report is divided into six sections and two appendices. Section 2 provides system background. Section 3 describes PNL's technical performance assessment of the system, while Section 4 discusses the life-cycle cost assessment. An operational assessment of the liquid-desiccant system is discussed in Section 5. Section 6 contains conclusions of this study. References are listed in Section 7. Appendix A contains a listing of the input data that describe the DOE-2 Loads, Systems and Plants models used to estimate the performance of the baseline system by simulation. Appendix B contains a preliminary cost and savings analysis of the control, conservation and energy efficiency opportunities that were identified in Sections 3 and 5.

2.0 BACKGROUND

This section provides background information that is useful in understanding the assessment that follows later in the report. Section 2.1 gives an overview of the building and its functions. Section 2.2 provides a summary of each of the primary components and subsystems at the MPF. A more detailed description of the MPF well layout and construction is included in Section 2.3, followed by an overview of the basic operating concept of the ATES system in Section 2.4. In Section 2.5, issues associated with modeling the performance of an ATES system when integrated with a building HVAC system are discussed. Finally, in Section 2.6, the application of computer simulation for the comparative performance assessment is presented.

2.1 FACILITY DESCRIPTION

The USPS Mid-Island Mail Processing Facility is a 260,000 ft^2 building located in Melville, New York. The facility services a population of 2.5 million with sophisticated, computerized mail-sorting equipment. Approximately 1200 employees work over the course of a 24-hr day, 7 days a week. The peak occupancy period is between 6 p.m. and 3 a.m., Monday through Friday. Various types of mail processing equipment at the facility sort, bundle and label approximately 5 million pieces of mail each day.

Lighting for the workroom area is provided by variable high-pressure sodium (HPS) fixtures, each controlled by a central computer-generated radio signal. The lighting intensity is dictated by USPS work program needs. However, energy use by the HPS lights can be reduced in response to daylighting provided by skylights, which comprise 4% of the roof area. Other energy conservation features of the MPF building envelope include white Hypalon membrane roofing mechanically fastened (to avoid use of conventional roof ballasts) over R-30 insulation; exterior walls are insulated precast concrete.

Heating, ventilation, and air conditioning of the MPF's main workroom (200,000 ft^2) is provided by a unique combination of ground water heating/cooling with a liquid-desiccant system for humidity control.

The MPF was designed in the early 1980s, at a time when future energy costs and availability were highly uncertain. Long Island Lighting Company (LILCO) was just completing a major nuclear facility and a substantial electricity rate hike was anticipated to pay for the $4.2 billion plant. Oil prices in the Middle East were still unstable, and political unrest in that region made the costs of additional petroleum products highly uncertain. Additionally, LILCO would not provide natural gas service to the facility. These factors played a significant role in designing the MPF.

2.2 HVAC DESIGN SUMMARY

A brief summary of each primary component or subsystem of the ATES/TEG HVAC system is given below:

- Aquifer well system - Three wells supply cold water to the MPF during the summer, and are recharged in the winter with water from the three "warm" wells after the water has been cooled via closed-circuit cooler operation. The three other wells that supply warm water also are used during the winter as a source for the heat pumps (chillers operated in heat recovery mode). During the summer, cold well water is circulated to the workroom air handlers (and chillers, as necessary), and then injected into the warm wells.

- Heat recovery chillers[a] - Two 100-ton chillers are configured to operate as heat pumps capable of meeting either heating or cooling loads imposed on the conditioner coils. In cooling mode the chillers only operate under heavy load to supplement the well water cooling effect. In heating mode warm well water is used as a heat source and the chillers operate in heat recovery mode to satisfy heating loads.

- Closed-circuit cooler - A roof-top closed-circuit cooler is used to cool water from the three warm wells during the winter for reinjection into the cold wells. Additional free cooling results from operation of the chillers in heat pump mode when a heating load exists.

- Liquid-desiccant system - The workroom supply air is air conditioned using "conditioners" (essentially air-handling units with an added liquid-desiccant dehumidification feature) supplied by the Niagara

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[a] While the term "heat pump" adequately describes the function of the equipment in question, it is customary to use the term "heat recovery chiller" because the selection of operating mode (heating or cooling) is effected by redirection of water-side flows rather than by use of a refrigerant reversing valve. The only obvious difference between the heat recovery chiller and a standard centrifugal chiller is the existence of an extra condenser.

Blower Company. Sprayed cooling coils, located in each conditioner, are supplied with chilled water during the summer cooling season. Triethylene glycol (TEG), a liquid desiccant, is sprayed over the cooling coils to absorb moisture from the supply air stream and thus dehumidify the air. The latent heat associated with this process, along with the sensible heat load in the air, is transferred to the water passing through the spray coil. Water absorbed by the TEG solution is removed at a central liquid-desiccant concentrator. In winter, the TEG cycle may also be reversed, with the liquid-desiccant system used to add water to the supply air stream for space humidification by spraying dilute TEG over unit heating coils through which hot water is flowing.

- Modular boilers - Heat required for operation of the liquid-desiccant concentrator is provided by two banks of modular propane boilers.

2.3 WELL DESCRIPTION

The ATES system consists of six 180 ft deep wells, each made up of a 1.0-ft diameter (I.D.) steel casing fitted with a 50-ft long 8-in. diameter screen. The screen is surrounded by a 1.5-ft diameter fine gravel envelope that extends 20 ft above the screen for a total height of 70 ft. The casing above the gravel pack is enclosed in cement grout. The well pump is located within the casing above the screen and within the water table at a sufficient depth to allow for drawdown. Clearance of 1 in. around the pump on all sides allows for recharge. A line shaft turbine pump is used. Thus, the pump drive motor is located above the well. The cross section of a typical well is shown in Figure 2.1. The subsurface environment penetrated by the wells is comprised of glacial outwash sand and gravel sediments known as manetto gravel. The ground water table is 40 ft below surface grade. At the level of the well screen the aquifer is classified as the upper glacial aquifer.

Wells 1A, 2A, and 3A are located on the eastern boundary of the USPS property, as shown in Figure 2.2. The reported temperature of water from these wells prior to ATES system operation was 63.5°F. This relatively high temperature is considered to be a result of warm water discharge into the ground from off-site wells located to the north that service air conditioners of large office complexes. Through regional flow of ground water, this discharge was expected to affect aquifer conditions at the USPS/MPF site. As a consequence, the warmest wells of the ATES system, (the east wells) are used

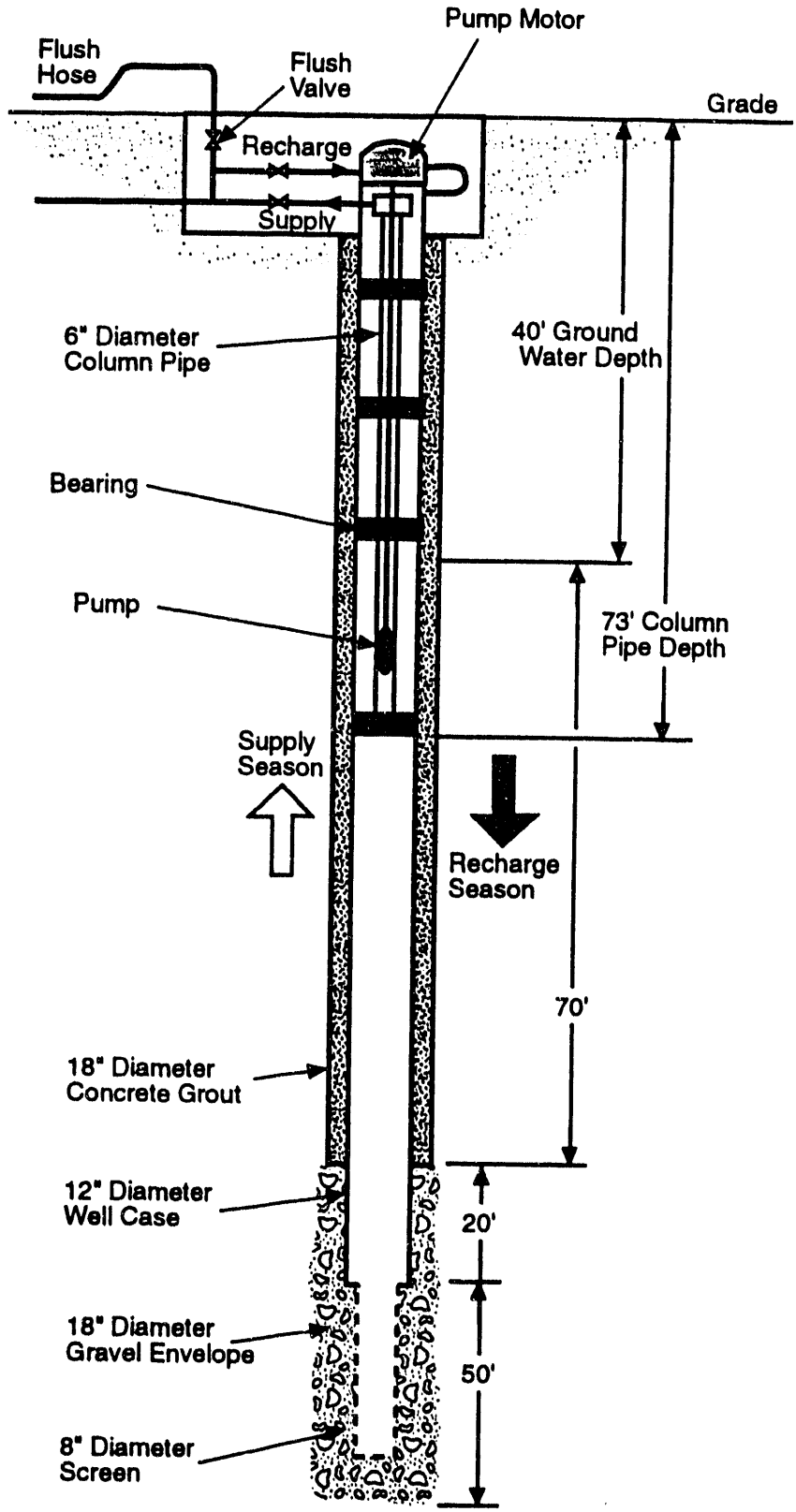

FIGURE 2.1. Typical Well Cross Section

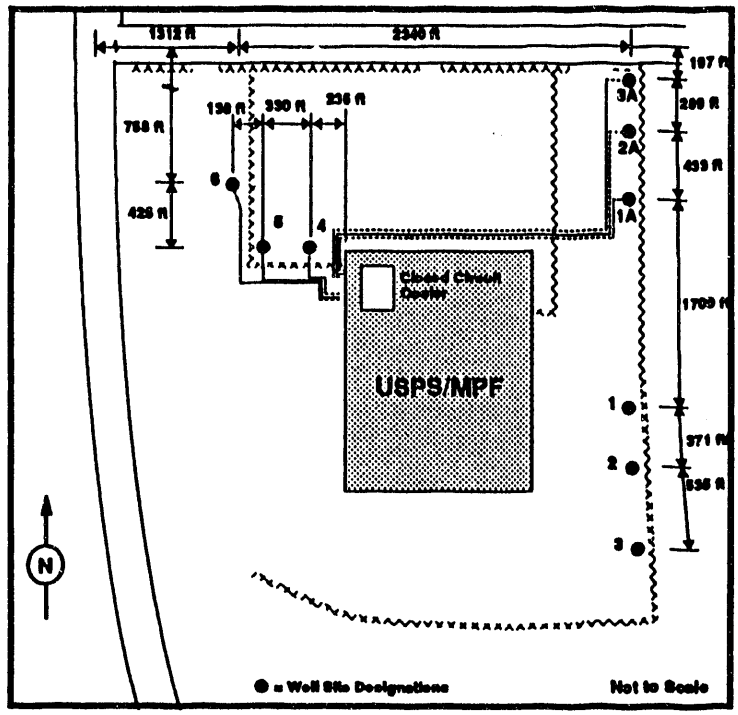

FIGURE 2.2. Site Map Showing Well Locations

as the winter heat source for the chillers. In summer, warm water return from the USPS/MPF is injected into the ground at these wells.

Wells 4, 5, and 6, also shown in Figure 2.2, are located on the west side of the USPS property. The reported temperature of water from these wells prior to ATES system operation was 57.9°F. The west wells are not in line with the regional flow of discharge water from the off-site wells to the north. These wells are therefore used for summer supply of cool water to the chillers and are reinjected with cool water during winter.

2.4 WELL SYSTEM OPERATING CONCEPT

The well system is designed to supply the MPF with cool water to meet the cooling load during summer and warm water for heating during winter. The basic concept is that of seasonally storing masses of warm and cool water in separate locations in the aquifer. As it is warmed in summer, the water is injected into east (warm) wells 1A, 2A, and 3A; as it is cooled in winter, the water is injected into west (cold) wells 4, 5, and 6.

Because the MPF's building envelope is well insulated and internal heat gains caused by equipment are high, the original design calculations indicate that more cooling energy is required on a seasonal basis than heating. To meet the additional cool water requirements beyond what can be produced during winter chiller (heat recovery mode) operation, water drawn from the east wells may also be passed through the roof-top closed-circuit cooler and reinjected into the west wells. The closed-circuit cooler is designed to reduce water temperatures in the winter prior to injection into cold water wells. This additional cooling capacity is required to offset the heat loading generated by the mail processing operation and to balance annual heating and cooling of the ground water thermal storage. Following seasonal changeover[a], a cool water mass should be available from the west wells for summer cooling, and a warm water mass should be available from the east wells for winter heating. The heat losses caused by (horizontal) regional water flow and vertical conduction, along with other geologic factors, limit the amount of stored heat or chill that can be recovered from the wells. As a result of the regional flow, the southernmost west (cool) wells supply the coolest water and southernmost east (heat) wells supply the warmest water.

The workroom summer cooling process was designed to use 50°F chilled water in combination with the liquid-desiccant system. When cool water recovered from the wells is above 50°F, the water chillers further chill the water to obtain the design temperature. The design goal is to be able to obtain direct cooling with well water without electric chillers, but this will depend on whether the amount of cold water stored during the previous season (less recovery losses) is adequate to meet the annual cooling load and whether recovery flow rates can be maintained at design levels. The design assumes wells will be pumped only as required to meet the heating or cooling needs of the MPF on a daily basis. Single-well operation occurs during moderate cooling and heating periods. Two-well supply operation occurs during periods of high cooling needs.

(a) According to the direct digital control schedule for the MPF, seasonal changeover occurs at a mean daily temperature threshold of 65°F. In the computer model, seasonal changeover times are determined by the procedure described in Section 3.2.4.3.

2.5 MODELING OF ATES AND BUILDING HVAC SYSTEMS

Several levels of sophistication are possible in the analysis of ATES systems. The connection between the aquifer model and the dynamic building load model is often simplified in the preliminary design phase. Total annual heating and cooling load estimates typically are used to estimate the "average" recovery flow rates from the aquifer for the entire heating/cooling season and the total "charge" of water that must be recovered over the season. Following a survey of the site, a hydrogeologic model is developed to predict how well a seasonal ATES system can meet these requirements. The aquifer models numerically solve for the thermal response of the subsurface environment (Vail 1989), accounting for horizontal transport and vertical conduction within the aquifer and for long-term heating or cooling in the storage zone. However, even if the aquifer model is highly accurate, a simplified, seasonal load approach to coupling the aquifer with the building can result in significant system performance prediction errors because it does not capture all of the dynamic effects of changing building loads (i.e., hourly, daily, monthly). Also, if average flow rates are used, the analysis might falsely predict that an ATES system is adequate when actually it might fail to produce sufficient flows at cold enough temperatures late in the summer, the time when peak cooling loads generally occur. An equally unsophisticated approach, and one that is often used to estimate chiller annual energy consumption for buildings with conventional HVAC systems, is the "equivalent full load hours" (EFLH) method (ASHRAE 1985, Chapter 25). Results of EFLH analyses are overly conservative if cooling loads vary much from the peak. Using peak recovery flow rates to estimate ATES seasonal performance might lead to overly conservative well designs.

On the other end of the spectrum, the most sophisticated method for modeling the USPS/MPF would be to link an ATES computational model created to predict aquifer performance (i.e., flow rates and heat/chill recovery efficiency) directly with a sophisticated transient simulation model of the building, so that the HVAC dynamic effects down to the zonal level might be determined. Such an approach is not justified in most design or case study settings because of the uncertainties in the parameters needed to specify a detailed engineering model. Instead, hourly total heating and cooling plant

loads were calculated using a dynamic building thermal model and these loads converted to required aquifer recovery flow rates. In this approach the loads are averaged over the larger time increments (approximately 12 hr) characteristic of the computational time step of the aquifer model (Vail, Kannberg, and Kincaid 1985). Return water temperatures to warm wells from the MPF are estimated by an energy balance based on recovery flow rate. Whenever the aquifer recovered chill is predicted to be inadequate to meet current building cooling needs, the balance is assumed to come from the water chillers.

The MPF calculated thermal loads and local weather data were also used to determine operation requirements for the closed-circuit cooler. Finally, predicted latent heat loads for the workroom were used to predict annual desiccant system performance, including the propane used by modular boilers for regeneration of desiccant in the concentrator.

2.6 COMPUTER SIMULATION

Though this study was focused on the performance of two specific HVAC subsystems--the aquifer and desiccant--the interactions and inherent interconnectedness of these systems with the building and its operation required that a building-level analysis be conducted. For such analyses, computer simulation is the most cost-effective tool, and often the most appropriate one. Although some field energy performance data were available on the energy performance of the MPF ATES/TEG system, direct comparisons between this measured data and simulated performance of the baseline system could not be justified because independent, uncontrolled variables such as weather and occupant behavior influence energy performance considerably. These variables must be assumed determinate in a computer model, but are rarely well understood in an actual facility. It can also be argued that computer models of complex systems (e.g., the aquifer) rarely perform exactly as the real system does in every detail even if the system is well understood. However, a computer simulation does provide a measure of control over uncertain or difficult-to-characterize parameters, and can thus provide a basis for unbiased comparative studies.

3.0 TECHNICAL PERFORMANCE ASSESSMENT

The objectives of the technical assessment were to estimate the annual electrical and propane energy use by the MPF HVAC system and compare this to the energy used by a hypothetical conventional HVAC system, henceforth referred to as the baseline HVAC system. The comparative study provides a gauge to the relative energy merit of the ATES/TEG HVAC system at the MPF. The study also provides a better understanding of design and operational problems to guide improvements in the MPF system and its operation, and serves as a guide to future applications of similar systems in other facilities.

The technical assessment approach included the following elements:

- Obtain information on the building (Section 3.1.1) and the HVAC system (Section 3.2.1) to define the building envelope, equipment, lighting, occupancy schedules, and HVAC system design parameters and operation modes.

- Obtain manufacturer's published data on the liquid-desiccant system to develop algorithms for modeling performance and energy consumption. This was supplied by Niagara Blower, and supplemented as needed by data collected for the operational assessment (Section 5) of the liquid-desiccant system.

- Develop a DOE-2 building load model (Section 3.1) to estimate block and zone hourly cooling and heating loads under typical weather conditions. DOE-2 is a widely used building energy simulation tool consisting of a Loads program, a Systems program and a Plants program (York et al. 1980).

- Develop "baseline" HVAC system[a] operation parameters and equipment selections for DOE-2 (Section 3.2.2) and from this, estimate baseline HVAC system annual energy use and total hourly plant heating, cooling, and dehumidification requirements for the workroom (Section 3.1.3).

- Create a geohydrologic model of the Melville aquifer system using the computer code ATESSS (Aquifer Thermal Energy Storage System Simulator) and calibrate it using well log performance data collected in 1992 (Section 3.2.4.1).

(a) Two baseline systems were ultimately developed: one using conventional chiller and air-handler equipment; the other using a desiccant wheel unit in parallel, and HEPA filters in series, with each air handler to provide air quality and humidity control comparable to that provided by a TEG system.

- Create a model for the above-ground portions of the ATES/TEG HVAC system that can be linked with the ATESSS model (Section 3.2.4.2). The resulting model is driven by the series of 8760 hourly loads computed by the DOE-2 Loads program.

- Simulate the integrated ATES/TEG HVAC system and from the simulation results, estimate the annual energy use of the ATES/TEG system (Section 3.2.5.2).

3.1 BUILDING LOAD MODEL

The DOE-2 Loads program was used to compute hourly block and zone heating, cooling, and dehumidification loads for the MPF for an entire year. DOE-2 mathematically represents the building's envelope thermal properties, occupant heat and moisture generation, light and equipment convective and radiative heat gains, solar radiation, and heat conduction to and from the environment. Hourly values of temperature, solar radiation, humidity, and wind speed from typical year weather data, along with 24-hr weekday, weekend, and holiday profiles describing internal heat gains from people, lights, and equipment, are used to drive the simulation.

Section 3.1.1 describes the data used in creating the building load model. Modeling assumptions used in developing a DOE-2 description of the MPF are reviewed in Section 3.1.2. Calculated peak cooling and heating loads for the MPF and its workroom are presented and discussed in Section 3.1.3.

3.1.1 Data Collection

The data used to generate the DOE-2 Loads model of the MPF were essentially all provided by Douglas A. Wilke. This included a limited number of reproductions from what were reportedly the original design calculations, and drawings and specifications. In addition[a], informal past reports prepared for the USPS and others were provided, along with data obtained

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(a) Because Wilke, MPF operations staff, etc., all concur that the main workroom, as currently configured and operated, differs significantly from the original design.

through on-site inspections or one-time field measurements by Wilke. These data are summarized below:

- Systems Operational Data - This 40-page table compiled by Wilke includes 24-hr work area schedules (derived in part from a USPS "Equipment Utilization Plan"), mail processing equipment operating schedules, and electrical consumption rates, as well operating schedules and electrical consumption rates on other equipment at the MPF. Electrical data on this schedule were reportedly obtained by one-time field measurements. Data on office equipment (e.g., personal computers, copy machines) were provided separately.

- USPS/MPF Melville Occupancy Schedule - This table provides the 24-hr weekday occupancy schedule for each HVAC system zone in the facility.

- Block cooling loading and closed-circuit cooler capacity - This 1987 informal report by Wilke for Spector/Hillier was (according to a reference within the report) based on a number of earlier documents. Included in this report are estimated design block and zone cooling load data that were useful for comparisons with DOE-2 results. Report appendices also contained well field and plant drawings, detailed space requirements (temperatures, ventilation rates, lighting, surface reflectance, occupants) and work area schedules. Even though listed space requirement and work schedules were obsolete, this report was a useful reference.

- "Cost Effective Daylighting for Large Industrial Buildings" - This paper by Wilke described in detail the workroom HPS lighting system, designed to save energy by taking advantage of daylighting from the roof skylights. Additional supporting drawings of the workroom lighting design were also provided with this paper.

- "U.S. Postal System Uses Thermal Energy" - This paper by D.L. Bratta of the USPS and Wilke summarizes the building energy design concept and key system operating parameters.

- Revised design criteria dated 4/27/84 - These documents included general design data and workroom zone load calculations used for sizing the liquid-desiccant system. Though primarily an aid during the development of the liquid-desiccant performance models and the HVAC system energy model, this also served as a reference for synthesis of the DOE-2 building load model.

- Fan schedules - This data was useful in the synthesis of a DOE-2 Systems model and was also used to help estimate infiltration rates for the DOE-2 Loads model.

Issues not covered by the above documents were resolved through phone conversations and written correspondence with Wilke.

3.1.2 Modeling Assumptions

A building's thermal behavior is a complex function of interactions between passive and active components and subsystems. Many of these interactions are not well understood, and can often only be estimated in models. Furthermore, developing a computer simulation that is detailed enough to capture all the complexities would be impractical, even if the available data on a particular building were detailed and of high quality and the understanding of this data complete enough to make the synthesis of such a model possible. For these reasons, a number of simplifications and assumptions are implicit in the DOE-2 simulation algorithms and building description formats. Additional assumptions made in developing the DOE-2 building load model for the MPF site are reviewed here.

The DOE-2 model of the MPF includes separate zones that were based on HVAC equipment service zones, rather than on individual thermostat-controlled spaces. For example, the building Zones 3a and 3b in the workroom, each serviced by liquid-desiccant conditioner AC-3, were combined into a single zone. This aggregation was required because the DOE-2 program limits the number of zones with unique operation schedules. in a model. Though modifications to the DOE-2 source code could have been made to overcome this limitation, detailed load calculation by control zone is not really necessary for the system-level assessment.

A total of 20 zones were established: Zones 1, 1A, 1B, 11, and 12 for the administrative area; Zone 2 for the kitchen, cafeteria, and locker rooms; Zones 3 through 9 for the workroom; Zone 8c for the medical unit; Zone 10 for the computer forwarding area; and separate zones for each of the two platforms (east and south), the mechanical room, the southeast storage area, and the plenum above the administrative area. The zone locations (except for the plenum) are identified in the floor plan shown in Figure 3.1.

To drive the DOE-2 building load and system models, a formatted hourly weather data file, which includes temperature, humidity, solar radiation, and wind speeds for an entire year, is required. Continuous and accurate weather data for the Melville area were unavailable. For this reason, Weather Year

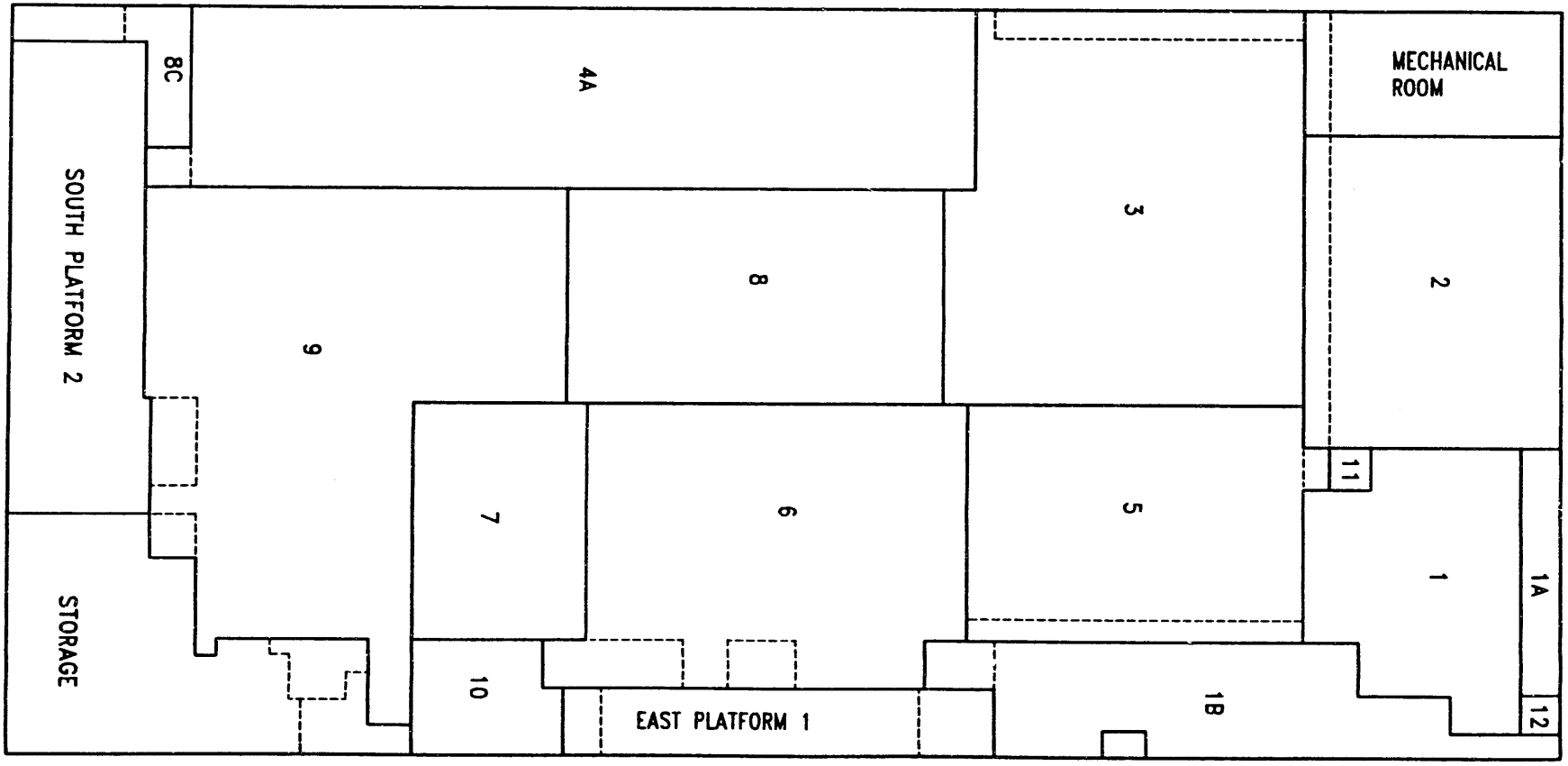

FIGURE 3.1. Floor Plan Showing Zones Used by the DOE-2 Loads Model

for Energy Calculations (WYEC) data for New York City's La Guardia Airport were used. The WYEC data file represents typical conditions and is based on over 15 years of historical data (Crow 1980).

Occupant behavior can represent perhaps the single greatest cause for uncertainty in building simulations. The presence and activity levels of occupants within the space directly affect the rates of heat and moisture gain. Equally pervasive are the effects of occupants' control over the operation of building equipment, lighting, and heating and cooling equipment. Significant deviations in the operation of this equipment from design intent can and do occur, and these deviations have a considerable effect on building performance. Because detailed log data were not available on all system operations and occupant activities, the schedules provided by Wilke were used in the DOE-2 model.

The design calculations for the desiccant system at MPF assumed workroom conditions of 78°F temperature and 45% relative humidity (RH) in summer and 65°F temperature and 30% RH in winter. (The Wilke report to Spector/Hillier listed 80°F and 55% RH summer, 68°F and 40% RH winter.) By contrast, thermostats in the building have reportedly been set at 72°F year-round. Because the DOE-2 building load model allows only a single interior temperature for the entire year, 72°F was used. For system equipment operation, however, the DOE-2 System program allows a variation in space setpoints. It was decided that to be true to the agreed philosophy of developing a "per-design" building model, the original design setpoints of 78°F in summer and 68°F winter should be used for the workroom.

Estimating the net cooling and heating load caused by infiltration at the MPF was another source of uncertainty. As originally designed, the building was intended to maintain a net positive pressurization (i.e., infiltration = 0.0). However, the volume of exhaust air is currently much larger than was originally expected. Three shrink-wrap machines with dedicated exhaust fans totalling 17,400 cfm, each operating about 8 hr/day, were introduced after the building was designed and occupied. In addition, air flow rates for some additional exhaust fans listed in the "System Operational Data" were not shown in the original "fan schedules". And, although the hr/day operating schedules were provided for the fans, the

corresponding times-of-day at which exhaust fans operated was uncertain. Assuming all fans (supply/return/exhaust) on the "fan schedule" operate at full capacity, the net flow imbalance that would be caused by infiltration may be as high as 30,000 cfm. Subtracting from this the flow rates for EF-5 (paint booth: 7,000 cfm) and other exhaust fans, which according to schedules are not used, reduces infiltration to about 20,000 cfm. To account for the expected diversity in fan operation and because some of this infiltration is to unconditioned spaces (e.g., the mechanical room), this number was further reduced by one-half in the DOE-2 Loads model.

The second question with regard to infiltration is where it occurs. Platform doors provide one likely path. Reportedly, these doors are intermittently propped open by staff. Windows in the administrative offices can also be opened. For the load analysis, the total estimated infiltration rate was apportioned on a volumetric basis (i.e., uniform air changes per hour) among all the conditioned zones except for the isolated Zones 11 and 12. Because the workroom is a common open area, this assumption is not unreasonable.

It would have been difficult--but fortunately was not necessary--to individually model each of the 364 skylights in the workroom area. To account for the effect of the skylights, two separate skylight models were developed for each workroom zone. The first model accounted for the thermal effect, and simply lumped together all the skylights into a single one with the sum total window area and identical heat transfer properties. The second skylighting model was for the daylighting calculations. A single skylight encompassing an area whose perimeter surrounds the entire array of skylights, thus including the roof area between skylights, was defined for each zone. This model had heat transfer properties artificially set so no thermal transmission would occur, and had a visible transmittance adjusted by a factor equal to the ratio of the actual skylight area to the larger (by the amount of included roof) artificial skylight area. This approach allowed DOE-2 to account for window position in the daylighting model, an effect that would otherwise have been lost.

3.1.3 Results

Once the DOE-2 Loads program input deck was completed, the heating, cooling, and dehumidification loads were estimated on a workstation computer using Version 2.1D of DOE-2. A copy of the input deck is included in Appendix A. Results are discussed for the summer cooling and winter heating loads. Note that because the energy analyses are concerned with net system and plant cooling and heating requirements, the loads include mechanical ventilation. This component of the load was actually computed using the DOE-2 Systems program.

The peak workroom block cooling load predicted using the DOE-2 building load model was 191 tons at 23:00 hr on July 19. Outdoor air conditions at this hour are 84°F dry bulb and 76°F wet bulb. This time corresponds to the peak activity period in the workroom, as indicated by equipment and occupancy schedules. By comparison, the design peak workroom load reported by Wilke in the Spector/Hillier report is 284 tons, and the original design calculations for the desiccant system reported 329 tons peak cooling load for the workroom. However, while the DOE-2 simulated total (sensible plus latent) design loads are much lower, the DOE-2 model predicts a much higher latent load fraction than the Spector/Hillier and desiccant system design estimates. One-third of the DOE-2 peak hour workroom load is latent load--equivalent to a required moisture removal rate of 727 lb/hr. Also the time of the DOE-2 peak latent load does not coincide with the time of the peak total load given by the Spector/Hillier or original design estimates. DOE-2 gives a peak latent load of 742 lb/hr on July 29 at 17:00 hr, when outside air temperatures are 84°F dry bulb and 78°F wet bulb. This is nearly 50% larger than the nominal concentrator capacity of 500 lb/hr, and more than double the 354 lb/hr estimate reported in the design calculations[a]. A total of 441 hr during

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[a] High as the estimate of workroom dehumidification load is, this high latent load may not be conservative relative to the actual peak. If the arbitrary diversity factors applied in estimating the infiltration rates are not valid, the load could be off by as much as 20 to 25%.

the cooling season are predicted by DOE-2 to have workroom dehumidification loads exceeding 500 lb/hr.

The differences between the DOE-2 predicted workroom block peak cooling loads and those given in the original design calculations for the desiccant conditioners can be attributed to:

- Higher sensible heat assumed in design calculations.

  1. 1,337,600 Btuh of equipment heat load was the basis for the design, but schedules indicate only 803,740 Btuh of equipment load is installed in the workroom.

  2. The design calculations assume all people are in the space and all equipment is "on" during the middle of the day (14:00 hr), when solar heat gains are at their highest. By contrast, because it followed the schedules indicated by the "Systems Operational Data," the DOE-2 model predicted that the peak load occurs at night when there is no solar gain.

  3. Workroom lighting in the original design calculations was assumed to be 0.9 W/ft^2, but based on the "Systems Operational Data" is actually only 0.72 W/ft^2.

  4. Although the original design calculations were based on building loads at 2:00 p.m., no lighting credit is taken for daylighting control.

- Higher latent loads predicted by DOE-2 building model.

  1. 338 people are assumed to be in the workroom in the original design calculations, but the "occupancy schedule" has a peak of 432 people.

  2. 7,260 cfm of the 10,000 cfm total infiltration air at the MPF is assumed to be introduced in the workroom in the DOE-2 model, but no infiltration is assumed in the original design calculations.

  3. Workroom ventilation air (13,360 cfm) at the peak time is 84°F dry bulb and 76°F wet bulb, while the original design assumed 90°F dry bulb and 74°F wet bulb. Thus, while sensible infiltration and envelope gains are lower, the moisture carried by the air at the DOE-2 design condition is higher by 0.004 lb moisture per lb dry air.

The "plant" at the MPF (i.e., that part of the HVAC system using the ATES water loop) provides heating and cooling to the workroom in addition to the administrative, lockers, and cafeteria areas. The annual plant cooling load predicted by DOE-2 was 5485.5 MBtu, or 457,000 ton-hr. The workroom

annual latent load, representing the total load imposed on the liquid-desiccant system, is 897.7 MBtu, or 846,000 lbm moisture.

As would be expected from the block loads, peak simulated zone loads in the workroom also differ greatly from the original design. Table 3.1 is a comparison between DOE-2 predicted workroom zone peak loads, the design zone peak loads, as given in the Spector/Hillier report, and the original design calculations for the desiccant system. The only zone load predicted to exceed the earlier design estimates of peak cooling load is the 44.9-ton load imposed on conditioner AC-9 (31.9 tons and 33.4 tons in the design calculations). Other zone cooling loads are predicted by DOE-2 to be between 33% and 60% of design. However, dehumidification loads (in lbm/hr moisture) were predicted to be 30% to 400% higher than design.[a]

The building envelope of the MPF is well insulated and internal heat gains caused by equipment are quite high relative to other commercial buildings. For these reasons, when the building was designed, the annual heating requirements were expected to be very low relative to annual cooling.

TABLE 3.1. Workroom Zone Cooling and Dehumidification Load Summary

Zone	Spector/Hillier Report, June 21, 3:00 p.m.	Liquid-Desiccant Design Calculations, July, 2:00 p.m.	DOE-2 Model Predictions, July 19, 11:00 p.m.
AC-3	42.5 tons	74.6 tons (40.5 #/hr)	35.56 tons (119.5 #/hr)
AC-4	43.5 tons	41.7 tons (92.1 #/hr)	24.94 tons (118.5 #/hr)
AC-5	36.4 tons	42.2 tons (26.5 #/hr)	21.6 tons (59.7 #/hr)
AC-6	52.0 tons	50.8 tons (63.9 #/hr)	17.0 tons (82.4 #/hr)
AC-7	27.6 tons	28.0 tons (14.9 #/hr)	11.26 tons (58.8 #/hr)
AC-8	48.1 tons	59.6 tons (77.4 #/hr)	35.89 tons (133.7 #/hr)
AC-9	33.4 tons	31.9 tons (38.7 #/hr)	44.9 tons (154.9 #/hr)
	283.5 tons	329.0 tons (353.9 #/hr)	191.2 tons (727.5 #/hr)

(a) The discrepancies in total (sensible plus latent) loads raised a separate issue with regard to the conditioners' ability to meet the higher latent loads while maintaining space dry bulb temperatures at the design setpoint. However, conditioner performance calculations showing that this is not an issue are discussed in Section 5.1.

The DOE-2 building load model confirmed this design expectation, predicting a peak plant heating load of 540,000 Btuh, and an annual plant heating load of 375 MBtu, which is only about 7% of the DOE-2 predicted seasonal plant cooling load. A margin of conservatism is clearly embodied in the original design of the MPF heating plant (and verbally confirmed by Wilke); the heating system was sized based on a peak plant heating load of 2,350,000 Btuh (300 gpm and 15°F temperature difference).

3.2 HVAC SYSTEM MODEL

Two HVAC system models were developed as a part of this assessment; a ATES/TEG model based on the actual design at the MPF and a baseline HVAC system, which is based on conventional 1982 HVAC technologies[a]. In Section 3.2.1, the sources of data that supported development of these systems models are described. Section 3.2.2 focuses on development of the baseline model. Sections 3.2.3 and 3.2.4 describe the liquid-desiccant system and ATES/HVAC integrated system simulators. A comparison of the predicted energy consumptions of the two models is presented in Section 3.2.5.

3.2.1 Data Collection

The majority of data used to generate the HVAC system models were, like the building load model data, provided by Douglas A. Wilke. The data sources include:

- Systems Operational Data (described in Section 3.1.1)
- "U.S. Postal System Uses Thermal Energy" (described in Section 3.1.1)
- Revised design criteria dated 4/27/84 (described in Section 3.1.1)
- Fan schedules (described in Section 3.1.1)

(a) Life-cycle cost of an alternate baseline system that provides better air filtration and humidity control but uses conventional chillers was estimated as well. However it was not necessary to model this system because its performance could be determined by hand calculation based on the simulated performance of the baseline system.

- System flow diagram - This drawing schematically depicts the water and desiccant piping circuits at the MPF, and was useful for both development of the system models described in this section, as well as for the economic assessment (Section 4) and desiccant system operational assessment (Section 5).

- Well operation logs - The USPS well operation logs for the spring and summer of 1992 were used to help calibrate the PNL aquifer model. These logs included flow rates, temperatures, and operational comments on the aquifer system.

- Pump schedule - This schedule was apparently the original schedule included with the plans and specifications for the MPF.

- Equipment notes - These notes included HVAC equipment performance specifications for (among other things) the propane boilers, closed-circuit cooler, and the heat recovery chillers.

- DDC description of system operation - These notes described the direct digital control (DDC) programming required for the different subsystems in the HVAC plant.

- Closed-circuit cooler operation logs - These logs tracked air and water temperature, water flow rates, and air relative humidity during the 1991-1992 winter charging of the cold wells using the closed-circuit cooler on the roof of the MPF. Together with the "well operation logs", this data provided a complete historical profile of the ATES system performance to assist in development of the PNL aquifer model.

In addition to the above data from Wilke, Niagara Blower furnished excerpts from their liquid-desiccant system design manuals and fan curves to support development of the desiccant system energy performance model created as part of the ATES/TEG HVAC system. These data also supported the desiccant system operational assessment (Section 5) and economic assessment (Section 4).

3.2.2 Baseline Model Development

The "baseline" HVAC system model was developed to estimate the annual energy use for a conventional HVAC system for comparison to the ATES/TEG HVAC system at MPF. Whereas the model of the system installed at MPF (Section 3.2.4) was based on the original design with refinements to approach "as-built" and "as-operated" conditions, the baseline system was a "paper" design.

The baseline model was, however, constrained by the following considerations:

- The baseline followed conventional system design practices of the early 1980s, the same time the MPF was designed. This vintage is reflected in the equipment efficiencies, as well as in how the equipment and systems are controlled.

- Sizing of the baseline equipment agreed with the original design calculations rather than the DOE-2 building load model. This was done primarily for the economic assessment (Section 4) because it was felt this yielded the fairest comparison between the baseline and ATES/TEG systems, even though it resulted in an oversized system.

Because the baseline HVAC system model was to use conventional technologies, it could be modeled using existing component models in the DOE-2 Systems and Plants programs. The baseline model was developed in two steps. Equipment and controls were first specified for the system subject to the constraints listed above. The DOE-2 Systems input deck was then prepared based on the model system specification.

Note that for the assessment, only differences between the baseline system and ATES/TEG system needed to be considered. This resulted in a more meaningful assessment because those HVAC subsystems and loads that could be treated as identical (e.g., packaged units AC-10, AC-11, and AC-12) were left out of the comparative energy and cost studies[a].

Because the existing ATES/TEG system has central chilled and hot water loops, a central system using a single electrically driven centrifugal water chiller and modular propane boilers was selected for the baseline system's heating and cooling plant, with summer heat rejection accomplished through an evaporative cooling tower.

The water chiller was sized for 343 tons refrigeration cooling, based on the original design chilled water flow of 824 gpm and a 10°F temperature difference. The chiller was assumed to produce 44°F chilled water, as opposed to the 50°F water design temperature used in the ATES/TEG system. These lower

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(a) Units AC-10, AC-11, and AC-12 and the zones they serve were included in the DOE-2 system model for completeness. The simulation model characteristics were based on existing plans and specifications. Because these units are independent from the central plant, they will not be discussed further in the report.

chilled water temperatures are required to achieve dehumidification of supply air in the air-handling units using standard chilled-water coils (described later). The chilled water flow rate was assumed to be constant, and driven by a centrifugal pump operating against 100-ft pumping head. The chiller efficiency was assumed to be 0.70 kW/ton at full load. The condensing water side is sized based on 3 gpm/ton, with a water temperature rise of between 85°F and 95°F at design conditions.

To reflect conventional design practices for a central chilled water plant, an evaporative cooling tower was chosen for the baseline model, rather than a closed-circuit cooler.[a] The evaporative cooling tower was assumed to have three cells, and was sized for a heat rejection rate based on the condensing water requirements of the water chiller at peak cooling load. Partload operation was accomplished by cycling fans in individual cells, and the minimum cold water temperature leaving the tower was specified as 75°F. The water piping loop interconnecting the tower and chiller's water-cooled condenser was serviced by a single pump operating at 100-ft pumping head.

The baseline system uses the modular propane boilers in the conventional function of generating hot water for space heating only. The ATES/TEG system presents a sharp contrast; the aquifer serves as a heat source for the chillers, when operated as heat pumps, and the propane boilers are used primarily for regeneration of dilute TEG during the summer cooling season. The boilers in the baseline system were sized in accordance with the DOE-2 calculated heating load. The hot water supply pump was, however, sized for the full original design flow rate (300 gpm) operating against a 100-ft head.

The baseline model's air system includes all units served by the central heating and cooling plant (described previously). These include units AC-1, AC-1A, AC-1B, and AC-2 serving the administrative area, lockers, and

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(a) Environmental restrictions on aquifer-injected water require that a closed-loop cooling tower be used for the ATES/desiccant system. All cooling towers have environmental impacts of some sort but not all are subject to EPA regulations and the cost of satisfying the regulations, when applicable, are generally negligible for conventional cooling systems.

cafeteria; and AC-3 through AC-9 serving the workroom and medical unit. The non-workroom units were assumed unchanged in the baseline system performance model from what is actually installed at the MPF; i.e., air-handling units without the liquid-desiccant technology. Thus, for the comparative analysis, only their contribution to hourly plant heating and cooling loads and water pumping head were considered.

The major difference between the air-side portions of the baseline and ATES/TEG systems was that, in the baseline system, standard central-station air-handling units replaced liquid-desiccant conditioners AC-3 through AC-9 of the ATES/TEG system. This decision is consistent with the objectives of the technology assessment, i.e., to compare the ATES/TEG system with a conventional HVAC system. From the standpoint of simple heating and cooling of the space, there should be no loss in capability in choosing one over the other. However, the liquid-desiccant system does potentially provide some additional capabilities, the benefits of which are difficult to quantitatively assess, including:

- The supply air humidity levels can be precisely controlled throughout the year. The baseline, by contrast, can dehumidify the air adequately for comfort, but is not designed for precise dewpoint control.
- Air quality may be enhanced through the use of the liquid-desiccant spray as an air scrubber to remove gaseous pollutants and particulate matter, and as a biocide capable of killing some airborne organisms, if the system is operated at high enough liquid-desiccant concentrations.

An alternate baseline system design, which provides air filtration and dewpoint control qualities comparable to those provided by the ATES/TEG system, was developed by adding HEPA filters and desiccant wheel dehumidifiers to the standard baseline system. The air-handler flow rates are the same as provided by the baseline system but the supply fan motors are assumed to be 50% larger to account for the additional pressure drop across the HEPA filters.

The baseline chilled water system was assumed to use 44°F chilled water to ensure adequate dehumidification of supply air. This results in a typical supply air temperature of around 55°F, compared to 63.5°F design for the ATES/TEG system. Therefore, for the same total cooling capacity, the conventional system can use lower air flow rates. Calculations based on the

original liquid-desiccant system design cooling loads were used to determine the air flow rates required for the two systems. These results are summarized in Table 3.2 for peak operating air flow rates.

Because the air flow rates assumed for the baseline units are different from the air flow rates assumed for the ATES/TEG units used to condition the workroom, an estimate of the fan energy consumption for both systems was required. As designed, the supply and return fans are intended to operate at three speeds, with speed reduction control based on the current activity level of the zone being served (i.e., either heavy, medium, or light). A three-speed fan was not available as a control option in the DOE-2 system model. For this reason, the baseline DOE-2 system model used for AC-3 through AC-9 was specified to use variable air volume space temperature control with a continuously variable-speed control for fan modulation. Using this assumption, total hourly heating and cooling loads needed by both the baseline and ATES/TEG system plant models could be calculated by DOE-2.

The energy consumption of the fans was calculated separately on a spreadsheet using the operation schedules defined in the "Systems Operational Data". For these calculations, the measured fan power draw listed for each fan speed for each of the units was taken directly from the "Systems Operational Data". For the baseline, these power draws were reduced to account for the lower air flow rates by using the fan laws. The duct system,

TABLE 3.2. Comparison of Workroom Supply Air Flows

Zone	Baseline Unit, cfm	ATES/TEG Unit, cfm
AC-3	24,100	39,400
AC-4	21,230	35,400
AC-5	14,170	23,600
AC-6	16,500	27,400
AC-7	10,710	19,500
AC-8	15,630	27,400
AC-9	18,790	32,400

however, was assumed unchanged from the ATES/TEG design, even though it likely would be smaller in a new design. Also, the fan laws were applied by assuming that the pressure drop characteristics of the air-handling units used in the conventional system and the liquid-desiccant system are the same[a].

The alternate baseline system uses the same air handler flow rates as the baseline but uses 50% more fan energy because of an additional 1.5-in. pressure drop across the HEPA filters. Additional fan energy required to operate the desiccant-wheel dehumidifiers' variable-speed fans is conservatively assumed to be directly proportional to the moisture load.

3.2.3 Liquid-Desiccant System Model Development

To support the technical assessment, an energy model for the liquid-desiccant system was developed, and then integrated into the complete model for the ATES/TEG system. The desiccant system energy model was developed to estimate the hourly and annual energy consumption of the desiccant system, if operated as originally designed, to dehumidify the workroom throughout the cooling season. To drive the model, the hourly latent loads for the workroom predicted by the DOE-2 System program (Section 3.1.3) were used.

It should be noted that the liquid-desiccant system can also be operated to humidify supply air in the winter without use of the concentrator. However, because performance data and system control and operational details needed to estimate conditioner hot water requirements and liquid-desiccant loss rates were unavailable, humidification was not included in the liquid-desiccant system energy model[b].

(a) The fan power savings attributable to the larger-than-standard-practice baseline duct system are more than offset by the first-cost savings for the smaller duct system that was not credited to the baseline system in the life-cycle cost comparison. The ATES/TEG system is therefore not unfairly penalized by the fan power assumptions.

(b) The heat required for humidification using the desiccant system is available at a much higher COP than in conventional humidifiers. It uses the condenser water from heat recovery chillers; the chillers use the warm well water on the low (evaporative) side as their heat source. This is, however, probably more than offset by the predicted high parasitic losses associated with operating the desiccant system and chiller water pumps (see Section 3.2.5.2).

The model tracks the energy consumption of each energy-consuming component in the liquid-desiccant system. These included all liquid-desiccant system pumps, the hot water and reflux water pumps, the concentrator fans, and the modular propane boilers. The operating kW power data for each of the electrically driven components was obtained either from the "Systems Operational Data" tables provided by Wilke or (where available) the metered data collected by PNL as part of the desiccant system operational assessment (Section 5).

A general, rigorous capacity model for a liquid-desiccant concentrator of the type used at the MPF must account for entering air humidity, air flow rate, hot water temperature and flow, reflux water temperature and flow, and desiccant concentration. It is not possible to build such a model using PNL's monitored test data (Section 5.1) and Niagara Blower's existing performance data because these data represent only two operating conditions. It was hoped that Niagara Blower could provide some new test data on a concentrator currently being tested at their laboratory so that such a model could be created. However, the new test data were unavailable at the time of this writing. Therefore, in the model of the concentrator, the capacity was held constant at its nominal rating of 500 lb/hr. Based on the original design documents, the heat input (i.e., neglecting parasitic pump and fan energy) required to achieve this capacity corresponds to a thermal coefficient of performance (COP) of 0.26. This COP and an assumed boiler efficiency of 0.7 provided a simple model for calculating propane use associated with operation of the concentrator.

The hourly duty fraction for the concentrator was used as the basis for calculating hourly energy consumption of the system, and was taken to be the ratio of latent load, for the hour in question, to concentrator nominal rating. The DOE-2 building load model predicted peak latent loads that exceeded the concentrator's nominal rating. The excess latent load during those simulation hours (over 500 lb/hr) was carried over to the next hour, and represents dilution of the liquid desiccant. In the physical system, this corresponds to an increased liquid volume in the storage tanks. Accumulation of excess latent load continues in the simulation until space latent loads drops below 500 lb/hr for a given hour, at which time the concentrator

continues to operate at 100% capacity until all of the accumulated dilute desiccant solution has been processed by the concentrator. The 500 lb/hr capacity limit, therefore, has no significant effect on predicted long-term (e.g., monthly) energy use.

Niagara Blower did provide detailed data on their conditioners. The data included tables and curves that are normally used for selecting conditioners, but can also be used for predicting exit air conditions given a known liquid-desiccant concentration, water flow rate, and chilled water entering temperature and flow rate. A model was developed from these data to support analyses of conditioner performance (Section 5.1). However, for the energy model, a detailed performance calculation was unnecessary because, as currently applied at the MPF, energy consumption by components associated with a conditioner (fan, spray, and return pump) is not dependent on the time-varying capacity of the conditioner. This is because:

- Conditioner fan and spray pumps are operated continuously during the summer cooling season regardless of moisture removal (hourly duty fraction = 1.0).
- The conditioner desiccant return pump cycles periodically to send dilute desiccant back to the concentrator storage tanks, and is, therefore, simply equal to the total rate of moisture removal (which can easily be calculated) and was found to be far below the pump's nominal capacity.

TEG losses were assumed to be 0.2% of conditioner capacity (i.e., 1 lb/500 lb). This is a rule-of-thumb used by Niagara Blower. Reported system losses as recorded in the desiccant system operation log kept by the USPS for the summer of 1992 confirmed this to be a reasonable approximation (Section 5.3.3).

Adjustments to the hourly plant cooling loads are also made by the liquid-desiccant system energy model to account for the heat of solution and circulation heat rise of the liquid desiccant. At the peak latent load of 742 lb/hr, this amounts to an additional 25 tons of plant cooling load.

3.2.4 ATES/HVAC Integration Model

The DOE-2 Loads program, described previously, was used to estimate hourly heating and cooling loads for the MPF. The baseline system was then modeled using the DOE-2 System and Plant programs to estimate annual energy

use for the comparative assessment. The liquid-desiccant system and the ATES system are, however, unconventional technologies combined at the MPF in a unique way, and DOE-2 does not have the capability to model such a system. Therefore, a separate model was developed. As discussed in Section 2.4, this integrated modeling presents special problems because the systems operate on very different time scales.

In Section 3.2.4.1, the well/aquifer computational model used for this analysis is described. Following that, the methodology used to integrate the aquifer model with the HVAC system is reviewed in Section 3.2.4.2. Finally, in Section 3.2.4.3, the major modeling assumptions used to develop this integrated model, including a review of HVAC system components and control elements, are discussed.

3.2.4.1 Aquifer Model

The Aquifer Thermal Energy Storage System Simulator (ATESSS) (Vail, Kannberg, and Kincaid 1985) was selected to model the well/aquifer system at Melville. The ATESSS code was developed by staff at Pacific Northwest Laboratory to facilitate evaluation and design of ATES systems. The numerical formulation used in ATESSS decouples the fluid and energy transport equations by assuming that the aquifer is perfectly stratified. This assumption is acceptable except in high-temperature (> 175°F) conditions. Decoupling the fluid and energy transport implies that buoyancy and viscous fingering effects are not significant. These assumptions are valid for the Melville site. Further simplification of the aquifer system results from considering only vertical conduction and horizontal advection. The resulting numerical scheme is unconditionally stable and very fast.

Well log data from 1992 was reviewed by a PNL staff geohydrologist to develop a description of the aquifer stratigraphy. These data were used to define the vertical discretization used in the simulations. The reference regional aquifer temperature used was 57.9°F.

3.2.4.2 Integration Model Methodology

The integrated model of the MPF HVAC and aquifer systems is driven by the precalculated DOE-2 hourly plant loads, weather data, and building design control criteria. In the simulation, this information is used at each ATESSS

computational time step to determine the required aquifer operating mode (i.e., recovery or injection). This mode is then assumed constant over the ATESSS time step (approximately 12 hr). Water flow requirements to the building are determined hourly, as required to meet the precalculated cooling or heating loads. In winter the injected water temperature is determined by the cold water temperatures that the closed-circuit cooler can produce. The cold water temperature is a function of water flow rate and outside air temperature and humidity. Return water temperatures to the ATES model are estimated for each hour based on an energy balance that accounts for heating and cooling loads, the performance of HVAC system component models[a], and well flow rate and supply temperature. The hourly flow rates and return temperatures are averaged over the ATESSS current time step to obtain the average constant return temperature and flow rate for the time step. The average temperature and flow rate are then used by the ATESSS program in calculations to update the aquifer computational grid.

Energy consumption is computed each hour for the energy-consuming components in the system. These components are assumed to operate as required by the building control system to meet cooling and heating demands. Hourly load/operation fractions are computed, and used in conjunction with equipment performance data to determine hourly electricity use. Fan energy use for the liquid-desiccant conditioners AC-3 through AC-9 are, as mentioned earlier in Section 3.2.2, computed separately on a spreadsheet based on workroom activity schedules. Predicted latent heat loads in the workroom are used by the liquid-desiccant system energy model to estimate hourly electrical and propane use. Adjustments to the hourly plant cooling loads are also made by the desiccant energy model to account for the heat of solution and the heat rise of the liquid desiccant as a result of circulation.

(a) In summer cooling mode, the cold well water is used directly to cool the air, and to absorb the latent heat load of the water absorbed by the liquid desiccant. This return water may also be used (as required) by the concentrator reflux coil and for water chiller heat rejection before being reinjected into the warm wells.

3.2.4.3 Modeling Assumptions

The HVAC system model developed for the MPF was essentially an energy rate model, as opposed to a dynamic temperature-control model used by other building simulation tools. However, the HVAC system model, as described above, is integrated with the ATES computational model, and estimates the required aquifer supply flow rates and return temperatures to the wells from the building. The major assumptions in the modeling approach are reviewed here.

HVAC Component Models

Individual component models are described below:

- Heat recovery chillers - The (two) heat recovery chillers were modeled based on performance specifications listed in the "equipment notes" provided by Wilke. Maximum summer cooling capacity for each chiller was assumed to be 100 tons, independent of operating temperatures, at a full load compressor power consumption of 71.4 kW. The chiller partload model assumed compressor power to vary linearly with load. This assumption could result in a small over- or under-estimation of annual chiller energy input, depending on the load distribution and actual partload chiller characteristic.[a] The maximum heat recovery capacity during winter was assumed to be 1,136,000 Btuh. Low-side water cooling at this capacity was 842,000 Btuh, implying a heat rejection fraction (HRF) of 1.35. This HRF was assumed to be constant for partload calculations, as was the listed heating season COP of 3.5, used to compute compressor kW in winter.

- Modular boilers - The (two) modular propane boilers used for regeneration of the liquid desiccant were assumed to have a total heating output of 1,440,000 Btuh each, in accordance with the "equipment notes". Based on verbal conversations with Wilke, a boiler efficiency of 70% was used rather than the originally specified 80%.

- Closed-circuit cooler - Detailed logs of the closed-circuit cooler operation during the winter of 1991-1992 were kept by the USPS at Wilke's direction. These logs included water flow rates and yinlet/outlet temperatures, along with outside air temperatures and

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(a) The partload characteristic is a second-order effect for this system (as it is for most situations in which the chiller is of conventional design and properly sized) because partload efficiencies are typically higher than full-load efficiency by 10 to 20% in intermediate modes and are correspondingly lower in the lowest capacity mode resulting in a integrated partload COP that is typically within 5% of the full-load COP.

relative humidities, thereby providing a fairly complete picture of actual tower performance. This information was correlated using a linear regression to develop a tower performance curve that could be used for predicting cold water temperatures during winter charging of the aquifer. Electrical power consumption rates for the three tower fans were taken from the "Systems Operational Data" tables.

- Pumps - Pump energy consumption rates were obtained from the "Systems Operational Data" tables. Water flow rates for the well system, however, were based on average measured flow rates as reported on USPS operational logs taken in 1992 rather than the original design specifications. This was done to better represent the ATES/TEG system, "as-built". During summer cooling, average flow rates for a single well pump plus a main supply pump were reported (based on logs) to be about 380 gpm on average (design is 412 gpm). Starting a second well pump in summer increases this flow to about 500 gpm, and starting a second main supply pump increases flow further to about 700 gpm (design is 824 gpm). By contrast, in winter, only a single well pump is operated most of the time (heating is negligible) to enable the closed-circuit cooler to make sufficiently cold water for later chill recovery. Based on the logs, the average flow with a single warm water well operating was about 185 gpm. Starting a main supply pump boosts this flow further, to about 300 gpm on average.

ATES/HVAC System Control Model

The simulated control sequence for the heating and cooling plant at the MPF combined automated and manual control decisions in a way that essentially followed the original design. Some simple modifications were made to improve the seasonal recovery efficiency of the aquifer system. The intent was not to develop an optimized system control model, but rather an effective control model constrained to the "as-built" system currently in operatior.

- Seasonal aquifer mode changes - Changing the operating mode of the ATES system from winter recharge to summer recovery is essentially a manual decision, made based on the weather (and thus heating or cooling needs of the building). Therefore, the model made use of its "knowledge" of the weather data. During the summer cooling season (May 15 through October 15), the ATES system was assumed to always be in recovery mode. During the winter (November 15 through April 15), the ATES system was assumed to always be in charging mode. To decide changeover during the interim seasons (defined as April 15 through May 15 and October 15 through November 15), outdoor air temperatures were looked at for the duration of the current ATES time step. If the outdoor air temperature was greater than 65°F the majority of the time, it was assumed the ATES system would be operated in recovery mode. Otherwise, the ATES system was operated in winter mode. A temperature of 65°F corresponds in the original "DDC sequence of operation" design documentation to the summer mode changeover temperature.

- Summer control mode - If weather data indicates the hourly outside air temperature equals or exceeds 65°F, a cold well and a main supply pump are started, along with the liquid-desiccant concentrator reflux pump. If the supply water temperature from the well is less than 50°F and the measured return water temperature from the building air conditioning units is greater than 60°F[a], a second well is started. If the return water temperature is still greater than 60°F, a second main supply pump is started. If the return water temperature is still above 60°F, a chiller is used to further cool the supply chilled water to whatever temperature is required to maintain 60°F return. If the supply water temperature from the well is greater than 50°F, a chiller is turned on to cool the water down to 50°F. If the return temperature is greater than 60°F, a second well is started, with the chiller continuing to operate. If the return water temperature is still above 60°F, the second main supply pump is started. Finally, if return water temperatures are still greater than 60°F, the second chiller is started. Whenever the simulator indicates the building is in summer mode, the dedicated chilled water pumps at the individual workroom conditioner units are cycled as necessary to meet the cooling load.

- Winter control mode - If weather data indicates the hourly outside air temperature is equal or less than 50°F, the hot water supply pump is started. A temperature of 50°F is the winter mode changeover temperature cited in the "DDC description of system operation". If hourly plant loads show there is a heating load, one of the chiller's pumps starts, along with a single well pump and the main supply pump. The chiller operates in heat pump heating mode as required to meet the heating load. Also, whenever the simulator indicates that the building is in winter mode, the dedicated hot water pumps for individual conditioner units are cycled on whenever there is a heating load indicated for the specific unit. Finally, the closed-circuit cooling tower model estimates the exiting cold water it could produce if operated for the current hour. If temperatures lower than 50°F can be produced, the closed-circuit cooler pumps are started and (provided there is no heating load) a well pump is started. A secondary check also estimates the cold water temperature that could be produced by the closed-circuit cooler with the higher supply flow rate (300 gpm versus 185 gpm) achievable by running a main supply pump. If the temperature is less than 50°F and the total net cooling of the water is enough to offset the cost of running the additional pump, the main well supply pump is started.

(a) Because the design temperature rise in the chilled water system is 10°F, if 50°F water (or less) is supplied to the conditioners, the cooling load is satisfied, as long as the return temperature doesn't exceed 60°F.

3.2.5 Technical Assessment Results

The simulation of the ATES/TEG system model was run for a total of 4 years simulated time, using the DOE-2 predicted loads each year. Output performance parameters for the wells, as well as HVAC system energy consumption, were found to be essentially steady after the third year of the simulation. Results reported here are for the fourth year of the simulation.

The predicted thermal performance of the ATES system in the integrated model is presented in Section 3.2.5.1. Following that, the estimates of annual energy consumption for the baseline and ATES/TEG systems are presented and discussed at length in Section 3.2.5.2. It must be emphasized that the numbers presented are simulated, and as such should not be directly compared to past building performance measures (e.g., propane use) because of the many assumptions, simplifications, and idealizations made in developing the models to approach the "as-built" design. The assessment approach, however, provides a reasonable basis for comparisons between the ATES/TEG system and the baseline, and gives a fair indication of which portions of the system are using the most energy.

3.2.5.1 Aquifer System Thermal Performance

The total volume of cold water injected during the winter aquifer recharge cycle in the fourth year was 65,000,000 gal. The total energy removed from the water by the closed-circuit tower was 6890 MBtu, or 574,000 ton-hr. The effective stored chill, based on the temperature difference between the injected water and the regional aquifer temperature (assumed to be 57.9°F) was 5890 MBtu, or 491,000 ton-hr. The total volume of water recovered during the summer cooling season was predicted to be 74,000,000 gal. The recovered chill, relative again to 57.9°F, totalled 4370 MBtu, or 364,000 ton-hr, for a net energy recovery efficiency of 4370/5890 = 74%. The building's annual sensible cooling load, as reported in Section 3.1.3, was 5485 MBtu. The calculated cooling load associated with operating the liquid-desiccant system, including heat of solution and circulation heat gain, adds an additional 347 MBtu to this total, for a total annual space cooling load of 5485 + 347 = 5832 MBtu. Another useful measure of overall system performance

is the TES load fraction: the simulation results indicate that recovered chill represents about 4370/5832 = 75% of the total space cooling load.

The high chill recovery efficiency (74%) predicted by the ATESSS model for the Melville ATES system was expected because the model was calibrated using the well log data collected by the USPS and Wilke at the site over the past year. The well logs suggest that the effect of local aquifer flow is negligible even though there is a general regional flow of 1.0 to 1.5 ft/day.

3.2.5.2 Energy Consumption: Baseline Versus ATES/TEG

Predicted annual and monthly energy consumption and demand numbers are shown in Tables 3.3, 3.4, and 3.5 for the baseline, alternate baseline, and ATES/TEG HVAC systems. The tables show total annual and monthly electrical and propane consumption and a breakdown of total electrical consumption for individual components and subsystems. In addition, monthly demand (peak hourly load) numbers for "on-peak" and "intermediate" demand charge time periods (see Table 4.2 in Section 4.1), as defined by the Long Island Lighting Company, are shown along with the total annual energy consumption during each of these charge periods.

Total annual energy consumption for the baseline system is predicted to be 1159 MWh and 6850 gal of propane for heating. Total annual energy consumption for the alternate baseline system is predicted to be 1375 MWh, 6850 gal of propane for heating, and 20,249 gal of propane for desiccant-wheel regeneration. By comparison, the predicted annual energy consumption for the ATES/TEG system is 1545 MWh, with 51,918 gal of propane used for liquid-desiccant regeneration.

Assuming 95,000 Btu/gal for propane, the total predicted site energy use per unit of conditioned floor area (227,125 ft^2) for the baseline system is 17,700 Btu/ft^2, while for the ATES/TEG system it is 45,000 Btu/ft^2. Considering only the workroom floor area (165,700 ft^2), these numbers are 27,300 Btu/ft^2 for the baseline system and 61,000 Btu/ft^2 for the ATES/TEG system. When designed, in 1982, the goal was to reduce energy consumption below 88,000 Btu/ft^2 for conditioned space.

TABLE 3.3. Baseline HVAC System Predicted Monthly and Annual Energy Performance

Month	Plant, kWh	Demand Peak, kW On-Peak	Demand Peak, kW Intermediate	Propane, gal
JAN	59,372	0	143	1987
FEB	54,377	0	144	1683
MAR	67,477	0	172	1064
APR	77,967	0	213	191
MAY	101,443	0	201	2
JUN	131,578	306	281	0
JUL	162,351	328	315	0
AUG	154,438	309	292	0
SEP	120,430	311	249	0
OCT	94,416	0	228	28
NOV	72,518	0	191	439
DEC	63,050	0	144	1456

Annual Plant, kWh	Demand Period, kWh On-Peak	Demand Period, kWh Intermediate	Demand Period, kWh Off-Peak	Propane, gal
1,159,417	346,679	424,011	388,727	6,850

Annual Component Energy Consumption: Baseline System, kWh

Chillers	313,950
Tower and tower pump	154,269
Hot water pump	58,003
Cold water pump	96,043
Workroom fans	537,152

TABLE 3.4. Alternate Baseline HVAC System Predicted Monthly and Annual Energy Performance

Month	Plant, kWh	Demand Peak, kW[a] On-Peak	Intermediate	Propane, gal
JAN	76,644	0	184	1987
FEB	69,988	0	185	1683
MAR	84,959	0	217	1064
APR	95,188	0	260	377
MAY	119,727	0	237	793
JUN	150,269	349	321	3504
JUL	182,480	369	354	6549
AUG	174,332	349	330	5591
SEP	138,788	358	287	2744
OCT	89,653	0	272	914
NOV	80,428	0	236	439
DEC	63,050	0	184	1456

Annual Plant, kWh	Time-of-Use Period, kWh[a] On-Peak	Intermediate	Off-Peak	Propane, gal
1,375,100	411,169	502,874	461,071	27,099

Annual Component Energy Consumption: Alternate Baseline System, kWh

Chillers	299,027
Tower and tower pump	146,936
Hot water pump	58,003
Cold water pump	96,043
Workroom fans	738,591
Dehumidifier fans	36,500

(a) Demand numbers and time-of-use distributions are based on the load factors and time-of-use distributions of the baseline system.

TABLE 3.5. ATES/TEG HVAC System Predicted Monthly and Annual Energy Performance

Month	Plant, kWh	Demand Peak, kW On-Peak	Demand Peak, kW Intermediate	Propane, gal
JAN	151,836	0	273	0
FEB	138,938	0	276	0
MAR	140,167	0	254	0
APR	110,780	0	241	476
MAY	95,103	0	231	2,027
JUN	112,825	226	226	8,983
JUL	132,496	228	228	16,790
AUG	152,649	310	310	14,335
SEP	138,533	351	354	7,036
OCT	106,540	0	319	2,271
NOV	115,775	0	246	0
DEC	149,445	0	265	0

Annual Plant, kWh	Demand Period, kWh On-Peak	Demand Period, kWh Intermediate	Demand Period, kWh Off-Peak	Propane, gal
1,544,979	348,674	709,399	486,906	51,918

Annual Component Energy Consumption: ATES/TEG, kWh	
Desiccant system	
Concentrator fan	8,630
Hot water pump	3,215
Desiccant pumps	86,258
Reflux pump	1,926
Workroom fans	940,260
Chiller	
Compressor	93,947
Cold water pump	21,206
Condenser pump	16,613
Closed-circuit cooler	
Fans	72,153
Pumps	94,863
Other plant pumps	
Well	86,552
Main supply	79,125
Conditioner chilled water	9,123
Conditioner hot water	973
Main hot water	30,125

The greatest single contributor to the increased electrical energy use of the ATES/TEG system relative to the baseline is the predicted energy consumed by the fans (940.3 MWh versus 537.2 MWh). Supply and return fan air flow rates vary with workroom activity levels in both the ATES/TEG system and the baseline cases, yet because the baseline system is designed to supply air at around 55°F versus the design 63.5°F in the ATES/TEG system, the baseline total air flow can be reduced and still provide the same cooling effect. The lower aggregate fan horsepower required to meet the reduced total supply air flow (outside air ventilation rates are not changed) results in the much lower annual energy consumption. Note that the baseline system would have used about the same fan energy as the ATES/TEG system if it had been required, by design criteria, to match the ATES/TEG system's high air circulation rates for the purpose of dust control.

Not counting the electrical consumption of conditioner fans, the annual electrical energy of the ATES/TEG heating and cooling plant is predicted to be 604.8 MWh, or about 3% lower than the baseline's 622.3 MWh. This modest net reduction in plant electrical energy use clearly comes at a high price because of conditioner fan energy and propane use. However, by focusing on this portion of the total energy bill, several things can be learned about the operating performance of the ATES and liquid-desiccant systems.

First, the ATES/TEG system does use far less mechanical refrigeration than the baseline system. The centrifugal water chiller in the baseline system is predicted to use 314 MWh annually, while the ATES/TEG heat recovery chillers only use 94 MWh for combined summer cooling and winter heating duty.

The standard baseline system evaporative cooling tower fans and the condenser water pump that moves water between the tower and the chiller use a total of 154.3 MWh to meet a annual cooling season chiller heat rejection load of 6556.6 MBtu. This is very comparable to the energy use associated with operating the closed-circuit cooler and spray and cooler water pumps of the ATES system for winter recharge of the cold wells (167 MWh). Such a result is not unexpected because the ATES system closed-circuit cooler is essentially performing the same function (heat rejection of the cooling load) as the evaporative cooling tower in the baseline.

The "other parasitic" electrical costs appear to be the biggest contributors (after conditioner fan energy) to the high predicted annual electrical consumption of the ATES/TEG HVAC system relative to the baseline. Water pumping energy (not including the cooling tower pump; see above) used annually by the baseline system is 154.1 MWh. Water pumping energy used annually by the ATES/TEG system (neglecting the closed-circuit cooler pumps) is 248.9 MWh. Adding the electrical energy consumed by the liquid-desiccant pumps and concentrator fan (94.9 MWh) results in an annual total of 343.8 MWh.

The largest energy expenses associated with operating the liquid-desiccant system are predicted to be for boiler fuel and liquid-desiccant pump energy. The predicted seasonal COP associated with operating the liquid-desiccant system, including all parasitic pumping and fan losses, is 0.171. Based on the assumed rate of 0.2% TEG loss relative to water moisture removal, the predicted annual TEG consumption is less than 200 gal[a]. Finally, in addition to these costs, the liquid-desiccant system also increases the annual plant cooling load by about 6.8%, from 5485 MBtu to 5832 MBtu.

The net COP associated with seasonally operating the ATES for both heating and cooling, including heat recovery chillers, closed-circuit cooler, and all associated water pumps, and excluding the liquid-desiccant system heat gains, energy consumption, and conditioner fans, is 3.4.

Although the primary goal of the technical assessment was to compare the energy performance of the ATES/TEG HVAC system to that of the baseline HVAC system, several potential operational or control improvements that could improve overall performance were identified in the process of running the simulations:

- Adjust conditioner fan speeds to reflect actual space loads. The DOE-2 building load model predicts that zone cooling loads in the workroom are considerably less than predicted in the original design, even though latent loads are higher. At these lower loads the conditioners might be

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(a) This estimate pertains to TEG losses from a properly controlled and operating system. Additional losses have occurred at the MPF from time-to-time when equipment failure or operator error resulted in concentrator or conditioner sumps overflowing and spilling quantities of TES solution on the floor. See Section 5.3.2.3 for a further discussion of system losses.

able to meet the loads at lower air flow rates, and possibly with less chilled water and/or lower concentrations of desiccant. Some performance analyses of conditioners based on Niagara Blower selection data are presented in Section 5.1.

- Control infiltration loads caused by shrink-wrap machines and other exhaust fans. A dedicated air supply for the machines will reduce infiltration by as much as 17,400 cfm when all three shrink-wrap machines are operating. This will reduce the moisture load on the liquid-desiccant system.

- Improve efficiency of the propane boiler from around 70% to 90% by either replacement with a new high-efficiency model or by reclaiming flue gas heat by injection into the air intake of the liquid-desiccant concentrator.

- Operate liquid-desiccant system main circulating and conditioner spray pumps only as needed (rather than continuously) or replace pump motors with variable- or two-speed pump models.

- Consider using variable-speed pumping on the well-water system to better match cooling loads (thus more effectively using limited resources of stored chill), increase the amount of chill produced by the closed-circuit cooler in winter, and decrease the total pumping energy.

4.0 LIFE-CYCLE COST ASSESSMENT

A life-cycle cost (LCC) assessment was conducted to compare the costs of owning and operating the ATES/TEG system with the two baseline HVAC technologies. Life-cycle cost is defined as the present value of all relevant costs (initial capital, annual maintenance, and annual energy costs) associated with each option. A 25-yr analysis period was used in all cases. This assessment addressed differential costs only; elements common to the three options were excluded. Unique elements resulting from differences in technology or size, were included. This approach highlights the key differences in the two systems by avoiding consideration of factors that provide no distinction.

Emphasis was placed on equitably evaluating the differences in cost among the systems rather than the absolute cost of each system. The same information sources were used for estimating the costs of common technologies such as pumps and pipes. The cost of conventional HVAC system elements was estimated from published data sources that included Means (1991), Richardson (1990), Colen (1990), Konkel (1987), Electric Power Research Institute (1988), and Gladstone, Humphreys, and Lunde (1987). Cost data were updated from the reference year for each source to 1992 dollars by applying the appropriate ratio from the Chemical Engineering Equipment Cost Index (McGraw-Hill 1992). Well cost data were provided by Delta Well and Pump, cost data for the liquid-desiccant system were provided by Niagara Blower, and cost data for the desiccant wheel units were provided by Bry-Air, Inc., and CargoCaire Engineering Corp.

4.1 COST ANALYSIS

Equipment requirements for the ATES/TEG systems and the two baseline systems are outlined in Table 4.1. This "bill of materials" was the basis for developing cost estimates. Again, HVAC system components common to the three systems (such as hot and cold water distribution piping) were excluded from the analysis.

TABLE 4.1. HVAC System Equipment Requirements

Equipment Category	Baseline System	Alternate Baseline	ATES/TEG System
Chiller	2 190-ton, centrifugal[a]	(same as baseline)	2 100-ton reciprocating with heat reclaim
Cooling tower	1020 gpm evaporative	(same as baseline)	565-gpm closed-circuit evaporative cooler
Propane water heater	2 600,000-Btuh	(same as baseline)	2 1,440,000-Btuh
Pumps	2 20-hp/condenser	(same as baseline)	2 5-hp/condenser 2 20-hp/tower supply 2 20-hp/tower spray 2 3/4-hp/reflux 2 3-hp/concentrator 6 20-hp/wells
Piping	condenser	(same as baseline)	condenser tower supply tower spray reflux concentrator wells desiccant
Air handlers	8 row, 14 fins per in. (fpi) coil; 1 row of 8-fpi hot coil; supply and return fans with variable-speed drive; Seven units in size range: 10,700 to 24,100 cfm supply and 9,200 to 22,140 cfm return	Same as baseline but with 50% larger supply fan motors and HEPA filters	8 row, 6 fpi cold coil; 1 row, 6 fpi hot coil; desiccant spray and return pumps; sprayers; supply and return fans with variable-speed drive; Seven units in size range: 19,500 to 39,400 cfm supply and 18,000 to 37,440 cfm return
Wells	not applicable	not applicable	Qty 6: 180-ft deep, 18-in. bore; 12-in. casing to 110 ft; 18-in. gravel packing, bottom 70 ft; 8-in. screen, bottom 50 ft.
Desiccant equipment	not applicable	packaged desiccant wheel dehumidifier with "regeneration energy control" by variable frequency drive of process & regeration fans and wheel indexing timer; 7 units: 1 1500 cfm, 3 2000 cfm, 3 3000 cfm	concentrator (including spray and return pumps, circulation pumps, hot and cold coils, fan, reclaimer, and demister); two storage tanks; heat exchanger

(a) At the December 2, 1992 review meeting it was agreed to size the baseline chillers so that one chiller alone could meet the cooling load at the Melville facility 90% of the time. This satisfies the USPS redundancy requirement and is consistent with the design criteria used in the design of the ATES/TEG system.

Direct installed costs (includes purchased equipment, materials, and direct installation labor) were estimated for each major component. National average data were adjusted to reflect conditions in Long Island, New York. Contingency, sales tax, and contractor overhead and profit were added to arrive at the total initial capital cost estimate. Annual non-energy operations and maintenance (O&M) costs and equipment lifetime were also estimated for each major component to complete the cost specifications required for the LCC analysis. The specific estimating approach for each major component is discussed and results presented in the following paragraphs.

Installed cost estimates for the 190-ton centrifugal chillers ranged from $390 to $420/ton. An average cost of $400/ton, or $76,000 per chiller was used in the analysis. Installed cost estimates for standard 110-ton reciprocating chillers ranged from $410 to $560/ton. Heat reclaim capabilities were estimated by Trane to add 20% to standard chiller equipment costs. An average installed cost of $560/ton or $56,000 was used in the analysis. O&M cost estimates varied considerably, ranging from $3,000 to $11,000/yr for a 100-ton reciprocating chiller, and $5,000 to $11,000/yr for a 190-ton centrifugal chiller. Average values of $7,000/unit for the reciprocating units and $7,500/unit for the centrifugal units were used. Although considerable uncertainty in the absolute level of O&M was indicated in the references, all sources agreed that reciprocating units were more expensive to operate and maintain than centrifugal units of the same size. Both units were estimated to have an expected life of 20 years, although at least one reference suggested a longer life for centrifugal units.

Installed cost estimates for a basic evaporative cooling tower ranged from $20,000 to $22,000. Chemical treatment and freeze protection equipment were estimated to cost an additional $11,000. An average installed cost of $32,000 was used in the analysis. Little data were found describing expected annual operating and maintenance costs or equipment lifetime. A general rule-of-thumb for the chemical process industry is that O&M is approximately 6% of initial capital per year for average processes, and ranges from 2 to 10% of initial capital depending on the severity of the operating condition (Ulrich 1984). An evaporative cooling tower was judged to operate under less severe

conditions than average by the chemical process industry, so annual O&M was estimated to be 4% of initial capital or $1300. The cooling tower was presumed to last at least as long as the economic life of the LCC analysis, or 25 years. The cost of the closed-circuit evaporative cooler was included with the liquid-desiccant equipment cost quoted by Niagara Blower and was not independently estimated.

Installed costs for propane water heaters were estimated to range from $3600 to $4800 for each of the six 200,000-Btuh units in the baseline system and from $4000 to $5400 for each of the twelve 240,000-Btuh units in the ATES/TEG system. Based on average costs of $4200 and $4700 per unit, respectively, the total installed cost for propane water heaters was calculated to be $25,200 for the baseline system and $56,400 for the ATES/TEG system. Annual O&M costs were estimated to be $150 for the smaller units and $175 for the larger units. The equipment was expected to last approximately 25 years.

Installed cost estimates for pumps and motors generally varied by a factor of two among the data sources consulted. The wide range can be attributed to uncertainty in the pump specifications (e.g., horizontal or vertical split casing; ANSI, American Petroleum Institute or other fabrication standards) and ancillary equipment (e.g., the amount of ancillary piping included in the installation), as well as the usual uncertainty in cost for a given specification. The average estimated cost was used for each horsepower rating. Installed pump and motor costs ranged from $1900 for a 3/4-hp unit up to $7600 for a 20-hp unit. Total installed costs for the pumps listed in Table 3.1 (not including the well pumps) were $15,200 for the baseline system and $47,600 for the ATES/TEG system. Annual pump O&M was estimated to be 6% of initial capital, which is consistent with average conditions in the chemical process industries (Ulrich 1984), where facilities operate 24 hr/day, 7 days/week. Pump life was assumed to be 20 years, the same as for the centrifugal and reciprocating chillers.

Piping costs were estimated based on the length, diameter, material, and complexity of the piping systems. Three distinct piping groups exist: 1) lengthy, low complexity buried steel piping connecting each of the wells to the mechanical room; 2) lengthy, moderate complexity copper piping connecting

the desiccant concentrator with the air handlers and other liquid-desiccant system components; and 3) short, high complexity steel piping connecting various equipment in and around the mechanical room. Well and desiccant piping lengths and diameters were based on specifications and dimensions shown on site blueprints. "Mechanical room" piping lengths were estimated based on visual inspection and judgment; diameters were calculated based on a design flow velocity of 5 ft/second.

Installed cost estimates for straight-run copper piping varied by a factor between 1.5 and 2, depending on the source. An average of the data obtained was used, resulting in an installed cost estimate of $7, $8, and $9 per ft of 1-in., 1.25-in., and 1.5-in. copper pipe. The variation in cost estimates for installed straight-run steel pipe was slightly less. Average installed costs per ft ranged from $10 for 2-in. pipe up to $53 for 10-in. pipe. Trenching, for the pipe running between the wells and the mechanical room, was estimated to cost about $3/ft. Labor for placing piping in a trench is approximately the same as for placing on hangers.

Fitting and valve costs were estimated by multiplying the straight-run piping cost by a factor that varied depending on the presumed complexity of the piping run. Empirical data describing valve and fitting requirements for a variety of piping installations (Hooper 1982) were used to help establish the following rules-of-thumb: 50% for low complexity piping; 100% for moderate complexity piping; and 200% for high complexity piping. An estimate based on an inventory of valves and fittings in the facility would be more accurate, but this degree of system specification was not available.

Total installed piping cost for the ATES/TEG system were estimated to be $240,000. Piping connecting the wells with the mechanical room accounted for about three-quarters of this total. About one-half of the remainder was for the desiccant piping. The total installed piping cost for the baseline system, which included only a relatively short line connecting the chiller condenser and cooling tower, was estimated to be about $16,000. Piping is expected to last at least 25 years and should require minimal maintenance compared to other mechanical equipment with moving parts. Annual O&M was estimated to be 2% of the initial capital cost.

Good agreement was found for conventional air handler installed costs estimated from several sources. In general, the range of estimates varied by less than a factor of 1.25 after adjusting to the same design and economic conditions. Average installed costs were about $0.81/cfm. Return air fans, required by both the baseline and ATES/TEG systems, but not included in the Niagara Blower estimate for the liquid-desiccant system air handlers, were estimated separately. Again, relatively little variation in estimated cost was found among the sources consulted, once differences in the estimating basis were accounted for. Installed return fan costs were estimated to range from $0.24/cfm at 10,000 cfm down to $0.18/cfm at 40,000 cfm.

The total installed cost for the baseline system air handlers, including return air fans, was estimated to be about $123,000. The return air fans for the ATES/TEG system were estimated to cost $36,000. Air-handler life data indicated that custom units operating indoors typically last 40 years or more, while packaged rooftop units may only last 10 to 15 years. Standard air handlers operating indoors were assumed to last 25 years. Annual O&M costs were judged to be 3% of initial capital based on a comparison of operating conditions and O&M costs already developed for the other HVAC system elements.

The total installed costs for the alternate baseline air handlers were assumed to equal the baseline air handler costs plus the incremental costs of larger supply fan motors and three-stage HEPA filters and racks. The estimated installed cost difference between 3- and 2-hp motors is $15, between 5- and 3-hp motors is $65, between 7.5- and 5-hp motors is $120, and between 10- and 7.5-hp motors is $90 (Nadel et al. 1991) for a total incremental installed motor cost of $395. Budgetary estimates received from Porter Air, Inc.[a] suggest incremental costs of $6500/10,000 cfm for filter racks and $1250/10,000 cfm HEPA filter media. The total incremental installed cost (relative to the baseline air-handlers), including installation of HEPA filter racks, filter media, and larger fan motors was, therefore, estimated to be $97,000. Additional annual O&M costs of $24,000 will be incurred for filter media and replacement labor. The installed costs of desiccant-wheel

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[a] Porter Air, Inc., Seattle, WA; personal communication with Mr. Chuck Anderson, Vice President.

dehumidifiers, including modulating controls, natural gas piping, and roof penetrations and vents for regeneration air exhaust, were estimated to be $36,900/3,000-cfm unit, $29,500/2,000-cfm unit, and $24,350/1,500-cfm unit[a]. A total of three 3000-cfm units, three 2000-cfm units and one 1500-cfm unit is required. Total materials cost for seven units is $215,350 and installation labor is an additional $8,200. Annual O&M is assumed to be of 4% of material cost, and the desiccant equipment is assumed to have a useful life of 25 years.

In general, well costs are extremely site-specific and are best estimated by consulting with local drilling contractors. Delta Well and Pump Company, the MPF well drilling contractor, reported that the cost for the wells drilled at the MPF site had been $63,500 at the time of construction. This figure included $41,000 for boring, screening, casing, gravel packing, and grouting. The pump, motor, column pipe, and drive shaft cost $13,000, and the valves and piping at the well head cost $9,500. Delta thought that changes in market conditions would probably result in the same cost being roughly applicable today. Consistent with the analysis of other system components, the well pump was assumed to last 20 years while the balance of the well (essentially piping components) was assumed to last 25 years. Annual O&M, expressed as a percent of initial capital, was assumed to be 6% for the pump, 4% for the well, and 2% for the valves and piping.

The unique nature of the liquid-desiccant dehumidification system made it critical to obtain an estimate from its vendor. Niagara Blower indicated that the equipment cost at the time of purchase (the mid-1980s) was $600,000 and that the same equipment might be purchased today for about $700,000. Equipment included in this estimate are the liquid-desiccant system air handlers, storage tanks, circulation pumps, concentrator, heat exchanger, and control panels. Also included is the closed-circuit cooling tower. Unfortunately, Niagara was unable to break the overall estimate into its component parts, which would allow direct comparison of similar components in

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(a) Bry-Air, Inc., Sunbury, OH; personal communication with Mr. Kenneth Baker. The purchase costs given by Bry-Air, Inc., were slightly lower than the costs given by the CargoCaire, Inc., distributor for comparable sized units.

the baseline system, namely the air handlers and cooling tower. The liquid-desiccant system installation cost was estimated by comparison to the installation costs for the baseline system air handlers and cooling tower. The desiccant-system air handlers (including the concentrator, which is similar in construction to an air handler) are larger than the conventional air handlers, and have more plumbing and electrical connections. Based on this comparison of requirements, installation of the desiccant system was assumed to cost twice as much as installation of the conventional air handlers. Installation of the closed-circuit cooling tower was assumed to cost the same as for the evaporative cooling tower. Thus, the total installed cost of equipment provided by Niagara Blower was calculated to be $745,000. The expected lifetime was assumed to be 25 years and annual O&M was assumed to be 4% of initial capital, which represents a composite of assumptions for similar equipment in the baseline system. In addition, TEG liquid desiccant is consumed at the rate of 200 gal/yr and is estimated to cost $15/gal.

Direct installed cost estimates prepared from national average cost data were adjusted to Long Island conditions based on construction cost indices published by Means (1991). Indices for New York and Yonkers, the two locations listed that were closest to Long Island were 126.8 and 116.7, respectively. The construction cost index applicable to the USPS project was assumed to be 115. This factor was not applied to the costs estimated by Delta Well and Pump or Niagara Blower.

In general, an allowance for contingency, sales tax, and general contractor overhead and profit must be added to the direct installed cost to arrive at the total capital investment. The contingency included here allows for unforeseen costs caused by the incomplete engineering and scope associated with a preliminary analysis of this kind. General contractor overhead items include temporary construction facilities, field engineering, construction management, and home office expenses. Means (1991) suggests a 10% contingency and 20 to 25% allowance for overhead and profit, including subcontractor overhead and profit. State and local sales tax for Long Island, applicable to materials and equipment only, is estimated to be about 4% of total installed cost. The product of these individual factors yields an overall multiplier of about 1.4. A reduced multiplier of 1.2 was applied to the well cost data

provided by Delta Well and Pump because their data already contained an allowance for (subcontractor's) overhead and profit and the scope was clearly defined, which reduces the necessary contingency.

Propane costs, based on the equivalent Btu input of natural gas and the late-1992 natural gas price for MPF, were taken to be $0.687/therm or $6.87/MBtu[a]. Electricity costs were based on the Long Island Lighting Co. rates applicable to the MPF in 1991. Schedule 284 (SC 2-MRP primary voltage customers) rates in effect between December 1990 and Novovember 1991 are defined in Table 4.2.

4.2 ECONOMIC ANALYSIS

The LCC analysis was conducted according to Federal Energy Management Program (FEMP) procedures established in Section 10, Part 436 of the Code of Federal Regulations (10 CFR 436 as revised by P.L. 100-615 1988). Although the USPS is not obligated to follow the FEMP procedures, Mr. Bob Paetzold of the USPS indicated that they generally do for building energy evaluations and that it would be appropriate to use these procedures for evaluating the MPF's HVAC system.

Currently prescribed assumptions for fuel cost indices, discount rates, and present worth factors are documented by the U.S. Department of Commerce (DOC) in Lippiatt (1992). Fuel cost indices and corresponding present worth

TABLE 4.2. Electric Rate Structure

	Off-Peak[a]	On-Peak[b]	Intermediate[c]
Demand charge, $/kW	-	19.05	4.56
Energy charge, $/kWh	0.0657	0.0872	0.0795

(a) "Off-peak" runs from midnight to 7:00 a.m.
(b) "On-peak" runs from 10:00 a.m. to 10:00 p.m., Monday through Saturday, June 1 through September 30.
(c) All other hours are "intermediate".

(a) The natural gas price was reported by Mr. Dominic Bratta of USPS, Melville, NY. The $0.687/therm is about 12% higher (inflation and Btu content adjusted) than the $0.56/gal paid for propane purchased by MPF between June, 1991 and July, 1992 as reported by Mr. John Wyder of USPS.

factors are presented by fuel type, sector (e.g., residential, commercial, industrial, transportation), and DOC Census Region. The specific economic assumptions used in this analysis are listed in Table 4.3. The nominal discount rate includes expected future inflation, while the real discount rate excludes expected future inflation. Present worth factors are multiplied by recurring annual cash flows to calculate an equivalent present value. For example, an annual O&M expenditure of $1 occurring every year for the next 25 years has a present value of $15.62. A present worth factor of 17.03, based on a nominal price escalation rate of 5%/year (i.e., somewhat higher than the general inflation rate), was used for electricity at the request of Mr. Dominic Bratta, USPS. Propane has a higher present worth factor because its price is expected to escalate at a rate significantly greater than the rate of inflation.

4.3 RESULTS

Initial capital costs, annual O&M costs, and equipment lifetimes are presented for all components of the baseline, alternate baseline, and ATES/TEG systems in Tables 4.4, 4.5, and 4.6. A summary of system-level annual costs and 25-yr LCCs is presented in Table 4.7 The tabulated costs show the ATES/TEG system to be significantly more expensive to own and operate than either the baseline or alternate baseline systems. Initial capital and annual

TABLE 4.3. LCC Analysis Parameters

Nominal discount rate	7.9%
Real discount rate	4.0%
Economic life	25 yr
Reference price year	1992
25-yr present worth factors at 4% discount rate	
Capital	15.62
O&M	15.62
Electricity	17.03
Propane[a]	20.21

(a) Based on the escalation rate for natural gas.

TABLE 4.4. Baseline System Component Cost Summary

Component	Direct Capital Cost, $K	Total Capital Cost, $K	Annual O&M Cost, $K	Life, Yr
Centrifugal chiller	175	245	17.3	20
Evaporative cooling tower	37	52	1.5	25
Propane water heaters	29	41	1.0	25
Cooling tower pumps	17	24	1.0	20
Cooling tower piping	18	26	0.3	25
Air handlers	141	198	4.3	25
Totals[a]	418	585	25.4	

(a) Sum of individual components may not add to the total because of rounding.

TABLE 4.5. Alternative Baseline System Component Cost Summary

Component	Direct Capital Cost, $K	Total Capital Cost, $K	Annual O&M Cost, $K	Life, Yr
Centrifugal chiller	175	245	17.3	20
Evaporative cooling tower	37	52	1.5	25
Propane water heaters	29	41	1.0	25
Cooling tower pumps	17	24	1.0	20
Cooling tower piping	18	26	0.3	25
Air handlers	141	198	4.3	25
HEPA filters	112	156	27.6	25
Desiccant wheel	257	360	10.3	25
Totals[a]	786	1101	63.3	

(a) Sum of individual components may not add to the total because of rounding.

TABLE 4.6. ATES/TEG System Component Cost Summary

Component	Direct Capital Cost, $K	Total Capital Cost, $K	Annual O&M Cost, $K	Life, Yr
Reciprocating chillers	129	180	16.1	20
Propane water heaters	65	91	2.4	25
Miscellaneous pumps	55	77	3.3	20
Well transmission piping	209	293	4.1	25
Desiccant piping	31	43	0.6	25
Miscellaneous piping	36	50	0.7	25
Return fans	41	58	1.3	25
Desiccant system and	745	1043	29.8	25
cooling tower wells	246	295	9.6	25
Well pumps	78	94	4.8	20
Wellhead piping	57	68	1.2	20
Liquid desiccant	-	-	3.0	-
Totals	1692	2292	76.9	

(a) Sum of individual cost components may not equal the total because of rounding.

TABLE 4.7. System Annual and Life-Cycle (Present Value) Costs

	Baseline, $K/yr	Alternate Baseline, $K/yr	ATES/TEG, $K/yr
Annualized capital	40	73	150
Initial annual O&M	25	63	77
Initial annual electricity	125	147	155
Initial annual propane	4	13	34
	$K	$K	$K
Capital LCC	625	1141	2344
O&M LCC	397	989	1202
Electricity LCC	2131	2508	2634
Propane LCC	90	267	685
Total LCC[a]	3243	4905	6865

(a) Sum of individual cost components may not equal the total because of rounding.

O&M costs for the ATES/TEG system are three to four times higher than for the baseline system. Compared to the alternate baseline system, initial capital costs for the ATES/TEG system are about double and annual O&M is about 20% higher. In addition, energy consumption and costs are about 50% higher than for the baseline and about 20% higher than for the alternate baseline system.

The ATES/TEG system offers little, if any, reduction in the costs of conventional HVAC system components, while adding ATES and liquid-desiccant components which, collectively, cost much more than the sum of the baseline system components. From an energy perspective, both electricity and propane consumption are higher for the ATES/TEG system. The increase in electricity consumption is primarily caused by the increased fan power required to circulate a greater volume of air through the building ducting. Increased propane consumption is caused by operation of the liquid-desiccant concentrator.

5.0 LIQUID-DESICCANT SYSTEM OPERATIONAL ASSESSMENT

The objectives of the operational assessment of the desiccant system were to characterize the performance of the system, identify operating anomalies, and provide recommendations (where possible) for improving and maintaining the system. Section 5.1 describes a performance prediction model that was developed for analyzing the liquid-desiccant conditioners. This task included on-site monitoring and analysis of other collected operational data (both logs and anecdotal data) on the liquid-desiccant system. Sections 5.2 and 5.3 discuss the short- and long-term concentrator monitoring tasks. Section 5.4 presents conclusions and recommendations.

5.1 MANUFACTURER-BASED PERFORMANCE MODEL

The operating capacities of the liquid-desiccant system concentrator and conditioners are complex functions of entering air humidity, air flow rate, spray coil temperatures and flow, and desiccant concentration. To help characterize these functional relationships, an objective of this assessment was to develop component computer models based on Niagara Blowers's published selection data.

Niagara's selection data for conditioners, in conjunction with TEG property data found in the literature, provide a basis for the simple model described in Section 5.1.1. Application of this model to evaluate the performance of workroom conditioners at the MPF is discussed in Section 5.1.2.

For the liquid-desiccant concentrator, the data were quite limited, consisting of a rating table with a single operating point listed for each concentrator in the Niagara Blower product line. It was hoped that Niagara Blower could provide some new test data on a concentrator currently being tested at their laboratory; however, new test data were unavailable at the time of this writing. Thus, no "manufacturer's performance data" based concentrator performance model was developed. A fixed capacity assumption was used instead in the Section 3 assessment. In-situ concentrator test results, reported later in this section, suggest that a simple humidity rise model, based on saturated leaving air, provides a reasonable estimate of capacity.

5.1.1 Model Development

The conditioner performance model requires the following as inputs:

- conditioner model number
- supply air flow (cfm)
- TEG concentration
- inlet air dry bulb temperature and relative humidity
- desiccant mass flow rate to the conditioner
- entering water temperature and flow rate
- fan power.

The model iteratively balances the air-side heat and mass transfer with the water-side heat transfer by finding the "spray contact temperature", the effective outside surface temperature of the air cooling coil on which the liquid desiccant is sprayed and through which the chilled water passes. The Niagara Blower ratings assume a single spray contact temperature is applicable over the entire coil surface, reducing the heat and mass transfer algorithm to (effectively) a one-dimensional numerical problem with a single computational cell.

The solution procedure used in the model is as follows. An initial "guess" is made as to the spray contact temperature. Using this spray contact temperature, interpolations are made using Niagara Blower curve-fit data for the appropriate TEG concentration to determine exit air humidity ratio and dry bulb temperature, from which the net cooling capacity is calculated. The heat of solution and circulation heat of the liquid desiccant is then added. This total air-side capacity is used to estimate the exiting water temperature on the water side. Next, the conditioner's basic rating, corrected for air and water flow rate, is computed using Niagara Blower tabular and graphic data. This basic rating is multiplied by the water-side mean exit temperature difference--calculated using the water-side temperature rise and spray contact temperature--to get conditioner "expected" capacity. If this expected capacity does not equal the previously calculated air-side capacity, the spray

contact temperature is updated using a secant scheme, and the process is repeated until convergence is reached.

Psychrometric properties of air that are required for the algorithm include enthalpy and humidity ratio. Relations for these calculations are taken from the ASHRAE Fundamentals (1989). The integral heat of solution of TEG and its specific heat are also needed to determine the desiccant contribution to the total conditioner cooling load. These are calculated from property curves for TEG (Peng and Howell 1981).

Outputs from the conditioner performance model include:

- total cooling capacity and water removal rate
- exit air dry bulb and humidity ratio (fan heat added)
- exit water temperature
- spray contact temperature.

5.1.2 Conditioner Performance Analysis

As reported earlier (Section 3.1.3), large differences were found between the workroom zone peak loads predicted by the DOE-2 Loads program and those used in original design calculations. The most significant difference between the DOE-2 and original design loads was a generally lower peak total cooling load for the units, but a much higher than expected latent load caused by high infiltration and ventilation rates. Because of this difference, the conditioner performance model was used to verify that zone conditioners could meet the higher latent loads while still maintaining space temperatures.

The performance of two conditioner units was studied: AC-8 and AC-9. AC-8 was selected because it is closest to its design load, yet still predicts the load trends observed for the majority of units. AC-9 raised some concern because, unlike all other conditioners, predicted DOE-2 cooling loads for this zone exceeded the numbers reported in the original design calculations.

AC-8 - This unit was originally designed for a maximum cooling capacity of 59.6 tons and 77.4 lb/hr moisture removal (see Table 3.1) but, according to the DOE-2 simulation, experiences a peak cooling capacity of approximately 36 tons and 133.7 lb/hr moisture removal. Using 1) the design chilled water flow

rate of 178.9 gpm, 2) 50°F entering chilled water temperature, 3) the design air flow rate of 27,400 cfm (high fan speed), and 4) the DOE-2 design outside air conditions (84°F dry bulb, 76°F wet bulb[a]), the conditioner model predicted that AC-8 would overcool the zone from the design indoor conditions (78°F dry bulb and 45% RH) to 67.9°F dry bulb at 40% RH, if 80% TEG solution is used. Because of this excess capacity, the model was again run at identical conditions at 19,100 cfm (medium fan speed). At this reduced air flow, AC-8 was predicted to still cool the space to 72.1°F dry bulb at 39% RH.

Additional calculations were then performed to see what minimal water conditions could still maintain the workroom design summer temperature and humidity. Results of these calculations are summarized in Table 5.1.

AC-9 - This unit was originally designed for a maximum cooling load of 32 tons cooling and 38.7 lb/hr moisture removal. All the other conditioners were found by the DOE-2 Loads model to be adequately sized for their peak workroom loads, but the peak cooling load for the zone served by AC-9 was found to be considerably larger than in the design calculations (44.9 tons and 155 lb/hr moisture). Although this initially caused concern, examination of the simulated hourly zone conditions on the peak cooling day showed that at the design water temperature of 50°F and flow of 76.5 gpm, the space could be maintained at 76.6°F and 38% RH with an 80% TEG solution. Space design temperature and humidity can in fact be maintained with the design incoming water flow of 76.5 gpm at 51.5°F. By increasing the chilled water flow to 95 gpm, a 54°F chilled water temperature suffices to maintain space conditions.

TABLE 5.1. Summary of Conditioner Performance Model Results for Conditioner AC-8 (80% TEG)

	Chilled Water		Exiting Air Conditions	
Fan Speed	Flow, gpm	Temperature, °F	Dry Bulb, °F	RH, %
Medium	130.0	55.0	78.0	36%
High	150.0	54.0	72.0	40%
Medium	178.9	54.0	76.0	38%

(a) These are the outdoor conditions at the time of the peak cooling load determined by the DOE-2 Loads simulation; see Section 3.1.3.

The foregoing load and capacity assessment results for AC-8, and even the worst case unit, AC-9, indicate that the conditioners have more than adequate capacity to handle existing cooling loads in the workroom. This is confirmed by historical anecdotal evidence, and also the DOE-2 Loads simulation, both of which indicate that the limiting factor in the liquid-desiccant system is an undersized concentrator. Although the conditioner performance model is based only on manufacturer-supplied selection tables and curves rather than measured performance of the installed units, the results are sufficiently credible to motivate the following preliminary recommendations for improving energy performance:

- Reduce air flow rates by reducing the supply and return fan speeds at units AC-3 through AC-8 to better match actual cooling loads. This should significantly reduce fan energy consumption, which is predicted to be the largest electrical energy use in the ATES/TEG system. As a very conservative estimate, a 10% reduction in speed can be made. This will reduce fan power by more than 25% and annual fan energy consumption on the order of 150,000 kWh. Further savings can be achieved by replacing motors with premium-efficiency motors, replacing blowers (or impellers only) with high efficiency units, replacing existing fan speed control with variable-speed drives and providing closed-loop room temperature/humidity control. Additional study of the individual units and workroom operation schedule is needed to assess the true potential for motor down-sizing and fan speed reduction; the total cost-effective energy savings are likely to be far greater than the conservative estimates given above.

- Change the design supply chilled water temperature from 50°F to 54°F[a]. Based on the chill recovery predicted using the ATESSS model of the Melville aquifer, this could reduce the need for mechanical refrigeration by as much as two-thirds. Including water pump energy, this translates to approximately 75,000 kWh annual savings.

- Use TEG concentrations of 80% to supply the conditioners instead of the 85% concentration that has been used historically at the Mid-Island MPF. This will reduce the heat-of-solution loads and the concentrator hot and reflux water requirements while still maintaining a workroom relative humidity of 40% at design conditions.

(a) This increase to 54°F can probably also be accompanied by a decreased design flow rate to several of the conditioners; however, the model indicates water flow to AC-9 should be increased to 95 gpm.

5.2 CONCENTRATOR ISOLATION TEST

A field test of the concentrator was undertaken because of a general perception that the concentrator's capacity was inadequate and because the performance data provided by Niagara Blower was very limited. The concentrator isolation test procedure and results are discussed in the next subsection.

5.2.1 Test Description

In the isolation test of July 2, 1992, the desiccant solution circulation pumps that normally operate continuously were shut down and hand operated valves in conditioner supply and return lines were closed (Figure 5.1). The diluted desiccant solution was supplied from a previously filled desiccant storage tank to the concentrator using the desiccant feed pump[a]. After passing through the concentrator, the desiccant solution was returned, via the sump float-level activated return pump, to the return storage tank. The duration of the test was limited to the time it takes the desiccant solution feed pump to empty the (initially) full storage tank. By measuring initial and final desiccant solution levels and concentrations in the tanks and sump, as well as water and air flows and temperatures, an assessment of the concentrator performance under a controlled, steady set of operating conditions was made.

A C180 data logger was used to monitor key concentrator operating parameters during the concentrator isolation test. Sensor outputs were sampled at least once every 10 seconds (hard-coded feature of the C180) and 1-minute average values were recorded. To track temperature, sheathed thermocouples (type T, ANSI special limits grade) were installed on inlet and outlet water and desiccant piping, as well as in the inlet and outlet of the concentrator air stream. The thermocouples were connected to the logger via differential thermocouple signal conditioner cards. To measure electric motor loads, current transformers were installed on the power leads of the fan and spray

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(a) The desiccant feed pump used during the isolation tests is one of three pumps that can be used, in normal operation, to feed weak desiccant solution to the concentrator. The other two pumps that can serve this function are the desiccant "circulation pumps". The circulation pumps were not operated during the isolation test.

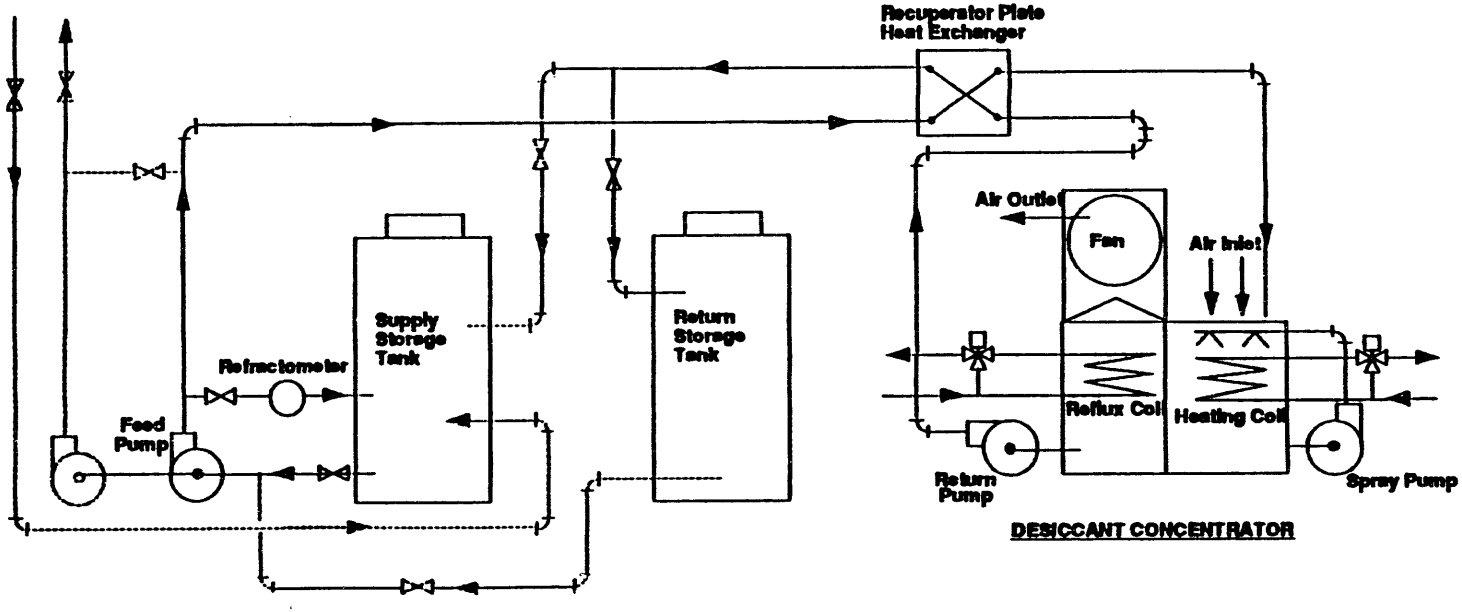

FIGURE 5.1. Concentrator Isolation Test System Schematic

pumps (combined load), the hot water pumps, the reflux water pumps, the desiccant feed pump, and the desiccant return pump. The hand-held refractometer used by USPS staff to monitor day-to-day operation of the concentrator was used to measure desiccant concentrations during the test. Outside air humidity (i.e., the supply air to the concentrator) was measured using a hand-held electronic temperature/humidity meter.

The desiccant feed pump was operated manually during the test. Start up of the concentrator was accomplished using the existing controls by adjusting the minimum concentration setting on the concentrator control panel to be above the concentration level measured by the electronic refractometer.

To prevent the hot water coil from experiencing a temperature/flow transient at the start of each test, its pump was manually activated to preheat the hot water coil prior to start up.

To ensure a uniform desiccant concentration in the supply tank at the start of each test, the solution was circulated through the sump and supply tank of the isolated concentrator subsystem for about 20 minutes. Also, the concentrator was operated in winter mode to avoid the time-delayed start.

The time of the test was 11:19 to 11:59 PST. A plot of the temperatures versus time is shown in Figure 5.2.

One-time measurements of air and water flows through the concentrator were needed to complete the isolation test energy balance. Using an on-site differential pressure (DP) gauge and previously installed venturi flowmeters, a hot water flow rate of 78 gpm was measured on July 2. The DP gauge was scaled too high to get a reliable reading of reflux water flow. The venturi pressure taps were cleaned out and a second set of measurements was made at PNL's request by USPS operations staff and Randy Schmitt of Douglas A. Wilke on July 10. The flows were reported to be 100 gpm for the hot water circuit and 45 gpm for the reflux water. Because the reflux coil throttling valve (used to maintain concentrator air discharge temperature) is inoperative, there should have been no change from the flow rate during the test.

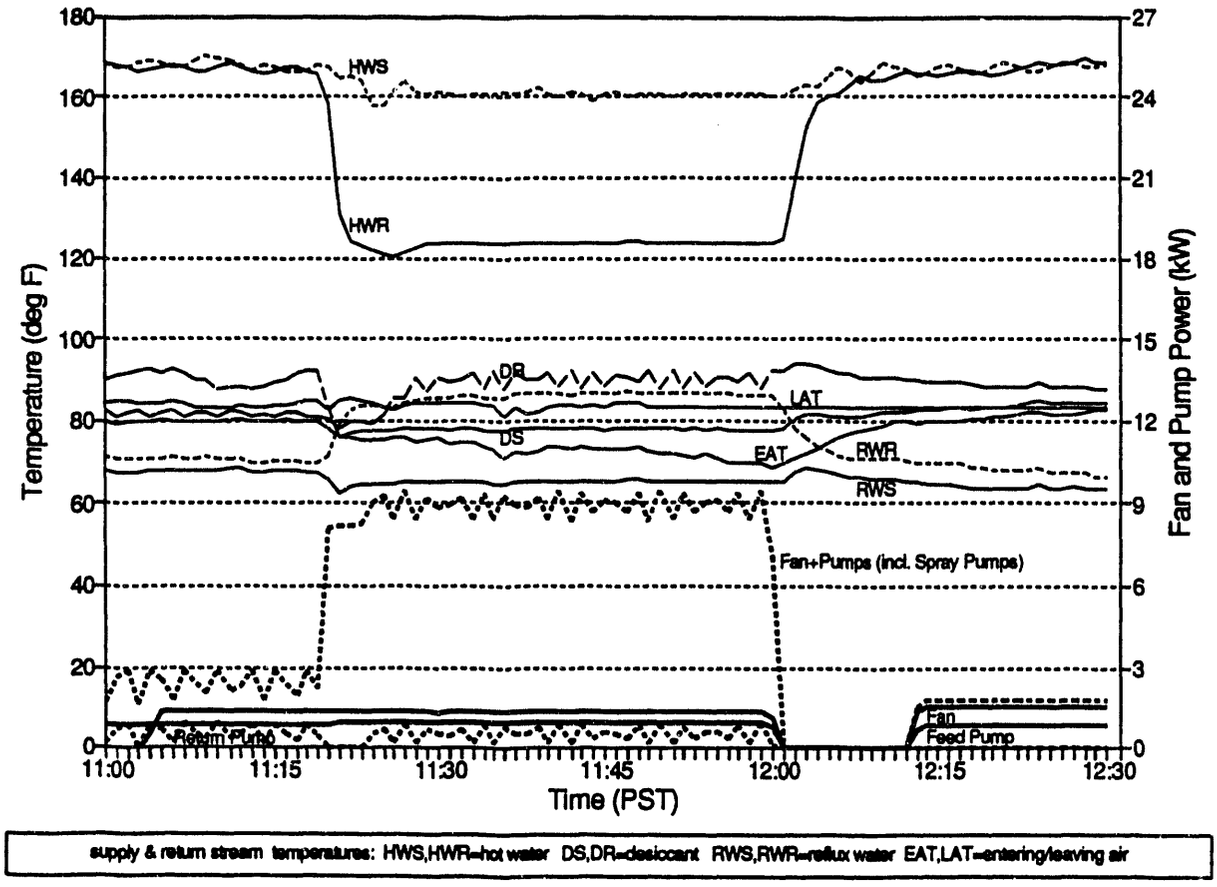

FIGURE 5.2. Conditions Observed in the Concentrator Test of July 2, 1992

A hot-wire anemometer was used to measure air velocities from which the concentrator air flow rates could be calculated. The velocities were measured three times at each of 12 uniformly spaced locations over a cross section at the throat of the discharge nozzle. A 15-second averaging interval was used at each location and replication. Taking the product of the average velocity and the cross-sectional area, the air flow rate at this location was calculated to be approximately 8280 cfm. A second set of measurements taken below (upstream of) the throat gave a flow rate of 9200 cfm but this value was not used because greater turbulence and velocity profile non-uniformity were observed at the second location.

5.2.2 Isolation Test Results

Average temperatures for the July 2 concentrator isolation test are shown in Table 5.2, along with fan motor and water pump input power levels and

TABLE 5.2. Concentrator Test Data Used for Analysis

Parameter	Value	
Inlet hot water (°F)	160.8	
Outlet hot water (°F)	124.0	
Hot water flow (gpm)	77.8	
Inlet reflux water (°F)	66.0	
Outlet reflux water (°F)	87.0	
Reflux water flow (gpm)	45.0	
Inlet air dry bulb (°F)	72.0	
Inlet air wet bulb (°F)	58.0	
Outlet air dry bulb (°F)	82.0	
Outlet air wet bulb (°F)	82.0	
Air flow (cfm)	8283	
Inlet solution (°F)	78.0	
Outlet solution (°F)	90.0	
Solution supply flow (gpm)	16.8	
Average return flow (gpm)	14.2	
Fan power (kW)	1.6	
Total power (fan + pumps, kW)	9.0	
	Initial	Final
Time (PST):	11:19	11:59
Concentrations:		
Tank 1 (%)	82	82
Tank 2 (%)	82	85.6
Sump (%)	82	86
Desiccant levels:		
Tank 1 (in. from bottom)	71.6	16.0
Tank 2 (in. from top)	83.8	36.9
Sump (in. from top)	4.75	3.15
Cross-sectional areas:		
Tanks 1 and 2 (ft^2)	19.6	
Concentrator sump (ft^2)	57.0	

air, desiccant solution, and water flow rates. Also summarized are measured TEG solution levels and concentrations before and after the test. Data in this table were used to estimate concentrator performance.

Because feed pump operation is continuous, desiccant flow to the concentrator was essentially constant during the test. The change in tank level implies a 16.8 gpm feed rate. This is less than the pump rating of 24 gpm because part of the flow bypasses the concentrator and goes through the electronic refractometer and back to tank 1. Return flow from the concentrator is cycled by the sump float switch. The return flow rate averaged 14.2 gpm during the test. The difference between supply and return flows is the result of evaporation of water and the higher final sump level.

The net decrease in the volume of the solution was found to be 50 gal during the test. Assuming the decrease is caused by evaporation, the mass of water rejected was 415 lbm, or 615 lbm/h.[a] As an independent check, the water evaporation rate was estimated based on air-side entering/exiting conditions and measured flow rate. The air-side calculation method indicates a water evaporation rate of 613 lbm/h. The closeness of the volume-based and air-side-based results (0.3%) should not, however, be taken as an indicator of their accuracy.[b]

Using the data in Table 5.2 and treating the concentrator as a control volume gives a heat energy balance error of about 8% of the energy supplied to the hot coil. Most of the heat balance error is probably caused by hot water flow rate measurement error. A summary of the computed thermal energy inputs and outputs used for this calculation is given in Table 5.3.

(a) Using measured TEG concentrations and the 50-gal volume change indicates about 17% of the losses (by weight) were TEG, and a water evaporation rate of 424 lbm/h instead of 615 lbm/h. Such high TEG losses are inconsistent with the TEG loss rate observed in long-term monitoring. Rather, the high implied TEG loss is caused by the hand-held refractometer resolution and repeatability error.

(b) Based on uncertainties in the individual measurements and assumptions, the volume-based result is probably a 3% number, the air-side result is probably a 6% number, and the refractometer-based result is probably a 15% number.

TABLE 5.3. Heat Energy Balance of July 2 Concentrator Test

Energy Input/Output Description	Average Rate, Btuh
Air	762,485
TEG heat of solution	29,279
TEG sensible heat	52,244
Reflux water	473,000
Hot water (energy input)	-1,433,037
Heat balance error (sum)	-117,376
Error divided by energy input	8%

COP is defined as latent heat of rejected water vapor divided by total energy required to operate the concentrator. The fan and pump power, expressed as heat energy, amounted to 0.02 MBtu during the test. Assuming a boiler efficiency of 70% gives an estimated boiler heat input of 1.38 MBtu. The resulting estimate of site energy input during the test is 1.4 MBtu[a].

The water evaporation rate of 613 lbm/h translates to a latent load of 0.6 MBtuh, or 0.41 MBtu over the test period. The concentrator COP, based on site energy input, was therefore about 0.3 during the isolation test. This COP is in reasonable agreement with the rated COP of 0.26 and can be expected to drop somewhat if concentrator air flow rate is increased. However, it is only about half the system COP typical of current off-the-shelf natural-gas-powered equipment. The short-term test also indicated that concentrator fan and pump energy requirements are a small fraction of the concentrator's thermal energy requirement and that the unit is in good operating condition.

(a) Note that energy to cool the reflux water was not included because the source of reflux water is the well water after it has passed through the conditioner coils. While this scheme might seem to offer a free source of cooling, there is, in fact, a small incremental cost that would accrue in the winter season when the cooling tower is operated to thermally recharge the aquifer.

5.3 LONG-TERM CONCENTRATOR MONITORING

The isolation test of July 2 provided data on concentrator performance under just one set of conditions. It did not provide information on the overall latent load of the building and did not answer questions about the need for greater concentrator capacity.

In the process of instrumenting the concentrator for the isolation test, transducers and cabling were installed in a manner that would not impede normal operation and maintenance of the desiccant system and it was learned that a rarely used phone line was available for remote logger communications. We were also successful in enlisting the cooperation of the operators in taking more detailed and frequent (4 hr versus 8 hr) logs than had been customary. It was, therefore, decided to continue monitoring the system, via a combination of automatic and manual logging, for at least 1 month.

The purposes of the long-term test were to confirm that performance observed in the isolation test was representative and to learn more about concentrator capacity and building latent load, variation of performance with conditions, and the magnitudes of some of the parasitic energy components that might lead to overall efficiency improvements. It was recognized that these objectives might not be achieved, given the relatively crude instruments involved (e.g., hand-held refractometer) and reliance on manual logging. We proceeded because the incremental cost to the project was very low.

5.3.1 Long-Term Monitoring Procedure

The concentrator was monitored from July 6 to October 20, 1992. The C180 data logger continued to monitor the variables observed during the short-term tests but the recording interval was increased from 1 minute to 15 minutes for the long-term monitoring period. The recording interval for variables that were monitored by hand was increased to 4 hr and the recording task was taken over by the operations staff. The flow of desiccant through the concentrator could not be measured in the long-term test by the tank transfer method that was devised for the short-term test. Instead, flow through the concentrator during the long-term test was inferred from return pump run time. The air-side flow rate measured in the short-term test was assumed to continue to be valid throughout the long-term monitoring period.

5.3.2 Observations

The conditions and performance of the units varied during the test period from July 6 through September 19, and are shown in the figures in this section. Data for September 13-15 was lost because of a data logger failure.

5.3.2.1 Parasitic Energy

Figure 5.3 shows daily average duty fraction (fraction of each day that the unit operated) and average motor power, during concentrator operation, for each fan and pump function. In some cases there are two pumps that serve a given function. In the case of the spray pumps, the motors operate together and the power is very constant. In the case of the hot water pumps, there is a lead-lag arrangement designed such that one pump provides the desired hot water flow while the other serves as a backup pump.

The motors most closely associated with the concentrator are those that drive the concentrator fan and spray pumps. The average power consumption from July 6 to August 17 was 7.75 $\pm$0.15 kW. This is consistent with the isolation test data. The feed pump power is also included in the fan and spray pump power history, but the feed pump was not operated prior to August 18. The increase in power after that date is not a constant; the reason for its variation is not clear but is probably caused by head variation. A similar behavior may be noted in the circulation pump trajectory. We postulate that the circulation head increased when a valve was closed, which allowed some of the output from the circulation pumps to branch to the concentrator at the time that the feed pump was brought on-line. The circulation, feed, and return pumps are all positive-displacement pumps, which makes their power draw almost directly proportional to head.

The reflux and desiccant return pump power trajectories are fairly constant and consistent with the isolation test results. Note, however, that the return pump values are slightly lower after August 17, indicating that the desiccant flow into the concentrator provided by the new feed pump is slightly lower than that provided by the circulation pumps. Because concentrator capacity is air-flow limited rather than desiccant-flow limited, the motivation for a separate feed pump that provides such a small increase in desiccant flow rate is unclear.

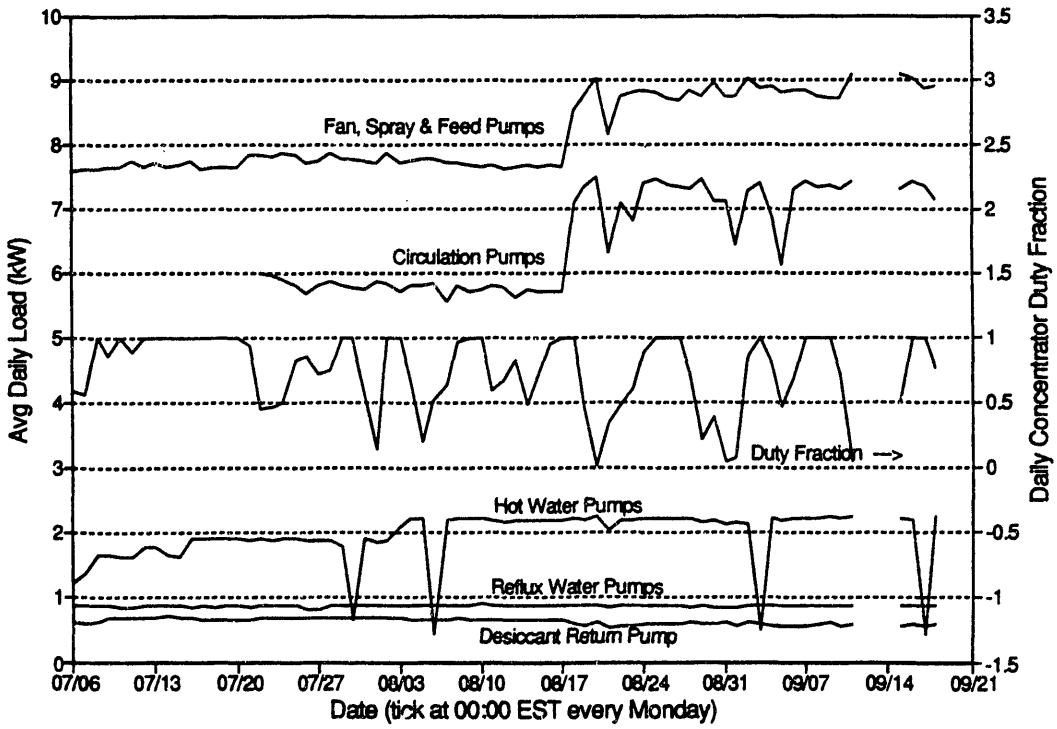

FIGURE 5.3. Average Motor Loads with Concentrator On

The concentrator duty fraction history, also shown in Figure 5.3, indicates that the machine is fully loaded (or overloaded) on nearly one-half of the days and that the average duty fraction for the entire monitoring period is about 0.75. The log of TEG concentration indicates extended periods of diluted solution, below refractometer setpoints, suggesting that the concentrator is not meeting the load. One must, therefore, conclude that the unit is either not operating at design capacity or is not adequately sized for the dehumidification loads that are currently experienced during the cooling season.

Table 5.4 provides a summary of motor loads. The table gives more detail than Figure 5.3 by listing the return and feed pump loads separately from the fan and spray pumps. Examination of the raw data shows that of the 7.04-kW load attributed to the fan and spray pumps, about 2.5 kW is for the fan and 4.5 kW is for the pumps (2.25 kW per pump).

TABLE 5.4. Average Motor Loads Observed in Long-Term Test

	Average kW		Electrical Energy	
	7/7-8/19, kW	8/31-9/13, kW	Concentrator On, kWh(b)	All Times, kWh(c)
Fan and spray	7.04	8.21	8,989	9,109
Return pump	0.67	0.58	780	789
Circulation[a]	5.77	7.23	7,224	14,705
Reflux water	0.87	0.87	1,054	1,364
Hot water	1.85	2.09	2,338	2,349
Total	16.20	18.98	20,385	28,316

(a) extrapolated from kWh measured in 7/23 through 9/13 period
(b) for periods of concentrator operation only
(c) includes periods when pump is running but concentrator is not

5.3.2.2 Process Summary

Figure 5.4 shows the daily temperatures of the fluid streams entering and leaving the concentrator, averaged over the period(s) of concentrator operation. The hot water supply and reflux water supply temperature are fairly constant but hot water supply temperature is a little lower, on the average, after August 3. The reflux water return and exit air temperatures track one another and are fairly constant because the reflux supply (well) water temperature is very constant; this is significant to the capacity model developed later. The desiccant supply and return temperatures (shown in Figure 5.5 for clarity) also track one another. These behaviors are expected from the concentrator design and system control strategy. The relationships among the temperatures plotted in Figure 5.4 are consistent with the concentrator behavior observed in the isolation test.

Figure 5.5 shows inlet-outlet temperature differences for the concentrator fluid streams. The hot water supply to hot water return temperature difference is fairly constant at about 40°F through August 3. After August 4, the temperature difference averages about 20°F and varies less from day to day. The magnitude of the reflux water temperature difference, on the other hand, is larger after August 4. These effects result from the hot water supply flow rate being approximately doubled on August 4. The increase in

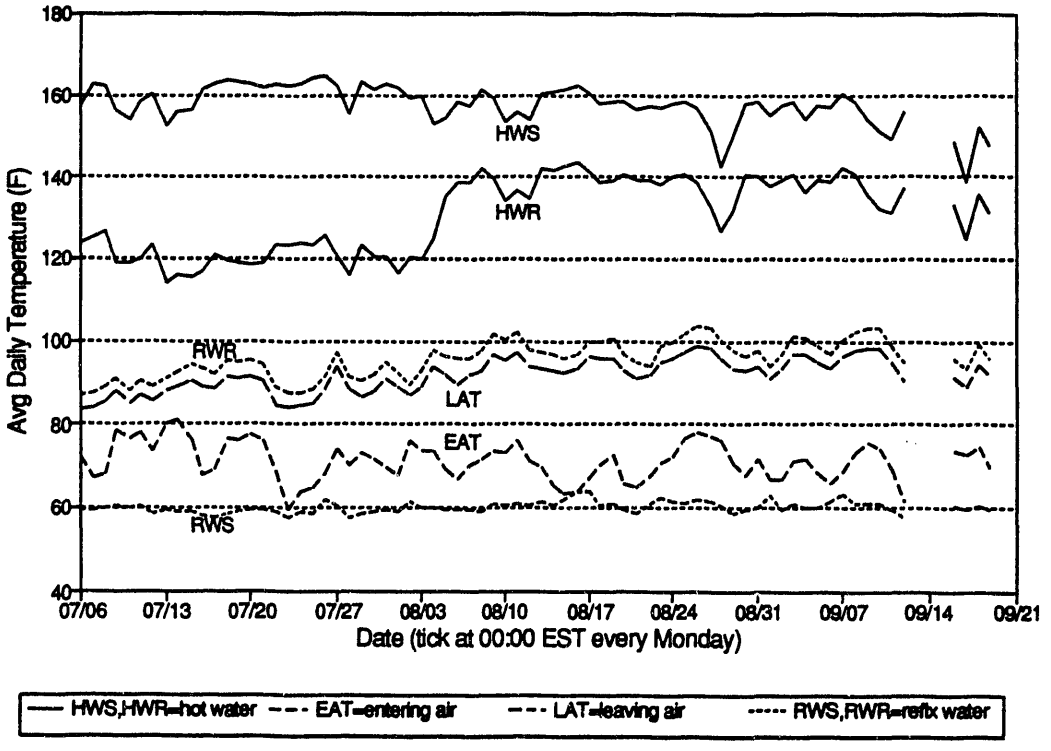

FIGURE 5.4. Average Stream Temperatures with Concentrator On

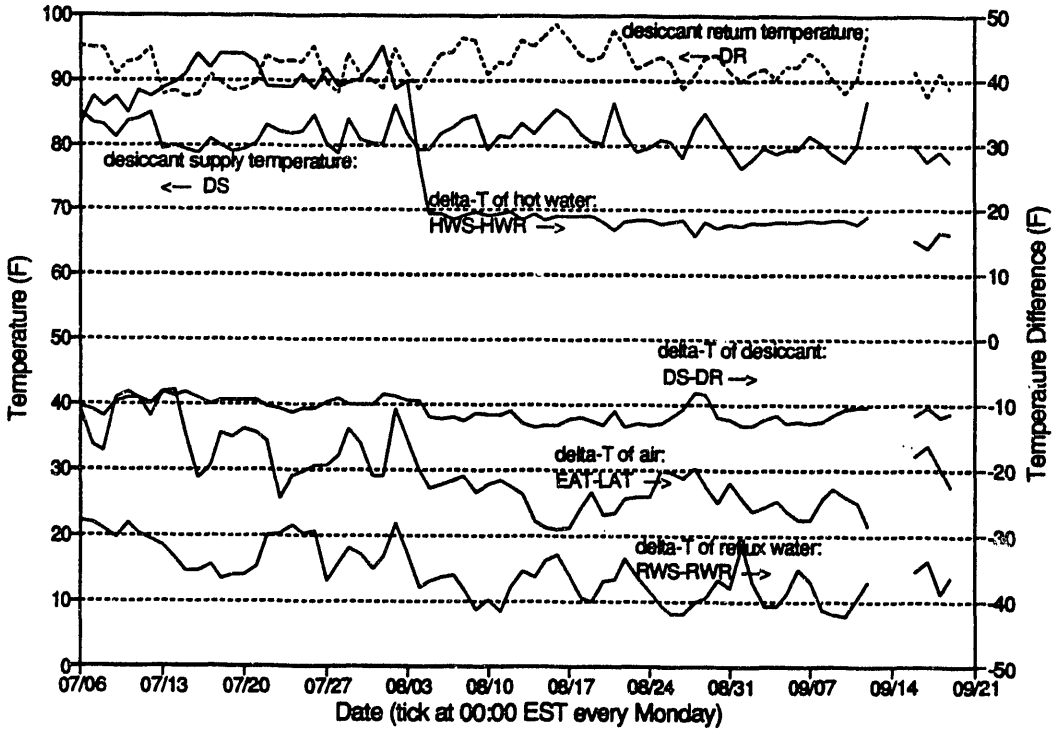

FIGURE 5.5. Average Stream Temperatures and Inlet-Outlet Temperature Differences with Concentrator On

flow rate results in increased heating of the entering desiccant and entering air, both of which contact the hot water coil in the spray section of the concentrator. This in turn imposes a higher load on the reflux coil and also raises the temperature of the exiting desiccant solution. From the change in load and hot water supply-return temperature difference, it is apparent that the flow rate increased by a factor of 2.6 from about 78 gpm to about 204 gpm.

Using the water-side temperature differences, hot water flow rate, and propane use reported during the monitoring period, an estimate of in-service boiler efficiency can be developed. Reported propane use was 31,365 gal from July 2 through September 15. Temperature data spans July 6 to September 15, and the concentrator operated 1220 hr over this period. It operated 1317 hr over the July 2 to September 15 period. Scaling propane use for the shorter time period results in an estimated propane input of 2673 MBtu between July 6 and September 15.

The concentrator operated 595 hr at the lower hot water flow rate with an average temperature difference of about 40°F and 625 hr at the higher flow rate with an average temperature difference of 18.4°F. These numbers imply a net boiler output of 2058.6 MBtu and an efficiency of 77%. This estimate seems high considering that the boilers are old and that the 2673 MBtu input includes fuel used during standby operation. A typical in-service efficiency for this type of boiler is about 70%. It is reasonable to conclude, based on the boiler efficiency discrepancy and the heat-balance error developed in Table 5.3, that the hot water flow rates referred to in Section 5.2.1 are 10% high. This error is quite plausible given the water flow rate measurement conditions.

5.3.2.3 TEG Loss

Table 5.5 summarizes the operator log entries pertaining to additions, losses, and transfers of the desiccant solution. Incidents of desiccant overflowing a conditioner sump were noted in the operators' log on several occasions, but these notes also indicated that most of the solution was recovered and returned to the system. Thus, there are no entries in the "Lost" column. Entries in the "Returned" column refer to desiccant solution

TABLE 5.5. Summary of Desiccant Solution Transfers by Plant Operators

Date	Returned, gal	Added, gal	Lost, gal	Tank Level, in.
7/3	55			
7/11				16
7/15	50			
7/17				12
7/18	46			
7/20	55			
7/23				
7/27				
7/31	80			25.5
8/7	55			
8/11	25			
8/14	15			
8/15	35			
8/17				
8/18	5	165		
8/20	70			
8/21	55			
8/27				24
9/9	110			

having overflowed a conditioner sump into a catch drum, the level in a catch drum is only recorded when its accumulated overflow is returned to one of the main tanks adjacent to the concentrator. The main tanks initially contained 515 gal at 85% concentration or 438 gal pure TEG. The final volume was 1105 gal at 71% concentration or 784 gal pure TEG. This implies an increase that exceeds the amount of pure TEG (165 gal) added to the system. However, the initial and final TEG volumes and concentrations in the conditioners and overflow sumps are unknown, so the net TEG loss during the monitoring period could actually have been as large as 165 gal of new TEG added. We have used this number because it is the best available, albeit probably conservative, estimate of TEG loss. Based on the air-side humidity rise method (see Section

5.3.3.1), the amount of water removed by the system between July 6 and September 12 was 500,000 lb. The upper-limit estimate of TEG loss is, therefore, about 0.3% of the moisture removed or about 1.5 times the 0.2% rule-of-thumb used by Niagara Blower.

5.3.3 Analysis and Modeling from Long-Term Observations

The objectives of our analysis were to obtain, if possible, empirical models of concentrator performance and building latent load.

5.3.3.1 Concentrator Performance

Three methods of measuring water rejection rate were presented in the isolation test analysis section. The change-in-volume method and air-side humidity rise method gave virtually the same result. The change-in-concentration method gave a number about 30% lower. One problem with the change-in-concentration number is that the change is small (3% to 4% concentration) and the hand-held refractometer used to measure concentration is graduated in whole degrees. (1 degree refractive index corresponds to about 1.2% concentration change.) The concentration measurement, therefore, has an inherent uncertainty of ±0.6% concentration (about 15% of the change in concentration of interest) from the instrument resolution alone. Other error sources include contamination of the sample, error in measured solution flow through the concentrator, and inhomogeneity in the sampled solution, particularly systematic inhomogeneities at the supply and discharge sampling locations.

The change-in-volume method used for the short-term tests could not be used in the long-term monitoring tests because it relied on the storage tanks to provide separate supply and discharge reservoirs where levels could be measured at various times.

The air-side humidity rise method uses the product of dry air mass flow rate and change in absolute humidity ($w_{exiting}$-$w_{entering}$) to estimate the

concentrator's water rejection rate.[a] The humidity of the entering air was based on ambient temperature and relative humidity logged by the facility energy management control system (EMCS) and the exit air temperature monitored by the PNL data logger. The exiting air was presumed to be saturated and free of entrained water or desiccant solution droplets.

Figure 5.6 shows the relation between desiccant concentration measured at the concentrator inlet weir box and sump outlet pan. A roughly 3% average concentration rise is clearly evident but the scatter in the data makes any analysis based on individual data points risky. Figure 5.7 shows the relation between the change in concentration and the humidity rise. The clustering of data points on incremental concentration differences in this figure is a result of limited resolution of the refractometer used by site personnel to measure the entering and leaving solution concentrations. The lack of correlation is consistent with the scatter seen in Figure 5.6. Figure 5.7 does seem to indicate that no significant increase in concentrator capacity occurred after the increase in supply flow rate on August 4. This implies that concentrator capacity is air flow or reflux water temperature limited and that the addition of more heat is a waste of propane. Beyond this, it is difficult to conclude anything except that the change-in-concentration data, while generally consistent with the isolation test results, contains too much random and round-off induced variation to serve as the basis for an empirical capacity model or even to qualitatively verify such a model.

5.3.3.2 Building Latent Load Model

The humidity rise model of concentrator capacity is the best model available from physical principles alone. We can tentatively accept the humidity rise model on the strength of its confirmation, for a single

---

(a) Humidity has been converted from relative humidity (which typically has a strong diurnal component that varies inversely with dry bulb temperature) to absolute humidity (the ratio of water to dry air mass, which has a much smaller diurnal variation). Relative humidity is a better measure of outdoor comfort than absolute humidity, but the two are equally good measures of indoor comfort when indoor temperature is constant. Absolute humidity is a better measure of the latent load produced by outside air infiltration.

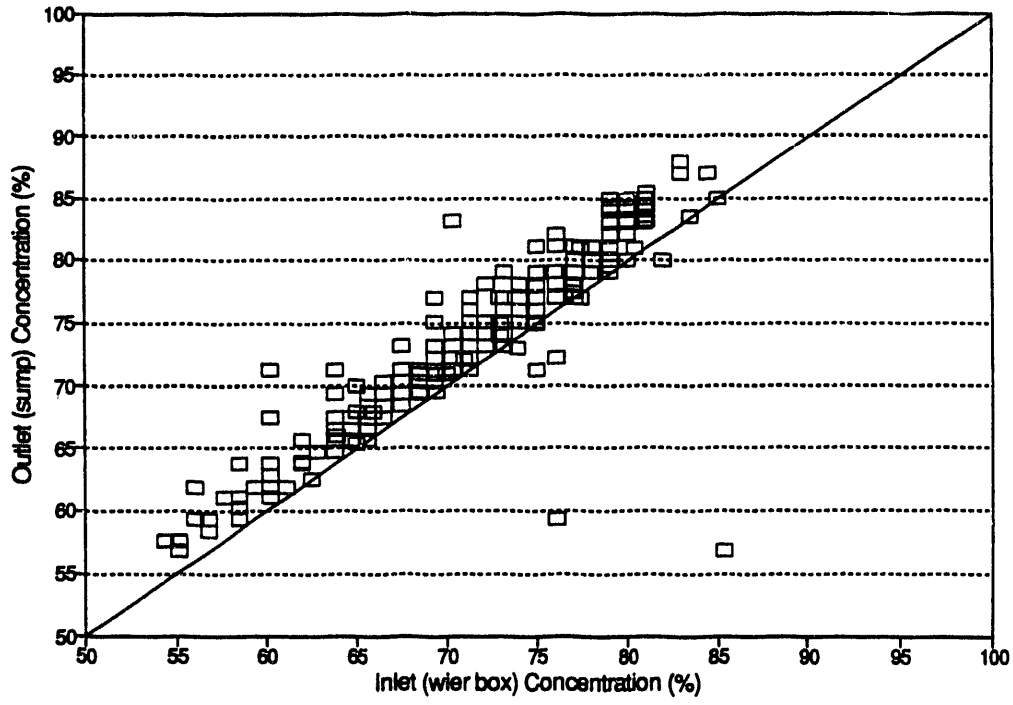

FIGURE 5.6. TEG Concentrations of Concentrator Inlet and Outlet (from Operator Log)

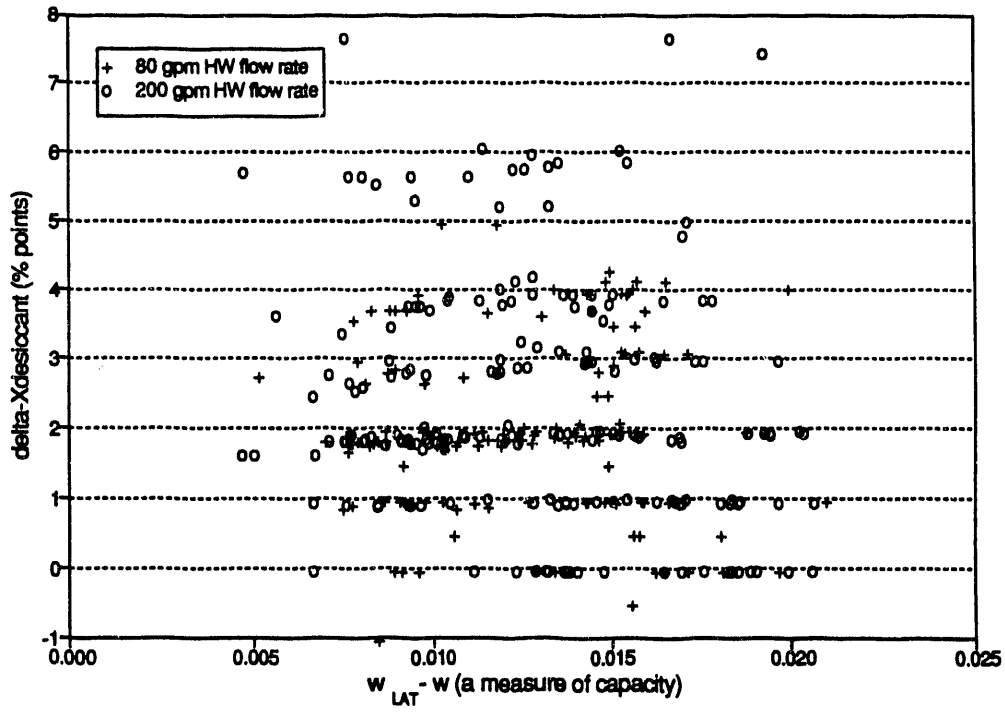

FIGURE 5.7. Change in TEG Concentration through Concentrator Versus Humidity Difference

operating point, obtained in the isolation test. This provides a basis for the development of an empirical building latent load model from the limited data obtained in the long-term monitoring.

Figure 5.8 shows the relation between capacity, as inferred from humidity rise and duty fraction product, and ambient humidity. Each point is based on the outside temperature and relative humidity, the daily average exiting air temperature, and the duty fraction for the 4-hr period ending at the time of the log entry. When consecutive periods of duty fraction = 0 or duty fraction = 1 were encountered, only the first and last points were retained. This rejection of selected points is necessary because there is no meaningful relation between duty fraction and outdoor humidity when the latent load is low (duty fraction = 0 because no dehumidification required); likewise there is no meaningful relation when the latent load exceeds the system's capacity (duty fraction = 1 because the system can't keep up).

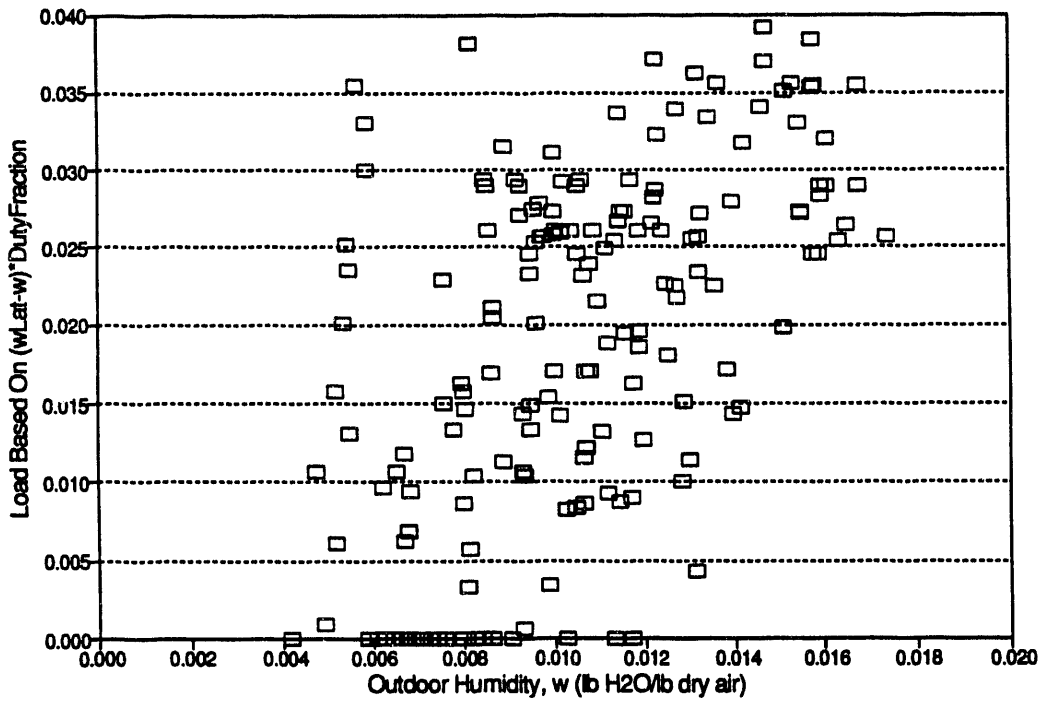

FIGURE 5.8. Combined Concentrator and Latent Load Model Based on Instantaneous Outdoor Humidity, w

A straight line fit to the data by ordinary least squares regression exhibits the expected slope (increasing capacity with increasing ambient humidity) and intercept (duty fraction = 0 at a humidity ratio of w = 0.07 where w = 0.07 corresponds to 40% relative humidity at 74°F). While there is considerable scatter around the line, this initial result was sufficiently promising to justify further model exploration.

Figure 5.9 shows the relation between capacity (humidity rise method) and the convolution of ambient humidity and a distributed-lag response factor vector obtained by linear regression. The best-fit line has a regression coefficient of $r^2 = 0.6$. The equation for this line is:

$$DF_t * (w_{LAT} - w_t) = -0.01691 + 3.40\ \bar{w}$$

where DF_t = duty fraction at current time, t,

w_t = outside ambient (and entering) air humidity ratio at time, t,

w_{LAT} = humidity ratio of saturated air based on the leaving air temperature at current time, t,

and $\bar{w} = 0.146w_t + 0.593w_{t-4} - 0.075w_{t-8} + 0.336w_{t-12}$

where w_{t-4} = ambient air humidity ratio 4 hours earlier,

w_{t-8} = ambient air humidity ratio 8 hours earlier, and

w_{t-12} = ambient air humidity ratio 12 hours earlier.

Note that $\bar{w}$ is a moving average humidity in which the moving average weights have been determined by ordinary least squares.

The concentrator's capacity in lbm/hr of water removed, based on the humidity rise model and concentrator air flow measured at the time of the isolation test, is 36,750 lbm/hr * DF * (w_{LAT} - w). The 36,750 factor and the slope of the regression line imply an average ventilation and infiltration rate of about 125,000 lbm/hr or 28,000 cfm. This corresponds to 0.5 air changes per hour in the spaces served by the conditioners. This compares reasonably well with the estimated air-change rate of 20,600 cfm used for the DOE-2 model (Section 3.1) that was based on a balance of supply and exhaust fan design air flow capacities.

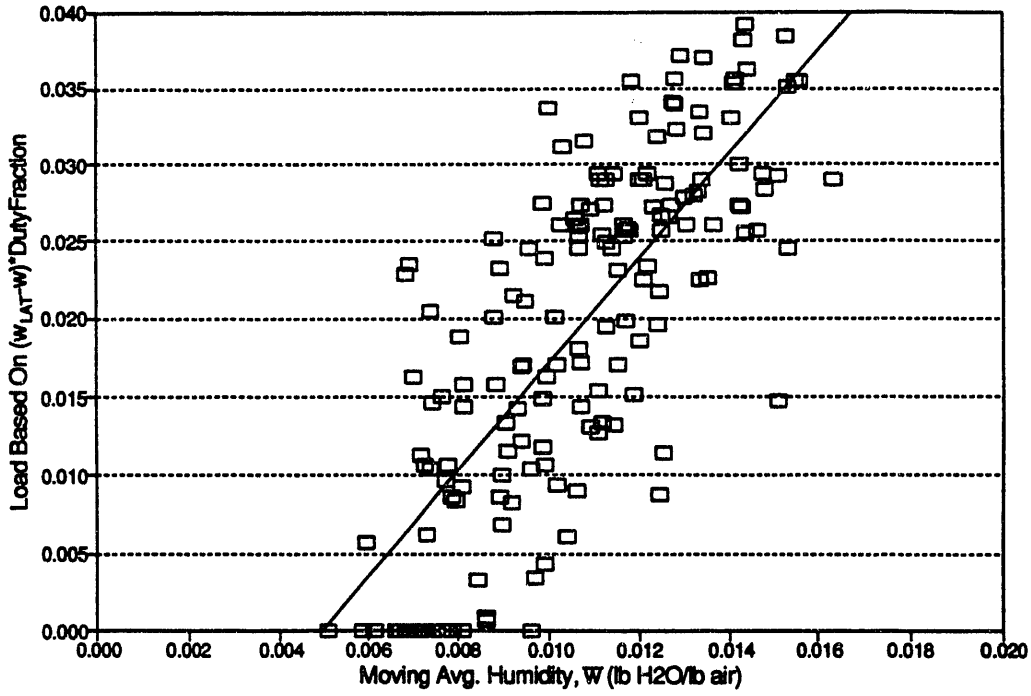

FIGURE 5.9. Combined Concentrator and Latent Load Model Based on Moving Average of Outdoor Humidity, $\bar{w}$

5.3.4 Long-Term Monitoring Summary

The concentrator's performance during the long-term test was consistent with the performance observed in the short isolation test in terms of parasitic power, thermal power, capacity, and COP. The TEG loss observed during the long-term test was not excessive--less than 0.3% of rejected liquid water volume. A building dehumidification load and capacity model was developed empirically. The infiltration rates implied by the model are consistent with the ventilation and infiltration rates developed from audit and facility management information about work force activity, mail processing equipment, and HVAC system operation summarized in Section 3.1. The model also confirmed the humidity rise model of concentrator capacity postulated in Section 5.2. There was no observed evidence that the dehumidifying capacity of the system is limited by conditioner capacity. The system capacity appears to be primarily or entirely controlled by the concentrator's capacity.

5.4 SUMMARY OF LIQUID-DESICCANT SYSTEM ASSESSMENT AND RECOMMENDATIONS

Results and recommendations pertaining to parasitic losses in the conditioners, concentrator and circulation system are described in Section 5.4.1. Results and recommendations pertaining to the thermal performance of the concentrator are summarized in Section 5.4.2.

5.4.1 Parasitic Losses

There are a number of opportunities for reducing parasitic losses. Two-speed or variable-speed pumps will reduce the annual energy use by the concentrator spray pumps and the reflux and hot water pumps from 10,500 kWh to less than 5,000 kWh. Two-speed or variable-speed circulation pumps would have reduced the circulation energy from 14,700 kWh to less than 6,000 kWh. Installing a larger fan motor to move 50% more air and making it a two-speed or variable-speed motor would increase fan energy from 3000 to 6000 kWh. This would boost concentrator capacity at the expense of a modest increase in overall parasitic power. If the moisture loads can be reduced to the point where the existing concentrator's capacity is sufficient, the two-speed or variable-speed fan motor retrofit would reduce concentrator fan energy from 3000 to less than 1500 kWh.

An efficient and low-cost approach to the two-speed retrofit of circulation, hot water, and reflux pumps is to simply replace one of the two lead-lag pumps with a smaller, high-efficiency pump/premium-efficiency motor set and retain the replaced unit as a spare.

5.4.2 Concentrator Thermal Performance

The concentrator has a thermal COP of about 0.3, which can probably be raised to about 0.7 by recovering heat from the concentrator exhaust stream and avoiding the mixing of desiccant streams of different temperature and concentration, that occurs in the current system. An additional 20% savings in propane use may be realized by recovering heat from the boiler flue-gas streams.

A set of heat exchangers to extract heat upstream of the reflux coil and add it to the intake air upstream of the hot water coil is the first priority. It is important to restore the reflux control that maintains constant

discharge air temperature to realize the potential reductions in well-water flow and pumping energy that are associated with this heat recovery measure.

The second priority is to recover heat from the boiler flue gas to preheat the concentrator intake air between the first stage preheat described above and the existing hot water coil. This can be accomplished by installing heat exchangers or by direct injection of the flue gas. There are three possible heat exchanger schemes: a water or glycol loop that transfers heat from a heat exchanger in the flue to a heat exchanger in the entering air stream, a heat wheel, and a flue gas spray that supplies hot water to the entering air heat exchanger.

Direct injection of the flue gas into the entering air stream can be used if a damper is provided to stop injection when the absolute ambient humidity is high. This will result in complete flue gas recovery at least half of the time, and a much higher fraction of the time if latent load reduction and increased concentrator air flow measures are implemented, without sacrificing capacity when it is most needed. Direct injection should be just upstream of the hot spray coil.

It is important that all heat exchangers used in the two heat recovery schemes mentioned above be designed for optimized counterflow operation. This can be accomplished by using multiple heat pipe systems or multi-row (at least four) coils with an intermediate heat transfer fluid.

A third opportunity for heat recovery is enlargement of the existing desiccant recuperator and optimal control of the desiccant flow. Flow control will be most beneficial if latent load reduction and increased concentrator air flow measures are implemented because there will then be more partload hours. Entering desiccant flow can be controlled by variable-speed circulation pumps with speed proportional to total building latent load. Return can be controlled with a two-speed pump controlled on sump level or by connecting the existing pump in parallel with the variable-speed drive circulator pumps and adapting the existing float control so it can serve as a variable-speed drive override that shuts off the return pump on low sump and runs it at full speed on high sump.

Modification of the concentrator sump to eliminate mixing of the solution that falls from the hot spray coil with the solution that falls from the reflux coil will reduce the entropy loss that currently occurs when the two streams are mixed and will also increase the temperature difference that drives heat recovery in the recuperator. The desiccant return stream should be taken from this new sump. The desiccant supply stream should continue to flow into the old sump near the spray pump pickup.

With the increased surface areas and improved counterflow properties in the hot coil section of the concentrator, it should be possible to reduce the flow of desiccant that wets the coil surface to equal the desiccant supply flow plus reflux recovery rate. This will result in the closest approach to theoretical COP for this (single-stage) type of system but may require a complete redesign of the hot coil wetted surface geometry and the subsystem that distributes the desiccant solution over the coil.

6.0 CONCLUSIONS

The results of the technical and economic assessments show the ATES/TEG system at MPF to be a relatively costly system to own and operate.

Performance and cost estimates were developed for three HVAC systems. The baseline system used conventional chillers and air handlers and a conventional wet cooling tower. The alternate baseline was an extension of the baseline system that used HEPA filters and desiccant wheel humidity control to provide air quality and humidity control similar to that provided by the ATES TEG system. Total annual energy consumption for the modeled baseline HVAC system was predicted to be 1159.4 MWh with an additional 6850 gal of propane for heating. The alternate baseline system consumed 1375 MWh in electrical energy, 6850 gal of propane for heating, and 20,249 gal of propane for desiccant-wheel regeneration. By comparison, the predicted annual energy consumption for the ATES/TEG system is 1545 MWh, with 51,918 gal of propane used for liquid-desiccant regeneration.

However, the cost to operate the ATES/TEG system is not excessive relative to other buildings of its vintage. The HVAC energy consumption per square foot of conditioned floor area was predicted to be 17,700 Btu/ft^2 for the baseline, 30,400 Btu/ft^2 for the alternate baseline, and 45,000 Btu/ft^2 for the ATES/TEG system. The USPS design goal for the facility in 1982 was to reduce energy consumption below 88,000 Btu/ft^2.

The higher electrical energy consumption is primarily caused by higher supply air flow rates for the ATES/TEG system relative to the baseline. Note that while USPS concerns for control of dust made the inherently high air circulation rates and air washing of the ATES/TEG system attractive, the high air circulation rate was not used in the conventional HVAC technology cases because it is relatively uncommon.

Parasitic pumping energy required to operate the ATES system and liquid-desiccant subsystems is the second highest contributor to the annual electrical load. Mechanical refrigeration requirements of the ATES/TEG system were reduced by more than two-thirds compared to the baseline but this was not nearly enough to compensate for the increased fan and pump energy. It appears that the parasitic energy use could be greatly reduced by careful adjustment

and balancing of the system and application of variable-speed drives and improved control. The unique nature of the system presents a dilemma because, while proper design of the control system will require extensive simulation, there are no existing building simulation programs that can simulate the system without extensive software development or modification.

Initial capital and O&M costs are approximately four times higher for the ATES/TEG system than for the baseline. The alternate baseline system has about one-half the initial capital cost and slightly less O&M costs than the ATES/TEG system. The ATES/TEG system offers little, if any, reduction in the costs of conventional HVAC system components, while adding ATES and liquid-desiccant system costs that individually are each more expensive than the sum of the baseline system components. Note, however, that the MPF is not an ideal case study because construction costs are generally higher for government projects. This could bias the life-cycle cost results to favor the lower first cost alternatives.

Because of the significant fraction of life-cycle costs attributed to the ATES and liquid-desiccant elements, they are clear targets for cost reduction to improve the competitiveness of the concept.

The results of the technical assessment suggest several operational modifications that could be made to improve site energy performance at relatively low cost. These include:

- Control infiltration loads caused by shrink-wrap machines and other exhaust fans, and keep the platform doors closed. Dedicated air supplies for the shrink-wrap machines will reduce most of the latent load associated with 17,000 cfm of humid outside air when all three machines are operating.

- Improve efficiency of propane boiler by either replacement with a new high-efficiency model or by reclaim of flue gas heat.

- Operate the liquid-desiccant system main circulating and conditioner spray pumps only as needed, rather than continuously, or replace pump motors with variable- or two-speed pump models.

- Consider variable-speed pumping of well water to reduce pumping energy, to better match cooling loads (thus more effectively using the limited resource of stored chill), and to increase the amount of chill produced by the closed-circuit cooler in winter.

The predicted workroom cooling loads are estimated to be quite different from those in the original design. These differences provide the following additional opportunities for further improving system energy performance:

- Reduce design supply and return fan air flow rates on the workroom conditioners to better match actual cooling loads. Preliminary conservative estimates indicate an aggregate fan horsepower reduction of 25%. Further savings can be achieved by replacing motors with premium-efficiency motors, replacing blowers (or impellers only) with high efficiency units, replacing existing fan speed control with variable-speed drives and providing closed-loop room temperature/humidity control.

- Change the chilled water supply temperature setpoint from 50°F to 54°F. These higher chilled water temperatures should still provide sufficient sensible cooling of the space and, based on the aquifer model, could reduce mechanical refrigeration by an additional two-thirds.

Both the technical assessment and operational assessment of the liquid-desiccant system indicate that the concentrator is undersized on the order of 50%. Control of outside air infiltration, as described above, is the preferred way to alleviate this condition. Concentrator capacity and energy efficiency may also be improved by:

- increasing concentrator air flow rate by either increasing blower capacity or relieving air restrictions in the duct system. The intake hood and exhaust nozzle are obvious restrictions; the intake hood should be replaced with a larger unit; the exhaust nozzle should be replaced by a diverging stack of sufficient height to ensure that no more than 5% of the exhaust air finds its way back to the intake hood in calm air conditions. Smooth transitions in flow direction and cross-sectional area between the hood and concentrator inlet and between the concentrator outlet and exhaust stack must be provided. A larger, more efficient blower should probably be installed. The motor and blower can then be re-sheaved to obtain the highest flow rate possible with the existing motor or a larger motor if still necessary.

- improving concentrator thermal COP by recovering heat and avoiding the mixing of desiccant streams of different temperature and concentration which occurs in the current system. The addition of a heat pipe heat exchanger to extract heat upstream of the reflux coil and add it to the intake air upstream of the hot water coil is the first priority.

- enlarging the existing desiccant plate heat exchanger and possibly using return reflux water to further cool the desiccant solution returned to the system from the concentrator.

Although not cost-effective as implemented at the Melville MPF (given today's energy costs and cost projections), it should be remembered that the

ATES/TEG system is a unique combination of technologies for which there was no previous operating experience. Improvements in the understanding of the technologies have been gained both through this assessment and through the USPS's continued operation of the system at the MPF.

Further activity focused on implementation of the recommendations presented here is warranted because it will help further understanding of the operation characteristics of ATES/TEG systems and it will significantly reduce energy consumption and operational problems at the MPF.

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7.2 SITE AND DESIGN DATA SOURCES

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"Well S0598 (#6) Log," from the files of Henry Ku, US Geological Survey, Water Resources Div, Syosset, NY, transmitted 8-8-91.

Letter Report from Henry Ku, US Geological Survey, Water Resources Div, Syosset, NY, to D.L. Bratta, USPS MPF, 8-8-91.

"Desiccant Concentration Monitor," schematic of MPF desiccant system with monitoring points, D. Wilke, 8-7-91.

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USPS/MPF Melville Occupancy Schedule by Zone.

Block cooling loading and closed-circuit cooler capacity, Wilke for Spector/Hillier (based on a number of earlier documents; appendices contain well field and plant drawings, zone set temperatures, ventilation rates, lighting specs, surface reflectances, occupant loadings and work area schedules) 1987.

"Cost Effective Daylighting for Large Industrial Buildings," D. Wilke.

"U.S. Postal System Uses Thermal Energy," D.L. Bratta and D. Wilke.

Fan schedules.

System flow diagram depicts water and desiccant piping circuits.

Well operation logs (flow, temperature, operation) USPS, spring-summer 1992.

Pump schedule from original construction documents.

Equipment notes performance specifications for (among other things) boilers, closed-circuit cooler, and heat recovery chillers.

DDC description of system operation program documentation for HVAC control.

Closed-circuit cooler operation logs air and water temperature, water flow, relative humidity during 1991-1992 winter; Together with "well operation logs", provides a 1-year profile of the ATES system performance.

APPENDIX A

DOE-2 INPUT DECK OF MELVILLE MPF

APPENDIX A

DOE-2 INPUT DECK OF MELVILLE MPF

```
TITLE LINE-1 = * MELVILLE MPF, USPS * ..
TITLE LINE-2 = * BASE MODEL *  ..
TITLE LINE-4 = * PACIFIC NORTHWEST LABORATORY *  ..

INPUT LOADS ..
ABORT ERRORS ..
LIST CAUTIONS ..

BUILDING-LOCATION  $ From DOE-2 manual for NYC $
ATM-MOISTURE =
   (0.34,0.33,0.40,0.54,0.76,1.02,1.18,1.16,1.01,0.69,0.55,0.42)
ATM-TURBIDITY =
   (0.11,0.11,0.12,0.15,0.17,0.22,0.21,0.24,0.2,0.15,0.11,0.11)  ..

RUN-PERIOD  JAN 1, 1990 THRU DEC 31 1990  ..

$ --------- SCHEDULES ---------- $

WORKRM-LIGHTS = SCHEDULE  THRU DEC 31 (ALL) (1,24) (1)  ..

$ Occupancy schedules :  $
##fileprefix /user0/d3c431/Melville/Doe/
##include occdat
$ Lights and equip. schedules : $
##include eqpdat

$ -- CONSTRUCTIONS -- $

L-EXTWALL = LAYERS
   MATERIAL = (CC26,IN33) ..

L-EXTROOF = LAYERS
   MATERIAL = (BR01,IN47,IN42,AS01)  ..

L-INTWALL2 = LAYERS
   MATERIAL = (GP01,AL21,IN01,GP01) .. $1/2" GWB,AIR,R-7 BATT,1/2" GWB$

L-INTWALL6 = LAYERS
   MATERIAL = (CB51)  .. $ 8" HOLLOW CONC. BLOCK $

L-INTWALL11 = LAYERS
   MATERIAL = (GP01,AL31,IN01,GP01)  .. $ 1/2" GWB,AIR,R-7 BATT,1/2" GWB $
```

```
L-CEILING = LAYERS
   MATERIAL = (GP02)  .. $ 5/8" GWB $

L-FLOOR = LAYERS
   MATERIAL = (IN43,CC15)  .. $ 1" EXP. POLY INS., 6" HVYWGHT CONC.$

EXTWALL = CONSTRUCTION
   LAYERS = L-EXTWALL  ..

EXTROOF = CONSTRUCTION
   LAYERS = L-EXTROOF
   ABSORPTANCE = 0.15
   ROUGHNESS = 5  ..

INTWALL2 = CONSTRUCTION
   LAYERS = L-INTWALL2  ..

INTWALL6 = CONSTRUCTION
   LAYERS = L-INTWALL6 ..

INTWALL11 = CONSTRUCTION
   LAYERS = L-INTWALL11  ..

EXTFLOOR = CONSTRUCTION
   LAYERS = L-FLOOR  ..

AIRWALL = CONSTRUCTION
   U-VALUE = 2.6 ..

CEILING = CONSTRUCTION
   LAYERS = L-CEILING  ..

GLASS1 = GLASS-TYPE
   PANES = 2
   GLASS-TYPE-CODE = 1  ..

SKY1 = GLASS-TYPE
   GLASS-CONDUCTANCE = 0.001
   SHADING-COEF = 0
   VIS-TRANS = 0.0257  ..

SKY2 = GLASS-TYPE
   PANES = 3
   GLASS-TYPE-CODE = 1
   VIS-TRANS = 0  ..

SET-DEFAULT FOR EXTERIOR-WALL
   CONSTRUCTION = EXTWALL
   HEIGHT = 20  ..

$ Non-Workroom Conditioned Spaces $
ZONE-CONDITIONS = SPACE-CONDITIONS
```

```
   TEMPERATURE = (72)  $ BLOCK COOLING LOADING DOC. $
   PEOPLE-HG-LAT = 200  $ assume moderately active office work $
   PEOPLE-HG-SENS = 250
   FLOOR-WEIGHT = 0
   FURN-FRACTION = 0.5
   FURNITURE-TYPE = LIGHT
   FURN-WEIGHT = 20
   INF-SCHEDULE = WORKRM-LIGHTS
   INF-METHOD = AIR-CHANGE
   INF-CFM/SQFT = 0.044  ..

WORKROOM = SPACE-CONDITIONS
   TEMPERATURE = (72)
   PEOPLE-HG-LAT = 435 $ from design docs. $
   PEOPLE-HG-SENS = 345
   DAYLIGHTING = YES
   LIGHT-SET-POINT1 = 50.0
   LIGHT-CTRL-TYPE1 = STEPPED
   LIGHT-CTRL-STEPS = 3
   LIGHTING-SCHEDULE = WORKRM-LIGHTS
   FLOOR-WEIGHT = 0
   FURN-FRACTION = 0.75
   FURNITURE-TYPE = HEAVY
   FURN-WEIGHT = 100
   INF-SCHEDULE = WORKRM-LIGHTS
   INF-METHOD = AIR-CHANGE
   INF-CFM/SQFT = 0.044  ..

PLAT-CONDITIONS = SPACE-CONDITIONS
   TEMPERATURE = (50)
   PEOPLE-HG-LAT = 435
   PEOPLE-HG-SENS = 345
   INF-SCHEDULE = WORKRM-LIGHTS
   INF-METHOD = AIR-CHANGE
   INF-CFM/SQFT = 0.044  ..

$ --- MECHANICAL ROOM --- $

MECH = SPACE
  X = 0 Y = 560
   AREA = 6600
   VOLUME = 132000
   ZONE-TYPE = UNCONDITIONED
   EQUIPMENT-KW = 24.16
   EQUIP-SCHEDULE = WORKRM-LIGHTS
$ EXTERIOR LIGHTING - FOR ACCOUNTING PURPOSES: $
   LIGHTING-KW = 24
   LIGHTING-SCHEDULE = WORKRM-LIGHTS
   LIGHT-TO-SPACE = 0.0
$ BUILDING ELECTRIC HWH & EXHAUST FANS - FOR ACCOUNTING PURPOSES: $
$ ELECTRIC HWH =  9.6 KW = 32,722 BTU/HR
$ EXHAUST FANS = 12.4 KW = 42,310 BTU/HR
```

```
   SOURCE-TYPE = ELECTRIC
   SOURCE-BTU/HR = 75033
   SOURCE-SCHEDULE = WORKRM-LIGHTS
   SOURCE-SENSIBLE = 0.0
   SOURCE-LATENT = 0.0
   INF-SCHEDULE = WORKRM-LIGHTS
   INF-METHOD = AIR-CHANGE
   AIR-CHANGES/HR = 0.50  ..

MECH-FLOOR = UNDERGROUND-FLOOR
   CONSTRUCTION = EXTFLOOR  TILT = 180
   AREA = 6600
   U-EFFECTIVE = 0.0034  ..

MECH-ROOF = ROOF
   CONS = EXTROOF  Z = 20  TILT = 0
   WIDTH = 81.24  HEIGHT = 81.24 ..

MECH-EW1 = EXTERIOR-WALL X = 0  Y = 110  Z = 4
   AZIMUTH = 270
   WIDTH = 110
   HEIGHT = 16
   ..

MECH-BERM = UNDERGROUND-WALL
   AREA = 440
   CONSTRUCTION = EXTWALL  ..

MECH-EW2 = EXTERIOR-WALL X = 60 Y = 110 Z = 4
   AZIMUTH = 0
   WIDTH = 60
   HEIGHT = 16
   ..

MECH-BERM2 = UNDERGROUND-WALL
   AREA = 240
   CONSTRUCTION = EXTWALL  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 110
   HEIGHT = 10
   NEXT-TO = ZONE2
   ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL11
   WIDTH = 60
   HEIGHT = 20
   NEXT-TO = ZONE3
   INSIDE-VIS-REFL = (0.70, 0.75)  ..
```

```
$ --- ZONE2 --- $

ZONE2 = SPACE  X = 60  Y = 560
   AREA = 18150
   VOLUME = 181500
   SPACE-CONDITIONS = ZONE-CONDITIONS
   PEOPLE-SCHEDULE = P-AC-2
   NUMBER-OF-PEOPLE = 150
   LIGHTING-W/SQFT = 1.4
   LIGHTING-SCHEDULE = EQ-AC-8C  $ COMMON SCHEDULE WITH DE-RATE FACTOR
   EQUIPMENT-KW = 37.5 $ DE-RATE FATOR = 0.375 $
$ KITCHEN EQUIPMENT KW BASED ON ENERGY USE AND SCHEDULES. $
   EQUIP-SCHEDULE = EQ-AC-8C
$ KITCHEN PROPANE USE:
   SOURCE-TYPE = GAS
   SOURCE-BTU/HR = 56301
   SOURCE-SCHEDULE = EQ-AC-8C
   SOURCE-SENSIBLE = 0.33
   SOURCE-LATENT = 0.05 ..

ZONE2-FLOOR = UNDERGROUND-FLOOR
   CONSTRUCTION = EXTFLOOR  TILT = 180
   AREA = 18150
   U-EFFECTIVE = 0.0087  ..

ZONE2-EW1 = EXTERIOR-WALL
   X = 160  Y = 110  Z = 4
   AZIMUTH = 0
   WIDTH = 165
   HEIGHT = 6  ..

ZONE2-BERM = UNDERGROUND-WALL
   AREA = 660
   CONSTRUCTION = EXTWALL  ..

ZONE2-CEILING = INTERIOR-WALL
   CONSTRUCTION = CEILING
   Z = 10  TILT = 0
   WIDTH = 134.72  HEIGHT = 134.72
   NEXT-TO = PLEN  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL11
   WIDTH = 140
   HEIGHT = 10
   NEXT-TO = ZONE3
   INSIDE-VIS-REFL = (0.70, 0.75)  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL11
   WIDTH = 35
   HEIGHT = 10
```

```
   NEXT-TO = ZONE5
   INSIDE-VIS-REFL = (0.70, 0.75)  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 15
   HEIGHT = 10
   NEXT-TO = ZONE1A  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 65  HEIGHT = 10
   NEXT-TO = ZONE1  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 10  HEIGHT = 10
   NEXT-TO = ZONE11  ..

$ --- ZONE1A --- $
ZONE1A = SPACE
  X = 220 Y = 650
   AREA = 1950
   VOLUME = 19500
   SPACE-CONDITIONS = ZONE-CONDITIONS
   PEOPLE-SCHEDULE = P-AC-1
   NUMBER-OF-PEOPLE = 28
   LIGHTING-W/SQFT = 1.4
   LIGHTING-SCHEDULE = EQ-AC-1
   EQUIPMENT-KW = 5.16
   EQUIP-SCHEDULE = EQ-AC-1 ..

ZONE1A-FLOOR = UNDERGROUND-FLOOR
   CONSTRUCTION = EXTFLOOR  TILT = 180
   AREA = 1950
   U-EFFECTIVE = 0.0087  ..

ZONE1A-EW = EXTERIOR-WALL X = 165  Y = 15  Z = 4
   WIDTH = 130
   HEIGHT = 6
   AZIMUTH = 0
   ..

WINDOW X = 20  Y = 0.5
   HEIGHT = 4.4375  WIDTH = 75
   GLASS-TYPE = GLASS1
   OVERHANG-W = 75
   OVERHANG-D = 4  ..

ZONE1A-BERM = UNDERGROUND-WALL
   AREA = 520
   CONSTRUCTION = EXTWALL  ..
```

```
ZONE1A-CEILING = INTERIOR-WALL
   CONSTRUCTION = CEILING
   Z = 10  TILT = 0
   WIDTH = 44.16  HEIGHT = 44.16
   NEXT-TO = PLEN  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL2
   WIDTH = 15  HEIGHT = 10
   NEXT-TO = ZONE12  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL2
   WIDTH = 130  HEIGHT = 10
   NEXT-TO = ZONE1  ..

$ --- ZONE12 --- $
ZONE12 = SPACE
  X = 350  Y = 650
   AREA = 150  VOLUME = 1500
   SPACE-CONDITIONS = ZONE-CONDITIONS
   EQUIPMENT-KW = 4.1
   EQUIP-SCHEDULE = WORKRM-LIGHTS
   INF-CFM/SQFT = 0.0  ..

ZONE12-FLOOR = UNDERGROUND-FLOOR
   CONSTRUCTION = EXTFLOOR  TILT= 180
   AREA = 150
   U-EFFECTIVE = 0.0087  ..

ZONE12-EW = EXTERIOR-WALL  X = 10  Y = 15  Z = 4
   HEIGHT = 6
   WIDTH = 10
   AZIMUTH = 0  ..

ZONE12-BERM = UNDERGROUND-WALL
   AREA = 40
   CONSTRUCTION = EXTWALL  ..

ZONE12-CEILING = INTERIOR-WALL
   CONS = CEILING  Z = 10  TILT = 0
   WIDTH = 12.25  HEIGHT = 12.25
   NEXT-TO = PLEN  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL2
   WIDTH = 10 HEIGHT = 10
   NEXT-TO = ZONE1 ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL2
   WIDTH = 15  HEIGHT = 10
```

```
   NEXT-TO = ZONE1B ..

$ --- ZONE11 --- $
ZONE11 = SPACE
  X = 220  Y = 570
   AREA = 400  VOLUME = 4000
   SPACE-CONDITIONS = ZONE-CONDITIONS
   EQUIPMENT-KW = 18
   EQUIP-SCHEDULE = WORKRM-LIGHTS
   INF-CFM/SQFT = 0.0  ..

ZONE11-FLOOR = UNDERGROUND-FLOOR
   CONSTRUCTION = EXTFLOOR  TILT = 180
   AREA = 400
   U-EFFECTIVE = 0.0001  ..

ZONE11-CEILING = INTERIOR-WALL
   CONS = CEILING  Z = 10  TILT = 0
   WIDTH = 20  HEIGHT = 20
   NEXT-TO = PLEN  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 20  HEIGHT = 10
   NEXT-TO = ZONE5
   INSIDE-VIS-REFL = (0.70, 0.75)  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL2
   WIDTH = 40  HEIGHT = 10
   NEXT-TO = ZONE1  ..

$ --- ZONE 1 --- $
ZONE1 = SPACE
  X = 240  Y = 590
   AREA = 10850  VOLUME = 108500
   SPACE-CONDITIONS = ZONE-CONDITIONS
   PEOPLE-SCHEDULE = P-AC-1
   NUMBER-OF-PEOPLE = 12
   LIGHTING-W/SQFT = 1.4
   LIGHTING-SCHEDULE = EQ-AC-1
   EQUIPMENT-KW = 7.93
   EQUIP-SCHEDULE = EQ-AC-1
   ..

ZONE1-FLOOR = UNDERGROUND-FLOOR
   CONSTRUCTION = EXTFLOOR
   AREA = 10850  TILT = 180
   U-EFFECTIVE = 0.0001  ..

ZONE1-CEILING = INTERIOR-WALL
   CONS = CEILING  TILT = 0  Z = 10
```

```
   WIDTH = 104.16  HEIGHT = 104.16
   NEXT-TO = PLEN  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 90  HEIGHT = 10
   NEXT-TO = ZONE5
   INSIDE-VIS-REFL = (0.70, 0.75)  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL2
   WIDTH = 135  HEIGHT = 10
   NEXT-TO = ZONE1B  ..

$ --- ZONE 1B --- $
ZONE1B = SPACE
  X = 320 Y = 395
   AREA = 12275  VOLUME = 209750  $ 8750 @ 20 AFF
   SPACE-CONDITIONS = ZONE-CONDITIONS
   PEOPLE-SCHEDULE = P-AC-1B
   NUMBER-OF-PEOPLE = 40
   LIGHTING-W/SQFT = 1.4
   LIGHTING-SCHEDULE = EQ-AC-8C  $ COMMON SCHEDULE WITH DE-RATE FACTOR
   EQUIPMENT-KW = 3.3075  $ DE-RATED ACTUAL KW BY 0.75
   EQUIP-SCHEDULE = EQ-AC-8C  ..

ZONE1B-FLOOR = UNDERGROUND-FLOOR
   CONS = EXTFLOOR  TILT = 180
   AREA = 12275
   U-EFFECTIVE = 0.0031  ..

ZONE1B-EW = EXTERIOR-WALL
   X = 60  Y = 225  Z = 4
   AZIMUTH = 0
   HEIGHT = 6
   WIDTH = 15  ..

ZONE1B-BERM = UNDERGROUND-WALL
   AREA = 60
   CONSTRUCTION = EXTWALL  ..

ZONE1B-EW2 = EXTERIOR-WALL
   X = 60  Y = 175
   AZIMUTH = 90
   WIDTH = 100
   HEIGHT = 10  ..

WINDOW  X = 15 Y = 4
   WIDTH = 80.5  HEIGHT = 4.4375
   GLASS-TYPE = GLASS1
   OVERHANG-W = 80.5
   OVERHANG-D = 4  ..
```

```
ZONE1B-EW3 = EXTERIOR-WALL
   X = 60  Y = 20
   AZIMUTH = 90
   WIDTH = 145  ..

ZONE1B-CEILING1 = INTERIOR-WALL
   CONS = CEILING  TILT = 0  Z = 10
   WIDTH = 59.79  HEIGHT = 59.79
   NEXT-TO = PLEN  ..

ZONE1B-ROOF = ROOF
   CONSTRUCTION = EXTROOF
   TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 60  HEIGHT = 145.83  ..

INTERIOR-WALL
   WIDTH = 150  HEIGHT = 20
   CONSTRUCTION = INTWALL6
   NEXT-TO = ZONE5
   INSIDE-VIS-REFL = (0.70, 0.75)  ..

INTERIOR-WALL
   WIDTH = 30  HEIGHT = 20
   CONSTRUCTION = INTWALL6
   NEXT-TO = ZONE6
   INSIDE-VIS-REFL = (0.70, 0.75)  ..

INTERIOR-WALL
   WIDTH = 55  HEIGHT = 20
   CONSTRUCTION = INTWALL6
   NEXT-TO = PLAT1  ..

$ --- PLEN ---$
PLEN = SPACE
 X = 60  Y = 560  Z = 10
   AREA = 35075  VOLUME = 350750
   ZONE-TYPE = UNCONDITIONED  ..

PLEN-EW1 = EXTERIOR-WALL
   X = 320  Y = 110
   CONSTRUCTION = EXTWALL
   HEIGHT = 10  WIDTH = 320
   AZIMUTH = 0  ..

PLEN-EW2 = EXTERIOR-WALL
   X = 320  Y = 0
   CONSTRUCTION = EXTWALL
   HEIGHT = 10  WIDTH = 110
   AZIMUTH = 90  ..

PLEN-ROOF = ROOF
```

```
   CONSTRUCTION = EXTROOF
   TILT = 0  Z = 10
   AZIMUTH = 180
   WIDTH = 320  HEIGHT = 110  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 110  HEIGHT = 10
   NEXT-TO = MECH  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL11
   WIDTH = 140  HEIGHT = 10
   NEXT-TO = ZONE3
   INSIDE-VIS-REFL = (0.70, 0.75)  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL11
   WIDTH = 120  HEIGHT = 10
   NEXT-TO = ZONE5
   INSIDE-VIS-REFL = (0.70, 0.75)  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL11
   WIDTH = 60  HEIGHT = 10
   NEXT-TO = ZONE1B  ..

$ --- ZONE 5 --- $
ZONE5 = SPACE
 X = 200  Y = 410
   AREA = 18200  VOLUME = 364000
   SPACE-CONDITIONS = WORKROOM
   PEOPLE-SCHEDULE = P-AC-5
   NUMBER-OF-PEOPLE = 33
   LIGHTING-W/SQFT = 0.9
   LIGHT-REF-POINT1 = (60,70,3.3)
   EQUIPMENT-KW = 34.72
   EQUIP-SCHEDULE = EQ-AC-5  ..

ZONE5-FLOOR = UNDERGROUND-FLOOR
   CONS = EXTFLOOR  TILT = 180
   AREA = 18200
   U-EFFECTIVE = 0.0001
   INSIDE-VIS-REFL = 0.40  ..

ZONE5-ROOF1 = ROOF
   CONSTRUCTION = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 118.7  HEIGHT = 118.7  ..

ZONE5-SKY1 = WINDOW  X = 10  Y = 10
   HEIGHT = 118.3  WIDTH = 118.3
```

```
   GLASS-TYPE = SKY1
   INSIDE-VIS-REFL = 0.0  ..

ZONE5-ROOF2 = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 120  HEIGHT = 151.67  ..

ZONE5-SKY2 = WINDOW
   WIDTH = 25.61  HEIGHT = 25.61
   GLASS-TYPE = SKY2  ..

INTERIOR-WALL  X = 0  Y = 0
   WIDTH = 120  HEIGHT = 20
   CONSTRUCTION = AIRWALL
   INT-WALL-TYPE = AIR
   NEXT-TO = ZONE6
   INSIDE-VIS-REFL = (0.0660, 0.0573)  ..

INTERIOR-WALL  X = 0  Y = 150
   WIDTH = 150  HEIGHT = 20
   CONSTRUCTION = AIRWALL
   INT-WALL-TYPE = AIR
   NEXT-TO = ZONE3
   INSIDE-VIS-REFL = (0.0516, 0.0822)  ..

$ --- ZONE 3 --- $
ZONE3 = SPACE
  X = 0  Y = 410
   AREA = 31100  VOLUME = 622000
   SPACE-CONDITIONS = WORKROOM
   PEOPLE-SCHEDULE = P-AC-3
   NUMBER-OF-PEOPLE = 84
   LIGHTING-W/SQFT = 0.9
   LIGHT-REF-POINT1 = (100,70,3.3)
   EQUIPMENT-KW = 51.92
   EQUIP-SCHEDULE = EQ-AC-3
   ..

ZONE3-FLOOR = UNDERGROUND-FLOOR
   CONS = EXTFLOOR  TILT = 180
   AREA = 31100
   U-EFFECTIVE = 0.0048
   INSIDE-VIS-REFL = 0.4  ..

ZONE3-EW1 = EXTERIOR-WALL
   X = 0  Y = 150
   WIDTH = 150
   AZIMUTH = 270
   INSIDE-VIS-REFL = 0.75  ..

ZONE3-ROOF1 = ROOF
```

```
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 163.2  HEIGHT = 163.2  ..

ZONE3-SKY1 = WINDOW LIKE ZONE5-SKY1
   HEIGHT = 163.1  WIDTH = 163.1  ..

ZONE3-ROOF2 = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 207.33  HEIGHT = 150  ..

ZONE3-SKY2 = WINDOW LIKE ZONE5-SKY2
   HEIGHT = 33.23  WIDTH = 33.23  ..

INTERIOR-WALL
   CONSTRUCTION = AIRWALL
   WIDTH = 100  HEIGHT = 20
   INT-WALL-TYPE = AIR
   NEXT-TO = ZONE4
   INSIDE-VIS-REFL = (0.0345, 0.0332)  ..

INTERIOR-WALL  X = -5  Y = 90
   CONSTRUCTION = AIRWALL
   WIDTH = 110  HEIGHT = 20
   INT-WALL-TYPE = AIR
   NEXT-TO = ZONE8
   INSIDE-VIS-REFL = (0.0379, 0.0613)  ..

INTERIOR-WALL  X = -5  Y = 200
   CONSTRUCTION = AIRWALL
   WIDTH = 10  HEIGHT = 20
   INT-WALL-TYPE = AIR
   NEXT-TO = ZONE6
   INSIDE-VIS-REFL = (0.0035, 0.0048)  ..

$ --- ZONE 4 --- $
$ From the revised design document, this is a manual sort area - there
$ is no equipment in use.
ZONE4 = SPACE
  X = 0  Y = 70
   AREA = 31000  VOLUME = 620000
   SPACE-CONDITIONS = WORKROOM
   PEOPLE-SCHEDULE = P-AC-4
   NUMBER-OF-PEOPLE = 92
   LIGHTING-W/SQFT = 0.9
   LIGHT-REF-POINT1 = (40,160,3.3)
   ..

ZONE4-FLOOR = UNDERGROUND-FLOOR
   CONS = EXTFLOOR  TILT = 180
   AREA = 31000
```

```
   U-EFFECTIVE = 0.0014
   INSIDE-VIS-REFL = 0.4  ..

ZONE4-EW1 = EXTERIOR-WALL
   X = 0  Y = 340
   WIDTH = 300
   AZIMUTH = 270
   INSIDE-VIS-REFL = 0.75  ..

ZONE4-EW2 = EXTERIOR-WALL
   X = 0  Y = 40
   WIDTH = 40
   HEIGHT = 16
   AZIMUTH = 270
   INSIDE-VIS-REFL = 0.75  ..

ZONE4-BERM = UNDERGROUND-WALL
   AREA = 160
   CONSTRUCTION = EXTWALL
   INSIDE-VIS-REFL = 0.75  ..

ZONE4-ROOF1 = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   HEIGHT = 162.5  WIDTH = 162.5  ..

ZONE4-SKY1 = WINDOW LIKE ZONE5-SKY1
   WIDTH = 162.48  HEIGHT = 162.48  ..

ZONE4-ROOF2 = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 90  HEIGHT = 344.44  ..

ZONE4-SKY2 = WINDOW LIKE ZONE5-SKY2
   WIDTH = 33.23  HEIGHT = 33.23  ..

INTERIOR-WALL  X = 90  Y = 165
   CONSTRUCTION = AIRWALL
   WIDTH = 160  HEIGHT = 20
   INT-WALL-TYPE = AIR
   NEXT-TO = ZONE8
   INSIDE-VIS-REFL = (0.0528, 0.0886)  ..

INTERIOR-WALL  X = 90  Y = 0
   CONSTRUCTION = AIRWALL
   WIDTH = 180  HEIGHT = 20
   INT-WALL-TYPE = AIR
   NEXT-TO = ZONE9
..

INTERIOR-WALL
```

```
   CONSTRUCTION = INTWALL6
   WIDTH = 90  HEIGHT = 8
   NEXT-TO = ZONE8C
   INSIDE-VIS-REFL = (0.70, 0.75)  ..

INTERIOR-WALL  X = 70  Y = 0
   CONSTRUCTION = INTWALL6
   WIDTH = 15  HEIGHT = 20
   NEXT-TO = PLAT2  ..

$ --- ZONE 8 --- $
ZONE8 = SPACE
  X = 90  Y = 230
   AREA = 18150  VOLUME = 363000
   SPACE-CONDITIONS = WORKROOM
   PEOPLE-SCHEDULE = P-AC-8
   NUMBER-OF-PEOPLE = 118
   LIGHTING-W/SQFT = 0.9
   LIGHT-REF-POINT1 = (50,75,3.3)
   EQUIPMENT-KW = 53.4
   EQUIP-SCHEDULE = EQ-AC-8  ..

ZONE8-FLOOR = UNDERGROUND-FLOOR
   CONS = EXTFLOOR  TILT = 180
   AREA = 18150
   U-EFFECTIVE = 0.0001
   INSIDE-VIS-REFL = 0.4  ..

ZONE8-ROOF1 = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   HEIGHT = 115.4  WIDTH = 115.4  ..

ZONE8-SKY1 = WINDOW LIKE ZONE5-SKY1
   HEIGHT = 115.3  WIDTH = 115.3  ..

ZONE8-ROOF2 = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 110  HEIGHT = 165  ..

ZONE8-SKY2 = WINDOW LIKE ZONE5-SKY2
   HEIGHT = 25.6  WIDTH = 25.6  ..

INTERIOR-WALL  X = 0  Y = 0
   CONSTRUCTION = AIRWALL
   WIDTH = 110  HEIGHT = 20
   INT-WALL-TYPE = AIR
   NEXT-TO = ZONE9
   INSIDE-VIS-REFL = (0.0613, 0.0385)  ..

INTERIOR-WALL  X = 110  Y = 15
```

```
   CONSTRUCTION = AIRWALL
   WIDTH = 150  HEIGHT = 20
   INT-WALL-TYPE = AIR
   NEXT-TO ZONE6
   INSIDE-VIS-REFL = (0.0832, 0.0714)  ..

INTERIOR-WALL  X = 110  Y = 0
   CONSTRUCTION = AIRWALL
   WIDTH = 10  HEIGHT = 20
   INT-WALL-TYPE = AIR
   NEXT-TO = ZONE7
   INSIDE-VIS-REFL = (0.0057, 0.0098)  ..

$ --- ZONE 6 --- $
ZONE6 = SPACE
  X = 200  Y = 245
   AREA = 21600  VOLUME = 414000
   SPACE-CONDITIONS = WORKROOM
   PEOPLE-SCHEDULE = P-AC-6
   NUMBER-OF-PEOPLE = 35
   LIGHTING-W/SQFT = 0.9
   LIGHT-REF-POINT1 = (60,80,3.3)
   EQUIPMENT-KW = 6.632
   EQUIP-SCHEDULE = EQ-AC-6
   ..

ZONE6-FLOOR = UNDERGROUND-FLOOR
   CONS = EXTFLOOR  TILT = 180
   AREA = 21600
   U-EFFECTIVE = 0.0001
   INSIDE-VIS-REFL = 0.4  ..

ZONE6-ROOF1 = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 122.5  HEIGHT =122.5  ..

ZONE6-SKY1 = WINDOW LIKE ZONE5-SKY1
   WIDTH = 122.47  HEIGHT = 122.47  ..

ZONE6-ROOF2 = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 135  HEIGHT = 160  ..

ZONE6-SKY2 = WINDOW LIKE ZONE5-SKY2
     WIDTH = 27.7  HEIGHT = 27.7  ..

INTERIOR-WALL  X = 0  Y = 0
   CONSTRUCTION = AIRWALL
   WIDTH = 120  HEIGHT = 20
   INT-WALL-TYPE = AIR
```

```
   NEXT-TO = ZONE7
   INSIDE-VIS-REFL = (0.0573, 0.1143)  ..

INTERIOR-WALL  X = 140  Y = 0
   CONSTRUCTION = INTWALL6
   WIDTH = 150  HEIGHT = 20
   NEXT-TO = PLAT1
   INSIDE-VIS-REFL = (0.75, 0.70)  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 30  HEIGHT = 15
   NEXT-TO = ZONE10
   INSIDE-VIS-REFL = (0.75, 0.70)  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 20  HEIGHT = 20
   NEXT-TO = ZONE7
   INSIDE-VIS-REFL = (0.75, 0.75)  ..

$ --- PLATFORM 1 --- $
PLAT1 = SPACE
  X = 350  Y = 230
   SPACE-CONDITIONS = PLAT-CONDITIONS
   AREA = 7600  VOLUME = 152000
   PEOPLE-SCHEDULE = P-PLAT
   NUMBER-OF-PEOPLE = 6
   LIGHTING-W/SQFT = 0.9
   LIGHTING-SCHEDULE = EQ-AC-PLAT1
   ..

PLAT1-FLOOR = UNDERGROUND-FLOOR
   CONS = EXTFLOOR  TILT = 180
   AREA = 7600  ..

PLAT1-EW1 = EXTERIOR-WALL
  X = 40 Y = 0
   AZIMUTH = 90
   WIDTH = 190 ..

PLAT1-ROOF = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 87.18  HEIGHT = 87.18  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 40  HEIGHT = 20
   NEXT-TO = ZONE10  ..

$ --- ZONE 10 --- $
```

```
ZONE10 = SPACE
  X = 320  Y = 165
   AREA = 3950  VOLUME = 59250
   SPACE-CONDITIONS = ZONE-CONDITIONS
   PEOPLE-SCHEDULE = P-AC-10
   NUMBER-OF-PEOPLE = 14
   EQUIPMENT-KW = 9.96
   EQUIP-SCHEDULE = P-AC-10
   LIGHTING-W/SQFT = 1.75
   LIGHTING-SCHEDULE = P-AC-10  ..

ZONE10-FLOOR = UNDERGROUND-FLOOR
   CONS = EXTFLOOR  TILT = 180
   AREA = 3950
   U-EFFECTIVE = 0.0021
   ..

ZONE10-EW1 = EXTERIOR-WALL
   X = 60  Y = 0  Z = 4
   HEIGHT = 16
   WIDTH = 65
   AZIMUTH = 90 ..

ZONE10-BERM = UNDERGROUND-WALL
   AREA = 260
   CONSTRUCTION = EXTWALL
   INSIDE-VIS-REFL = 0.75  ..

ZONE10-ROOF = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 62.85  HEIGHT = 62.85  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 55 HEIGHT = 15
   NEXT-TO = ZONE7
   INSIDE-VIS-REFL = (0.70, 0.75)  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 40  HEIGHT = 15
   NEXT-TO = ZONE9  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 20  HEIGHT = 15
   NEXT-TO = STOR  ..

$ --- ZONE 7 --- $
ZONE7 = SPACE
  X = 200  Y = 170
```

```
   AREA = 9600  VOLUME = 192000
   SPACE-CONDITIONS = WORKROOM
   PEOPLE-SCHEDULE = P-AC-7
   NUMBER-OF-PEOPLE = 18
   LIGHTING-W/SQFT = 0.9
   LIGHT-REF-POINT1 = (60,38,3.3)
   EQUIPMENT-KW = 5.25
   EQUIP-SCHEDULE = EQ-AC-7 ..

ZONE7-FLOOR = UNDERGROUND-FLOOR
   CONS = EXTFLOOR  TILT = 180
   AREA = 9600
   U-EFFECTIVE = 0.0001
   INSIDE-VIS-REFL = 0.4  ..

ZONE7-ROOF1 = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 81.3  HEIGHT = 81.3  ..

ZONE7-SKY1 = WINDOW LIKE ZONE5-SKY1
   WIDTH = 81.24  HEIGHT = 81.24  ..

ZONE7-ROOF2 = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 120  HEIGHT = 80  ..

ZONE7-SKY2 = WINDOW LIKE ZONE5-SKY2
   WIDTH = 18.33  HEIGHT = 18.33  ..

INTERIOR-WALL
   CONSTRUCTION = AIRWALL
   WIDTH = 190  HEIGHT = 20
   INT-WALL-TYPE = AIR
   NEXT-TO = ZONE9
   INSIDE-VIS-REFL = (0.1780, 0.0661)  ..

$ --- ZONE 9 --- $
ZONE9 = SPACE
  X = 90  Y = 50
   AREA = 33650  VOLUME = 649800  $ 1400 @ 8 AFF, 800 @ 12 AFF $
   SPACE-CONDITIONS = WORKROOM
   PEOPLE-SCHEDULE = P-AC-9
   NUMBER-OF-PEOPLE = 92
   LIGHTING-W/SQFT = 0.9
   LIGHT-REF-POINT1 = (90,80,3.3)
   EQUIPMENT-KW = 83.63
   EQUIP-SCHEDULE = EQ-AC-9
   ..

ZONE9-FLOOR = UNDERGROUND-FLOOR
```

```
   CONS = EXTFLOOR  TILT = 180
   AREA = 33650
   U-EFFECTIVE = 0.0001
   INSIDE-VIS-REFL = 0.4  ..

ZONE9-ROOF1 = ROOF
   CONS = EXTROOF TILT = 0  Z= 20
   AZIMUTH = 180
   WIDTH = 87.2  HEIGHT = 87.2  ..

ZONE9-SKY1 = WINDOW LIKE ZONE5-SKY1
   WIDTH = 87.18  HEIGHT = 87.18  ..

ZONE9-ROOF2 = ROOF
   CONS = EXTROOF TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 130.4  HEIGHT = 130.4  ..

ZONE9-SKY2 = WINDOW LIKE ZONE5-SKY1
   WIDTH = 130.38  HEIGHT = 130.38  ..

ZONE9-ROOF3 = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 220  HEIGHT = 152.95  ..

ZONE9-SKY3 = WINDOW LIKE ZONE5-SKY2
   WIDTH = 34.64  HEIGHT = 34.64  ..

INTERIOR-WALL  X = 90 Y = 0
   CONSTRUCTION = INTWALL6
   WIDTH = 235  HEIGHT = 20
   NEXT-TO = STOR
   INSIDE-VIS-REFL = (0.75, 0.70)  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 170  HEIGHT = 20
   NEXT-TO = PLAT2
   INSIDE-VIS-REFL = (0.75, 0.70)  ..

$ --- STOR --- $
STOR = SPACE
  X = 260  Y = 0
   AREA = 13700  VOLUME = 274000
   SPACE-CONDITIONS = ZONE-CONDITIONS
   LIGHTING-W/SQFT = 0.9  ..

STOR-FLOOR = UNDERGROUND-FLOOR
   CONS = EXTFLOOR  TILT = 180
   AREA = 13700
   U-EFFECTIVE = 0.0027
```

```
   ..

STOR-EW1 = EXTERIOR-WALL
   X = 120  Y = 0  Z = 4
   HEIGHT = 16
   AZIMUTH = 90
   WIDTH = 165  ..

STOR-BERM = UNDERGROUND-WALL
   AREA = 660
   CONSTRUCTION = EXTWALL  ..

STOR-EW2 = EXTERIOR-WALL
   X = 0  Y = 0
   AZIMUTH = 180
   WIDTH = 120  ..

STOR-ROOF = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 117.05  HEIGHT = 117.05  ..

INTERIOR-WALL
   CONSTRUCTION = INTWALL6
   WIDTH = 55  HEIGHT = 20
   NEXT-TO = PLAT2  ..

$ --- PLATFORM 2 --- $
PLAT2 = SPACE
 X = 20  Y = 0
   AREA = 12000  VOLUME = 240000
   SPACE-CONDITIONS = PLAT-CONDITIONS
   PEOPLE-SCHEDULE = P-PLAT
   NUMBER-OF-PEOPLE = 10
   LIGHTING-W/SQFT = 0.9
   LIGHTING-SCHEDULE = EQ-AC-PLAT2
   ..


PLAT2-FLOOR = UNDERGROUND-FLOOR
   CONS = EXTFLOOR  TILT = 180
   AREA = 12000
   ..

PLAT2-EW1 = EXTERIOR-WALL
   WIDTH = 240
   AZIMUTH = 180  ..

PLAT2-ROOF = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 109.54  HEIGHT = 109.54  ..
```

```
INTERIOR-WALL
   CONSTRUCTION = EXTWALL
   WIDTH = 120  HEIGHT = 20
   NEXT-TO = ZONE8C  ..

$ --- ZONE 8C --- $
ZONE8C = SPACE
   AREA = 2400  VOLUME = 24000 $ 10' AFF
   SPACE-CONDITIONS = ZONE-CONDITIONS
   LIGHTING-W/SQFT = 0.917
   LIGHTING-SCHEDULE = EQ-AC-8C
   EQUIPMENT-KW = 1.73
   EQUIP-SCHEDULE = EQ-AC-8C
   NUMBER-OF-PEOPLE = 3
   PEOPLE-SCHEDULE = EQ-AC-8C  ..

ZONE8C-FLOOR = UNDERGROUND-FLOOR
   CONS=EXTFLOOR  TILT = 180
   AREA = 2400
   U-EFFECTIVE = 0.0049
   ..

ZONE8C-EW1 = EXTERIOR-WALL
   X = 0  Y = 70  Z = 4
   HEIGHT = 16
   WIDTH = 70
   AZIMUTH = 270  ..

ZONE8C-BERM = UNDERGROUND-WALL
   AREA = 280
   CONSTRUCTION = EXTWALL  ..

ZONE8C-EW2 = EXTERIOR-WALL
   WIDTH = 20
   AZIMUTH = 180  ..

ZONE8C-ROOF = ROOF
   CONS = EXTROOF  TILT = 0  Z = 20
   AZIMUTH = 180
   WIDTH = 48.99  HEIGHT =48.99  ..

$ --- BUILDING RESOURCE --- $

$ --- REPORTS --- $
LOADS-REPORT
   VERIFICATION (LV-B)
   SUMMARY (LS-A,LS-B,LS-C,LS-D,LS-F,LS-K)  ..

END ..
COMPUTE LOADS  ..

$ ------------------------------------------------------------ $
```

```
$ --------------  SYSTEM INPUT  ---------------------------- $

INPUT SYSTEMS ..

$ --- SCHEDULES --- $
HEAT-TEMP = SCHEDULE THRU DEC 31 (ALL) (1,24) (65)  ..
COOL-TEMP = SCHEDULE THRU DEC 31 (ALL) (1,24) (78)  ..
PLAT-TEMP = SCHEDULE THRU DEC 31 (ALL) (1,24) (50)  ..

$ --- ZONES --- $
ZONE5 = ZONE
   DESIGN-HEAT-T = 65
   DESIGN-COOL-T = 78
   HEAT-TEMP-SCH = HEAT-TEMP
   COOL-TEMP-SCH = COOL-TEMP
   THERMOSTAT-TYPE = PROPORTIONAL
   THROTTLING-RANGE = 4
   MIN-CFM-RATIO = 0.3
   OUTSIDE-AIR-CFM = 700
   ..

ZONE1A = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 240
   ..

ZONE1 = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 3000
   ..

ZONE1B = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 1200
   ..

ZONE2 = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 6800
   ..

ZONE3 = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 1960
   ..

ZONE4 = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 2650
   ..

ZONE8 = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 1845
   ..

ZONE8C = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 205
   ..
```

```
ZONE6 = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 2050
   ..

ZONE7 = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 1500
   ..

ZONE9 = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 2450
   ..

ZONE10 = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 1680
   ..

ZONE11 = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 0.0
   ..

ZONE12 = ZONE LIKE ZONE5
   OUTSIDE-AIR-CFM = 0.0
   ..

STOR = ZONE
   DESIGN-HEAT-T = 50
   HEAT-TEMP-SCH = PLAT-TEMP
   DESIGN-COOL-T = 78
   OUTSIDE-AIR-CFM = 0.0
   ..

PLAT1 = ZONE LIKE STOR  ..
PLAT2 = ZONE LIKE STOR  ..

MECH = ZONE  ZONE-TYPE = UNCONDITIONED  ..

PLEN = ZONE  ZONE-TYPE = UNCONDITIONED  ..

AC-1 = SYSTEM
   SYSTEM-TYPE = VAVS
   MIN-SUPPLY-T = 55  $ RESET FROM 55 F FOR HUMIDITY CONTROL $
   SUPPLY-STATIC = 2.0
   SUPPLY-EFF = 0.65
   OA-CONTROL = ENTHALPY
   FAN-CONTROL = SPEED
   MOTOR-PLACEMENT = OUTSIDE-AIRFLOW
   RETURN-STATIC = 1.0
   RETURN-EFF = 0.65
   ZONE-NAMES = (ZONE1,MECH,PLEN)  ..

AC-1A = SYSTEM LIKE AC-1
   OA-CONTROL = ENTHALPY
```

```
   SUPPLY-STATIC = 2.5
   RETURN-STATIC = 0.75
   ZONE-NAMES = (ZONE1A)  ..

AC-1B = SYSTEM LIKE AC-1
   OA-CONTROL = ENTHALPY
   SUPPLY-STATIC = 2.0
   RETURN-STATIC = 1.0
   ZONE-NAMES = (ZONE1B)  ..

AC-2 = SYSTEM LIKE AC-1
   OA-CONTROL = ENTHALPY
   ZONE-NAMES = (ZONE2)
   ..

AC-3 = SYSTEM LIKE AC-1
   ZONE-NAMES = (ZONE3)  ..

AC-4 = SYSTEM LIKE AC-1
   ZONE-NAMES = (ZONE4)  ..

AC-5 = SYSTEM LIKE AC-1
   ZONE-NAMES = (ZONE5)  ..

AC-6 = SYSTEM LIKE AC-1
   ZONE-NAMES = (ZONE6)  ..

AC-7 = SYSTEM LIKE AC-1
   ZONE-NAMES = (ZONE7)  ..

AC-8 = SYSTEM LIKE AC-1
   ZONE-NAMES = (ZONE8,ZONE8C)  ..

AC-9 = SYSTEM LIKE AC-1
   ZONE-NAMES = (ZONE9)  ..

AC-10 = SYSTEM
   SYSTEM-TYPE = PSZ
   MAX-SUPPLY-T = 105
   MIN-SUPPLY-T = 55
   SUPPLY-CFM = 8400
   RETURN-CFM = 6720
   MIN-OUTSIDE-AIR = 0.2
   ZONE-NAMES = (ZONE10)
   HEAT-SOURCE = ELECTRIC
   COOLING-CAPACITY = 176900
   HEATING-CAPACITY = -85303  ..

AC-11 = SYSTEM LIKE AC-10
   SUPPLY-CFM = 3350
   RETURN-CFM = 3350
   COOLING-CAPACITY = 71400
```

```
   HEATING-CAPACITY = -34121
   ZONE-NAMES = (ZONE11)
   MIN-OUTSIDE-AIR = 0.0  ..

AC-12 = SYSTEM LIKE AC-10
   SUPPLY-CFM = 700
   RETURN-CFM = 700
   COOLING-CAPACITY = 16000
   HEATING-CAPACITY = -15100
   ZONE-NAMES = (ZONE12)
   MIN-OUTSIDE-AIR = 0.0  ..

AC-STOR = SYSTEM
   SYSTEM-TYPE = UHT
   MAX-SUPPLY-T = 105
   RATED-CFM = 2200
   HEATING-CAPACITY = -118500
   SUPPLY-STATIC = 0.8
   SUPPLY-KW = 0.00051
   FURNACE-HIR = 1.266
   HEAT-SOURCE = GAS-FURNACE
   ZONE-NAMES = (STOR)  ..

AC-PLAT = SYSTEM LIKE AC-STOR
   ZONE-NAMES = (PLAT1,PLAT2)  ..

SYSTEMS-REPORT
   HOURLY-DATA-SAVE = YES
   VERIFICATION = (SV-A,SV-B)
   SUMMARY = (SS-A,SS-B,SS-D,SS-F,SS-I,SS-N)  ..

##include tomsys

END  ..
COMPUTE SYSTEMS  ..

$ ------------------------------------------------------------------ $
$ ---------------------- PLANT INPUT ------------------------------ $

INPUT PLANT  ..

BOILER = PLANT-EQUIPMENT
   TYPE = HW-BOILER
   INSTALLED-NUMBER = 1
   MAX-NUMBER-AVAIL = 1
   SIZE = -999  ..

CHILL = PLANT-EQUIPMENT
   TYPE = HERM-CENT-CHLR
   SIZE = -999
   INSTALLED-NUMBER = 1
   MAX-NUMBER-AVAIL = 1  ..
```

```
CTOWER = PLANT-EQUIPMENT
   TYPE = COOLING-TWR
   SIZE = -999    ..

PART-LOAD-RATIO
   TYPE = HERM-CENT-CHLR
   MIN-RATIO = 0.2
   ELEC-INPUT-RATIO = 0.20  ..

PLANT-PARAMETERS
   CHILL-WTR-THROTTLE = 1.7
   HERM-CENT-COND-TYPE = TOWER
   HERM-CENT-UNL-RATIO = 0.2
   TWR-PUMP-HEAD = 100.
   TWR-DESIGN-WETBULB = 74.0
   TWR-FAN-CONTROL=TWO-SPEED
   TWR-TEMP-CONTROL=FIXED
   TWR-WTR-SET-POINT=75
   BOILER-FUEL = LPG
   HW-BOILER-HIR = 1.429
   CCIRC-HEAD = 100.
   HCIRC-HEAD = 100.
   CCIRC-DESIGN-T-DROP = 10
   HCIRC-DESIGN-T-DROP = 2  ..

PLANT-REPORT
   HOURLY-DATA-SAVE = YES
   VERIFICATION = (PV-A)
   SUMMARY = (PS-A,PS-B,PS-C,PS-G,PS-H,BEPS)  ..

##include tomplt

END  ..
COMPUTE PLANT  ..

STOP  ..
```

Auxiliary File: OCCDAT

```
P-AC-1 = SCHEDULE THRU DEC 31
  (WD) (1,24) (0.14,0.14,0.14,0.14,0.04,0.14,0.14,0.14,1.00,1.00,1.00,0.89,
               0.21,0.89,1.00,1.00,0.14,0.14,0.14,0.14,0.04,0.14,0.14,0.14)
  (WEH) (1,24) (0.07,0.07,0.07,0.07,0.02,0.07,0.07,0.07,0.50,0.50,0.50,0.45,
               0.11,0.45,0.50,0.50,0.07,0.07,0.07,0.07,0.02,0.07,0.07,0.07)
..
P-AC-1B = SCHEDULE THRU DEC 31
  (WD) (1,24) (0.08,0.08,0.08,0.08,0.25,0.08,0.08,0.08,0.50,0.50,0.50,0.50,
               1.00,0.50,0.50,0.50,0.10,0.10,0.10,0.10,0.03,0.10,0.10,0.10)
  (WEH) (1,24) (0.04,0.04,0.04,0.04,0.13,0.04,0.04,0.04,0.25,0.25,0.25,0.25,
               0.50,0.25,0.25,0.25,0.05,0.05,0.05,0.05,0.01,0.05,0.05,0.05)
..
P-AC-2  = SCHEDULE THRU DEC 31
  (WD) (1,24) (0.67,0.13,0.13,0.13,1.00,0.13,0.13,0.67,0.67,0.13,0.13,0.13,
               1.00,0.13,0.13,0.67,0.67,0.13,0.13,0.13,1.00,0.13,0.13,0.67)
  (WEH) (1,24) (0.33,0.07,0.07,0.07,0.50,0.07,0.07,0.33,0.33,0.07,0.07,0.07,
               0.50,0.07,0.07,0.33,0.33,0.07,0.07,0.07,0.50,0.07,0.07,0.33)
..
P-AC-3  = SCHEDULE THRU DEC 31
  (WD) (1,24) (1.00,1.00,1.00,1.00,0.59,1.00,0.68,0.68,0.23,0.23,0.23,0.23,
               0.05,0.23,0.23,0.23,0.52,0.52,0.52,0.52,0.25,0.52,0.52,1.00)
  (WEH) (1,24) (0.50,0.50,0.50,0.50,0.30,0.50,0.34,0.34,0.11,0.11,0.11,0.11,
               0.02,0.11,0.11,0.11,0.26,0.26,0.26,0.26,0.13,0.26,0.26,0.50)
..
P-AC-4  = SCHEDULE THRU DEC 31
  (WD) (1,24) (0.83,0.83,0.83,0.83,0.60,1.00,0.70,0.70,0.62,0.62,0.62,0.62,
               0.04,0.33,0.33,0.00,0.41,0.41,0.41,0.41,0.20,0.41,0.41,0.83)
  (WEH) (1,24) (0.41,0.41,0.41,0.41,0.30,0.50,0.35,0.35,0.31,0.31,0.31,0.31,
               0.02,0.16,0.16,0.00,0.21,0.21,0.21,0.21,0.10,0.21,0.21,0.41)
..
P-AC-5  = SCHEDULE THRU DEC 31
  (WD) (1,24) (1.00,1.00,1.00,1.00,0.58,1.00,0.36,0.36,0.48,0.48,0.48,0.48,
               0.06,0.48,0.48,0.48,0.79,0.79,0.79,0.79,0.36,0.79,0.79,1)
  (WEH) (1,24) (0.50,0.50,0.50,0.50,0.29,0.50,0.18,0.18,0.24,0.24,0.24,0.24,
               0.03,0.24,0.24,0.24,0.39,0.39,0.39,0.39,0.18,0.39,0.39,0.50)
..
P-AC-6  = SCHEDULE THRU DEC 31
  (WD) (1,24) (0.46,0.46,0.46,0.46,0.29,0.46,0.29,0.29,0.17,0.17,0.17,0.17,
               0.03,0.17,0.17,0.17,1.00,1.00,1.00,1.00,0.46,1.00,1.00,0.37)
  (WEH) (1,24) (0.23,0.23,0.23,0.23,0.14,0.23,0.14,0.14,0.09,0.09,0.09,0.09,
               0.01,0.09,0.09,0.09,0.50,0.50,0.50,0.50,0.23,0.50,0.50,0.19)
..
P-AC-7  = SCHEDULE THRU DEC 31
  (WD) (1,24) (0.89,0.89,0.89,0.89,0.50,0.89,0.44,0.44,1.00,1.00,1.00,1.00,
               0.17,1.00,1.00,0.00,0.61,0.61,0.61,0.61,0.28,0.89,0.89,0.89)
  (WEH) (1,24) (0.44,0.44,0.44,0.44,0.25,0.44,0.22,0.22,0.50,0.50,0.50,0.50,
               0.08,0.50,0.50,0.00,0.31,0.31,0.31,0.31,0.14,0.44,0.44,0.44)
```

```
..
P-AC-8  = SCHEDULE THRU DEC 31
  (WD) (1,24) (1.00,1.00,1.00,1.00,0.60,1.00,0.86,0.86,0.16,0.16,0.16,0.16,
               0.03,0.16,0.00,0.00,0.92,0.92,0.92,0.92,0.42,0.92,0.92,1)
  (WEH) (1,24) (0.50,0.50,0.50,0.50,0.30,0.50,0.43,0.43,0.08,0.08,0.08,0.08,
               0.01,0.08,0.00,0.00,0.46,0.46,0.46,0.46,0.21,0.46,0.46,0.50)
..
P-AC-9  = SCHEDULE THRU DEC 31
  (WD) (1,24) (0.96,0.96,0.96,0.96,0.55,0.96,0.43,0.43,0.58,0.58,0.58,0.58,
               0.08,0.58,0.58,0.00,0.43,0.43,0.43,0.43,0.20,1.00,1.00,1)
  (WEH) (1,24) (0.48,0.48,0.48,0.48,0.28,0.48,0.22,0.22,0.29,0.29,0.29,0.29,
               0.04,0.29,0.29,0.00,0.22,0.22,0.22,0.22,0.10,0.50,0.50,0.5)
..
P-AC-10 = SCHEDULE THRU DEC 31
  (WD) (1,24) (1.00,1.00,1.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
               0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,1.00,1.00,1)
  (WEH) (1,24) (0.50,0.50,0.50,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,
               0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.00,0.50,0.50,0.5)
..
P-PLAT = SCHEDULE THRU DEC 31
  (WD) (1,24) (1.00,1.00,1.00,1.00,0.17,1.00,1.00,1.00,1.00,1.00,1.00,1.00,
               0.17,1.00,1.00,1.00,1.00,1.00,1.00,1.00,0.17,1.00,1.00,1)
  (WEH) (1,24) (0.50,0.50,0.50,0.50,0.08,0.50,0.50,0.50,0.50,0.50,0.50,0.50,
               0.08,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.08,0.50,0.50,0.5)
..
```

```
EQ-AC-8C     = SCHEDULE THRU DEC 31
 (WD)  (1,24)(1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0,
              1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0)
 (WEH)  (1,24)(.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,
              .50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50)
..

EQ-AC-PLAT1 = SCHEDULE THRU DEC 31
 (WD)  (1,24)(1.0,1.0,1.0,0.0,0.0,0.50,0.50,0.75,0.75,0.75,0.75,0.75,
              .75,0.75,0.75,0.75,1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0)
 (WEH)  (1,24)(.50,0.50,0.50,0.0,0.0,0.25,0.25,.375,.375,.375,.375,.375,
              .375,.375,.375,.375,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50)
..
EQ-AC-PLAT2 = SCHEDULE THRU DEC 31
 (WD)  (1,24)(1.0,1.0,1.0,1.0,1.0,0.0,0.50,0.50,0.50,0.50,0.50,0.50,
              .50,0.50,0.50,0.50,0.50,1.0,1.0,1.0,1.0,1.0,1.0,1.0)
 (WEH)  (1,24)(.50,0.50,0.50,0.50,0.50,0.0,0.25,0.25,0.25,0.25,0.25,0.25,
              .25,0.25,0.25,0.25,0.25,0.50,0.50,0.50,0.50,0.50,0.50,0.50)
..

EQ-AC-1      = SCHEDULE THRU DEC 31
 (WD)  (1,24)(.50,0.50,0.50,0.50,0.50,0.50,1.0,1.0,1.0,1.0,1.0,1.0,
              1.0,1.0,1.0,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50)
 (WEH)  (1,24)(.25,0.25,0.25,0.25,0.25,0.25,0.50,0.50,0.50,0.50,0.50,0.50,
              .50,0.50,0.50,0.25,0.25,0.25,0.25,0.25,0.25,0.25,0.25,0.25)
..

EQ-AC-3      = SCHEDULE THRU DEC 31
 (WD)  (1,24)(1.0,1.0,1.0,1.0,1.0,1.0,1.0,0.125,0.125,0.125,0.125,0.50,
              .375,.375,0.50,0.50,0.50,1.0,1.0,1.0,1.0,1.0,1.0,1.0)
 (WEH)  (1,24)(.50,0.50,0.50,0.50,0.50,0.50,0.50,.063,.063,.063,.063,0.25,
              .188,.188,0.25,0.25,0.25,0.50,0.50,0.50,0.50,0.50,0.50,0.50)
..

EQ-AC-5      = SCHEDULE THRU DEC 31
 (WD)  (1,24)(1.0,1.0,1.0,1.0,1.0,1.0,0.50,0.50,0.50,0.50,0.50,0.50,
              .50,0.50,1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0)
 (WEH)  (1,24)(.50,0.50,0.50,0.50,0.50,0.50,0.25,0.25,0.25,0.25,0.25,0.25,
              .25,0.25,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50)
..

EQ-AC-6      = SCHEDULE THRU DEC 31
 (WD)  (1,24)(.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50,
              .50,0.50,1.0,1.0,1.0,1.0,0.75,0.75,0.75,0.75,0.50,0.50)
 (WEH)  (1,24)(.25,0.25,0.25,0.25,0.25,0.25,0.25,0.25,0.25,0.25,0.25,0.25,
              .25,0.25,0.50,0.50,0.50,0.50,.375,.375,.375,.375,0.25,0.25)
..
```

```
EQ-AC-7      = SCHEDULE THRU DEC 31
 (WD)  (1,24)(1.0,1.0,1.0,1.0,1.0,1.0,0.75,0.75,1.0,1.0,1.0,1.0,
              1.0,0.75,0.75,0.75,0.75,1.0,1.0,1.0,1.0,1.0,1.0,1.0)
 (WEH)  (1,24)(.50,0.50,0.50,0.50,0.50,0.50,.375,.375,0.50,0.50,0.50,0.50,
              .50,.375,.375,.375,.375,0.50,0.50,0.50,0.50,0.50,0.50,0.50)
..

EQ-AC-8      = SCHEDULE THRU DEC 31
 (WD)  (1,24)(1.0,1.0,1.0,1.0,1.0,1.0,1.0,0.125,0.125,0.50,0.50,0.50,
              .50,0.50,0.50,.375 1.0,1.0,1.0,1.0,1.0,1.0,1.0,1.0)
 (WEH)  (1,24)(.50,0.50,0.50,0.50,0.50,0.50,0.50,0.063,0.063,0.25,0.25,0.25,
              .25,0.25,0.25,.188,0.50,0.50,0.50,0.50,0.50,0.50,0.50,0.50)
..

EQ-AC-9      = SCHEDULE THRU DEC 31
 (WD)  (1,24)(.75,0.75,0.75,0.75,0.75,0.75,0.75,0.5,0.5,0.75,0.75,0.75,
              .75,0.75,0.75,0.75,0.75,0.75,0.75,0.75,0.75,0.75,0.75,0.75)
 (WEH)  (1,24)(.375,.375,.375,.375,.375,.375,.375,.25,.25,0.375,0.375,0.375,
              .375,.375,.375,.375,.375,.375,.375,.375,.375,.375,.375,.375)
..

EQ-AC-10     = SCHEDULE THRU DEC 31
 (WD)  (1,24)(1.0,1.0,1.0,0.0,0.0,0.0,0,0,0,0,0,0,
              0,0,0,0.0,0.0,0.0,0.0,0.0,0.0,1.0,1.0,1.0)
 (WEH)  (1,24)(0.50,0.50,0.50,0.0,0.0,0.0,0.0,0,0,0,0,0,
              0,0,0,0.0,0.0,0.0,0.0,0.0,0.0,0.50,0.50,0.50)
..
```

Auxiliary File: TOMSYS

```
$ *** add  HOURLY-DATA-SAVE=YES command to the SYSTEMS-REPORT command block
***
$ *** below that command block, add the following:
***

$ SYSTEMS HOURLY REPORTS

FULL-YEAR = SCHEDULE THRU DEC 31 (ALL) (1,24) (1)        ..

REPORT-SG1  = REPORT-BLOCK
              VARIABLE-TYPE        = GLOBAL
              VARIABLE-LIST        = (7,8)            ..
$                                      WBT,DBT
REPORT-S3   = REPORT-BLOCK
              VARIABLE-TYPE        = AC-3
              VARIABLE-LIST        = (6,48,5,10,11,17,20,32) ..
$                                      TCLG,LCL,CHT,PHT,QHUM,CFM,RCFM,SKW
REPORT-S4   = REPORT-BLOCK
              VARIABLE-TYPE        = AC-4
              VARIABLE-LIST        = (6,48,5,10,11,17,20,32) ..
$                                      TCLG,LCL,CHT,PHT,QHUM,CFM,RCFM,SKW
REPORT-S5   = REPORT-BLOCK
              VARIABLE-TYPE        = AC-5
              VARIABLE-LIST        = (6,48,5,10,11,17,20,32) ..
$                                      TCLG,LCL,CHT,PHT,QHUM,CFM,RCFM,SKW
REPORT-S6   = REPORT-BLOCK
              VARIABLE-TYPE        = AC-6
              VARIABLE-LIST        = (6,48,5,10,11,17,20,32) ..
$                                      TCLG,LCL,CHT,PHT,QHUM,CFM,RCFM,SKW
REPORT-S7   = REPORT-BLOCK
              VARIABLE-TYPE        = AC-7
              VARIABLE-LIST        = (6,48,5,10,11,17,20,32) ..
$                                      TCLG,LCL,CHT,PHT,QHUM,CFM,RCFM,SKW
REPORT-S8   = REPORT-BLOCK
              VARIABLE-TYPE        = AC-8
              VARIABLE-LIST        = (6,48,5,10,11,17,20,32) ..
$                                      TCLG,LCL,CHT,PHT,QHUM,CFM,RCFM,SKW
REPORT-S9   = REPORT-BLOCK
              VARIABLE-TYPE        = AC-9
              VARIABLE-LIST        = (6,48,5,10,11,17,20,32) ..
$                                      TCLG,LCL,CHT,PHT,QHUM,CFM,RCFM,SKW


SYSTEMS-HOURLY = HOURLY-REPORT
              REPORT-SCHEDULE      = FULL-YEAR
              REPORT-BLOCK         = (REPORT-SG1,
                                      REPORT-S3,REPORT-S4,
                                      REPORT-S5,REPORT-S6,REPORT-S7,
                                      REPORT-S8,REPORT-S9)        ..
```

Auxiliary File: TQMPLT

```
$ PLANT HOURLY REPORTS

FULL-YEAR = SCHEDULE THRU DEC 31 (ALL) (1,24) (1)        ..

REPORT-SG1  = REPORT-BLOCK
              VARIABLE-TYPE        = GLOBAL
              VARIABLE-LIST        = (2,1)              ..
$                                      WBT,DBT
REPORT-P1   = REPORT-BLOCK
              VARIABLE-TYPE        = HW-BOILER
              VARIABLE-LIST        = (1,3,4)            ..
$                                      EQDEM(1,2),EQDEM(3,2),EQDEM(4,2)
REPORT-P2   = REPORT-BLOCK
              VARIABLE-TYPE        = HERM-CENT-CHLR
              VARIABLE-LIST        = (1,3)              ..
$                                      EQDEM(1,10),EQDEM(2,10)

REPORT-P3   = REPORT-BLOCK
              VARIABLE-TYPE        = PLANT
              VARIABLE-LIST        = (3,10)             ..
$                                     ENGYLD(3,IHR),PDEM(3)

PLANT-HOURLY = HOURLY-REPORT
              REPORT-SCHEDULE      = FULL-YEAR
              REPORT-BLOCK         = (REPORT-SG1,REPORT-P1,REPORT-P2,
                                      REPORT-P3)  ..
```

APPENDIX B

PRELIMINARY COST/SAVINGS ANALYSIS OF HVAC SYSTEM IMPROVEMENTS

APPENDIX B

PRELIMINARY COST/SAVINGS ANALYSIS OF HVAC SYSTEM IMPROVEMENTS

The material presented in this appendix was developed to help the Mid-Island MPF plant engineer select energy efficiency improvements for the ATES/TEG system. The results are based on many assumptions which, while believed accurate, are not sufficient for design purposes. The cost effectiveness of the measures should be rechecked after design work is completed and decisions to proceed made at that time.

Material and labor costs are from Means Mechanical Cost Data 1992 (published 1991, 10-digit code) or WW Grainger (1991; 5-digit stock number) and do not include overhead or profit unless otherwise noted. A burden factor of 1.4, to cover contingency, overhead and profit, is applied to all raw material and labor costs. Energy savings are based on the 1991 rates used in the body of the report.

DEDICATED MAKE-UP AIR FOR SHRINK-WRAP VENTILATION

Exhaust fans were installed when the three shrink-wrap machines were added at the MPF. The fans remove heat and organic vapors generated in the shrink-wrap process. The ambient air conditions required for operation of these machines are much less stringent than the 70 to 76°F, 40 to 60% RH conditions that are generally maintained in the workroom. Direct ventilation with outside air could therefore be used, except in very cold weather, to provide make-up air in the immediate vicinity of the shrink-wrap machines. This will eliminate the sensible and latent cooling loads currently imposed by the 17,400 cfm flow of outside air introduced to the conditioned space when the shrink-wrap machine exhaust fans are operating. These fans operate approximately 8 hours/day.

Description. Each of three proposed make-up air systems will consist of a roof mounted air intake hood, a make-up air fan (flow capacity equal to that of the existing exhaust fan) and motorized damper operated in parallel with

the existing exhaust fan, an over-ride thermostat to disable the fan and shut the damper when the outside temperature is below 45°F, and a simple duct system to distribute outside air to four registers that will direct the air downward at four locations, about 12 ft above the floor, around the periphery of the shrink-wrap machine.

Cost. The three systems are assumed to be the same size. The cost of each system is based on the following elements:

Description and Quantity	Cost($)
36-in. 3/4-hp three-phase ventilator with motor, drive, roof hood (7F548)[a]	840
36x36 in. 12-in. high curb (3C438)	185
Installation labor (157-440-6280 and 157-490-5700)	190
36x36 in. opposed blade damper and motor, installed (157-480-5150 and 6140)	360
Line-voltage thermostat and three-pole motor contactor (2E728, 5X464)	70
Installation (157-420-5000 and 163-310-0100)	80
100-ft 24-in. spiral 20 ga. steel duct, installed (157-250-5520)	1750
Four 45- and four 90-degree elbows, installed (157250-8560 & 8800)	1090
Total x 1.4 x 3 =	$19,170

Savings. The energy savings are based on the following assumptions. Shrink-wrap ventilation represents 20% (annual average) of all infiltration and ventilation for spaces served by the ATES/TEG system. The load reductions are estimated to be 10% of latent load (90 MBtu/yr), 5% of sensible cooling load (300 MBtu/yr) and 7% of the heating load (25 MBtu/yr). Each MBtu of latent load requires 55 therms of boiler fuel, based on a COP of 0.26 and a boiler efficiency of 0.7) and 110 kWh of desiccant pump energy and each MBtu of latent load also contributes 0.4 MBtu to the sensible cooling load or 25 kWh of cooling plant electrical consumption (based on a system COP, which accounts for energy used by fans, pumps, and compressors associated with the cooling tower, chillers, wells and chilled water distribution, of 4.8).

---

[a] A filtered intake ventilator (7C378) will not add appreciably to the initial cost but will have twice the operating cost.

The resulting savings are tabulated below. Note that relatively small components of the savings (e.g. pumping energy associated with heating load and desiccant pumping energy associated with latent cooling load) have been ignored, making the overall savings estimate a conservative one.

Savings basis:	load		site energy		
Load component:	MBtu/yr	kWh/yr	$	therm/yr	$
latent cooling	90	12,150	950	4,950	3,370
sensible cooling	300	7,500	590	0	0
heating	25	0	0	360	245
new fans	0	-5,500	-420	0	0
Total					$4,700

The simple payback, based on the foregoing estimates, is 4.1 years.

Implementation notes.

- While the payback for this measure not especially attractive, it has the important side effect of reducing peak latent loads. This will ameliorate the problem of insufficient TEG concentrator capacity.
- The ventilation fan motors should be specified as premium efficiency models.
- The duct sizes in the cost estimate were selected for low velocity and low pressure loss; to ensure an energy efficient ventilation system, the contract for this work should specify the make-up air flow rates (nominal or actual, whichever is larger) of the existing exhausts and the maximum acceptable aggregate fan motor power (2.5 with no filters and 4.5 with filters in dirty condition).

CONDITIONER FANS: REDUCED FLOW AND CONVERSION TO VARIABLE AIR VOLUME

The DOE-2 simulation indicated that conditioner fans account for 940,260 kWh/yr or over 60% of the total electrical energy used by the ATES/TEG system. The simulation also indicated that conditioners could satisfy the peak cooling loads with their air flow rates reduced on the order of 10%.

Description. The energy savings from a 10% conditioner air flow reduction alone would reduce the fan energy by 25%[a] at practically no cost. However additional savings can be obtained by installing variable-speed drives (VSDs) and controlling fan speed to just meet the load in each zone. Both load criteria, temperature and relative humidity, must be satisfied for comfort. Proper ventilation must also be ensured.

Cost. Each conditioner requires one VSD of sufficient ampacity to handle both supply and return fans at 90% of current maximum flow rate. The VSD costs are based on budgetary figures provided by Utech Systems Associates, Milwaukee, Wisconsin. The costs to retrofit the seven conditioners with VSDs, and provide proportional-plus-integral control of temperature, humidity and CO^2, are estimated to be as follows:

Description and Quantity	Cost($)
30-hp VSD, disconnect and transfer switch/bypass (Qty 2)	8,740
20-hp VSD, disconnect and transfer switch/bypass (Qty 3)	9,660
15-hp VSD, disconnect and transfer switch/bypass (Qty 2)	4,900
Installation (163-130-0640, Qty 7)	1,610
Humidity and temperature (General Eastern, Qty 7)	1,750
CO^2 sensor (Gaztech, Qty 7)	2,800
Installation (3x7=21 units, 157-400-5000)	560
Total x 1.4 =	$42,028
LILCO rebate ($140/hp x 150 hp)	21,000
Total after rebate	$21,028

Savings. The energy savings are based on a 25% savings for VSD operation of all conditioner supply and return fans, on top of a 25% savings from reducing maximum air flow by 10%. This results in a 100*(1 - (1-0.25)*(1-0.25)) = 44%

(a) Fan power varies with air flow, Q, according to the distribution of pressure drop between turbulent and laminar flow. The power varies with Q^3 in pure turbulent flow and with Q^2 in pure laminar flow. An exponent of 3 (i.e. power varies with Q^3) is often assumed, but to be conservative we have used an exponent of 2.7 in the analysis.

savings overall or 411,000 kWh/yr fan input energy savings. 411,000 kWh/yr represents 9% of the sensible cooling load but credit has not been taken for the cooling input savings because the interactive savings are not fully realized when multiple conservation and energy efficiency measures are implemented and because the cooling savings are partly balanced by additional heating costs. The value of the fan input energy savings is $31,500/yr.

The simple payback based on the foregoing estimates, is 1.3 years without the LILCO rebate and 8 months with the rebate.

Implementation notes.

- Outside air control should maintain an average CO^2 concentration of less than 800 ppm. Because the technology for CO^2 control of outside air is relatively new and because occupant's are sensitive to air quality issues, it is important that plant operators check proper functioning of the system at least monthly for the first year. A precision CO^2 analyzer accurate to 50 ppm can be purchased for about $7000. This cost, and the labor cost for monthly surveys, do not make the proposed measure much less attractive because the annual electric energy savings are so large. Also, LILCO offers a $0.20/avoided kW custom HVAC DSM rebate, which is probably applicable to the CO^2 sensors.

ENERGY-EFFICIENT MOTORS FOR CONDITIONER SUPPLY AND RETURN FANS

Variable-speed drives are generally more cost-effective, and generally produce greater savings, than replacement of standard efficiency motors by premium efficiency motors. For this reason, it is standard practice, when evaluating both measures, to evaluate the VSD measure first. This tends to make the economics of the efficient motor retrofit even less attractive, with respect to the VSD retrofit, but this does not mean that the efficient motor retrofit will always turn out to be economically unattractive.

Description. Each conditioner requires one supply and one return fan motor.

Cost. The costs to retrofit the motors, assuming a 25% average size reduction, are as follows:

Qty	Current Size (hp)	Retrofit Size (hp)	Material ($/unit)	Labor ($/unit)	Rebate ($/unit)	Total ($)
2	25	20	647	221	-320	1,096
3	20	15	527	221	-240	1,524
2	15	10	388	132	-160	720
1	10	7.5	327	132	-120	339
4	7.5	5	231	132	-80	1,132
2	5	5	231	132	-80	566
Total	190	142.5	5,364	2,293	-2,280	5,377

Savings. The energy savings are based on an aggregate energy consumption (after converting to variable air volume, balancing for minimum fan speed, and making air flow improvements in selected conditioners) of 56% of the estimated current use of 940,260 kWh/yr, or about 529,000 kWh/yr. The estimated savings, by AHU, are as follows.

Qty	Current Size (hp)	Retrofit Size (hp)	% of Fan Load	Annual Load (kWh)	---------Savings--------- (%)	(kWh/yr)	($/yr)
2	25	20	28	148,000	5.6	8,288	638
3	20	15	32	169,000	6.9	11,661	898
2	15	10	16	85,000	7.3	6,205	478
1	10	7.5	5	26,000	7.3	1,898	146
4	7.5	5	14	74,000	8.0	5,920	456
2	5	5	5	27,000	8.0	2,160	166
Total	190	142.5	100	529,000	6.8	36,132	2,782

The simple payback of 2 years makes the efficient motor retrofit a very good investment. Even the payback for the smallest return fan motors, 3.5 years, is attractive.

Implementation notes.

- The motor sizes used in the cost/benefit analysis are based on a generalization of the simulation results. The actual sizes required to meet the coincident peak zone loads efficiently can only be determined by executing an iterative balancing procedure. Procedures may vary somewhat among competent balancing contractors but should include most or all of the following elements:

- The subsystems of each conditioner should be checked for proper operation. Primary and secondary chilled water flow rates, air-side coil and filter pressure drops, and air distribution balance should all be checked and deficiencies corrected.

- The chilled water setpoint should be raised to 58°F and the existing energy management and control system set to print maximum daily space temperature and RH for each zone. Occupant complaints should also be logged. After several hot days it should be possible to determine which zones have more than sufficient air flow and which have insufficient air flow and to adjust the sheaves accordingly. The chilled water setpoint should be dropped 2°F and the process continued until some hotter days are encountered. It may be found that a chilled water setpoint greater than 54°F can satisfy the peak load, or it may be determined that a lower setpoint is necessary[a].

- If the system is extremely unbalanced (e.g., one or more fans are at or near their maximum continuous duty load while an equal number or more are at 50% or less of maximum load), it may be possible to justify replacing sections of the air distribution system with larger ducts and registers and low pressure loss fittings. (LILCO's $0.20/avoided kW rebate should apply to this work.) If just one air handler is a problem, addition of a parallel air handler or rezoning of existing air handlers may be appropriate.

- Once system deficiencies have been corrected and the system properly balanced, the proper motor sizes can be readily determined. The life-cycle economics of replacing existing motors with properly sized energy-efficient motors should be determined (by measuring the electric power required by each fan to satisfy peak load) on a case-by-case basis and the VAV and efficient motor retrofits executed in any order or, if funding priorities permit, simultaneously. The life-cycle cost of replacing supply and return fan impellers (or entire blower assemblies) with units that have energy efficient air-foil blades should also be considered on a case-by-case basis.

(a) The procedure may indicate insufficient air flow in a fan that has already been sheaved to its maximum rated continuous duty load; one shouldn't immediately conclude that it is necessary to lower the chilled water setpoint. Instead, a thorough check of primary chilled water flow rate, secondary flow rate, tube fouling, air flow rate, fan static pressure, external static pressure, coil and filter pressure drops, and air distribution balance should be made and deficiencies corrected or appropriate corrective measures taken. Lighting, ventilation air, and equipment operating in the zone should also be checked to determine if failure to satisfy load is caused by some fault outside the chilled water, terminal unit, or air distribution systems. In other words, the chilled water temperature should be lowered only as a last resort.

VARIABLE-SPEED DRIVE FOR CHILLED WATER, COOLING TOWER, AND ATES WELL PUMPS

The DOE-2 simulation indicated that chilled water pumps, cooling tower fans and pumps, and ATES well pumps account for 370,500 kWh/yr or about 25% of the total electrical energy used by the ATES/TEG system. Savings can be obtained by installing VSDs and controlling pump speeds based on temperature rise through the load.

Description. Each pump function requires one VSD of sufficient ampacity to handle all pumps that serve the function and are to be controlled in parallel. Note that pumps associated with a given pump function are, in most cases, currently controlled in a sequence such that one pump operates under light load conditions and all pumps operate under full load conditions.

Cost. The VSD costs are based on budgetary figures provided by Utech Systems Associates, Milwaukee, Wisconsin. The costs to retrofit the seven conditioners with VSDs and provide PI controls to act on load temperature difference are estimated to be as follows:

--Pump--			Energy	--VSD---		Annual Savings			Cost
Qty	(hp)	Function	(kWh/y)	Qty	(hp)	(%)	(kWh)	($)	($)
2	5	Condenser	16,600	1	10	50	8,300	636	2320
2	20	Tower Pump	94,900	1	40	50	47,450	3633	5100
2	20	Tower Fan	72,100	1	40	25	18,025	1380	5100
6	20	Well Pump	86,600	1	120	40	34,640	2653	9300
2	20	ATES Water	79,100	1	40	40	31,640	2423	5100
2	10	ChW Pump	21,200	1	20	20	4,240	325	2450
	Installation (at $500 ea)			6					3000
Totals:			370,500	6	270		144,295	11051	32370
Total burdened installed cost:									45318

Savings. The total savings is 144,300 kWh/yr in pump input energy. This represents 3% of the sensible cooling load but credit has not been taken for the cooling input savings because the interactive savings are not fully

realized when multiple conservation and energy efficiency measures are implemented. The value of the pump input energy savings is $11,050/yr.

The simple payback based on the foregoing estimates is 4.1 years without the LILCO rebate and less than 9 months with the rebate.

Implementation notes.

- The cooling tower pumps and fans will operate in parallel from a single variable-frequency driver. A frequency-proportional-to-temperature-difference control law can be used. Additional savings are possible if a more advanced control strategy (to account for the trade-off between tower pump and fan energy and chiller energy in the cooling mode, and the trade-off between ATES injection temperature and volume and tower and well pump energy in the cold injection mode) is developed and implemented.

- The well recovery and injection pumps will operate in parallel from a single variable-frequency driver. A frequency-proportional-to-temperature-difference control law can be used based on hot well minus cooling tower discharge temperature in cold charge mode and chilled water return minus cold well temperature difference in cooling mode. Additional savings are possible if a more advanced control strategy (to account for the trade-off between pump and conditioner fan energy and chiller and cooling tower energy) is developed and implemented.

- The ATES chilled water pumps will operate in parallel with well pumps in discharge mode only.

- Chiller water pumps will operate in parallel from a single variable-frequency driver. A frequency-proportional-to-chiller capacity control law can be used. Additional savings are possible if a more advanced control strategy (to account for the trade-off between pump and conditioner fan energy and chiller and cooling tower energy) is developed and implemented. Note, however, that to prevent evaporation of TEG into the air stream, the chilled water temperature should never be allowed to exceed 62°F unless the desiccant concentration is also modulated (in which case the maximum allowable chilled water temperature is a function of desiccant concentration) or the conditioner spray pumps are shut off under low latent-load conditions.

EFFICIENT BOILERS

The DOE-2 simulation results indicated propane use of 52,000 gal or about 5000 MBtu per year. The first year fuel cost was estimated to be about $34,000. These estimates are based on a boiler *annual fuel utilization efficiency* of 70%.

Description. Fuel consumption can be reduced by replacing the boilers with high-efficiency boilers.

Cost. Means did not list a cost for the 240,000 Btuh pulse boiler but did report the installed cost of a standard 240,000 Btuh boiler to be $4,700. The unburdened installed cost of a 44,000 Btuh pulse boiler is reported by Means (155-115-8000) to be $2,100 and the cost of a 134,000 pulse boiler (155-115-8000) is reported to be $3,235. From the smaller pulse units reported in Means, the burdened cost of a 240,000 Btuh pulse boiler can be extrapolated to be about $6,400. Ten pulse boilers are required to meet the current concentrator load.

Savings. A pulse boiler will provide an AFUE of about 94%, reducing annual fuel use to 5000 x 70/94 = 3,720 MBtu at a cost of $25,300/yr. This results in annual savings of $8,700 and a simple payback of 7.5 years. Smaller savings but quicker payback will result if fewer of the boilers are replaced.

Implementation notes.

- This measure interacts strongly with the concentrator improvement measures. New estimates of the payback should therefore be recalculated after decisions have been made about which concentrator measures to implement.

CONCENTRATOR IMPROVEMENTS

The DOE-2 simulation indicated that the concentrator currently uses 3,450 MBtu/year in hot water.

Description. It is estimated that the average concentrator COP can be improved from 0.3 to 0.7 by installing air-side recuperator coils, providing separate sumps for the solution streams that fall from the hot sprayed coil and the reflux coil, and increasing the size of the solution recuperator.

Cost. The air-side recuperator coils will have to be carefully designed to provide the necessary heat transfer area without excessive pressure drops. For the purpose of a preliminary cost estimate, however, we assume the use of two 6-row coils with 1/2-in. tubes, 8 fins per inch (fpi) aluminum fins, and a face area of 48 ft^2. The unburdened installed cost of two such coils (based on Means 157-201-1420 and -1630) is $10,730. A 1/12-hp, 32-gpm circulator (152-410-2040) and expansion tank (155-671-2000) are also needed. The burdened installed equipment cost, therefore, totals $16,761. Additional costs for design, labor to modify the concentrator enclosure, and commissioning are estimated to be $6,000 resulting in a total first cost to $22,761.

The cost to provide a separate sump for the desiccant solution that falls from the spray coil and rearrange the desiccant return pump pickup and float control cannot be reliably estimated without a more detailed design[a]. However, a ballpark estimate of $15,000, including design, commissioning, and 1.4 burden factor, can be used to estimate feasibility.

Cost of doubling the size of the desiccant solution recuperator[b] is estimated to be $6,000.

Savings. With the increase in COP from 0.3 to 0.7 the concentrator will use 1,480 MBtu/yr and save 1,970 MBtu/yr in hot water or 2,100 MBtu in fuel at an AFUE of 94%. The reduction in operating cost will, therefore, be about $14,400 per year giving a simple payback of about 3 years.

Implementation notes.

- The concentrator should be thoroughly inspected and serviced during the implementation of this measure since it will have to be flushed prior to welding sump baffles anyway. Operation of the system for an extended period prior to the installation of filters at appropriate locations in

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(a) Niagara Blower may be the only qualified party available to develop the split sump design, estimate its cost, and recommission the concentrator after modification at reasonable cost.

(b) Niagara blower has used plate heat exchangers manufactured by APV Crepaco, Tonawanda, NY, and Graham Manufacturing Co. Inc., Batavia, NY. The plate cost estimates are for the model HX-25 from Graham Mfg.

the desiccant solution distribution system may have resulted in significant fouling of the heat transfer surfaces. Clean coils, clean drift eliminators, and clean desiccant recuperator plates will all contribute to efficient operation.

- The air-side recuperator pump should be made to operate at variable speed by connecting it in parallel with the fan if the VSD measure is also implemented.

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