Continuous Commissioning

Release 0.0.1

Jul 19, 2019

Contents

CHAPTER 1

Ch 1: Introduction to Continuous Commissioning ^{SM}

With the energy crisis of the early 1970s came the realization that buildings could be made more efficient without sacrificing comfort. Over the last 30 years, the building industry has made revolutionary changes: chiller systems have decreased their power requirements by a factor of two, from more than 1 kW/ton to less than 0.5 kW/ton; the use of variable air volume systems has become common practice; and the use of building automation systems has become the norm, with digital controls increasingly replacing pneumatics. Advances in HVAC technology have greatly improved building comfort and significantly decreased building energy consumption. The technological advances have increased the importance of proper operational practices in achieving the efficiency potential of the HVAC systems. While reducing energy use is a noble goal, it must not compromise comfort. Occupant comfort and productivity must be maintained or improved.

> **NOTE**
>
> **Building commissioning has emerged as the preferred method of ensuring that building systems are installed and operated to provide the performance envisioned by the designer.**

Building commissioning has emerged as the preferred method of ensuring that building systems are installed and operated to provide the performance envisioned by the designer. ASHRAE has detailed the commissioning process in Guideline 1-1996 [ASHRAE 1996]. A number of other building commissioning guidelines have been developed by different organizations, with the same basic objective as the ASHRAE guideline, i.e., to ensure proper operation of the building according to the design intent.

In 1999, DOE developed a practical guide for commissioning existing buildings [Haasl and Sharp 1999]. Several building commissioning processes are discussed in this guide. These processes include: new building commissioning, existing building commissioning or retro-commissioning, Continuous Commissioning ^{SM} and recommissioning. While most commissioning processes focus on bringing building operation to the original design intent, Continuous Commissioning ^{SM} is different[1]. Continuous Commissioning ^{SM} (CC ^{SM}) focuses on optimizing HVAC system operation and control for the existing building conditions. This is an important distinction. Based on Continuous Commissioning ^{SM} results from more than 130 buildings, the average measured utility savings are about 20%, with simple paybacks often in less than two years. These results are based on the experience of the Texas Engineering Experiment

[1] The terms Continuous CommissioningSM and CCSM have been service marked by the Texas Engineering Experiment Station Energy Systems Laboratory to ensure a consistent meaning for this terminology, as described in this guidebook.

Station's Energy Systems Laboratory at Texas A&M University during the last 10 years [Liu et al. 1994, Claridge et al. 1994, Liu et al. 1999, Claridge et al. 2000]. Continuous Commissioning ^{SM} maintains long-term savings by ongoing monitoring of energy savings with followup commissioning, as needed; improves the system reliability and building comfort by optimizing system operation and control schedules based on actual building conditions; upgrades the operating staff's skills by allowing direct participation of O&M staff; and reduces O&M costs.

This chapter defines CC^{SM} *and provides an introduction to the* CC^{SM} *process. The detailed objectives, methods, and procedures are discussed for each step of the* CC^{SM} *process. This discussion includes the major goals of each step, the methods needed to achieve the specific goals and the procedures needed to conduct* CC^{SM} *wisely and efficiently.*

1.1 1.1 Definition

Continuous Commissioning ^{SM} (CC ^{SM}) is an ongoing process to resolve operating problems, improve comfort, optimize energy use and identify retrofits for existing commercial and institutional buildings and central plant facilities.

CC ^{SM} focuses on improving overall system control and operations for the building, as it is currently utilized, and on meeting existing facility needs. It goes beyond an operations and maintenance program. It does not ensure that the systems function as originally designed, but ensures that the building and systems operate optimally to meet the current requirements. During the CC ^{SM} process, a comprehensive engineering evaluation is conducted for both building functionality and system functions. The optimal operational parameters and schedules are developed based on actual building conditions and current occupancy requirements. An integrated approach is used to implement these optimal schedules to ensure local and global system optimization and persistence of the improved operational schedules.

NOTE

The CC ^{SM} team consists of a project manager, one or more CC ^{SM} engineers and CC ^{SM} technicians, and one or more designated members of the facility operating team.

The CC ^{SM} team consists of a project manager, one or more CC ^{SM} engineers and CC ^{SM} technicians[2], and one or more designated members of the facility operating team. The primary role of the project manager is to coordinate the activities of building personnel and the commissioning team and schedule project progress. The project manager can be an owner representative or a CC ^{SM} contractor representative. The primary responsibilities of the CC ^{SM} engineer are to:

- Develop metering and field measurement plans
- Develop improved operational and control schedules and set points
- Make necessary programming changes to the building automation system
- Supervise technicians implementing mechanical systems changes
- Estimate potential performance changes and energy savings
- Conduct an engineering analysis of the system changes
- Write a project report

The CC ^{SM} engineer should have the qualifications to perform the work specified. The CC ^{SM} technician will conduct field measurements and implement mechanical, electrical, and control system program modifications and changes, under the direction of the CC ^{SM} engineer.

[2] This guidebook will speak of a single CC engineer and a single CC technician for simplicity. However, there may be more than one CC engineer and more than one CC technician on large projects.

1.2 1.2 Process

The CC ^{SM} process consists of two phases. The first phase is the project development phase, that identifies the buildings and facilities to be included in the project and develops the project scope. At the end of this phase, the CC ^{SM} scope is clearly defined and a CC ^{SM} contract is signed. The second phase implements CC ^{SM} and verifies project performance.

This phase includes six steps:

- Develop the CC ^{SM} plan and form the project team
- Develop performance baselines
- Conduct system measurements and develop CC ^{SM} measures
- Implement CC ^{SM} measures
- Document comfort improvements and energy savings
- Keep the commissioning continuous

Phase 1: Project Development

Step 1: Identify Buildings or Facilities

Objective

Screen potential CC ^{SM} targets with minimal effort to identify buildings or facilities that will receive a CC ^{SM} audit. The CC ^{SM} target can be a building, an entire facility, or a piece of equipment. If the building is part of a complex or campus, it is desirable to select the entire facility as the CC ^{SM} target since one mechanical problem may be rooted in another part of the building or facility.

Method

The initial CC ^{SM} targets can be selected based on one or more of the following criteria:

- The target provides poor thermal comfort
- The target consumes excessive energy, and/or
- The design features of the facility HVAC systems are not fully used

If one or more of the above criteria fits the description of the facility, it is likely to be a good candidate for CC ^{SM}. CC ^{SM} can be effectively implemented in buildings that have received energy efficiency retrofits, in newer buildings, and in existing buildings that have not received energy efficiency upgrades. In other words, virtually any building can be a potential CC ^{SM} candidate.

The CC ^{SM} targets can be selected by the building owner or the CC ^{SM} contractor. However, the building owner is usually in the best position to select the most promising targets because of his or her knowledge of the facility operation and costs. The CC ^{SM} contractor should then perform a preliminary assessment to check the feasibility of using the CC ^{SM} process on the target facilities before conducting a CC ^{SM} audit.

Information needed for preliminary assessment:

- Actual monthly utility bills (both electricity and gas) for at least 12 months (preferable to just a table of historical energy and demand data because meter reading dates are needed)
- General building information: size, function, major equipment, and occupancy schedules
- O&M records, if available
- Description of any problem in the building, such as thermal comfort, indoor air quality, moisture, mildew

An experienced engineer should review this information and determine the potential of the CC ^{SM} process to improve comfort and reduce energy cost. The CC ^{SM} projects often improve building comfort and reduce building energy consumption at the same time. However, some of the CC ^{SM} measures may increase building energy consumption in order to satisfy room comfort and indoor air quality requirements. For example, providing building minimum outside air will certainly increase the cooling energy consumption during summer and winter than providing no outside air to the building. If the potential justifies a CC ^{SM} audit, a list of preliminary commissioning measures for evaluation in a CC ^{SM} audit should also be developed. If the owner is interested in proceeding at this point, a CC ^{SM} audit may be performed.

Step 2: Perform CC :sup:'SM' Audit and Develop Project Scope

Objectives

- Define owner's requirements
- Check the availability of in-house technical support such as CC ^{SM} technicians
- Identify major CC ^{SM} measures

Method

The owner's representative, the CC ^{SM} project manager and the CC ^{SM} engineer will meet. The expectations and interest of the building owner in comfort improvements, utility cost reductions and maintenance cost reductions will be discussed and documented in detail. The availability and technical skills of in-house technicians will be discussed. After this discussion, a walkthrough must be conducted to identify the feasibility of the owner's expectations for comfort performance and improved energy performance. During the walkthrough, major CC ^{SM} measures will be identified by the CC ^{SM} engineer and project manager. An in-house technician should participate in the walkthrough. The CC ^{SM} project manager will organize the audit and document the expectations of the building owner.

Special Considerations:

- A complete set of mechanical and control system design documentation is needed
- The CC ^{SM} engineer and technician will take preliminary measurements of equipment operating parameters
- Any available measured whole building level or sub-metered energy consumption data from stand-alone meters or the building automation system should be utilized

A CC ^{SM} audit report must be completed that lists and describes preliminary CC ^{SM} measures, the estimated energy savings from implementation and the cost of carrying out the CC ^{SM} process on the building(s) evaluated in the CC ^{SM} audit. There may be more than one iteration or variation at each step described here, but once a contract is signed, the process moves to Phase 2 as detailed below.

NOTE

System problems should be documented based on interviews with occupants and technical staff, combined with field observations and measurements.

Phase 2: CC :sup:'SM' Implementation and Verification

Step 1: Develop CC :sup:'SM' plan and form the project team

Objective

- Develop a detailed work plan
- Identify the entire project team
- Clarify the duties of each team member

Method

The CC ^{SM} project manager and CC ^{SM} project engineer develop a detailed work plan for the project that includes major tasks, their sequence, time requirements and technical requirements. The work plan is then presented to the building owner or representative. During the meeting, the owner's representative and in-house technicians who will work on the project should be identified. If in-house technicians are going to conduct measurements and system adjustments, additional time should be included in the schedule unless they are to be dedicated full-time to the CC ^{SM} project. Typically, in-house technicians must continue their existing duties and cannot devote full time to the CC ^{SM} effort, which results in project delays. In-house staff may also require additional training. The work plan may need to be modified, depending on the availability and skill levels of in-house staff.

Special Issues:

- Availability of funding to replace/repair parts found broken
- Time commitment of in-house staff
- Training needs of in-house staff

Deliverable:

CC ^{SM} report part 1: CC ^{SM} plan that includes project scope and schedule, project team and task duties of each team member.

Step 2: Develop performance baselines

Objectives:

- Document existing comfort conditions
- Document existing system conditions
- Document existing energy performance

Method:

Precisely document all known comfort problems in individual rooms resulting from too much heating, cooling, noise, humidity, odors (especially from mold or mildew) or lack of outside air. Also, identify the following HVAC system problems:

- Valve and damper hunting
- Disabled systems or components
- Operational problems
- Frequently replaced parts

An interview and walk-through may be required, although most of this information is collected during the CC ^{SM} audit and step 1. Room comfort problems should be quantified using hand-held meters or portable data loggers. System problems should be documented based on interviews with occupants and technical staff and combined with field observations and measurements.

Baseline energy models of building performance are necessary if energy savings are to be documented after commissioning the building. The baseline energy models can be developed using one or more of the following types of data:

- Short-term measured data obtained from data loggers or the EMCS system
- Long-term hourly or 15-minute whole building energy data, such as whole building electricity, cooling and heating consumption, and/or
- Utility bills for electricity, gas and/or chilled or hot water

The whole building energy baseline models normally include whole building electricity, cooling energy and heating energy models. These models are generally expressed as functions of outside air temperature since both heating and cooling energy are normally weather dependent. Any component baseline models should be represented using the most relevant physical parameter(s) as the independent variable(s).

For example, the fan motor power should be correlated with the fan airflow and the pump motor energy consumption should be correlated with water flow. Short-term measured data are often the most cost-effective and accurate if the potential savings from CC ^{SM} measures are independent of the weather. For example, a single true power measurement can be used to develop the baseline fan energy consumption if the pulley were to be changed in a constant air volume system. Shortterm data are useful to determine the baseline for specific pieces of equipment, but are not reliable for baselining overall building energy use.

Long-term measurements are normally required since potential savings of CC ^{SM} measures are weather dependent. These measurements provide the most convincing evidence of the impact of CC ^{SM} projects. Long-term data also help in continuing to detect/diagnose system faults during CC ^{SM} follow-up. Although more costly than short-term measured data, long-term data often produce additional savings, making them the preferred data type. For example, unusual energy consumption patterns can be identified easily using long-term, short-interval measured data. “Fixing” these unusual patterns can result in significant energy savings. Generally speaking, long-term interval data for electricity, gas and thermal usage are preferred.

NOTE

Utility bills may be used to develop the energy-use baseline if the CC ^{SM} process will result in energy savings that are a significant fraction (>15%) of baseline use, and if the building functions and use patterns will remain the same throughout the project.

The CC ^{SM} engineers should provide the metering option(s) that meet the project requirements to the building owner or representative. A metering method should be selected from the options presented by the CC ^{SM} engineer and a detailed metering implementation plan developed. It may be necessary to hire a metering subcontractor if an energy information system is installed prior to implementing the CC ^{SM} measures.

Special Considerations:

- Use the maintenance log to help identify major system problems
- Select a metering plan that suits the CC ^{SM} goals and the facility needs
- Always consider and measure weather data as part of the metering plan
- Keep meters calibrated. When the EMCS system is used for metering, both sensors and transmitters should be calibrated using field measurements.

Deliverables:

CC ^{SM} report part 2: Report on Current Building Performance, including current energy performance, current comfort and system problems, and metering plans if new meters are to be installed. Alternatively, if utility bills are used to develop the baseline models, the report should include baseline energy models.

Step 3: Conduct System Measurements and Develop Proposed CC ^{SM} *Measures*

Objectives:

- Identify current operating schedules, set points, and problems
- Develop solutions to existing problems
- Develop improved operation and control schedules and set points
- Identify potential cost effective energy retrofit measures

Method:

The CC ^{SM} engineer should develop a detailed measurement cut-sheet for each major system. The cut-sheet should list all parameters to be measured and all mechanical and electrical parts to be checked. The CC ^{SM} engineer should also provide the technician with measurement training if a local technician is used to perform system measurements. The CC ^{SM} technician should follow the procedures on the cut-sheets to obtain the specified measurements using appropriate equipment. The CC ^{SM} engineer conducts an engineering analysis to develop solutions for the existing problems; develops improved operation and control schedules and set points for terminal boxes, air handling units (AHUs), exhaust systems, water and steam distribution systems, heat exchangers, chillers, boilers and other components, as appropriate; and identifies potential cost effective energy retrofit measures.

Special Considerations:

- Trend main operational parameters using the EMCS and compare with the measurements from the hand meters
- Print EMCS control sequences
- Verify system operation in the building and compare to EMCS schedules

Deliverables:

CC ^{SM} report part 3: Existing System Conditions. This report includes:

- Existing control sequences and set points for all major equipment, such as AHU supply air temperatures, AHU supply static pressures, terminal box minimum airflow and maximum airflow values, water loop differential pressure set points and equipment on/off schedules
- List of disabled control sequences
- List of malfunctioning equipment and control devices
- Engineering solutions to the existing problems and a list of repairs required
- Improved control and operation sequences

Step 4: Implement CC ^{SM} *Measures*

Objectives:

- Obtain approval for each CC ^{SM} measure from the building owner's representative prior to implementation
- Implement solutions to existing operational and comfort problems
- Implement and refine improved operation and control schedules

Method:

The CC ^{SM} project manager and project engineer should present the engineering solutions to existing problems and the improved operational and control schedules to the building owner's representative in one or more meetings. The in-house operating staff should be invited to the meeting(s). All critical questions should be answered. It is important, at this point, to get "buy-in" and approval from both the building owner's representative and the operating staff. The meeting(s) will decide the following issues:

- Approval, modification or disapproval of each CC ^{SM} measure
- Implementation sequence of CC ^{SM} measures
- Implementation schedules

> **NOTE**
>
> CC ^{SM} implementation should start by solving existing problems.

The existing comfort and difficult control problems are the first priority of the occupants, operating staff and facility owners. Solving these problems improves occupant comfort and increases productivity. The economic benefits from comfort improvements are sometimes higher than the energy cost savings, though less easily quantified. The successful resolution of the existing problems can also earn trust in the CC ^{SM} engineer from the facility operating staff, facility management, and the occupants. This can lead to the team receiving support in a variety of unexpected ways.

Implementation of the improved operation and control schedules should start at the end of the comfort delivery system, such as at the terminal boxes, and end with the central plant. This procedure provides benefits to the building occupants as quickly as possible. It also reduces the overall working load. If the process is reversed, the chiller plant is commissioned first. The chiller sequences are developed based on the current load. After building commissioning, the chiller load may be decreased by 30%. The chiller operating schedules are then likely to need revision. The CC ^{SM} engineer should develop a detailed implementation plan that lists each major activity. The CC ^{SM} technician should follow this plan in implementing the measures.

The CC ^{SM} engineer should closely supervise the implementation and refine the operational and control schedules as necessary. The CC ^{SM} engineer should also be responsible for key software changes as necessary.

Special Considerations:

- Ensure that the owner's technical representative understands each major measure
- Encourage in-house technician involvement in implementation and/or have them implement as many measures as possible
- Document improvements in a timely manner

Deliverables:

CC ^{SM} Report part 4: CC ^{SM} Implementation. This report includes detailed documentation of implemented operation and control sequences, maintenance procedures for these measures, detailed O&M guidelines and additional measures recommended for implementation.

Step 5: Document comfort improvements and energy savings

Objectives:

- Document improved comfort conditions
- Document improved system conditions
- Document improved energy performance

Method:

The comfort measurements taken in step 2 (Phase 2) should be repeated at the same locations under comparable conditions to determine impact on room conditions. The measured parameters, such as temperature and humidity, should be compared with the measurements from step 2.

The measurements and methods adopted in step 4 should be used to determine post- CC ^{SM} energy performance. Energy performance should be compared using appropriate occupancy and weather normalization. Typically, building energy consumption is regressed as a function of outside air temperature if annual projections are desired from short-term data. When hourly or daily models are used, separate models are generally developed for weekends and weekdays.

Special Considerations:

- Savings analyses should follow accepted measurement and verification protocols such as the International Performance Measurement and Verification Protocol [IPMVP 2001] or an agreed upon alternate method
- Comfort conditions should conform to appropriate guidelines/design documents such as ASHRAE standards

Deliverables:

CC ^{SM} Report, Part 5: Measurement and Verification. This report includes results of detailed measurements of room conditions and energy consumption after CC ^{SM} activities, and retrofit recommendations. The room conditions and energy consumption should be compared to those during the pre-CC ^{SM} period. The annual energy savings are projected from the available measured data.

Step 6: Keep the Commissioning Continuous

Objectives:

- Maintain improved comfort and energy performance
- Provide measured annual energy savings

Method:

The CC SM engineers should review the system operation periodically to identify any operating problems and develop improved operation and control schedules as described below.

> **NOTE**
>
> The CC SM engineer should provide follow-up phone consultation to the operating staff as needed, supplemented by site visits.

This will allow the operating staff to make wise decisions and maintain the savings and comfort in years to come. If long term measured data are available, the CC SM engineers should review the energy data quarterly to evaluate the need for a site visit. If the building energy consumption has increased, the CC SM engineers determine possible reasons and verify with facility operating staff. Once the problem(s) is identified, the CC SM engineer should visit the site, develop measures to restore the building performance, and supervise the facility staff in implementing the measures. If the CC SM engineer can remotely log into the EMCS system, the CC SM engineer can check the existing system operation quarterly using the EMCS system. When a large number of operation and control measures are disabled, a site visit is necessary. If the CC SM engineer cannot evaluate the facility using long-term measured energy data and EMCS system information, the CC SM engineer should visit the facility semi-annually.

One year after CC SM implementation is complete, the CC SM engineer should write a project follow-up report that documents the first-year savings, recommendations or changes resulting from any consultation or site visits provided, and any recommendations to further improve building operations.

Special Considerations:

- Operating personnel often have a high turnover rate. It is important to train new staff members in the CC SM process and make sure they are aware of the reasons the CC SM measures were implemented
- Ongoing follow-up is essential if the savings are to be maintained at high levels over time

Deliverables: Special CC SM Report, which documents measured first-year energy savings, results from first year follow-up, recommendations for ongoing staff training, and a schedule of follow-up CC SM activities.

References

ASHRAE, 1996. ASHRAE Guideline 1-1996: The HVAC Commissioning Process, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, Ga.

Claridge, D.E., Haberl, J., Liu, M., Houcek, J. and Athar, A., "Can You Achieve 150% of Predicted Retrofit Savings: Is It Time for Recommissioning?" ACEEE 1994 Summer Study on Energy Efficiency In Buildings Proceedings: Commissioning, Operation and Maintenance, Vol. 5, American Council for an Energy Efficient Economy, Washington, D.C., pp. 73-87, 1994.

Claridge, D.E., Culp, C.H., Liu, M., Deng, S., Turner, W.D. and Haberl, J.S., 2000. "Campus- Wide Continuous Commissioning SM of University Buildings," Proc. of ACEEE 2000 Summer Study on Energy Efficiency in Buildings, Pacific Grove, Cal., Aug. 20-25, pp. 3.101-3.112.

Haasl, Tudi and Sharp, Terry, 1999. A Practical Guide for Commissioning Existing Buildings, Portland Energy Conservation, Inc. and Oak Ridge National Laboratory for U.S. DOE, ORNL/TM-1999/34, 69 pp. + Appendices.

IPMVP, 2001. IPMVP Committee, International Performance Measurement & Verification Protocol: Concepts and Options for Determining Energy and Water Savings, Vol. 1, U.S. Dept. of Energy, DOE/GO-102001-1187, 86 pp., January.

Liu, M., Houcek, J., Athar, A., Reddy, A. and Claridge, D., 1994. "Identifying and Implementing Improved Operation and Maintenance Measures in Texas LoanSTAR Buildings," ACEEE 1994 Summer Study on Energy Efficiency In Buildings Proceedings: Commissioning, Operation and Maintenance, Vol. 5, American Council for an Energy Efficient Economy, Washington, D.C., pp. 153-165, 1994.

Liu, Mingsheng, Claridge, David E. and Turner, W.D., 1999. "Improving Building Energy System Performance by Continuous Commissioning ^{SM}," Energy Engineering, Vol. 96, No. 5, pp. 46-57.

CHAPTER 2

Ch 2: Benefits of Continuous Commissioning SM

The term Continuous Commissioning (CC SM) was first used by engineers at the Texas Engineering Experiment Station's Energy Systems Lab (ESL) at Texas A&M University to describe an ongoing process that improves building operation using measured hourly energy use and environmental data. The first buildings to undergo a Continuous CommissioningSM process were in the Texas LoanSTAR program [Liu, et al, 1994, Claridge, et al, 1994].[1] These buildings were retrofitted with various energy efficiency improvements; measured hourly data were available to verify that the retrofits were performing as desired and analyze the overall building performance. The ESL engineers applied the CC SM process to LoanSTAR buildings where it resolved major comfort complaints and decreased whole building energy consumption by another 15% to 30% [Gregerson, 1997]. The ESL engineers quickly applied the CC SM process to other types of buildings, and the results were comparable in savings and comfort improvements.

In this chapter, the economics and other benefits of the CC SM process are discussed based on the experience gained from implementing the process in more than 130 buildings. Actual cost and savings data are provided and a detailed example is presented.

Note

The ESL engineers applied the CC process to LoanSTAR buildings where it resolved major comfort complaints and decreased whole building energy consumption by another 15% to 30%.

2.1 2.1 Economics of CC SM

The energy savings and costs are well documented in more than 130 buildings commissioned by the ESL. CC SM costs consist of commissioning labor costs, metering costs, and energy savings determination and reporting costs. Note that the costs of replacing broken or damaged parts are generally covered by the maintenance budget and have not been charged to the CC SM effort. The energy savings are normally determined using measured hourly energy consumption data. No credit is given to productivity increases by occupants as a result of improved comfort and lower

[1] LoanSTAR is an acronym for Loans to Save Taxes and Resources, a major public building retrofit program administered by the Texas State Energy Conservation Office.

absenteeism because of lack of documentation of this benefit. Credible anecdotal evidence, however, suggests that occupant productivity can increase substantially when facility IAQ and comfort improve.

The costs are tracked using actual labor costs. Table 2-1 summarizes the commissioning labor for Phase 2 and energy savings for 28 of the commissioned buildings. This table shows the annual savings, the cost of the CC ^{SM} labor (if the combined time cost of CC ^{SM} engineers and technicians is charged at $100/hour) and the simple payback.

It should be noted that these costs represent the initial commissioning effort. They do not include metering and reporting costs or cost of the time devoted to the effort by building facility staff. Nor do they include follow-up labor costs to find and fix problems related to subsequent component failures or other problems. The simple payback varies from 0.3 years to 2.0 years. The cost of metering is discussed in detail later in the chapter.

Table 1: Table 2-1. Savings, Labor Costs and Simple Payback by Building Type for the Commissioning of 28 Buildings

Building Type	Number of Buildings	Savings $/1000ft2/yr	Labor Cost $/1000ft2/yr	Simple Payback Years
Hospitals	6	430	474	1.1
Laboratory/Offices	7	1260	368	0.3
Classroom/Offices	5	430	226	0.5
Offices	8	220	329	1.5
Schools	2	170	336	2.0
Averages/Total	28	540	359	0.7

Table 2-2 provides more-detailed information on the initial commissioning labor costs and energy savings for each of the 28 commissioned buildings. It provides information on building type, location, floor area, measured annual savings, savings per square foot of floor area, commissioning labor and the year initial commissioning was completed. The measured annual savings varied from $10,000/yr to $395,000/yr with an average of $90,000, or $0.64/ft2/yr for the 28 buildings. The measured savings per square foot were strongly dependent on the building type. Average savings were $1.26/ft2/yr for seven medical research laboratory buildings, $0.43/ft2/yr for six hospitals, $0.43/ft2/yr for five university teaching and office buildings, $0.22/ft2/yr for seven office buildings and $0.17/ft2/yr for two school buildings. It should be noted, however, that the variation between buildings within a single category is often as great as the variation between building types.

The CC ^{SM} costs are strongly dependent on the building type and building condition, HVAC system type and size, HVAC system condition and the type and condition of the building automation system. Generally speaking, higher CC ^{SM} costs are directly associated with more existing mechanical and comfort problems, more complex system types, smaller HVAC system sizes, poorly maintained systems and old building automation systems. The CC ^{SM} cost per unit floor area for particular office buildings may be higher than the cost of some hospitals or laboratory buildings due to old building automation systems (See Table 2-2 for details). The labor cost is a complex matrix of the factors mentioned above. The cost of implementing CC ^{SM} must be determined case by case, based on individual building information.

The commissioning effort in these buildings focused strongly on resolving known comfort problems and optimizing energy use in the buildings. In some cases, building owners asked that the commissioning engineers go through the buildings and examine the operation and control of all parts of the system to specifically identify and replace any malfunctioning components in addition to dealing with known comfort problems. In these cases, the labor costs are significantly higher. These problems must be addressed before complete operational savings potential can be realized.

Table 2: Table 2-2. Summary of Measured Savings and Cost for 28 Buildings Commissioned According to ESL Continuous CommissioningSM Guidelines

Approach	Metering Cost	Reporting Cost	Total Cost

Metering is important to develop energy baselines, evaluate CC ^{SM} energy savings and identify operation and control problems. The parameters metered vary with the type of metering used. Use of utility bills requires both monthly electrical and gas bills. Use of short-term monitored data requires hourly chilled water energy or chiller electricity, heating water energy or gas consumption, possibly whole building electricity consumption, and/or electricity consumption of selected end uses such as fan motors.

Use of long-term hourly data requires chilled water energy or chiller electricity, hot water/steam energy or gas consumption and building electricity consumption. Lighting and motor control center electricity may be sub-metered in some cases. However, the additional costs of metering the lighting and motor control center electricity limit access to this data.

Metering costs depend on the type of metering used: utility bills, short-term monitoring with dedicated meters, long-term monitoring with existing meters or long- term monitoring with dedicated meters. The reporting cost also often correlates with the type of metering used. Table 2-3 lists the average metering and reporting costs for a typical building. Although costs will vary greatly depending on the complexity of the building and the location of the building, the average cost of metering and reporting varies from $1,500 to $20,500. Some methods may not be suitable for a particular project due to specific accuracy requirements. In addition, use of utility billing data (see chapter 1) will not provide diagnostic benefits for follow-up services. The type of metering should be selected based on the interests of the owner, potential savings opportunities, and the overall CC ^{SM} plan.

Table 3: Table 2-3. Typical Costs for Metering a Building and Reporting Savings for One Year

Approach	Metering Cost	Reporting Cost	Total Cost
Utility Bills		$1500	$1500
Short Term dedicated Meters	$5000	$5000	$10000
Long Term with Existing Meters	$4000	$3500	$7500
Long Term with Dedicated Meters	$17000	$3500	$20500

If the savings are determined from the utility bills before and after Continuous CommissioningSM, there is no metering cost. The staff may directly compare similar months of utility bills before and after CC ^{SM}. This is the least-accurate approach and should not be used unless it can be shown that the facility energy use is independent of weather, has the same occupancy level before and after CC ^{SM}, uses the same operational schedule before and after CC ^{SM} and the expected savings are greater than 15% of the total energy consumption. If a more formal comparison is done that considers billing period length, corrects for weather variations and provides a formal report, a typical cost would be $1,500.

When portable meters or loggers are used to measure whole-building electricity, chilled water, and hot water consumption for a short time period (one or two weeks), the cost varies from $3,000 to $10,000 with a typical cost of $5,000. This cost includes labor and travel (2 trips) to the building, meter hook-up, recording data for two weeks and the associated meter rental.

When short-term data are used to determine annual savings, a simplified hourly simulation model must be calibrated to the short-term data and a longer period of utility billing data. The modeling effort varies greatly according to available data, building type and size, and HVAC system and operational schedules. This cost can vary from $3,000 to $10,000 per building, but is typically $5,000.

When an EMCS is used to measure whole building electricity, chilled water, and hot water consumption with existing meters, the cost may vary from essentially zero (if all sensors are already present) to $20,000 or more if a complete set of new sensors and channels must be added to the system to record heating, cooling and other energy use. The cost of $4,000 shown in the table assumes that a minimal amount of new equipment is needed and that existing equipment is calibrated. New meters with dedicated data loggers can be installed to measure whole building heating, cooling and electricity use. The cooling use may come from metering the chiller and associated parasitic electricity use, or metering chilled water energy consumption. Likewise, heating consumption may come from metering heating water energy consumption or boiler gas consumption. The metering cost may be below $10,000 when only whole-building electricity and gas consumption are measured. Note that gas is typically measured by adding a signal splitter to the existing gas meters due to the higher cost associated with installing a dedicated gas meter. The metering cost is approximately $15,000 for a building where chilled water and hot water are to be measured. When there are multiple transformers, metering costs can easily exceed $20,000 including data acquisition and analysis.

When hourly data are available, a statistical regression model can be used to determine the savings accurately. The cost varies from $3,000 to $4,000 per building. This cost includes the baseline model development, database management, savings analysis and reporting. The metering cost is likely to decrease as more meters are installed as part of building automation systems. When existing meters are used, sensor calibration should be conducted first to verify the sensor accuracy and operating range. For example, a differential pressure transmitter with a 100 in H2O range may be used to transfer a signal with a maximum value of 10 in H2O. In this case, the sensor should be replaced due to a mismatch of the signal range with the sensor range.

2.2 2.2 Benefits of CC ^{SM}

The CC ^{SM} process uniquely combines four features that make it an attractive engineering process. These four features are (1) sustainable engineering solutions to operational problems, (2) superior energy and comfort performance, (3) increased staff skills and (4) service as an enabling factor for a comprehensive facility overhaul. Each feature is discussed below and examples are given.

2.2.1 Sustainable Engineering Solutions for Operational Problems

Sustainable engineering solutions are often not used to solve existing operational and comfort problems in buildings. Part of the reason is a lack of understanding of the problems and a lack of engineering knowledge by some facility operations staff. During the CC ^{SM} process, a thorough engineering inspection is conducted and measurements are made. Sustainable engineering solutions based on fundamental engineering principles are developed and implemented. This will generally solve the existing problems and decrease the maintenance cost.

CASE STUDY EXAMPLE:

Four large hearing rooms in a new state building could not maintain room temperature at the required set point (72°F) when a large number of people used the facility. In an attempt to resolve the problem, users of the facility were required to inform the operating staff 24 hours before each scheduled use. The operating staff then precooled the room temperature to temperatures as low as 66°F before a meeting. The room temperature could then be maintained below 74°F if the meeting lasted less than three hours. However, cold complaints often occurred at the beginning of the meetings and hot complaints occurred later, particularly in longer meetings.

After a thorough analysis, the CC ^{SM} engineers developed the following solutions. The supply air static pressure and temperature were reset based on the maximum terminal box damper position. If the maximum terminal damper position is less than 80% open, the supply air temperature is reset to a higher value but should not exceed the high limit determined by the room relative humidity requirement. The static pressure set point is decreased but may not decrease below a low limit setpoint. If the maximum terminal box damper position is more than 80% open, the supply air temperature is reset to a lower value and the static pressure is reset to a higher value. The room temperature set point remains at 72°F. Since the building has a modern EMCS, this type of sophisticated control and schedule were easily implemented by building operators. This control schedule provides more than the design cooling capacity to the hearing room when it is required. The "additional capacities" come from the system diversity. When a hearing room required maximum cooling, other rooms served by the same unit had less than the design cooling load. This improved schedule made the occupants more comfortable, decreased energy consumption and reduced complaints for the operating staff .

Note

The modeling effort varies greatly according to available data, building type and size, and HVAC system and operational schedules. This cost can vary from $3,000 to $10,000 per building, with $5,000 being typical.

2.2.2 Superior Energy and Comfort Performance

Design engineers face uncertainties in building design because they lack knowledge of actual occupancy levels and construction quality. To insure a workable building, HVAC systems are often designed with more capacity than required. Conservative

operation and control schedules are recommended. The HVAC engineers often pay little attention to part-load control and operation. Consequently, working as designed, a system may have poor energy performance. In addition, an excessively large system often creates comfort problems since it may not control well under very low load conditions.

During the CC ^{SM} process, accurate occupancy and operational information are available. The CC ^{SM} engineers can develop an improved or practical optimal operation and control schedule based on the information gathered during the CC ^{SM} assessment. Moreover, the CC ^{SM} engineers can fine-tune their schedules to ensure the best performance. Implementing the CC ^{SM} process often results in additional energy savings for both retrofit projects and new construction projects [Claridge et al. 1996, Liu et al. 1998].

CASE STUDY EXAMPLE:

Figure 2-1 compares the annual energy costs before retrofit, after retrofit and after CC ^{SM} in three major medical facilities. The measured savings from CC ^{SM} are higher than the retrofit savings in two of the facilities and slightly less than the retrofit savings in the third facility. The CC ^{SM} costs were a small fraction of the retrofit (capital) costs. The equipment replaced or upgraded was important to these facilities. However, in each of these three cases the retrofits were considered "completed" even though significant savings ($2,471,000/yr) resulted from commissioning of the retrofits and the facilities.

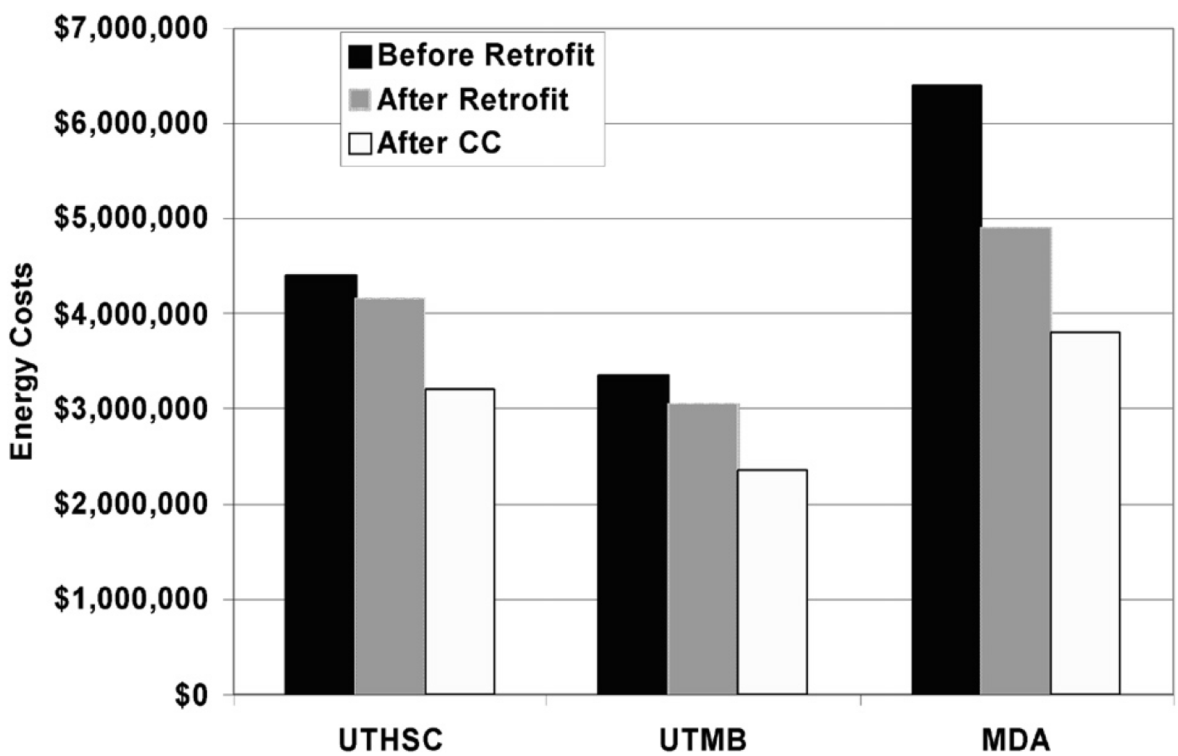

Fig. 1: Figure 2-1. Energy Costs Before Retrofit, After Retrofit, and After Retrofit and CC ^{SM} at University of Texas, Health Science Center (UTHSC), University of Texas Medical Branch at Galveston (UTMB), University of Texas, M. D. Anderson Cancer Center (MDA)

2.2.3 Increased Staff Skills

During the CC ^{SM} process, the CC ^{SM} engineers work closely with the building staff to identify operating problems and determine optimal operating strategies. The staff is also heavily involved in the decisions to implement specific commissioning measures and may actually implement these measures. Hence, they gain a higher level of skill and understanding of the engineering principles involved in optimal building operation. In addition, the staff may attend a training workshop as part of the CC ^{SM} process as was done at the Fairview University Medical Center in the case study described later.

2.2.4 An Enabling Factor for Comprehensive System Overhaul

A comprehensive system overhaul or major system upgrade is often delayed or canceled due to lack of funding. This problem can be resolved if savings from the CC ^{SM} process are used to fund the upgrade project.

The CC ^{SM} process requires minimal initial investment and produces significant energy savings as soon as the process starts. Positive cash flow is often achieved in less than two years. This creates the opportunity to use utility savings to support major retrofit projects. The CC ^{SM} process can have another significant impact on retrofit projects. It reduces the audit and engineering analysis cost since major cost-effective measures are normally identified during the CC ^{SM} process. It also decreases the risk of engineering mistakes during retrofits since the commissioning process results in an excellent understanding of the characteristics and operation of the existing systems. Consequently, retrofit costs can be controlled to a minimum.

CASE STUDY EXAMPLE:

Texas A&M University initiated campus-wide CC ^{SM} in 1995. A comprehensive energy information system was installed in 80 major buildings and five central plants at a total cost of nearly $1,000,000. The CC ^{SM} started in May 1996 after energy baselines were established for several buildings. The annual budget since 1996 for CC ^{SM}, data acquisition, and reporting is approximately $600,000. Figure 2-2 presents the accumulated savings, cost, and net cash flow from May 1995 to May 2001. Positive cash flow was achieved in approximately two years. In smaller projects, positive cash flow has often been achieved in less than one year.

The positive results of the CC ^{SM} project accelerated the process of upgrading the campus EMCS systems. Since 1996, central EMCS systems have been installed in 45 buildings and nine pumping facilities.

2.3 2.3 CC ^{SM} Project Example

Riverside North is an extended-care mental health facility built in 1962 as one of the Fairview University Medical Center hospitals in Minneapolis, Minnesota. The gross floor area is approximately 37,300 sq. ft. on four floors, excluding the basement. The building receives steam from a district steam plant and chilled water from a central campus plant. Two major air-handling units (AHUs) serve the entire building with induction terminal units. The AHUs each have 20 hp. supply fans and 10 hp. return fans operated at a constant speed 24 hours per day. Hot water supplied by steam to hot water converters is circulated to reheat coils and supplemental perimeter radiation by four 7.5 hp. pumps. The supplemental perimeter radiation system was turned on manually at outdoor temperatures below approximately 10°F and was automatically reset over a range of 120°F to 180°F when the outside air temperature varied from 10°F to -20°F. Chilled water is circulated through the building by a 10 hp. pump with a variable frequency drive located in the basement. The building has unitary controllers and a supervisory control building automation system.

The CC ^{SM} engineers were responsible for training technicians, identifying CC ^{SM} measures, conducting the engineering analysis and supervising in-house technicians during the field measurement and implementation phases. As part of the training, a two-day workshop was provided to participating technicians and other key facility staff. The CC ^{SM} engineers participated in the initial field measurement process since the building is relatively small.

After the engineering analysis, six CC ^{SM} measures summarized in Table 2-4 were identified.

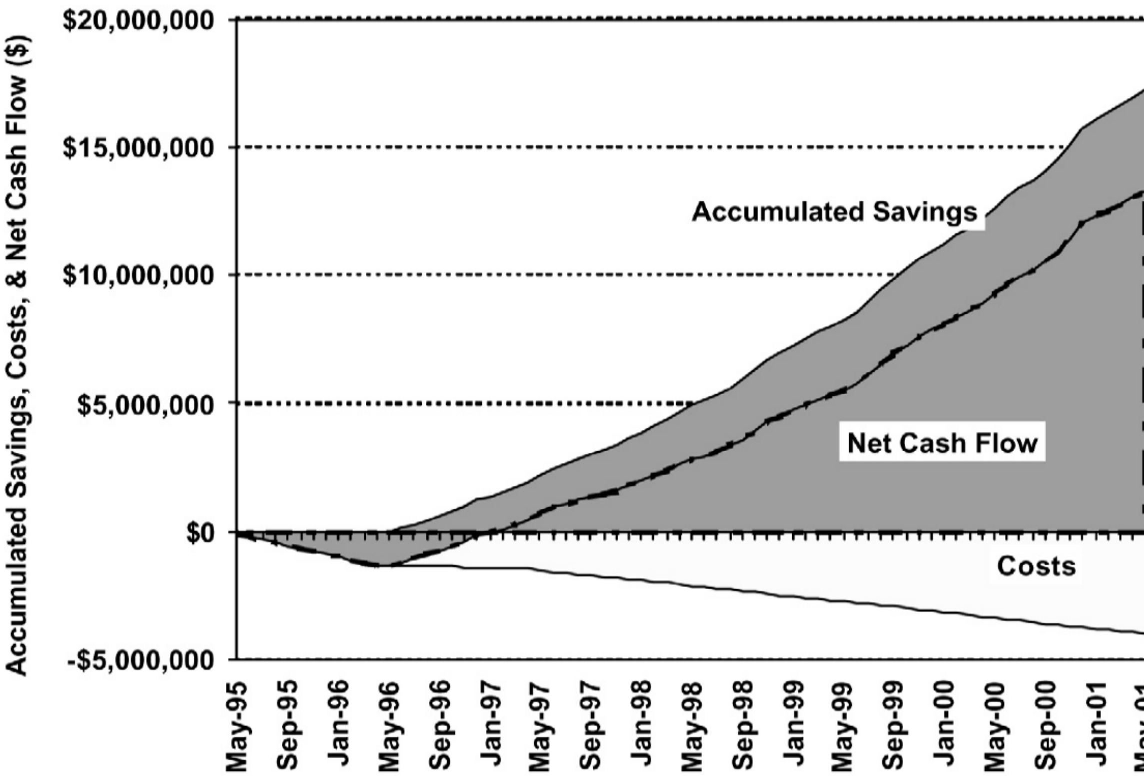

Fig. 2: Figure 2-2. Accumulated Project Cost, Energy Cost Savings and Positive Cash Flow at Texas A&M University, College Station Campus

Table 4: Table 2-4. CC Measures Identified/Implemented at Riverside North

Measure and Details
• Reduce total supply airflow from 1.7 to 1.1 cfm/sq.ft. – Change supply and return motor sheaves, reduce fan speeds – Follow up with necessary fixes to maintain comfort • Reduce outside air from 17,300 cfm to 6,660 cfm – Reduce fan speeds and properly match supply and return speeds – Repair OA dampers and actuators – Readjust minimum OA damper position based on measured OA flow • Reset supply air temperature as a function of outside air temperature – Replace constant temperature pneumatic controller with Trane stand-alone controllers tied into the existing Trane supervisory control – Switch from constant 55°F supply air to a reset schedule between 53°F and 65°F – Follow up with necessary fixes to maintain comfort • Automate and increase reset of reheat water temperature as function of outside airtemperature, reduce reheat water flow rate – Switch from manual reset between 180°F water in winter and 150°F to 160°F in summer to automatic reset between 180°F water at -20°F OAT and 110°F at 90°F OAT, using spare points on existing unitary controller – Reduce reheat pump flow with manual valve • Automatically cut off perimeter radiation above 10ºF outside air temperature using spare points on existing unitary controller • Implement zone-level fixes to assure comfort – Clean nozzle plates in induction units – Repair leaking reheat valves – Calibrate thermostats – Spot-balance air flows

At the owner’s request, a new unitary Trane controller replaced the existing pneumatic controller. All existing sensors of the AHU systems also were replaced with new ones. The energy savings were determined using short-term measured data. Figure 2-3 compares the measured heating energy consumption before and after CC ^{SM} measures were implemented. A significant reduction in cooling energy use was also achieved as noted in Table 2-5.

Table 2-5 summarizes the measured energy cost savings, cost and payback. The measured annual energy cost savings are $45,512 including $10,906 for fan power, $5,700 for chiller power, $65 for radiation pump and $28,841 for heating. The total project costs were $40,146 including $10,850 for controller and sensors, $19,296 for CC ^{SM} engineers and approximately $10,000 for in-house labor for technician participation in training, field measurement and implementation. The simple payback was 0.9 years.

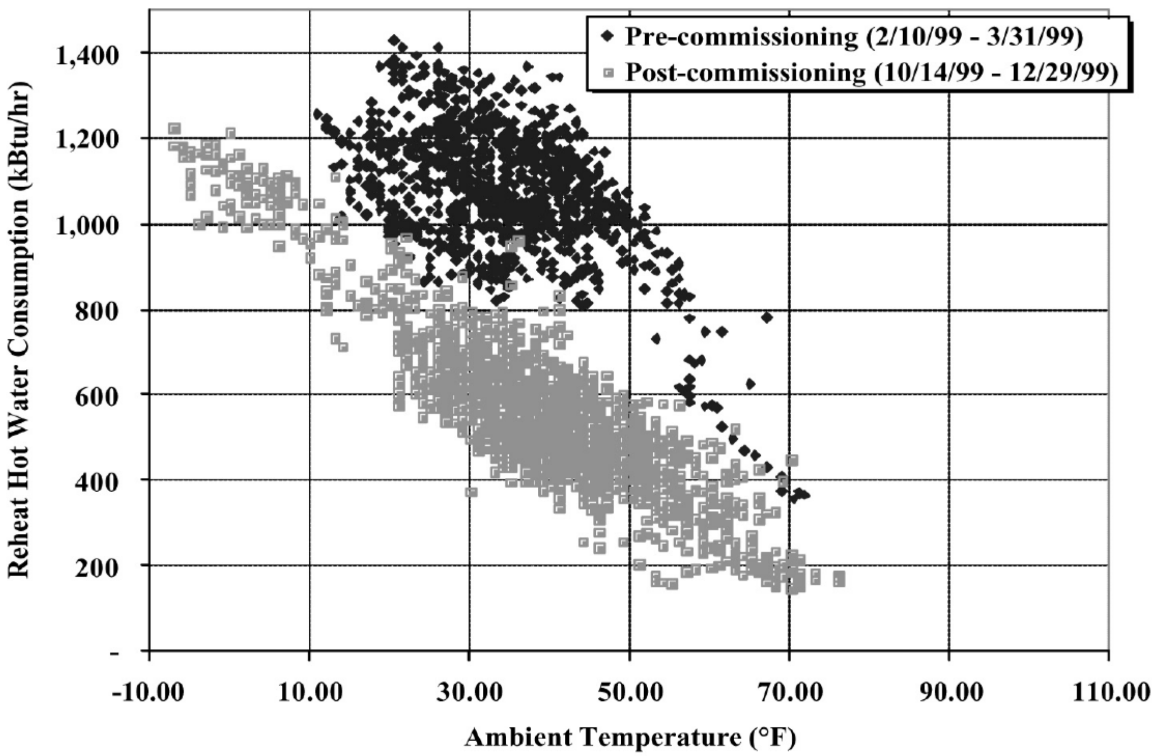

Fig. 3: Figure 2-3. Hourly Reheat Energy at Riverside North (15-minute data)

Table 5: Table 2-5. Savings and Costs at Riverside North

Savings					
END USE	Source	Electricity	Demand	Steam	Dollars
		kWh/y	kW/mo	MMBTU	$
Fan	Measured True Power Pre - Post	218,124	24.9	–	10,906
Chiller	Modeled based on SA rate and SAT	114,000	33.4	–	5,700
Radiation Pump	Estimated	1,300		–	65
Reheat	Measured energy bin analysis			3,770	28,841
Total		333,424	58.3	3,770	45,512
Costs					
Equipment&materials (Trane Controller and a few sensors)					10,850
Engineering					19,296
In-house labor					10,000
Total					401,46

2.4 2.4 Summary

The Continuous CommissioningSM process typically provides payback in less than three years and often in less than two years. Project duration typically varies from three to six months per building.

The CC SM process improves building comfort and decreases maintenance cost. It also provides significant benefits to the owners by (1) identifying potential energy retrofits, (2) upgrading the technical level of in-house staff and (3) providing energy savings that may be used to finance a comprehensive facility overhaul or upgrade.

References

Claridge, D.E., J.S. Haberl, M. Liu, J. Houcek and A. Athar, 1994. "Can You Achieve 150% of Predicted Retrofit Savings: Is It Time for Recommissioning?" ACEEE 1994 Summer Study on Energy Efficiency In Buildings Proceedings: Commissioning, Operation and Maintenance, Vol. 5, American Council for an Energy Efficient Economy, Washington, D.C., pp. 73-87.

Claridge, D.E., M. Liu, Y. Zhu, M. Abbas, A. Athar and J.S. Haberl, 1996. "Implementation of Continuous CommissioningSM in the Texas LoanSTAR Program: Can You Achieve 150% of Estimated Retrofit Savings Revisited," Proceedings 4, Commercial Buildings: Technologies, Design, and Performance Analysis. ACEEE 1996 Summer Study on Energy Efficiency In Buildings, American Council for an Energy Efficient Economy, Washington, D.C., pp. 4.59-4.67.

Gregerson, Joan, 1997. "Cost Effectiveness of Commissioning 44 Existing Buildings," Proceedings Fifth National Conference on Building Commissioning, Huntington Beach, CA April 28-30.

Liu, M., A. Athar, D.E. Claridge, J. S. Haberl and E. White, 1994a. "Reducing Building Energy Costs Using Optimized Operation Strategies for Constant Volume Air Handling Systems," Proceedings of the Ninth Symposium on Improving Building Systems in Hot and Humid Climates, Arlington, Texas, May 19-20, pp. 192-204.

Liu, M., J. Houcek, A. Athar, A. Reddy and D.E. Claridge, 1994b. "Identifying and Implementing Improved Operation and Maintenance Measures in Texas LoanSTAR Buildings," ACEEE 1994 Summer Study on Energy Efficiency In Buildings Proceedings: Commissioning, Operation and Maintenance, Vol. 5, American Council for an Energy Efficient Economy, Washington, D.C., pp. 153-165.

Liu, M., Y. Zhu, T. Powell and D.E. Claridge, 1998. "System Optimization Saves $195,000/yr. in a New Medical Facility," Proceedings of the 6th National Conference on Building Commissioning, Lake Buena Vista, FL, May 18-20, pp. 14.2.1-14.2.11.

CHAPTER 3

Ch 3: Basic CC ^{SM} Measures

The measures considered in this chapter are largely the classical "shut it off if it isn't needed" measures, supplemented with "slow it down if you can." It is important to remember that any measure must not degrade the occupant's comfort and must not negatively impact productivity.

3.1 3.1 Remove Foot Heaters and Turn Off Desk Fans

The presence of foot heaters and desk fans indicates an unsuitable working environment and an energy waste as well. To turn off foot heaters and desk fans, the following actions should be taken:

- Adjust the individual zone temperature set point according to the occupant's desires
- Balance zone airflow if foot heaters are used in a portion of the zone
- Adjust AHU supply air temperature and static pressure if the entire building is too cold or too hot
- Fix existing mechanical and control problems such as replacing diffusers of the wrong type and relocating return air grilles in order to maintain a comfortable zone temperature

Different people have different temperature requirements to feel comfortable. Some organizations, however, mandate the zone temperature set point for both summer and winter. This often leads to comfort complaints and negatively impacts productivity. The operating staff must place comfort as a priority and adjust the room temperature set point as necessary. Workers should be asked to dress appropriately during the summer and winter to maintain their individual comfort if set points are centrally mandated for a facility. Most complaints can be eliminated when the room temperature is within the range of ASHRAE's recommended comfort zone.

3.2 3.2 Turn off Heating Systems During Summer

Heating is not needed for most buildings during the summer. When the heating system is on, the hot water or steam often leaks through control valves, causing thermal comfort problems and consuming excessive cooling and heating energy. To improve building comfort and decrease heating and cooling energy consumption, the following actions should be taken:

- Turn off boilers or heat exchangers if the entire building does not require heating
- Manually valve off heating and preheating coils if the heating system must be on for other systems
- Reset differential pressure of the hot water loop to a lower value to prevent excessive pressure on control valves during the summer
- Trouble-shoot individual zones or systems that have too many cold complaints * Do not turn heating off too early in the summer in order to avoid having to turn the system on and off repeatedly

This measure may be applied in constant air volume systems in dry climates. When the reheat system is shut off, room comfort may be maintained by increasing supply air temperature. This measure is not suitable for other climates where the temparature of the air leaving the cooling coil must be controlled below 57°F to control room humidity levels.

This simple measure results in significant energy savings as well as improved comfort in most buildings. Figure 3-1 compares the measured heating energy consumption before and after manually shutting off AHU heating valves in a building in Austin, Texas.

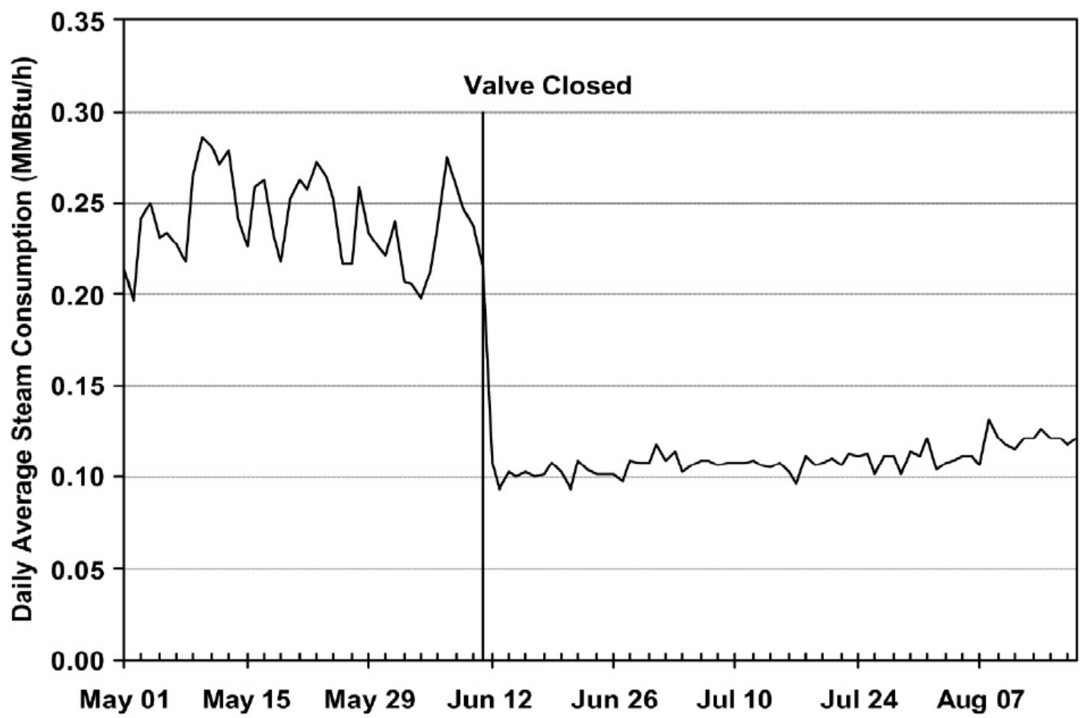

Fig. 1: Figure 3-1 Comparison of Measured Daily Average Steam Consumption Before and After Manually Shutting Off Heating Coil Valves in the Business Administration Building at University of Texas at Austin

The building has a floor area of 147,000 sq.ft. with two dual duct VAV systems. Before closing the heating coil manual valves, the average daily steam consumption varied from a low of 0.2 MMBtu/hr to a high of approximately 0.28 MMBtu/hr. After the manual valves were closed, steam leakage was eliminated through the heating coil. The steam consumption immediately dropped to slightly above 0.10 MMBtu/hr. Since the manual valves in this building can stay closed for more than seven months, the annual steam savings are 756 MMBtu/yr. The same amount of chilled water will be saved if the building remains at the same temperature, creating cooling energy savings of 756 MMBtu/yr as well. The annual energy cost savings is $7,560 at an energy price of $5/MMBtu. This savings is not huge, but the only action required was shutting two manual valves.

3.3 3.3 Turn Off Systems During Unoccupied Hours

If a building is not occupied at night or on weekends, the HVAC system often may be turned off completely during these periods. With a properly designed warm-up/cool-down, building comfort can be maintained properly with significant energy savings. In a commercial or institutional building, office equipment and lighting make up a large portion (50% or more) of the electrical system requirements. However, a significant portion of a building (15% or more) is unoccupied during office hours due to travel, meetings, vacations, and sick leave. Turning off systems during unoccupied hours results in significant energy savings without degrading occupant comfort. This measure can be achieved by the following actions:

- Turn off lights, computers, printers, fax machines, desk fans and other office equipment when leaving the office
- Turn off lights and set back room thermostats after cleaning
- Turn off AHUs at nights and on weekends. A schedule must be developed for each zone or air handling unit. Turning off the system too early in the evening or turning the system on too late in the morning may cause comfort problems. Conversely, turning off a system too late in the evening or turning the system on too early in the morning may lose considerable savings.
- Turn off the boiler hot water pump at night during the summer when AHUs are turned off
- Turn off chillers and chilled water pumps when free cooling is available or when AHUs are turned off

Note

With a properly designed warm-up/cool-down, building comfort can be maintained properly with significant energy savings.

Figure 3-2 presents the measured building electricity consumption, excluding chiller consumption, before and after implementation of AHU and office equipment turn-off on nights and weekends in the Stephen F. Austin Building in Austin, Texas.

The Stephen F. Austin Building has 470,000 sq.ft. of floor area with 22 dual duct AHUs. During the first phase of implementation, 16 AHUs were turned off from midnight to 4 a.m. weekdays and weekends. During the second phase, 22 AHUs were turned off from 11:00 p.m. to 5 a.m. during weekdays and weekends. During the second phase, all occupants were asked to turn off office equipment when they leave their office.

The measured results show that the nighttime whole building electricity use decreased from 1,250 kW to 900 kW during the first phase. During the second phase, the nighttime minimum electricity decreased to 800 kW.

It was observed that the daily peak electricity consumption after night shutdowns began is significantly lower than the base peak. For example, the lowest peak during the second phase is 1,833 kW, which is 8% lower than the base peak. The lower electricity peak indicates that some office equipment remained off during the daytime or employees were more conscientious in turning off lights and equipment when they left the office. The annual energy cost savings, including electricity, heating and cooling, were determined to be $100,000/yr using measured hourly data.

3.4 3.4 Slow Down Systems During Unoccupied/Lightly-Occupied Hours

Most large buildings are never completely unoccupied. It is not uncommon to find a few people working regardless of the time of day. The zones that may be used during the weekends or at nights, are also unpredictable. System shut down often results in complaints. Substantial savings can be achieved while maintaining comfort conditions in a building by an appropriate combination of the following actions:

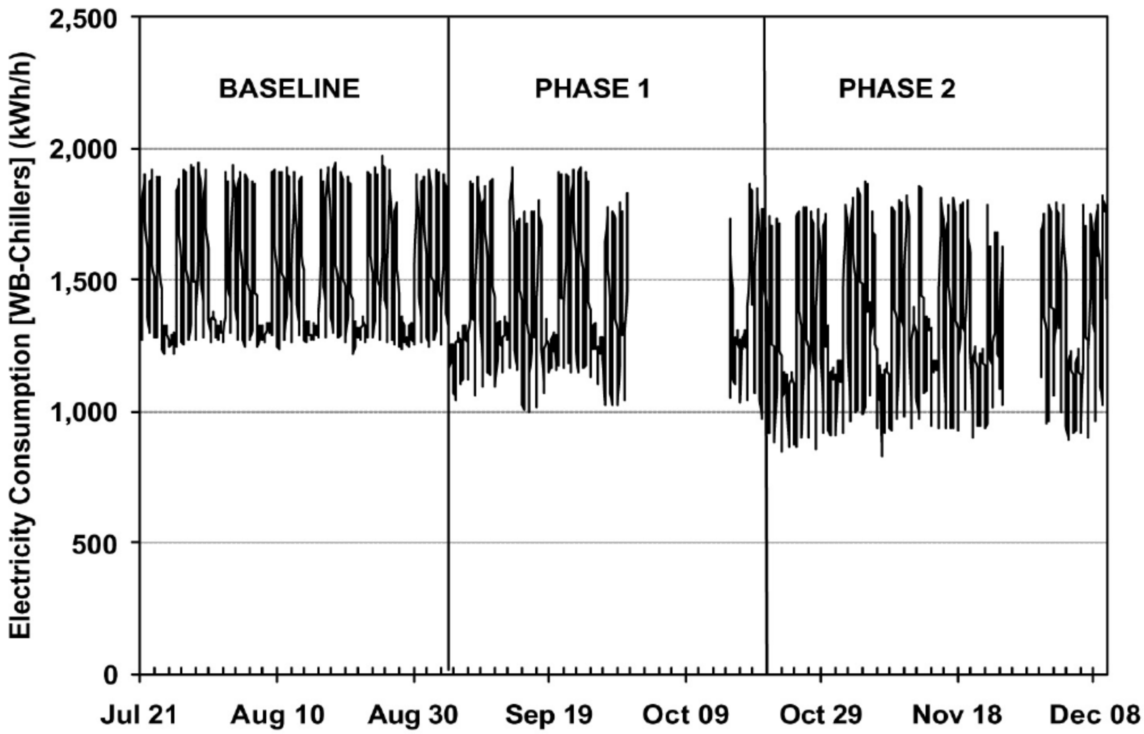

Fig. 2: Figure 3-2. Hourly Whole Building Electricity Consumption at the Stephen F. Austin (SFA) Building Before and After Night Shut Down of AHUs Was Initiated

- Reset outside air intake to a lower level (0.05 cfm/sq.ft.) during these hours during hot summer and cold winter weather. Outside air can be reduced since there will be very few people in the building. Check outside and exhaust air balance to maintain positive building pressure.
- Reset the minimum airflow to a lower value, possibly zero, for VAV terminal boxes
- Program constant volume terminal boxes as VAV boxes and reset the minimum flow from the maximum to a lower value, possibly zero, during unoccupied hours
- Reset AHU static pressure and water loop differential pressure to lower values
- Set supply air fan at lower speed

These measures maintain building comfort while minimizing energy consumption. The savings are often comparable with the shutdown option. Figure 3-3 presents the measured hourly fan energy consumption in the Education Building at the University of Texas at Austin. The Education Building has 251,000 sq.ft. of floor area with eight 50 hp. AHUs that are operated on VFDs. Prior to the introduction of this measure, the motor control center (MCC) energy consumption was almost constant. The CC ^{SM} measure implemented was to set the fan speed at 30% at night and on weekends. The nighttime slowdown decreased the fan power from approximately 50 kW to approximately 20 kW while maintaining building comfort.

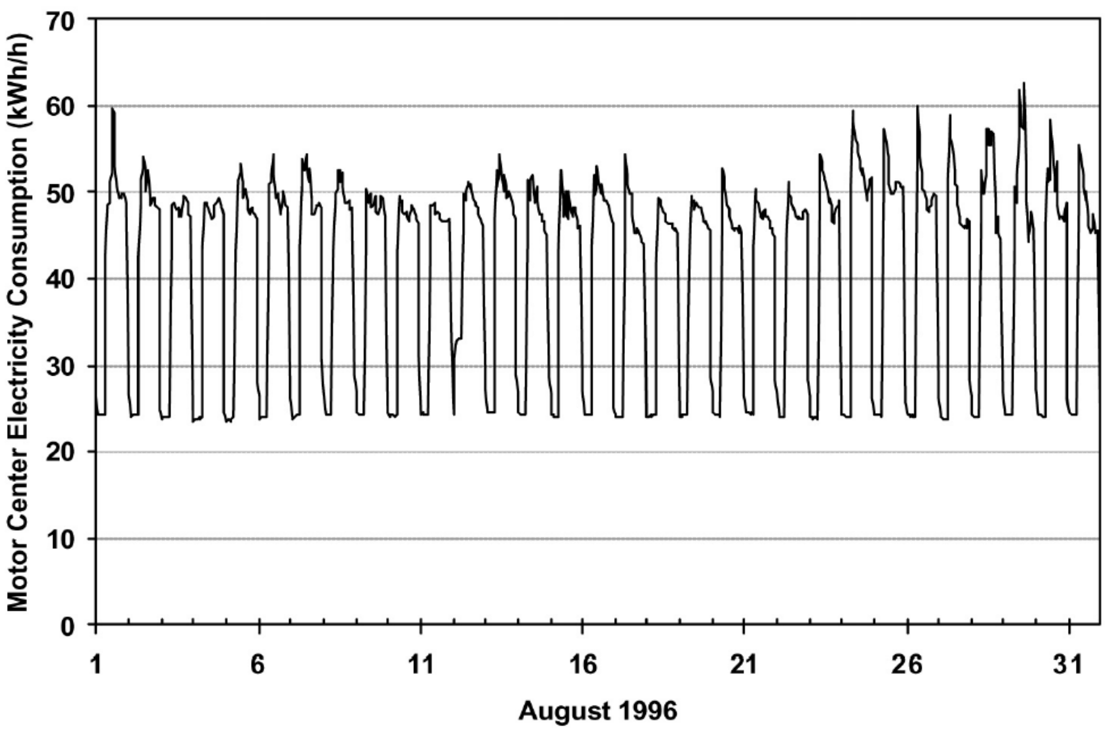

Fig. 3: Figure 3-3. Measured (Post-CC ^{SM}) Hourly Supply Fan Electricity Consumption in the Education Building

3.5 3.5 Limit Fan Speed During Warm-Up and Cool-Down Periods

If nighttime shut down is implemented, warm-up is necessary during the winter and cool-down is required during the summer. During warm-up and cool-down periods, fan systems are often run at maximum speed since all terminal boxes require either maximum heating or maximum cooling. A simple fan speed limit can reduce fan power significantly.

This principle may also be used in other systems such as pumps. The following actions should be taken to achieve the fan energy savings:

- Determine the optimal start up time using 80% (adjustable) fan capacity if automatic optimal start up is used
- Set the fan speed limit at 80% (adjustable) manually and extend the warm up or cool down period by 25%. If the speed limit is set at another value (x), determine the warm up period using the following equation:

  $T_n = \frac{T_{exist}}{x}$
- Keep outside air damper closed during warm-up and cool-down periods

The fan energy savings increase significantly as the fan speed limit decreases. Figure 3-4 presents the theoretical fan power savings. When the fan speed limit is 50% of design fan speed, the potential fan energy savings are 75% of the fan energy even if the fan runs twice as long. The theoretical model did not consider the variable speed drive loss. The actual energy savings will normally be somewhat lower than the model projected value.

Note that if the outside air damper cannot be closed tightly, extra thermal energy may be required to cool or warm outside air that leaks through the damper. This factor should be considered when this measure is used.

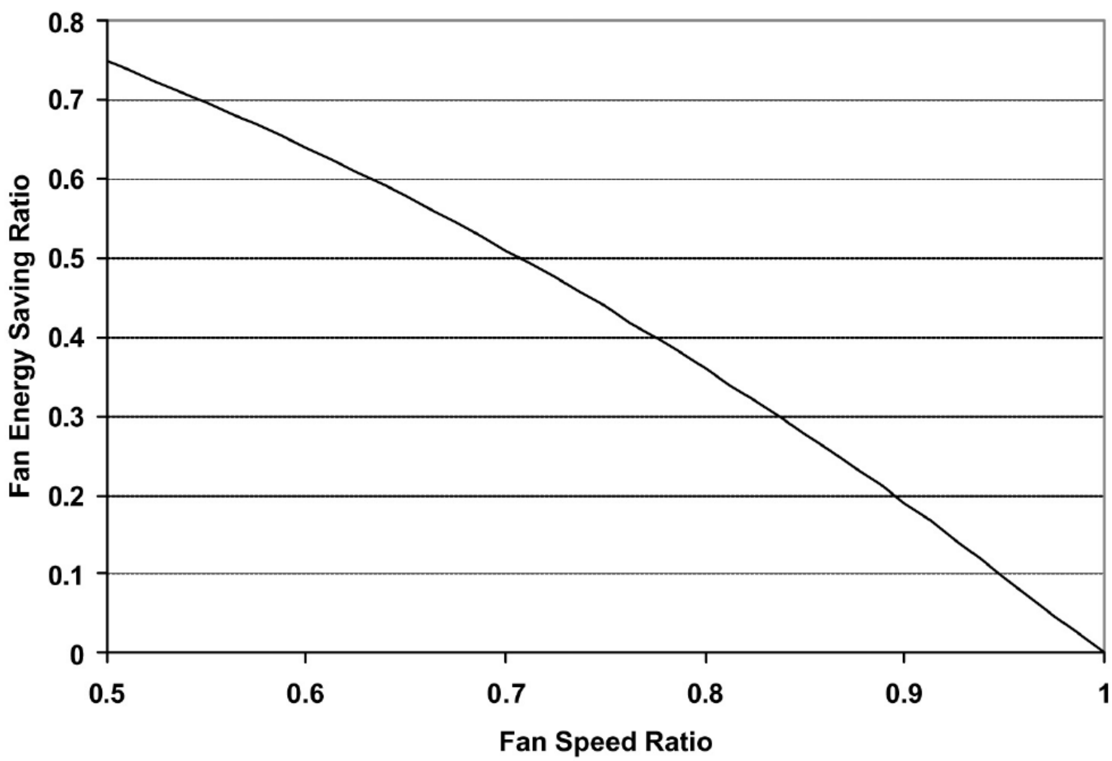

Fig. 4: Figure 3-4. Potential Fan Energy Savings Using Fan Speed Limiting

3.6 3.6 Summary

Significant amounts of energy can be saved by implementing the basic CC ^{SM} measures. More advanced CC ^{SM} measures can be used to improve the building energy performance; however, this chapter is limited to the simple measures. Before implementing these measures, the building and HVAC system must be in good condition. Local water and air balances may be required to solve existing mechanical problems.

CHAPTER 4

Ch 4: CCSM Measures for AHU Systems

Air handler systems normally condition and distribute air inside buildings. A typical AHU system consists of a combination of heating and cooling coils, supply and return air fans, filters, humidifiers, dampers, ductwork, terminal boxes and associated safety and control devices, and possibly an economizer. As the building load changes, AHUs change one or more of the following parameters to maintain building comfort: outside air intake, total air flow, static pressure, supply air temperature, and humidity. Both operating schedules and initial system set up, such as total air flow and outside air flow, significantly impact building energy consumption and comfort. This chapter presents 10 major CCSM measures used to optimize AHU operation and control schedules:

- Adjust total air flow for constant air volume systems
- Set minimum outside air intake correctly
- Improve static pressure set-point and schedule
- Optimize supply air temperatures
- Improve economizer operation and control
- Improve coupled control AHU operation
- Valve off hot air flow for dual duct AHUs during summer
- Install VFD on constant air volume systems
- Install airflow control for VAV systems
- Improve terminal box operation

4.1 4.1 Adjust Total Air Flow and Fan Head for Constant Air Volume Systems

Airflow rates are often significantly higher than required in buildings primarily due to system over-sizing. In some large systems, an oversized fan causes over-pressurization in terminal boxes. This excessive pressurization is the primary cause of room noise. The excessive air flow often causes excessive fan energy consumption, excessive heating and cooling energy consumption, humidity control problems and excessive noise in terminal boxes [Liu, et al. 1999].

For constant volume systems, excessive air flow can be identified using one of the following methods:

- Measure zone supply air temperatures under peak building load conditions. If the zone supply air temperature (Tz - the temperature in the duct down stream of the reheat coil and ahead of the diffuser for a single duct system or the temperature down stream of the terminal box and before the diffuser for a dual duct system) of the average zone is more than 2°F higher than the design supply air temperature (Tc - the temperature specified by the designer as the required system supply air temperature leaving the fan or the cooling coil), the actual air flow is higher than required by the thermal load. The airflow reduction ratio (α) can be determined by equation (1), where Tr is the room return air temperature. The new fan speed should be (1-α) times the existing fan speed.

$$\alpha = 1 - \frac{T_r - T_z}{T_r - T_c}(4.1)$$

Measure actual air flow and compare with the design airflow rate. If the air flow rate is higher than the design value, the air flow can be adjusted to the design value. Experience shows that evaluating the actual air flow required in the space will find that most zones will function properly with 80% of the design flow due to system over-sizing and load diversity factor. The new fan speed is (cfm_n / cfm_e) times the current speed.

Excessive fan head for single duct systems can be identified using one of the following methods:

- Select remote terminal boxes that have the longest duct runs from the supply air fan or that have the maximum damper open positions
- Measure the static pressure at the front of the remote terminal boxes and select the lowest value as (p_e)
- Force the most remote terminal box (where p_e is measured) damper to the 95% open position by adjusting balancing damper
- Measure the static pressure at the front of the most remote terminal box again (p_d)
- Calculate the potential static pressure reduction using p_e-p_d

For dual duct constant air volume systems, the following procedures are recommended:

- Select remote terminal boxes which have the longest duct runs from supply air fan or which have the maximum cold air damper open positions
- Force the zone to full cooling using the zone thermostat. Measure the cold air static pressure in front of the selected remote terminal boxes and determine (p_e).
- Force terminal box cold air damper where pe was measured to the 95% open position by adjusting balancing damper
- Measure the cold air static pressure at the front of the remote terminal box again (p_d)
- The potential static pressure reduction is calculated as p_e-p_d

If the static pressure at the most remote terminal box is higher than necessary, the new fan speed can be determined based on the potential static pressure reduction (p_e-p_d) and the fan curve. Figure 4-1 presents the procedure. Locate the fan working point first. The new working point is (p_e-p_d) down from the existing working point. Identify the new fan speed from the fan curve. This method is conservative since the air flow is often lower than the current value after fan head reduction. This airflow reduction is directly associated with terminal box leakage that is reduced by the pressure reduction. It will not cause comfort problems.

The airflow reduction can result in significant fan energy savings and reduced noise level. The following tips will help the implementation process:

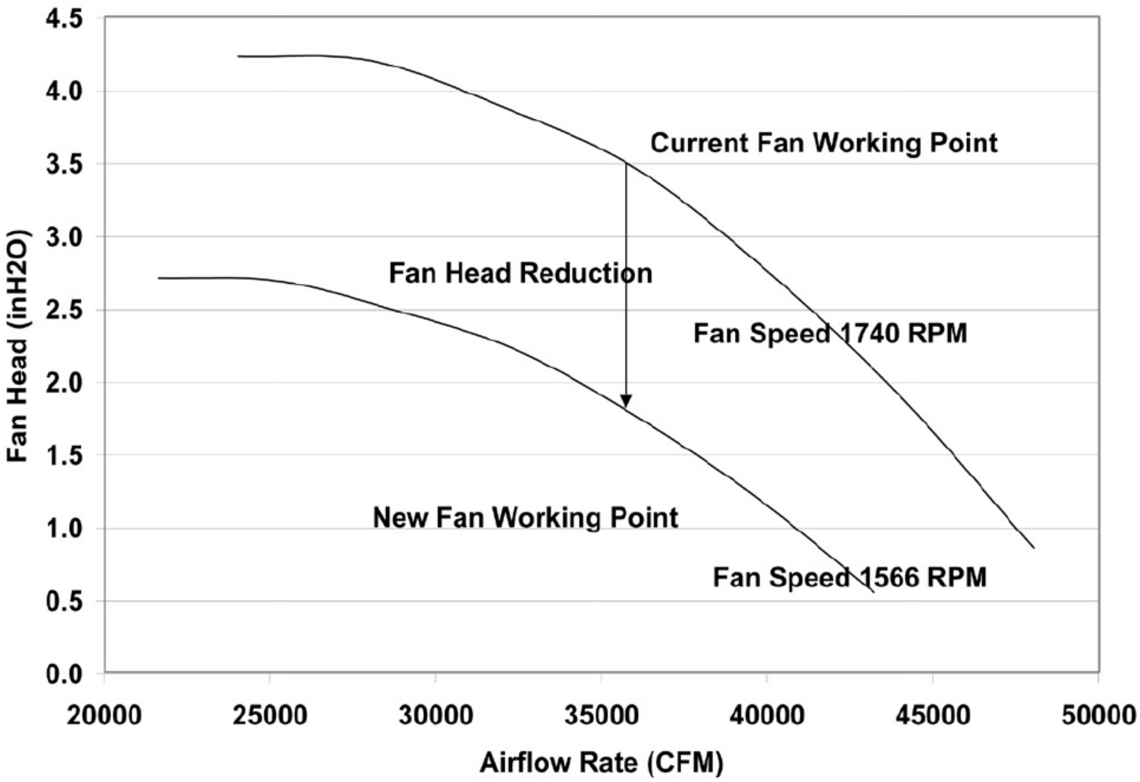

Fig. 1: Figure 4-1. Determining New Fan Speed Using Measured Fan Head Reduction and Fan Performance Curve

- VFD or New Pulley: The fan speed can be changed by modifying the fan pulley or using a variable frequency drive. A VFD is generally recommended. The installation of a VFD allows further refinement of fan speed and slowing of the AHU during nights and weekends. The additional savings can generally offset the VFD cost within two years. In some cases, changing fan pulleys is more attractive because it avoids the VFD power loss and the cost is much lower.
- Air Circulation and Thermal Comfort: For hospitals and similar facilities, air flow must satisfy both air circulation and thermal comfort requirements. Make sure that any airflow reduction will not violate ventilation code requirements.
- Sequence of Implementation: Before reducing fan speed, existing mechanical problems should be addressed at the zone level. If a zone lacks air due to distribution problems such as a kinked flex duct or a damper stuck closed, solve these distribution problems first. After these air balance problems are solved at the zone level, the fan speed adjustment can be performed.
- Readjustment of Outside Air Flow: For most existing systems, the minimum out side air damper position should be adjusted after adjusting total air flow. Make sure that the minimum outside air intake satisfies IAQ requirements. See next section for details.

More information can be found in “Continuous CommissioningSM of a Hospital Complex” [Wei et al. 2001].

EXAMPLE:

The Materials Research Institute (MRI) Building in State College, Pa., has 50,000 square feet of floor area that includes offices, laboratories and classrooms.

Fig. 2: Figure 4-2. The Materials Research Institute Building in State College, Pa.

There are three major AHUs in the building. AHU1 is a DDVAV unit with inlet guide vanes and supplies air to the offices in the building. AHU3 and AHU4 are 100% outside air dual duct constant volume units that supply air to the laboratory areas in the building. The minimum static pressure was measured to be 2.68 in. H2O at the entrance to the last terminal box. Pulley sizes on AHU3 and AHU4 were reduced to lower the static pressure to approximately 1.0 in. H2O. A number of other measures were also implemented in the summer of 1998.

Figure 4-3 compares the monthly gas and electricity consumption before and after implementation of fan pulley changes and other CCSM measures in the MRI building. Utility bill analysis showed a 40% reduction in annual gas use with a cost savings of $52,382/yr. The annual electricity use was reduced by 12% with cost savings of $34,250/yr. The total annual cost savings were $86,632/yr, or 21% of the total utility cost.

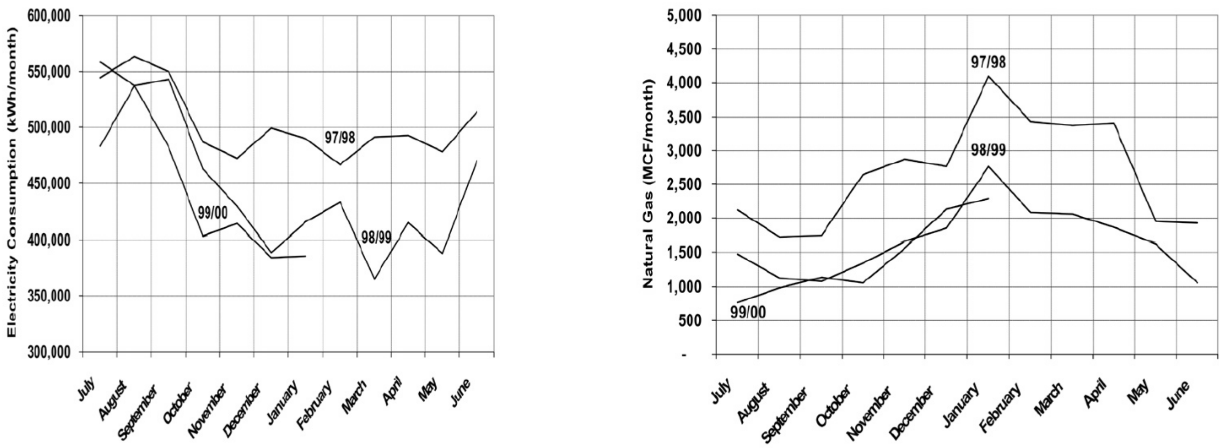

Fig. 3: Figure 4-3. Natural Gas and Electricity Consumption at the Materials Research Institute Before and After Implementation of CCSM Measures

4.2 4.2 Set Minimum Outside Air Intake Correctly

Outside air intake rates are often significantly higher than design values in existing buildings due to lack of accurate measurement, incorrect design calculation and balancing, and operation and maintenance problems. Excessive outside air intake isoften directly caused by one or more of the following:

- Mixed air chamber pressure is lower than the design value. For example, the static pressure often varies from –0.2 in. H2O to –1.0 in. H2O when the design assumed –0.1 in. H2O or higher.
- Significant outside air leakage through the maximum outside air damper on systems with an economizer. Due to the large size of this damper, the air leakage can be significant.
- Minimum outside air intake is set using minimum total air flow for a VAV system. For most existing systems, the minimum outside air damper position is set at a fixed position. When the total air flow is higher than the minimum system flow, the pressure in the mixed air chamber becomes more negative. Consequently, the outside air intake is higher than the minimum required when total air flow is higher than the minimum air flow.
- Lower than expected/designed occupancy. For example, the outside air intake is often determined based on space peak occupancy schedule. However, when a meeting is held in a conference room, several offices are not occupied. It is estimated that 10% or more of the occupants will not be present in their working place at any given time due to travel, meetings, vacations or sick leave. Hence the minimum outside air flow is often significantly over- designed.

The excessive outside air intake consumes a significant amount of extra heating and cooling energy. Each extra cfm of outside air intake typically costs from $1 to $3 per year depending on location and energy cost. If there is too much outside air, the AHU may lose the ability to control room humidity and temperature.

Excessive outside air intake can be identified by one of the following methods:

- Measure CO2 level of the return air for critical zones. For a typical office building under normal occupied conditions, the return air CO2 level should be 500 ppm to 600 ppm higher than the outside air CO2 level when minimum design outside air is used. If the CO2 level of return air increases by less than 500 ppm under normal occupancy, it indicates excessive outside air intake. Do not apply this criterion during economizer cycle operation.
- Measure outside air intake and compare with the design value. If a section of straight ductwork carries the outside air, a direct flow measurement is recommended. The airflow measurement may also be performed using

turbine flow sensors in the inlet of the outside air intake grilles. However, this measurement should be performed when wind speed is lower than 15 FPM.

- Measure air flow indirectly using temperature measurements. In most cases, the outside air intake goes directly or almost directly into the mixing chamber. Therefore, outside air flow is determined using measured total air flow and temperatures (mixed, outside and return):

$$CFM_{oa} = CFM_t \frac{T_m - T_r}{T_{oa} - T_r}$$

This method can be used only when the temperature difference between return and outside air is greater than 10°F. To improve the measurement accuracy, one probe should be used to measure all temperatures. Be cautious about using control system sensors for these measurements. The typical measurement errors of control sensors are +1.5°F, which will, in many cases, significantly lower the accuracy of the outside air flow determined. The location of the sensors may also cause problems. The return air temperature sensor must be located on the discharge side of the return air fan. The mixed air temperature sensor must be located before the supply air fan. The temperature measurement methods must ensure true average temperature.

Note

Be cautious about using control system sensors for these measurements. The typical measurement errors of control sensors are +1.5°F, which will, in many cases, significantly lower the accuracy of the measurement.

Minimum outside air control can be implemented using one of the following methods:

- For constant air volume systems, the minimum outside air intake should be adjusted using the outside air damper. After adjusting the outside air damper position, the air flow should be measured again to verify the flow. A seasonal inspection and adjustment is suggested since the air leakage through the maximum outside air dampers changes significantly after economizer operation.
- For VAV systems, the outside air damper position corresponding to correct minimum flow should be determined at both the minimum and the maximum total air flows. The minimum outside air damper position can be modulated between these two positions when the total air flow varies from the minimum to the maximum value.
- For VAV systems, where the minimum outside air damper is not controlled by an independent actuator, the following action can improve outside air control and minimize excessive thermal energy consumption associated with outside air intake:
  - Set the minimum outside air intake position when the building is normally occupied and the outdoor temperature is about 65°F
  - Reset static pressure to a lower value when the total air flow is lower than the design value. Decrease the static pressure from the design value to 50% of the design value as the total air flow decreases from 100% to 70%. For example, assume the design static pressure set point is 1.2 in. H2O at maximum air flow. When the AHU air flow is 70% of the design value, the static pressure would be reset to 0.6 in. H2O.

The minimum outside air requirement actually depends on both occupancy and building exhaust air flow. The outside air requirement decreases as the number of occupants decreases. However, the outside air intake must be slightly higher than the common exhaust air flow in order to maintain positive building pressure. Dynamically adjusting outside air intake based on the occupancy can result in significant building energy savings while maintaining satisfactory indoor air quality. The optimal outside air control (demand control) can be implemented using one of the following methods:

- Install CO2 sensor(s) to measure return air or representative zone CO2 level. Modulate the outside air damper to maintain the set point. When this method is used, a low limit must be set to make sure that the outside air intake is higher than the common exhaust air flow.

- Develop an outside air damper reset schedule based on the time of the day. The occupancy level has a strong correlation with the day of the week and the time of day. When occupancy information is available, a damper schedule can be developed and implemented using the building automation system.

EXAMPLE:

The Starr Building in Austin, Texas, is a typical older state government office building. It consists of a three-story section and a six-story section with a combined floor area of 99,000 ft2. The HVAC systems included two 175-ton hermetic centrifugal chillers, two 2.4 MMBtu/hr gas fired boilers and four multi-zone AHUs.

Due to excessive negative pressure in the mixed air chambers, the building outside air intake (79,950 cfm) was nine times higher than the required minimum outside air intake (9,000 cfm). Due to a number of control problems, the total air flow (140,700 cfm) was 25% higher than the design value (112,225 cfm) and 53% higher than the required value (92,130 cfm).

The excessive outside air and total air flows caused significant building comfort problems as well as substantial energy waste. Since 1985 the building had experienced garage air backflow into the building though AHUs, high room temperatures in numerous rooms, high relative humidity (up to 80%) during summer, cold rooms during winter, and high building positive pressure.

After adjusting both outside air flow and total air flow to the required level, the building comfort problems were solved. Significant energy savings were also achieved. Table 4-1 compares the measured room temperature, relative humidity, and CO2 level in 18 pre-selected rooms before and after outside air and total airflow reduction. After implementing airflow reduction, the building comfort was controlled properly. The maximum room relative humidity decreased from 69% to 55%. The building positive pressure decreased from 0.1 in. H2O to 0.02 in. H2O.

Table 1: Table 4-1. Comparison of Room Comfort Parameters Before and After CCSM Implementation

Condition	Before CC	After CC
Room CO2 Level	400 – 500 ppm	650 - 800 ppm
Room Temperature	67 - 74.5 °F	72 – 75 °F
Room Relative Humidity	58% - 69%	50% - 55%
Building Positive Pressure	0.1 in. H2O	0.02 in. H2O

Note: The ambient air temperature was 88°F on June 8, 1995 when the pre-CCSM test was performed. The ambient temperature was 99°F on June 14, 1996 when the post CCSM test was performed.

Figure 4-4 compares the measured monthly average hourly electricity consumption before and after the implementation of the CCSM measures. The simple linear regression model shown was created based on the measured data. When the ambient temperature was low, the post-CCSM electricity consumption was higher than that of the pre-CCSM period due to the reduced outside air flow. When the ambient temperature was high, the post CCSM electricity consumption was significantly lower than during the pre-CCSM period. The measured electricity demand savings were 90 kW when the ambient temperature was 85°F.

Figure 4-5 presents the measured monthly average hourly heating energy consumption versus the monthly average ambient temperature. Regression models of the data are also presented. The measured heating energy savings varied from 0.1 MMBtu/hr to 0.7 MMBtu/hr when the monthly average hourly temperature was between 75°F and 52°F. The measured gas savings varied from 0.12 MMBtu/hr to 0.88 MMBtu/hr when the monthly average hourly temperature varied from 75°F to 52°F.

The measured annual energy savings are 4,940 MMBtu/yr which includes 1,640 MMBtu/yr of electricity savings and 3,300 MMBtu/yr of gas savings. The annual energy use index decreased from 150,800 Btu/ft2/yr to 101,000 Btu/ft2/yr. More detailed information can be found in “An O&M Story in An Old Building” [Liu et al. 1996].

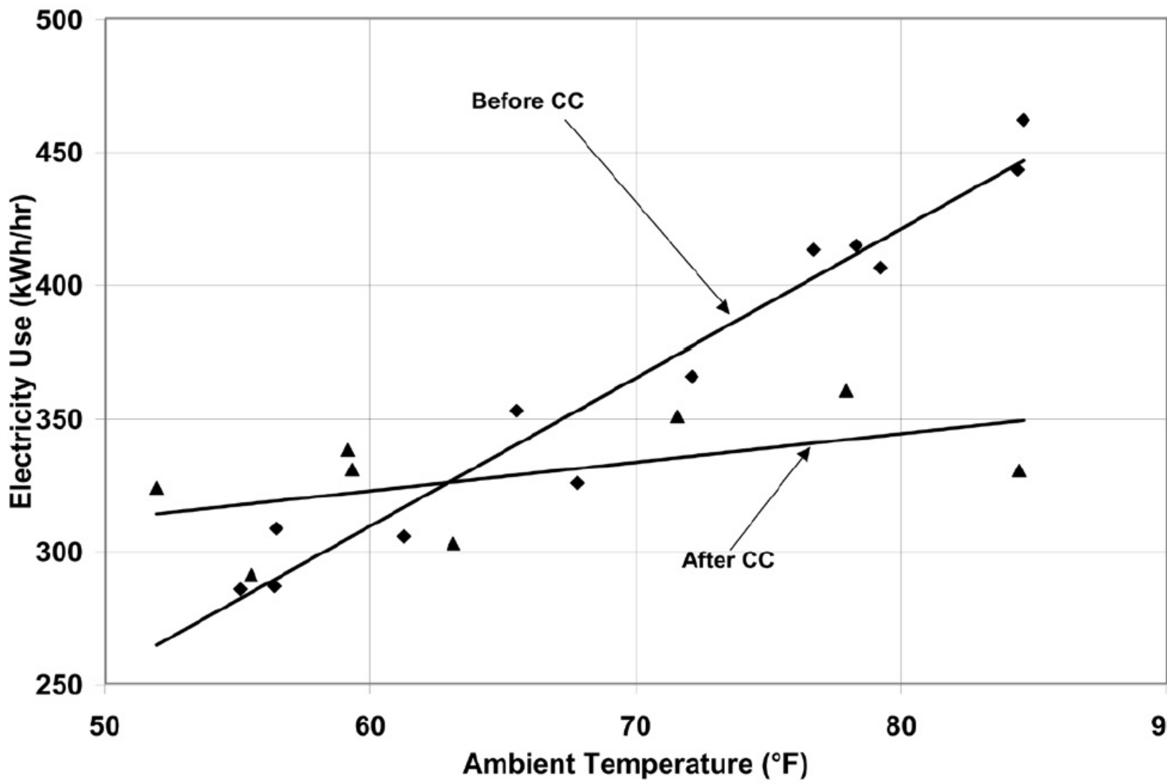

Fig. 4: Figure 4-4. Measured Monthly Average Hourly Whole Building Electricity Consumption Versus the Monthly Average Ambient Temperature

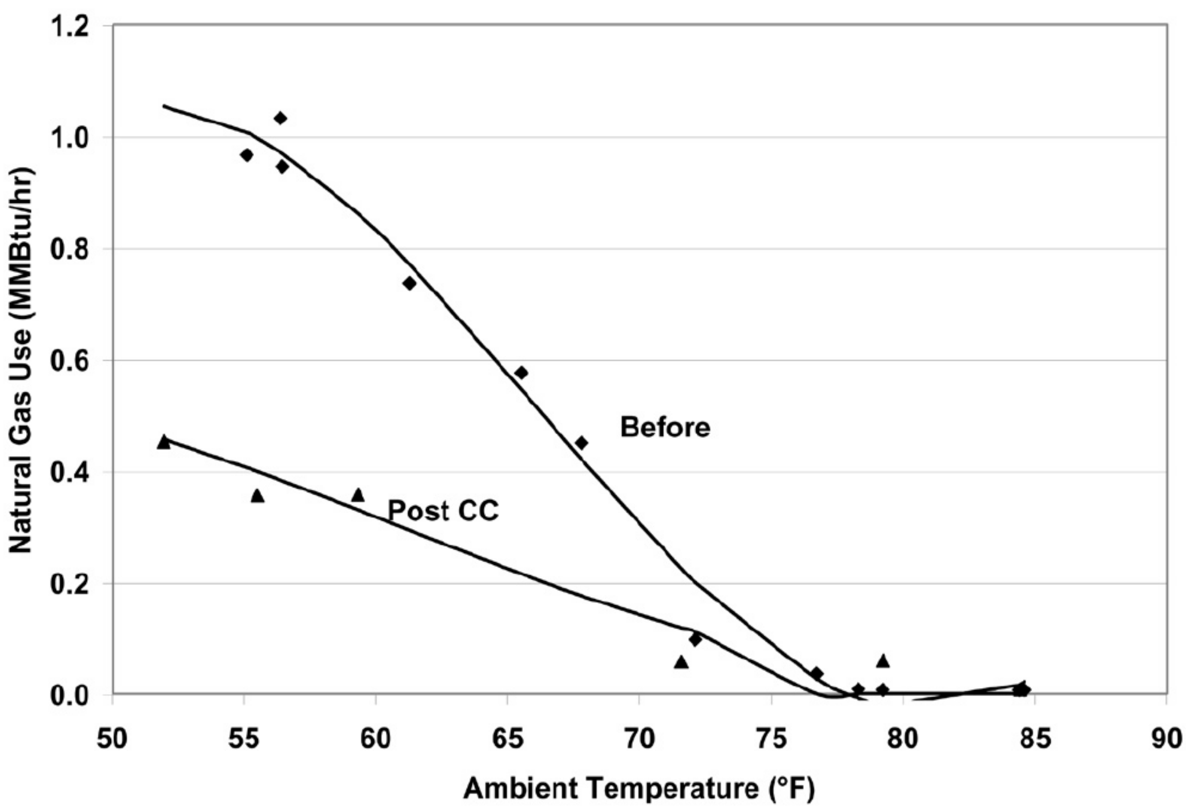

Fig. 5: Figure 4-5. Measured Heating Energy Consumption Versus the Monthly Average Hourly Ambient Temperature

4.3 4.3 Improve Static Pressure Set Point and Schedule

The supply air static pressure is often used to control fan speed and ensure adequate air flow to each zone. If the static pressure set point is lower than required, some zones may experience comfort problems due to lack of air flow. If the static pressure set point is too high, fan power will be excessive. In most existing terminal boxes, proportional controllers are used to maintain the airflow set point. When the static pressure is too high, the actual air flow is higher than its set point. The additional air flow depends on the setting of the control band. Field measurements have found that the excessive air flow can be as high as 20% [Liu et al. 1997b].

Excessive air flow can also occur when terminal box controllers are malfunctioning. For pressure dependent terminal boxes, high static pressure causes significant excessive air flow. Consequently, high static pressure often causes unnecessary heating and cooling energy consumption. A higher than necessary static pressure set point is also the primary reason for noise problems in buildings.

The static pressure set point is often determined under the maximum cooling load condition. The value may be determined by the design engineer using a theoretical calculation or a rule of thumb. The operating staff may increase the value to "eliminate" hot spots. The static pressure set point is often significantly higher than required. Accurately determining the maximum static pressure set point is critical for both thermal comfort and fan energy consumption [Zhu et al. 1998]. The maximum static pressure set point can be identified using the following procedures:

- Determine the maximum static pressure requirement of the terminal box. Set the terminal box to full cooling. Modulate static pressure at the front of the terminal box using a balance damper or VFD. When the terminal box is 95% open, record the static pressure in front of the terminal box. This pressure is considered to be the maximum static pressure required for the terminal box. The measurement should be conducted for each type of terminal box if more than one type of terminal box is used.
- Measure the duct pressure loss. Select the remote terminal box that has the lowest static pressure at the inlet of the terminal box. Measure the static pressure at the entrance of each remote terminal box. Pick the minimum static pressure value measured at a remote terminal box as the terminal box static pressure (pt). If there is more than one type of terminal box on a single AHU, determine the minimum remote box pressure for each box type and use the highest of these values as the terminal box static pressure (pt). Measure the static pressure (ps) at the location of the static pressure sensor. The pressure loss is defined as the difference between the static pressure at the static pressure sensor and the terminal box static pressure. Make sure all dampers are completely open between the static pressure sensor and the terminal box. Other flow blockages must be removed as well. Measure air flow through the AHU .
- Determine the maximum static pressure set point. The following steps should be followed:
  - Determine the airflow ratio (β) defined as the ratio of the measured air flow to the maximum air flow reached at the design condition by this AHU. Many AHUs never reach 100% design air flow, so do not assume the maximum flow is the design flow. It may only be 70% or 80% of the design flow.
  - Calculate the maximum static pressure set point using the equation below based on the measured terminal box static pressure requirement (pt) and the duct pressure loss between the location of the static pressure sensor and the remote terminal box (ps-pt)

$$p_{s,max} = p_t + \frac{1}{\beta^2}\left(p_s - p_t\right)$$

The maximum static pressure determined by equation 4-3 will provide reliable system operation under both peak and partial load conditions. Under partial load conditions, the duct pressure losses are lower due to decreased airflow rate. If the maximum static pressure is used, the terminal box dampers must provide the pressure drop no longer occurring in the duct. This causes higher fan power than necessary and sometimes causes noise problems in the terminal box due to excessive pressure drop. Therefore, the static pressure set point should be decreased when the air flow decreases. This is called static pressure reset.

The static pressure set point, under partial load conditions, depends on a number of parameters such as the zone load distribution and duct layout. If all zones have the same load ratio, the static pressure set point under partial load is proportional to the square of the airflow ratio.

$$p_s = \beta^2 p_{max}$$

If the zone load ratios are different, the static pressure set point should be higher than the set points given by equation 4-4. The accurate determination of the set point is a complex task. Generally, the following method can be used to determine the reset schedule.

- Set the minimum static pressure based on the minimum air flow ratio (determined at the minimum flow setting in equation 4-5), the maximum static pressure value (ps,max) determined from equation 4-3, and the terminal box minimum static pressure requirement (pt). For example if the maximum static pressure, the minimum airflow ratio and the terminal box minimum static pressure are 1.2 in. H2O, 50% and 0.5 in. H2O respectively, the minimum static pressure is 0.43 in. H2O. It is assumed that the terminal box will require at least half of the minimum pressure requirement (pt) under partial air flow.

$$p_{s,min} = 0.5p_t + \beta_{min}^2 \left(p_{s,max} - p_t\right)$$

- Due to the uncertainty of the duct layout and the load diversity factor, it is recommended that the static pressure be reset linearly between ps,min and ps,max as a function of the air flow (cfm):

$$p_s = p_{s,min} + \frac{\dot{V} - \dot{V}_{min}}{\dot{V}_{max} - \dot{V}_{min}} \left(p_{s,max} - p_{s,min}\right)$$

When air flow is not measured, the VFD speed may be used to represent the airflow ratio. For example, if the VFD control command is 50 Hz, the fan speed is approximately 80% of its maximum speed. The air flow can be assumed to be 80% of the design flow. This is only an approximation due to changes in terminal box damper positions.

When modern control systems are installed on both the AHU and terminal boxes, the fan may be directly controlled by the damper positions in the terminal boxes. The fan speed control should maintain at least one selected terminal box at the maximum open position [Hartman 1989]. When all terminal boxes are functioning properly, this method uses the least fan power. However, when a terminal box is malfunctioning, this method may not produce the expected savings. For example, malfunctioning flow stations may force dampers to the full open position. The fan will run at full speed to satisfy the requirement of these malfunctioning sensors. Therefore, this control method should be integrated with a static pressure reset schedule [Wei et al. 2000] to minimize the fan energy. The fan speed is modulated to maintain one or more terminal dampers at full open position. If the static pressure is lower than the reset schedule set point, modulate the fan using the damper position. If the damper position signal modulates the static pressure to the reset schedule set point, use the reset schedule to prevent the static pressure from going higher.

Note

The fan speed control should maintain at least one selected terminal box at the maximum open position [Hartman 1989]. When all terminal boxes are functioning properly, this method uses the least fan power.

EXAMPLE:

AHU-P2, serving the 11th floor of an M. D. Anderson Hospital facility in Houston, Texas, is a dual duct VAV system with design air flow of 19,650 cfm. A VFD is installed on the 40 hp. supply air fan. The static pressure was set at 2.5 in. H2O according to the design specifications in May 1997.

A static pressure reset schedule was developed and implemented during the building commissioning process [Liu et al. 1998a]. Figure 4-6 presents the reset schedules implemented and compares the measured values with the set points. The static pressure is reset according to the fan speed. When the VFD speed is less than 60%, the static pressure is set at 0.5 in. H2O. As the VFD speed increases from 60% to 90%, the static pressure set point increases from 0.5 in. H2O to 0.8 in. H2O. As the VFD speed increases from 90% to 100%, the static pressure set point increases from 0.8 in. H2O to 1.25 in. H2O. The measured static pressure set point closely follows the reset schedule.

When the VFD speed is less than 60%, the static pressure set point reduction is 2.0 in. H2O or 80% of the initial set point. As the VFD speed increases from 60% to 80%, the static pressure set point reduction decreases from 80% to 68%. The VFD speed is rarely higher than 90%. The static pressure reset saves about 68%-75% of the annual fan power.

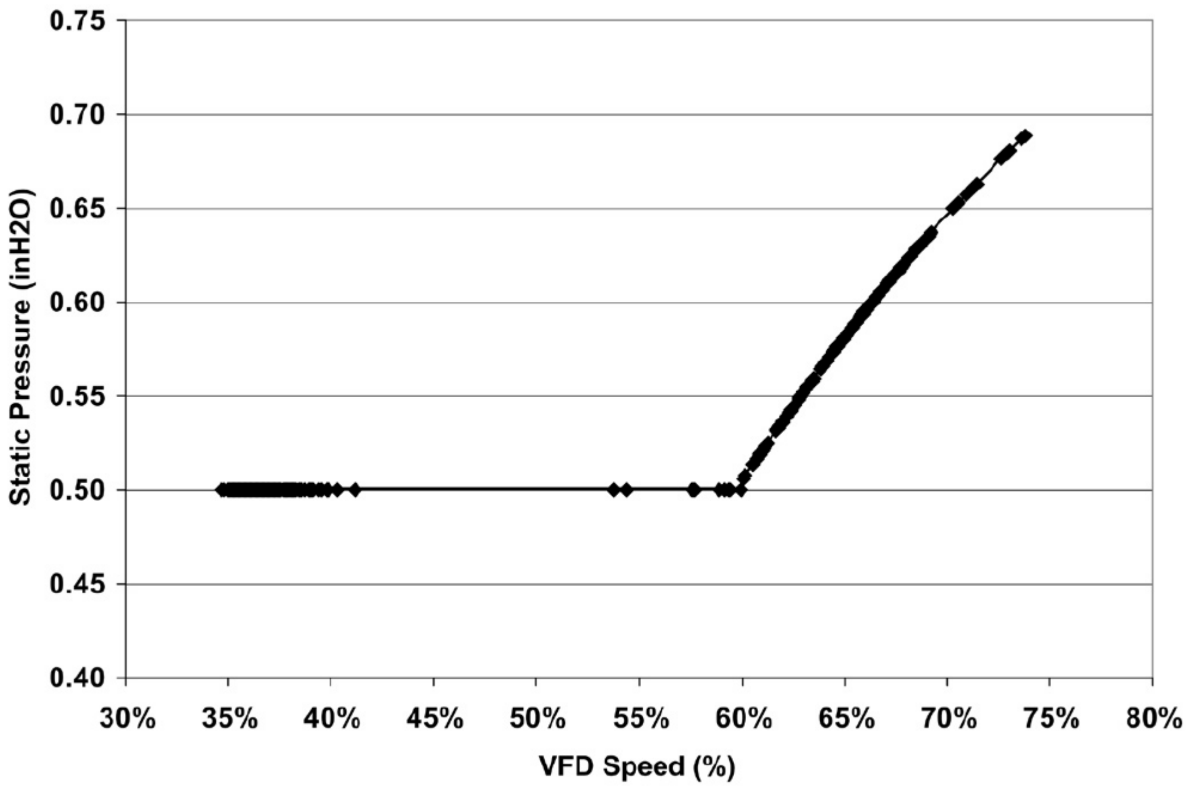

Fig. 6: Figure 4-6. Optimal Static Pressure Reset Schedule and Measured Static Pressure Versus the VFD Speed for AHU-P2 at the CSF Building, M.D. Anderson Cancer Center, Houston, Texas

4.4 4.4 Optimize Supply Air Temperatures

Supply air temperatures, cooling coil discharge air temperature for single duct systems or cold deck and hot deck temperatures for dual duct systems, are the most important operation and control parameters for AHUs. If the cold air supply temperature is too low, the AHU may remove excessive moisture during the summer using mechanical cooling. The terminal boxes must then warm the over-cooled air before sending it to each individual diffuser for a single duct AHU. More hot air is required in dual duct air handlers. The lower air temperature consumes more thermal energy in both systems. If the cold air supply temperature is too high, the building may lose comfort control. The fan must supply more air to the building during the cooling season; therefore fan power will be higher than necessary. The goal of optimal supply air temperature schedules is to minimize combined fan power and thermal energy consumption or cost. Although developing optimal reset schedules requires a comprehensive engineering analysis, improved, near optimal, schedules can be developed based on several simple rules. Guidelines for developing improved supply air temperature reset schedules are provided below for four major types of AHU systems.

For single duct, constant air volume systems, the following guidelines are recommended:

- Maintain the supply air temperature no higher than 57°F if the outside air humidity ratio is higher than 0.009 or the dew point is higher than 55°F. This is required to properly control room humidity level. Both humidity ratio and dewpoint can be determined using dry bulb temperature and relative humiditydata. Most building automation systems can calculate humidity ratio and dewpoint temperature. The psychrometric chart can also be used to determine the humidity ratio and the dew point.

- When outside air humidity ratio is lower than 0.009, the supply air temperature can be reset to a higher temperature using one of the following parameters: outside air temperature; minimum reheat valve position; or return air temperature.
  - The supply air temperature is often linearly reset using outside air temperature. A reset schedule that may be used as a convenient starting point is ts = 65°F if toa < 30°F ts = [55 + 0.333(60 – toa)] °F if 30°F < toa< 60°F ts = 55°F if toa > 60°F If you wish to determine a more aggressive reset schedule, the following procedure may be used. ts = Min (65°F, t1 – ϕ(toa,1 – toa)) when toa < toa,1 ts = td when toa > toa,1 td is the design supply temperature, typically 55°F. As noted, the supply temperature is maintained at this value when outside air temperature is high enough that humidity levels above approximately 0.009 are likely to occur. toa,1 is often selected as 55°F in humid climates and increases to 65°F in relatively dry climates. When outside air temperatures are below toa,1, supply air temperature reset to higher temperatures will not impact room humidity control. t1 is the supply air temperature set point determined by the sensible load at toa,1. The supply temperature begins to increase as toa decreases below toa,1 until it reaches 65°F and is maintained at 65°F for lower outside air temperatures. A field measurement should be performed when the outside air tempera- ture is at toa,1 to determine the optimal supply air temperature t1. Under normal occupancy conditions, increase the supply air temperature gradually until at least one reheat valve is fully closed. This supply air temperature is t1. The same measurement can be performed to determine the optimal supply air temperature (t2) at an outside air temperature (toa,2) at least 10°F below (toa,1). The reset rate ϕ is then determined as:

    $$\phi = \frac{t_2 - t_1}{t_{oa,2} - t_{oa,1}}$$

    Note that ϕ, as defined in equation 4-9, will be negative. t1 is often sign- ificantly higher than 55°F and the control must be properly set to avoid unstable switching between t1 and td when the outside air temperature is near toa,1. When the outside air temperature is higher than toa,1, the supply air temperature is based on the need for dehumidification. When the outside air temperature is lower than toa,1, the supply air temperature is based on the sensible load. Resetting the supply air temperature to a higher value, such as t1, can reduce reheat without humidity control problems.
  - The supply air temperature may be reset using the minimum reheat valve position. The supply air temperature should maintain at least one reheat valve in the closed position. If all reheat valves are open, the supply air temperature should be increased and vice versa. When this method is used, high and low limits should be used to prevent incorrect set points caused by a faulty control valve.
  - The supply air temperature may also be reset using the return air temperature when all room temperatures are controlled and monitored by the central control system. If occupants can change room temperature set points, this method should be combined with the reset schedule defined above. This method can only be used when the outside air temperature is lower than 60°F or another value of toa,1 is determined above to be a suitable starting temperature for increasing the supply temperature.

For single duct VAV systems, the following guidelines are recommended:

- Maintain the air temperature no higher than 57°F if the outside air humidity ratio is higher than 0.009 or the dew point is higher than 55°F. Both humidity ratio and dew point can be determined using dry bulb temperature and relative humidity data. Most building automation systems can calculate humidity ratio and dew point temperature. The psychrometric chart can also be used to determine the humidity ratio and the dew point.
- Maintain the supply air temperature no higher than 57°F if the fan air flow is higher than 70% of the air flow under the maximum load conditions. This is often significantly smaller than 70% of the design air flow. When the air flow is higher than 70%, increased air flow has a significant impact on fan power. For example, resetting the supply air temperature from 55°F to 57°F can potentially increase the air flow by 10%. This will increase fan power from 34% to 51% of the maximum value.

- When the outside air humidity ratio is lower than 0.009 and the air flow is lower than 50%, the supply air temperature can be modulated to maintain total airflow at 50% or lower. If the air flow is lower than 50%, the supply air temperature can be increased. However, the supply air temperature must be lower than a high limit, which can be set to 65°F. When static pressure reset is applied, the air flow ratio can be estimated using the supply air fan speed ratio.

More detailed information on these procedures can be found in "Optimize the Supply Air Temperature Reset Schedule for Single Duct VAV Systems" [Wei et al. 2000a].

For dual duct constant air volume systems, the following guidelines are recommended:

When the mixed air temperature is lower than the cold deck set point, set the hot deck temperature based on zone comfort requirements. Set the cold deck temperature at the mixed air temperature. In theory, the hot deck set point has no impact on thermal energy consumption. However, a higher hot deck temperature may cause higher thermal energy consumption due to hot air leakage in interior zone terminal boxes. Therefore, the hot deck temperature should be set as low as possible provided the room comfort is maintained properly.

- When the mixed air temperature is higher than the hot deck temperature set point or the heating coil is shut off, set the cold deck temperature at the design value (55°F). Resetting the cold deck temperature higher does not reduce cooling energy consumption.
- When the mixed air temperature is between the cold and hot deck temperature set points, the reset should narrow the difference between the cold and hot deck temperatures. The closer the cold and hot deck temperatures, the lower the thermal energy consumption. However, setting the hot deck too low or the cold deck too high can cause building comfort problems. Therefore, the following suggestions are given:
  - Reset the cold deck temperature using the same procedure as used for the constant air volume single duct system: ts = Min (65°F, t1 – ϕ(toa,1 – toa)) when toa < toa,1 ts = td when toa > toa,1 provided the mixed air temperature, tma, is greater than ts . t1 is the supply air temperature set point determined by sensible load when the outside air temperature is toa,1. toa,1 is often selected as 55°F for humid climates and increases to 65°F for relatively dry climates. The field measurement that determines t1 is slightly different from that for the single duct system. Under normal occupancy conditions, increase cold deck temperature gradually until at least one hot air damper is fully closed. The cold air temperature is now at t1. The same measurement can be performed to determine the optimal supply air temperature (t2) at another outside air temperature (toa,2) at least 10°F below (toa,1). The reset rate is then determined as: It is recommended that the discussion for the single duct system be read before implementing this schedule for a dual duct system. It contains additional detail that may be helpful.
  - The cold deck supply air temperature may be reset using the maximum cold air damper position of the terminal boxes. If the maximum cold air damper position is less than 100% open, the cold deck temperature should be increased and vice versa. The cold air temperature should be limited to less than 65°F. This method can only be used when the outside air temperature is lower than t1 described above.
  - Reset the hot deck temperature based on outside air temperature. The hot deck temperature should not be higher than 75°F when the outside air temperature is 70°F or higher. The supply air temperature should be determined through testing under typical local winter conditions. Under typical local winter conditions, adjust the hot deck temperature until the supply air temperature of one zone approaches within 2°F of the hot air temperature (th,max). The hot deck temperature should be reset linearly between 75°F and th,max as the outside air temperature decreases from 70°F to typical local winter conditions.
- For dual duct variable air volume systems, the cold and hot deck resets should consider both thermal and fan power. The optimal temperature reset schedules should minimize total air flow when the building cooling load or heating load requires more than the minimum airflow ratio. When minimum air flow is reached under low load conditions, hot air mixes with cold air to satisfy the minimum airflow requirement. To minimize thermal energy consumption, the difference between the cold and hot deck temperatures should be minimized.

The following guidelines are recommended for dual duct VAV systems:

- When the outside air temperature is higher than approximately 70°F, set the cold deck temperature at the design value (55°F) and shut off the hot deck control valve. Since the building has significant cooling load, this cold deck temperature set point will decrease the total air flow and save fan power.

- When the outside air temperature is lower than 70°F but higher than 55°F, set the cold deck temperature at 55°F and set the hot deck temperature in a range of 75°F to 80°F.
- When the outside air temperature is lower than 55°F, reset the cold and hot deck temperature to keep at least one hot damper and one cold damper fully open. If the damper positions are not available, the reset schedule for the dual duct constant air volume system can be used.

More information can be found in "The Maximum Potential Energy Savings from Optimizing Cold and Hot Deck Reset Schedules for Dual Duct VAV Systems [Liu and Claridge 1999], "Impacts of Optimized Cold and Hot Deck Reset Schedule On Dual Duct VAV Systems-Theory and Model Simulation" [Liu and Claridge, 1998], "Impacts of Optimized Cold and Hot Deck Reset Schedule On Dual Duct VAV Systems-Application and Results" [Liu et al. 1998b] and "Reducing Building Energy Cost Using Optimized Operation Strategies for Constant Air Volume Systems" [Liu et al. 1995].

EXAMPLE:

Optimal cold and hot deck reset schedules were implemented in a major engineering education building with 324,400 square feet of gross floor area located on the Texas A&M Campus in College Station, Texas. The building houses classrooms, laboratories, computer facilities and offices. There are also clean rooms for solid state electronics studies. The building is open 24 hours per day and all AHUs operate 24 hours daily to satisfy fume hoods, late-night studying, research activities and computer facility operations. There are 12 dual-duct variable air volume systems, each with a single supply air fan (12-40 hp.) installed in the basement to serve about 90% of the total building floor area. These 12 AHUs are spaced uniformly around the exterior wall. Each AHU has two risers from the basement to the third floor and serves approximately the same amount of area on each floor.

The building has a total of 384 terminal boxes. The zone load varies significantly from zone to zone due to occupancy, usage and exterior envelope load. Some of the terminal boxes serve only interior space. The total maximum air flow was determined to be 240,789 cfm, or 1.00 cfm/ft2, for the net usable floor area. The minimum airflow ratio varied from 0.3 to 0.7 with an average of 0.4.

The building used a constant cold deck temperature set point, even though it varied from 52°F to 55°F from one AHU to another. The hot deck set point varied from 110°F to 80°F as the outside air temperature increased from 40°F to 65°F.

note

When the ambient temperature is higher than 70°F, the set point of each hot deck will force the hot water valve closed and the hot deck temperature will be at the mixed air temperature. Generally speaking, the building needs cooling when the outside air temperature is higher than 60°F. Therefore, the hot deck temperature at 70°F will not cause cold complaints.

The improved cold and hot reset schedules were determined using a calibrated simulation model [Liu et al. 1998b]. The improved cold deck temperature varies from 60°F to 54°F as the ambient temperature increases from 55°F to 90°F. The set point of the hot deck varies from 90°F to 70°F as the ambient temperature increases from 55°F to 70°F. When the ambient temperature is higher than 70°F, the set point of each hot deck will force the hot water valve closed and the hot deck temperature will be at the mixed air temperature.

Generally speaking, the building needs cooling when the outside air temperature is higher than 60°F. Therefore, the hot deck temperature at 70°F will not cause cold complaints. When a cold complaint occurs, it often indicates a malfunctioning terminal box, such as a cold air damper stuck open. The hot air temperature set point is often determined by a single zone, such as a corner office with many windows. Other CCSM measures were also implemented in this building, but the reset schedules had the greatest impact on energy use.

Figure 4-7 compares the measured daily average chilled water energy consumption. Before the implementation of the improved reset schedules, the measured daily average chilled water consumption (per hour) varied from 2.4 to 7.5 MMBtu/hr. After implementation of the improved reset schedule, the measured hourly daily average chilled water energy consumption varied from 0.9 to 7.5 MMBtu/hr. Simultaneous heating and cooling has been reduced significantly when the daily average temperature is lower than 75°F.

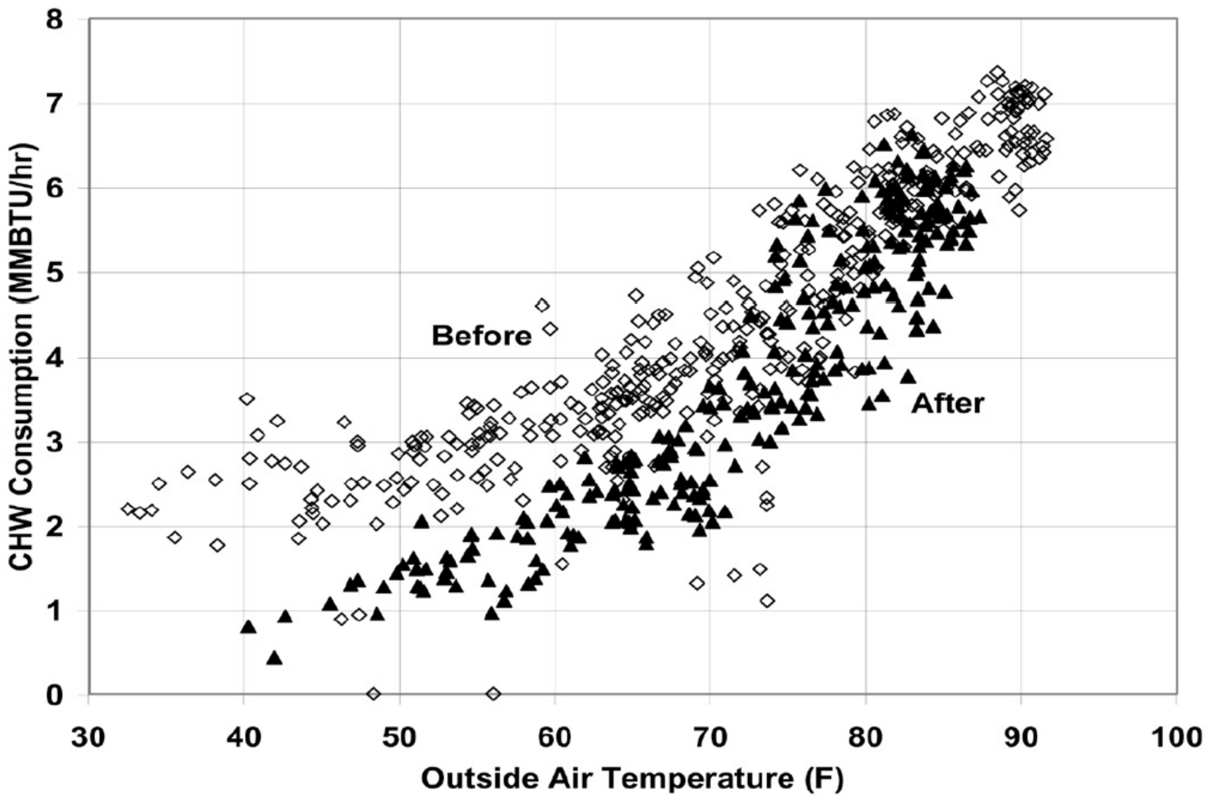

Fig. 7: Figure 4-7. Comparison of Measured Daily Average Chilled Water Consumption Before and After Implementation of the Optimal Hot and Cold Deck Temperature Reset Schedules

4.5 4.5 Improve Economizer Operation and Control

An economizer is designed to eliminate mechanical cooling when the outside air temperature is lower than the supply air temperature set point and decrease mechanical cooling when the outside air temperature is between the cold deck temperature and a high temperature limit or return air conditions, typically less than 70°F. An economizer should control the supply-air temperature by modulating the o/a damper when the o/a temperature is lower than supply-air temperature set point. However, economizer control is often implemented to maintain mixed air temperature at 55°F. This control algorithm is far from optimum. It may, in fact, actually increase the building energy consumption. Economizer operation can be improved using the following steps:

- Integrate economizer control with optimal cold deck temperature reset. It is tempting to ignore cold deck reset when the economizer is operating, because the cooling is free. However, cold deck reset normally saves significant heating.
- For a draw-through AHU, set the mixed air temperature 1°F lower than the cold deck temperature set point. For a blow-through unit, set the mixed air temperature at least 2°F lower than the supply air temperature set point. This will eliminate chilled water valve hunting and unnecessary mechanical cooling.

- For a dual duct AHU, the economizer should be disabled if the hot air flow is higher than the cold air flow because the heating energy penalty is then typically higher than cooling energy savings.
- Set the economizer operating range as wide as possible. For dry climates, the economizer should be activated when the outside air temperature is between 30°F and 75°F, between 30°F and 65°F for normal climates and between 30°F and 60°F for humid climates. When proper return and outside air mixing can be achieved, the economizer can be activated even when the outside air temperature is below 30°F.
- Measure the true mixed air temperature. Most mixing chambers do not achieve complete mixing of the return air and outside air before reaching the cooling coil. It is particularly important that mixed air temperature be measured accurately when an economizer is used. An averaging temperature sensor should be used for the mixed air temperature measurement.
- Use the economizer to directly control supply air temperature when the outside air temperature is lower than the cold deck set point. The chilled water valve should be closed to avoid damper and valve fighting. The mixed air chamber pressure should also be monitored to prevent freezing.

More detailed information on economizer control and optimization can be found in "Economizer Application in Dual-Duct Air Handling Units" [Joo and Liu 2002].

EXAMPLE #1:

Economizers were set to control mixed air temperature at 55°F for a major university, located in central Pennsylvania, with approximately 20 million square feet of floor area. Observations and field measurement indicated that, on average, none of the zones required the 55°F air supplied by the economizer to handle the internal gains when the outdoor temperatures were below 55°F. Thus, it was suggested that the mixed air temperature set point be increased to 60°F during the fall, winter and spring months, which our testing indicated was the lowest required supply air temperature under those conditions. It was recommended that the cold deck temperature be reset to the mixed air temperature. This change eliminated the need for the zone terminal units to reheat the air from 55°F to 60°F during these months, essentially saving 5°F of reheat for all hours that the system operated during these months. The savings were estimated to be 490,000 MMBtu/yr (20 million ft2×7 month/yr × 30 day/month × 24 hr/day × 0.9 cfm/ft2 × 60 min/hr × 5°F × 0.24 Btu/lbm.°F × 0.075 lbm/ft3). At a heating energy cost of $5/MMBtu to the end user, the annual energy cost reduction was estimated to be $2,450,000/yr.

EXAMPLE #2:

The Nursing Building, located in Austin, Texas, is a five story building with 95,000 square feet of floor area. It includes nursing classrooms, lecture halls, workshops, lounges and faculty offices. In 1991, VFDs were installed in two dual-duct AHUs (100 hp. each). Terminal boxes were converted into VAV boxes. As part of the retrofit, economizers were installed as well. During the winter of 1991/1992, the fan power was five times higher than the summer fan power consumption. The HVAC systems were unable to maintain the room temperature set point in many rooms.

In the winter of 1992/1993, the steam pressure was increased and the steam supply pipe was enlarged. The same problems continued with $2,000 of additional energy cost. Prior to performing CCSM, a field inspection and engineering analysis found that the hot deck air flow was three times the cold deck air flow. The recommendation was to disable the economizer. In the winter of 1993-94, two economizers were disabled. The fan power was kept below 40 kW. Steam consumption was decreased. The chilled water consumption increased slightly. The comfort problems disappeared. The annual energy cost savings were measured to be $7,000/yr.

Figures 4-8, 4-9 and 4-10 present the hourly fan power, heating and cooling energy consumption from August 1991 to March 1994. More detailed information can be found in "An Advanced Economizer Controller for Dual Duct Air Handling Systems with a Case Study" [Liu et al. 1997a].

4.6 4.6 Improve Coupled Control AHU Operation

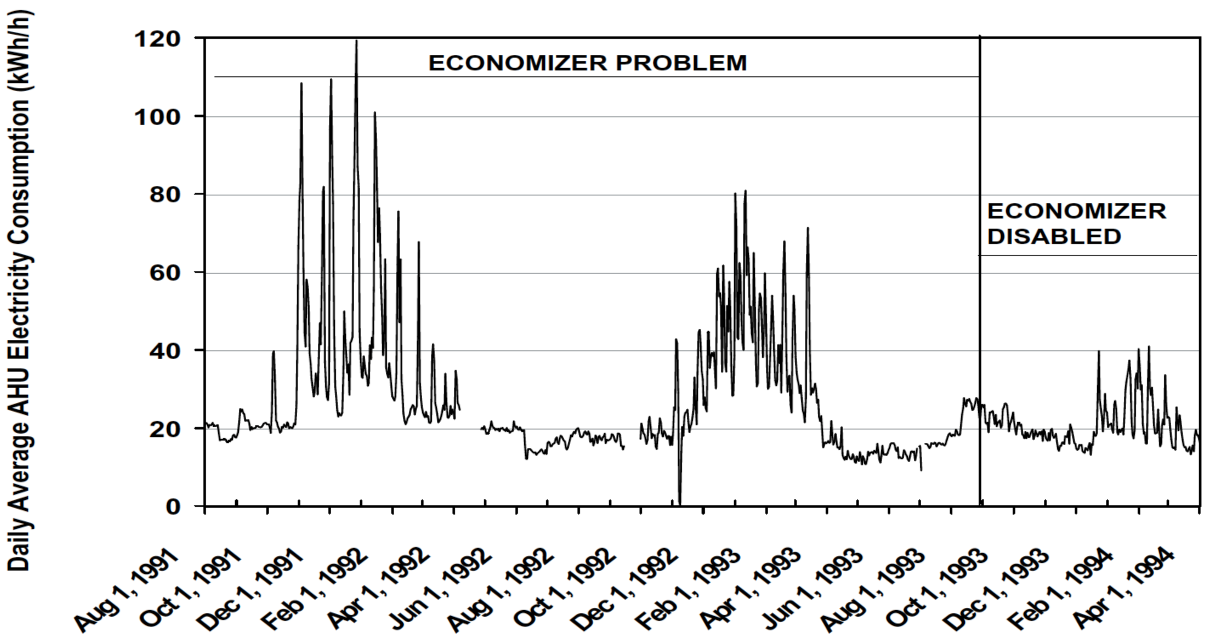

Fig. 8: Figure 4-8. Measured Hourly Fan Power for the Nursing Building in Austin, Texas

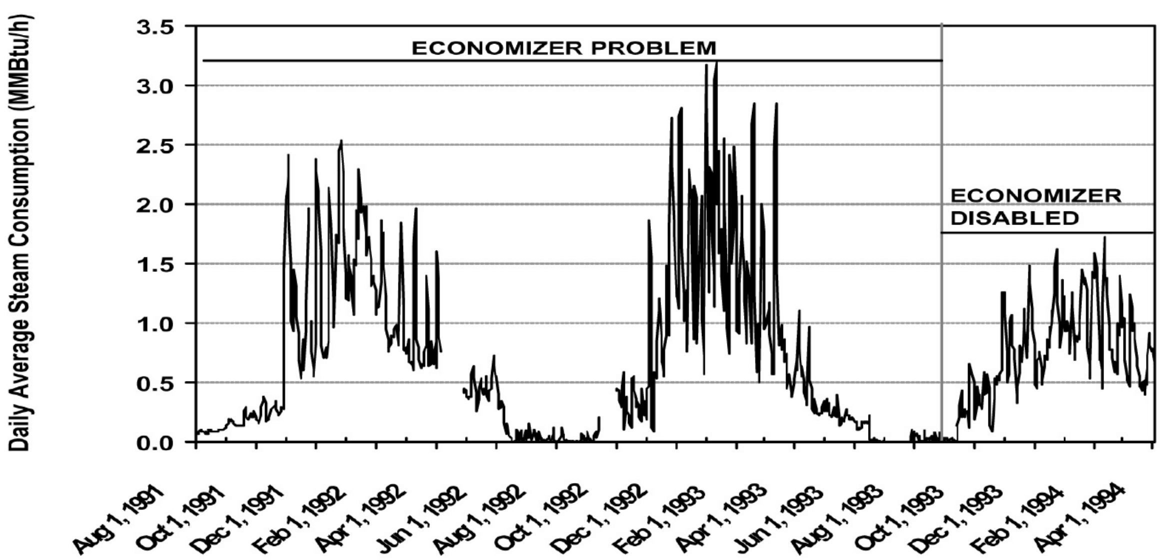

Figure 4-9. Measured Hourly Heating Energy Consumption of the Nursing Building in Austin, Texas

Fig. 9: Figure 4-9. Measured Hourly Heating Energy Consumption of the Nursing Building in Austin, Texas

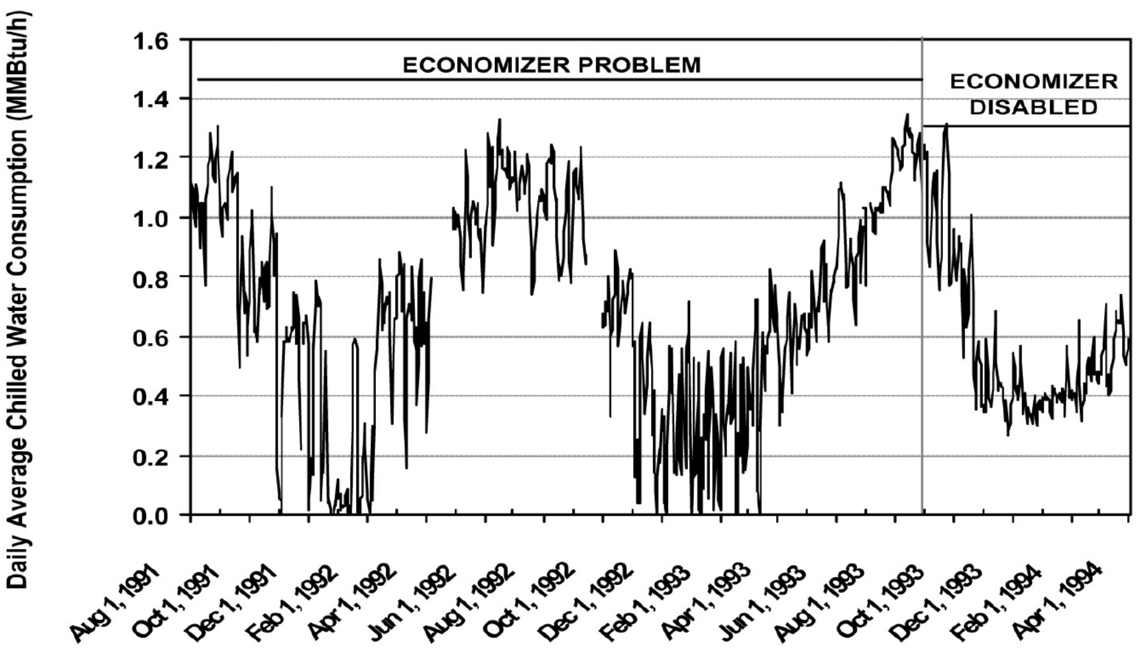

Fig. 10: Figure 4-10. Measured Hourly Cooling Energy Consumption of the Nursing Building in Austin Texas

Note

To control room relative humidity level, the control signals or spring ranges are overlapped. Simultaneous heating and cooling occurs almost all the time.

Coupled control is often used in single zone single duct, constant volume systems. Figure 4-11 presents the schematic diagram of a typical system. Conceptually, this system provides cooling or heating, as needed, to maintain the set point temperature in the zone. It uses simultaneous heating and cooling only when the humidistat indicates that additional cooling, followed by reheat, is needed to provide humidity control. However, the humidistat is often disabled for a number of reasons. To control room relative humidity level, overlap the control signals or spring ranges (See Figure 4-12). Simultaneous heating and cooling occurs almost all the time. Liu and Wang discuss the detailed optimization of this system [2001]. The key elements are listed below:

- Locate the humidistat in an appropriate position. Avoid placing it above the ceiling or near a bathroom. Due to the local humidity environment, the humidistat often calls for full cooling even when the room relative humidity level is low.
- Set the humidity level properly. Most humidistats have three set points: low, medium and high. For commercial building applications, the low level should be avoided except in dry or heating dominated climates. The low setting is equivalent to 30% room relative humidity. It is impossible to reach this value in humid climates even when the cooling valve is full open.
- If the humidistat is disabled, repair is recommended. After the humidistat is repaired, a dead band (2 psi) should be set between actuation of the hot water and chilled water control valves for humid climates. For dry climates, the dead band is not necessary.
- If a humidistat cannot be installed, the overlap should be less than 20% of the valve spring range. The following water side management measures should be implemented:
  - Manually valve off heating water during the summer if heating water is supplied to the building. This will eliminate simultaneous heating and cooling.

- Manually valve off chilled water during the winter if chilled water is supplied to the building. This will eliminate simultaneous heating and cooling.
- Maintain stable differential pressure across the control valves under partial load conditions. Reset loop differential pressure based on chilled water or hot water flow rate (see Chapter 5 "CCSM Measures for Distribution Systems" for details). Excessive differential pressures in the water loops can increase heating and cooling energy consumption by as much as 20%.
- Separate chilled water and hot water valve control signals. This often requires an added control point in the control system. Force the chilled water valve closed during the heating season. Force the hot water valve closed during the cooling season.

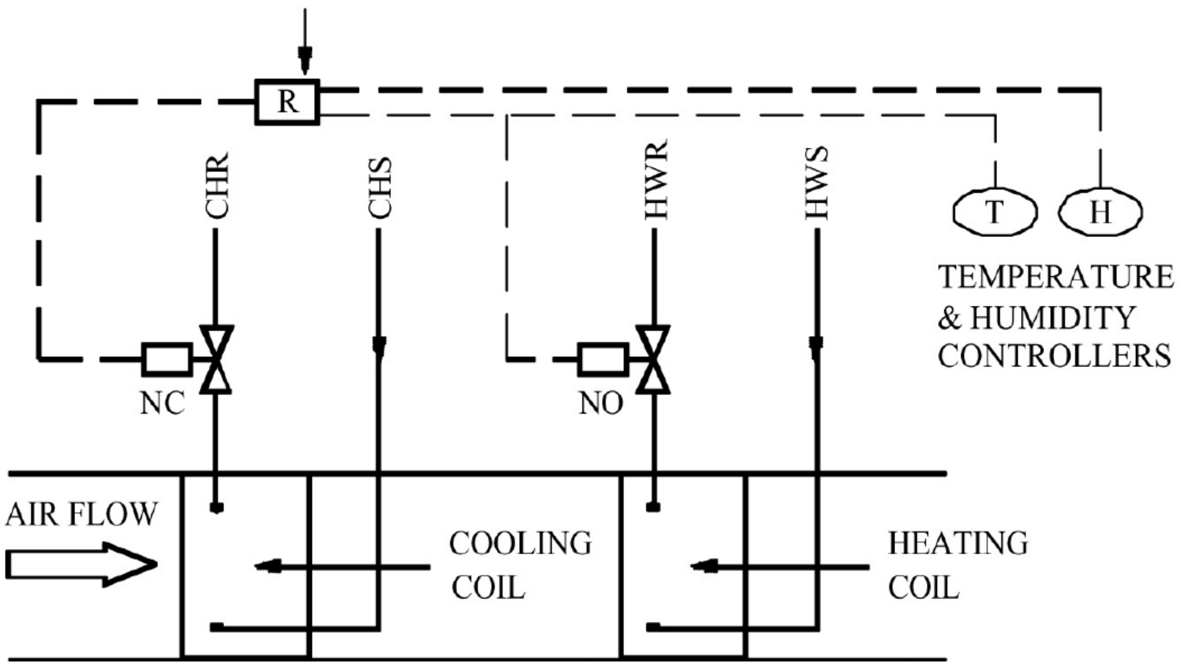

Fig. 11: Figure 4-11. Schematic Diagram of a Coupled Control System

EXAMPLE:

The Memorial Student Center, located in College Station, Texas, is a multi-purpose facility with 368,935 sq.ft. of space on three floors. The space includes cafeterias, banquet facilities, a bookstore, student activity rooms, meeting rooms, hotel rooms, a bowling alley, game rooms, an art gallery and other facilities. The first section of the building was built in the 1960s with additions over the next 30 years. It has 40 AHUs, a majority of which are coupled control units.

The building had comfort problems in a dozen areas. After solving these problems through air balance and other measures, the following measures were implemented in the coupled control AHUs:

- Chilled water loop differential pressure was reset based on water flow rate. The chilled water loop differential pressure was decreased from 30 psi to a range of 6.5 psi to 15 psi.
- Hot water temperature was reset from 180°F to 140°F since a VFD was not installed on the hot water loop
- Overlaps of the spring ranges were adjusted and calibrated to 20%
- High selector pneumatic thermostat settings were enabled for several multi-zone reheat AHUs

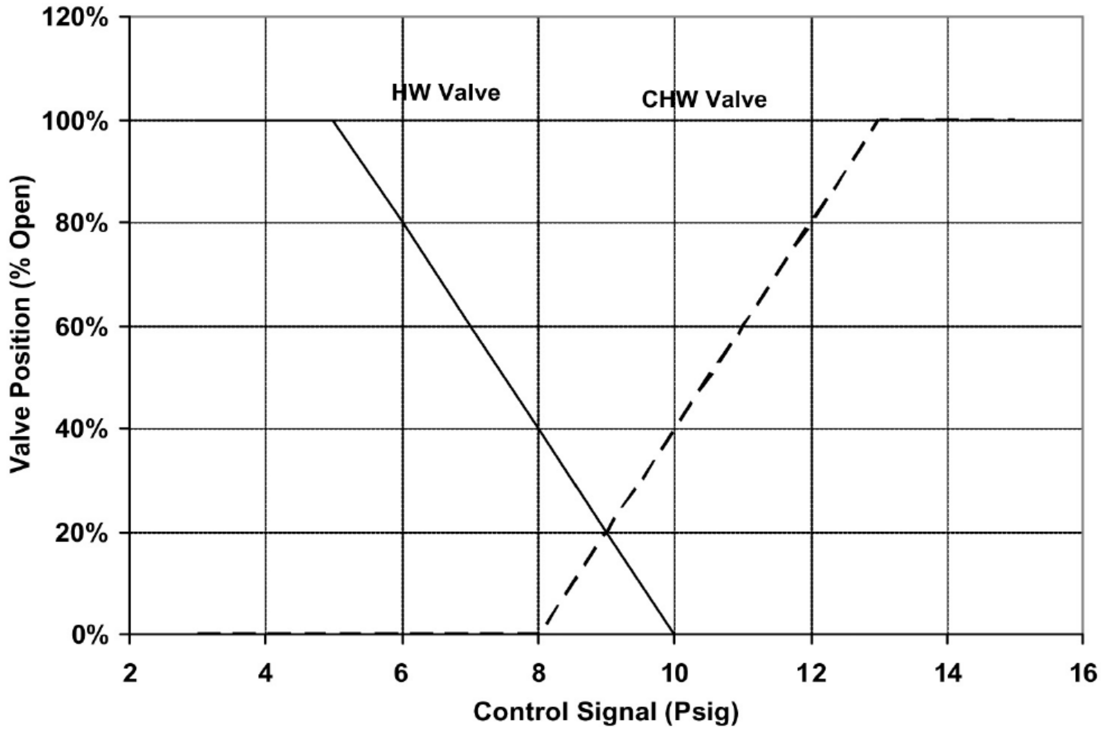

Fig. 12: Figure 4-12. Typical Valve Overlap for a Coupled Control System

Figure 4-13 compares the measured hourly chilled water energy consumption before and after commissioning. The cooling energy consumption is plotted as a function of ambient temperature. The consumption for approximately four months prior to implementation of the CCSM measures is shown in the open rectangles while the consumption after implementation of the CCSM measures is shown in the open triangles. Cooling consumption was reduced by approximately 32% at the same time that comfort in the building was substantially improved. Figure 4-14 is a similar plot of heating consumption and shows a 27% reduction in heating after the CCSM measures were implemented.

4.7 4.7 Valve Off Hot Air flow for Dual Duct AHUs During Summer

During the summer, most commercial buildings do not need heating. Theoretically, hot air should be zero for dual duct VAV systems. However, hot air leakage through terminal boxes is often significant due to excessive static pressure on the hot air damper. For constant air volume systems, hot air flow is often up to 30% of the total air flow. During summer months, hot air temperatures as high as 140°F have been observed due to hot water leakage through valves [Liu et al. 1998b]. The excessively high hot air temperature often causes hot complaints in some locations. Eliminating this hot air flow can improve building thermal comfort, reduce fan power, cooling consumption and heating consumption [Liu and Claridge 1999]. This measure should be implemented using one of the following methods:

- Use an automatic hot air damper for single fan systems. The procedures below should be followed:
  - Install an automatic hot air damper on the main hot air duct
  - Identify the minimum outside air temperature at which heating is not required. Operating staff can start with 70°F and refine as necessary.

If the outside air temperature is 3°F higher than the minimum value identified above, close the hot air damper. If the outside air temperature is 3°F below the minimum value, open the automatic damper. It is important to set the

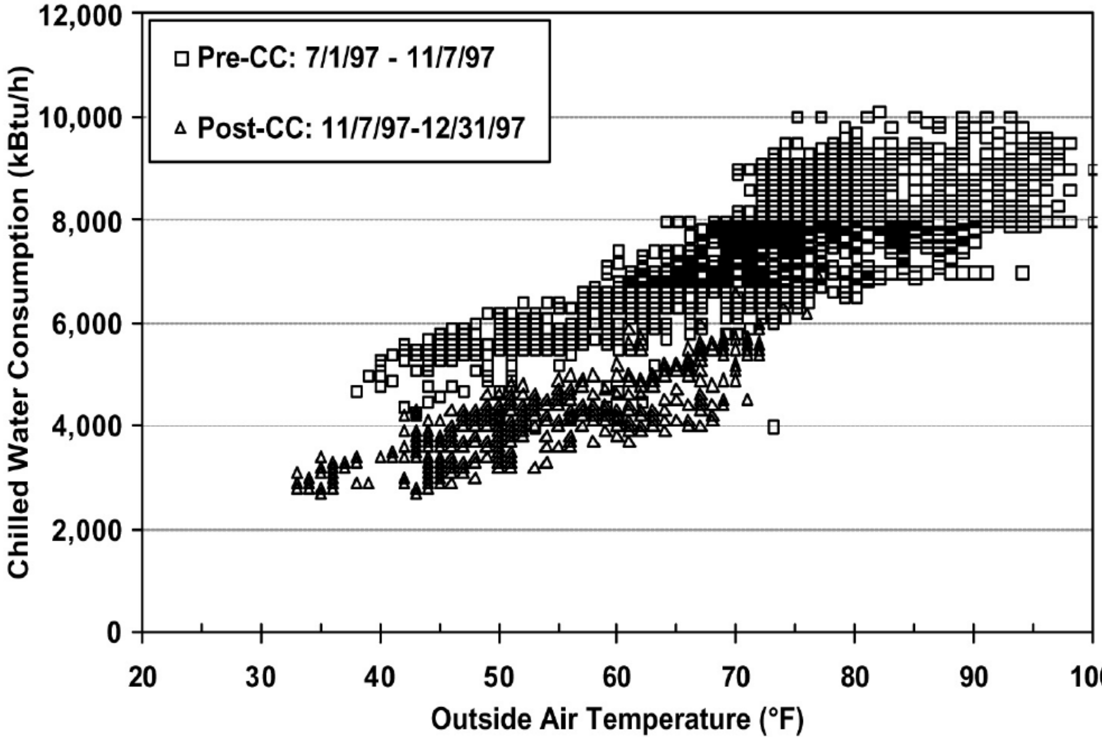

Fig. 13: Figure 4-13. Comparison of Chilled Water Energy Consumption Before and After Implementation of CCSM Measures

damper cycle time properly to maintain control stability. A two-minute cycle time is suggested. On large systems with significant air flows, significant damage can occur if the dampers are closed or opened too quickly. Figure 4-15 illustrates pressure rise vs. time. The static pressure can be as high as 13 in. H2O, which is high enough to damage the ductwork.

- Use existing fire dampers for single fan systems
  - Identify the summer period, during which heating is not needed
  - Manually valve off fire dampers on the hot air duct at the beginning of the summer period
  - Manually open the fire dampers on the hot air duct at the end of the summer period. The procedures should be documented properly so they can be integrated as part of the NFPA bi-yearly fire damper inspection.
  - The fire damper control may also be performed automatically with minimal effort or cost. However, the fire damper must be checked to ensure it functions properly in case of emergency.
- Use a VFD on hot air fans for dual fan systems
  - Identify the minimum outside air temperature when heating is not required. Operating staff can start with 70°F and refine as necessary.
  - If the outside air temperature is 3°F higher than the minimum value identified above, turn off the hot air fan and close the discharge air damper of the hot air fan. If the outside air temperature is 3°F below the minimum value, turn on the hot air fan. It is important to set the damper cycle time properly to maintain control stability. A two-minute cycle time is suggested. A field inspection must be conducted to ensure that the air backflow through the fan does not drive the fan to run backward. If this occurs, either DC braking or a damper with better seals should be used to prevent the fan from turning backward. The backward motion of the fan could damage the VFD during the start-up process.

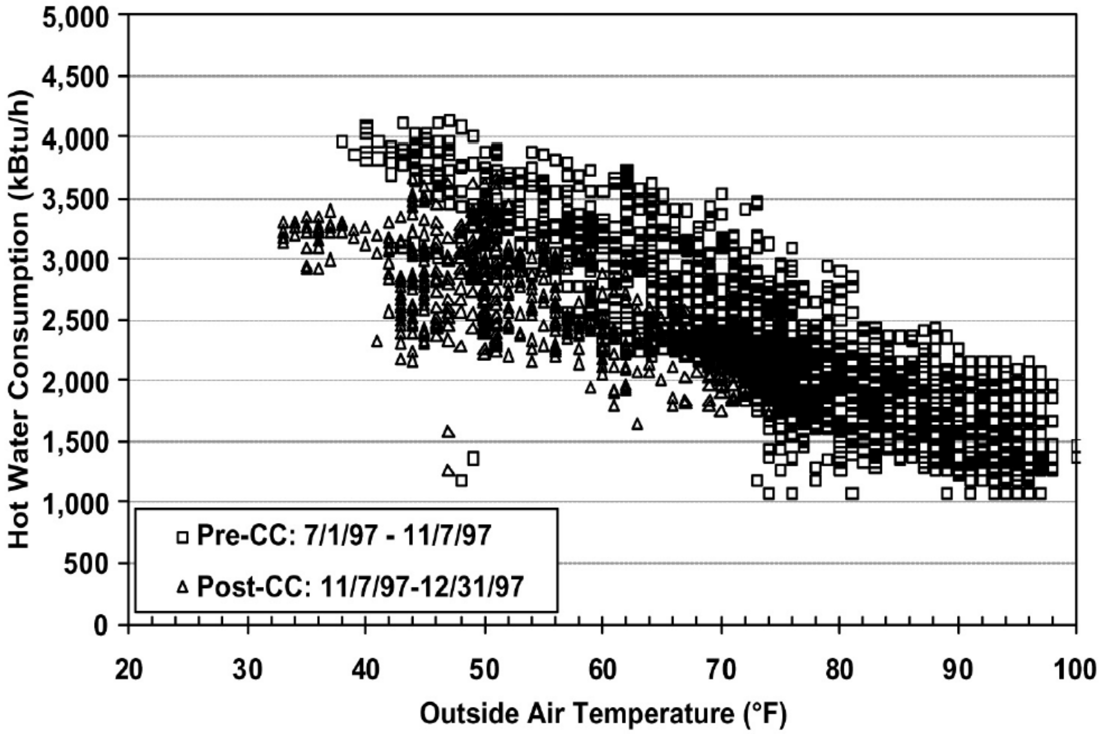

Fig. 14: Figure 4-14. Comparison of Hot Water Energy Consumption Before and After Implementation of CCSM Measures

EXAMPLE:

The James E. Rudder building, in Austin, Texas, is a five-story office building with a total floor area of 170,000 sq.ft. It has two 75 hp. cold air fans on VFDs with a 50 hp. hot air fan without a VFD. Two computer rooms and a print shop are conditioned by separate AHUs. Constant volume terminal boxes are used in the building.

Prior to CCSM, the cold air static pressure was controlled at 2.5 in. H2O while the hot air static pressure varied from 2 in. H2O to 3.5 in. H2O, depending on the building heating load. The cold deck discharge air temperature was controlled at 52°F. The hot deck discharge air temperature varied from 90°F to 120°F as the outside air temperature varied from 80°F to 40°F. The average building temperature was approximately 75°F. Operating staff frequently received hot complaints during the summer months.

The following changes were recommended after a CCSM field visit and engineering analysis. A VFD was installed on the hot air fan. During the summer months (April 1 to October 3), the hot air fan and the hot air damper were shut off. The cold air fan maintained the cold air static pressure at 1.2 in. H2O. The cold deck temperature was maintained in the range from 55°F to 58°F.

During the winter months (November 1 to March 31), the hot air damper was open and the hot air fan was turned on to maintain a static pressure of 0.7 in. H2O in the hot air duct. The cold air static pressure was reset to 1.0 in. H2O. The cold deck discharge air temperature was reset in the range from 58°F to 60°F. The hot deck discharge air temperature was reset from 75°F to 95°F as the outside air temperature varied from 60°F to 30°F.

After implementing the control schedules described, hot and cold complaints were significantly reduced. The AHU system maintained room temperature set points well; they vary from 70°F to 76°F, depending on occupants' preferences. The average building temperature decreased from 75°F to 73°F. The building relative humidity level was maintained between 50% and 58% during summer months.

Figures 4-16, 4-17 and 4-18 present the measured energy use for fans, heating and cooling, respectively. The pre-

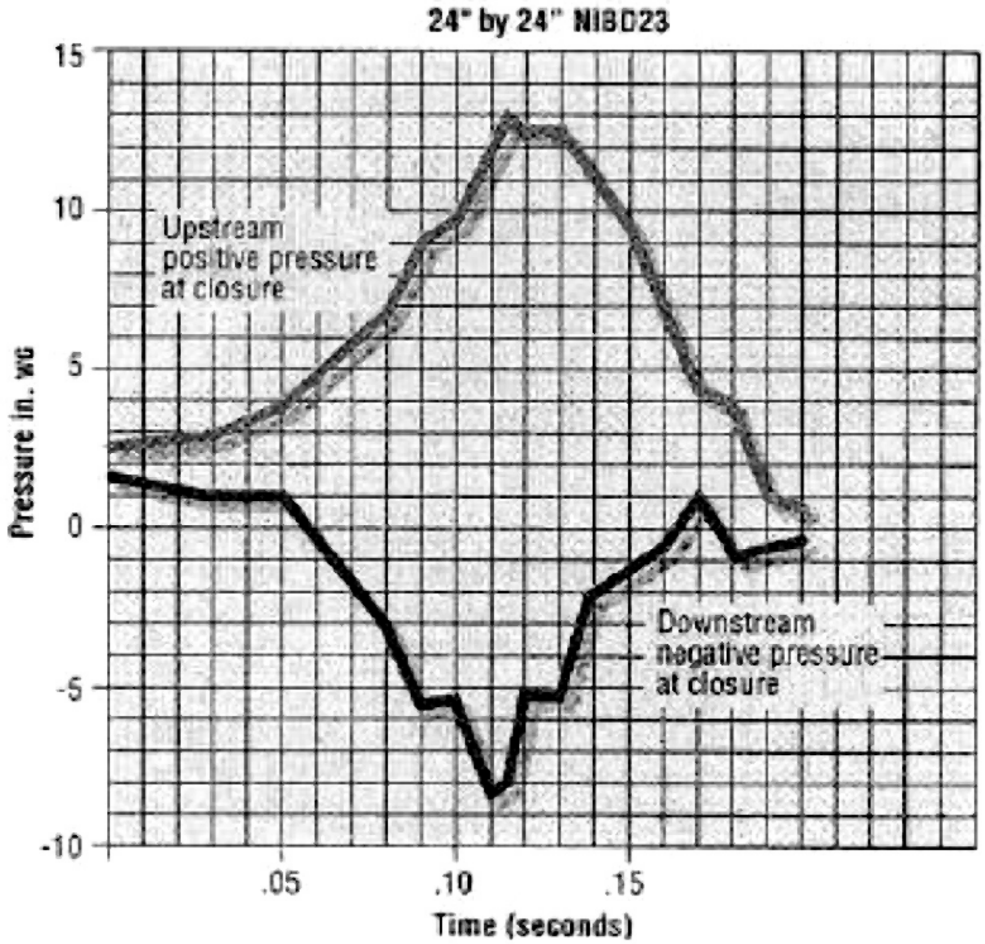

Fig. 15: Figure 4-15. Static Pressure Versus Time Upstream and Downstream of a Fire Damper When Suddenly Closed

conversion period was from June 15, 1995, to June 14, 1996. The post-conversion period was from June 15, 1996, to June 14, 1997.

Before implementing CCSM measures, the daily average fan power varied from 90 kW to 125 kW. After implementing CCSM measures, the daily average fan power varied from 20 kW to 85 kW. The average measured fan power savings were 56 kW or 53% of the average pre-conversion fan power.

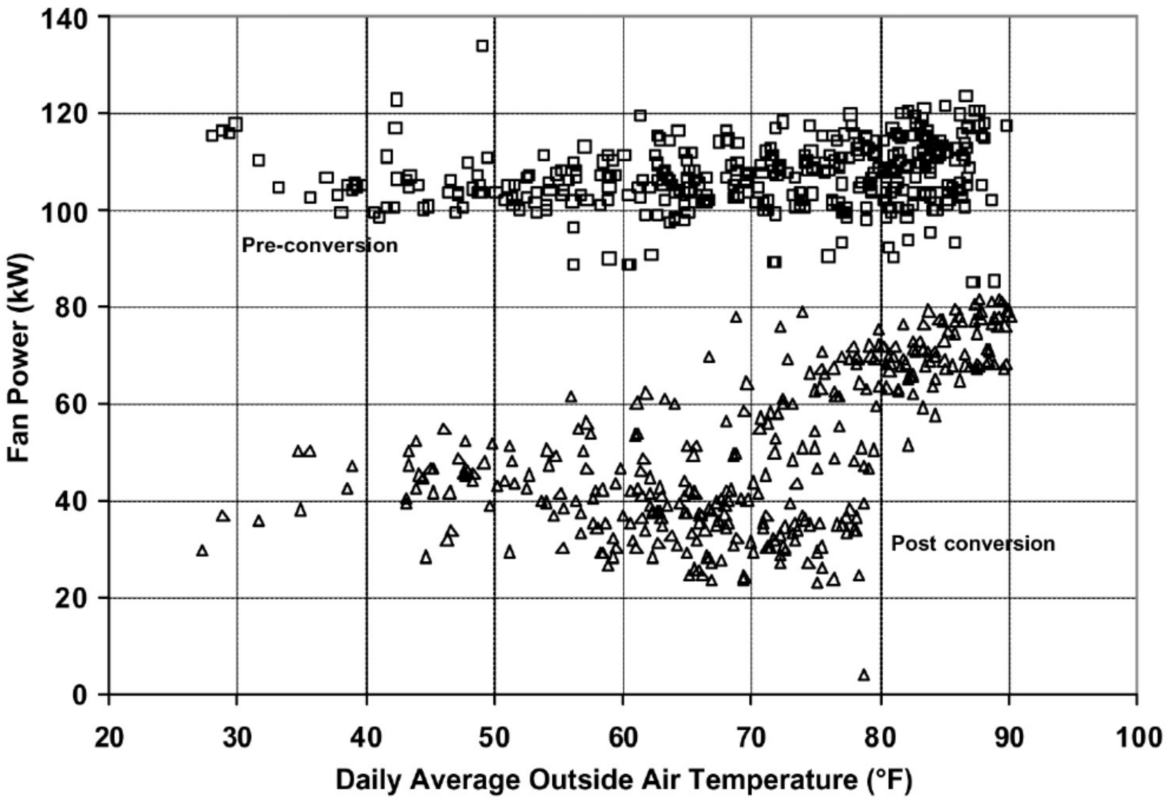

Fig. 16: Figure 4-16. Measured Fan Power Before and After Implementing the CCSM Measures

Figure 4-17 shows the measured daily average chilled water consumption. Before implementing the CCSM measures, the daily average chilled water consumption varied from 1.2 MMBtu/hr to 3.0 MMBtu/hr as the outside air temperature varied from 40°F to 90°F. Since implementing the CCSM measures, the daily average chilled water consumption has been reduced to a range of 0.3 MMBtu/hr to 2.0 MMBtu/hr. The measured daily average chilled water savings have averaged 0.68 MMBtu/hr or 36% of the daily average consumption (1.85 MMBtu/hr).

Figure 4-18 shows a similar plot of the measured daily average hot water consumption. Before implementing the CCSM measures, the measured daily average hot water consumption varied from 0.8 MMBtu/hr to 1.8 MMBtu/hr as the outside air temperature varied from 40°F to 90°F. Since implementing the CCSM measures, the daily average hot water consumption has been reduced to the range from 0.1 MMBtu/hr to 1.5 MMBtu/hr. The measured daily average hot water savings are 0.56 MMBtu/hr or 45% of the pre-CCSM consumption.

Table 4-2 summarizes the measured annual energy savings. The measured cost savings are $62,550/yr which includes $19,200/yr for chilled water, $18,670/yr for hot water and $24,680 for electricity.

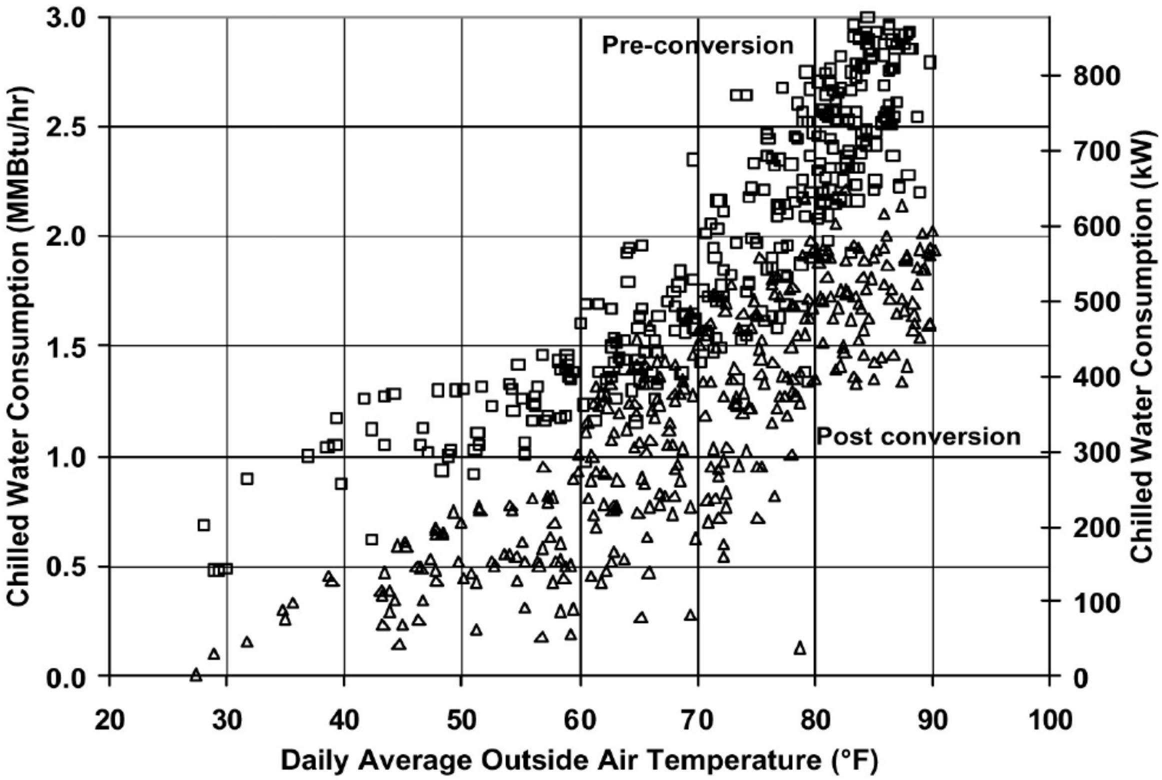

Fig. 17: Figure 4-17. Measured Chilled Water Consumption Versus the Daily Average Outside Air Temperature

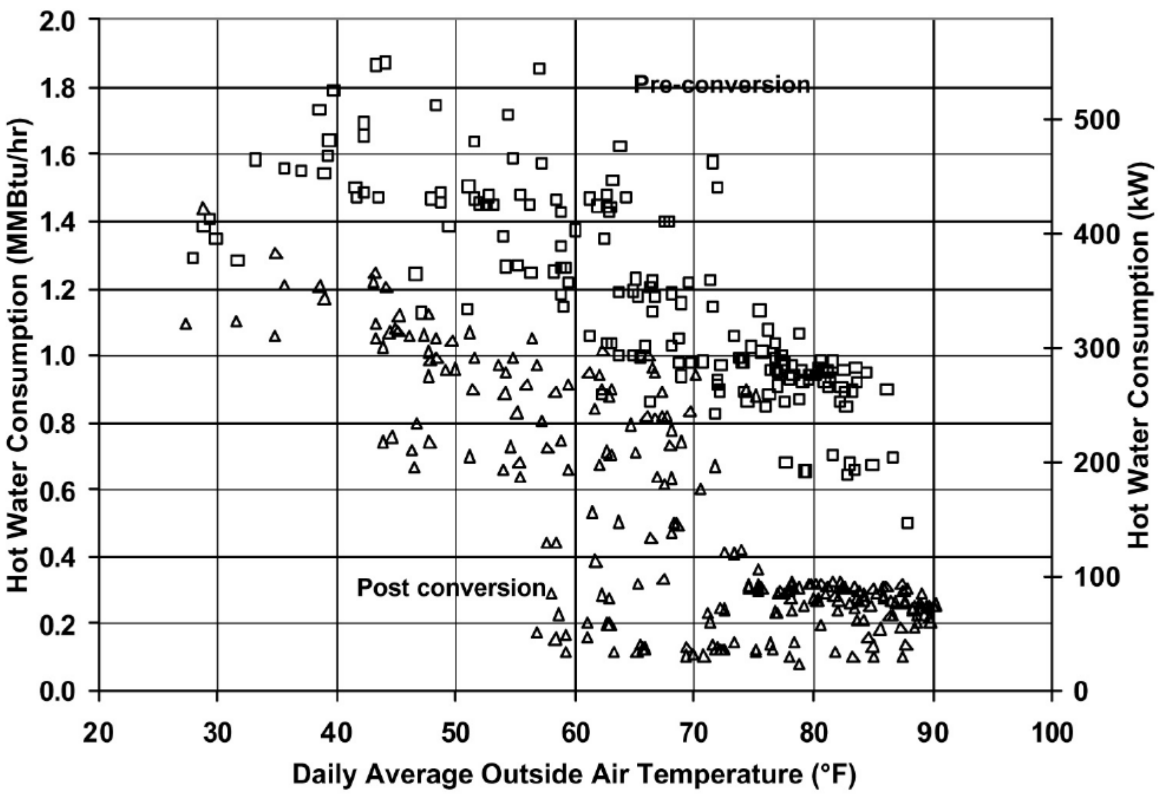

Fig. 18: Figure 4-18. Measured Hot Water Consumption Versus the Daily Average Outside Air Temperature

Table 2: Table 4-2. Summary of Measured Annual Energy Savings

Item	MMBtu	MWh	Demand(kW/Mon)	Cost($/yr)	Percentage(%)
Chilled Water	5,910	1,730	–	$19,200	36%
Hot Water	4,862	1,430	–	$18,670	45%
Fan Power	1,694	497	60	$24,680	53%
Total	12,466	3,657	60	$62,550	41%

Note: Energy prices for electricity: $0.03470/kWh, $10.32/kW; chilled water price: $3.25/MMBtu; and hot water price: $3.84/MMBtu

4.8 4.8 Install VFD on Constant Air Volume Systems

The building heating load and cooling load varies significantly with weather and internal occupancy conditions. In constant air volume systems, a significant amount of energy is consumed unnecessarily due to humidity control requirements. Most of this energy waste can be avoided by simply installing a VFD on the fan without a major retrofit effort. Guidelines for VFD installation are presented separately for dual duct, multi-zone and single duct systems.

- For a single fan dual duct constant air volume system, a VFD and two static pressure sensors should be installed
  - During normal operating hours, the fan speed should maintain the static pressure set point in both ducts. When the building thermal load is less than the design value, both ducts carry air flow. The pressure loss through the main duct is significantly less than the design value. The use of a VFD can avoid having this reduced duct loss show up as additional pressure loss at the terminal boxes, thus saving fan power. More importantly, over-pressurization of the terminal box dampers with accompanying leakage is significantly decreased. Noise problems may also decrease significantly.
  - During weekends or at night, flow can easily be reduced using the VFD. Compared with running the fan at full speed, this saves significant amounts of fan power, heating energy and cooling energy. More information can be found in “Variable Speed Drive Application in Dual Duct Constant Air Volume Systems” [Joo et al. 2002].
- For multi-zone systems, the VFD can be controlled using zone dampers
  - During the cooling season, the VFD should maintain at least one zone cooling damper at 95% open or at another chosen value
  - During the heating season, the VFD should maintain at least one zone heating damper at 95% open or at another chosen value
  - A minimum fan speed should be used to prevent air circulation problems in the zones during the swing seasons. Generally speaking, the minimum fan speed should be no less than about 50%, but may vary depending on the type of diffusers.
  - The installation of the VFD combined with the control recommended achieves true VAV operation for multi-zone systems. Ideally, the multi-zone system with a VFD supplies cooling to at least one zone during the summer and heating to at least one zone during the winter. The airflow to each zone changes proportionally with the zone load assuming constant supply air temperature.
- For a single duct constant air volume system, the VFD should be installed if nighttime shut down cannot be implemented. The VFD can be used to reduce flow at night and on weekends.
- For a single zone constant air volume system, VFD installation may be feasible if the system operates for at least 5000 hours per year

EXAMPLE:

A single-fan, dual-duct constant air volume system serves a four-story building with a gross floor area of 68,000 sq.ft. The unit was installed in the 1960s in the attic. The initial design airflow rate was 57,000 cfm supplied with a 100 hp. fan. The motor was later downsized to 60 hp. to reduce the noise level. The total airflow rate is now 48,000 cfm.

A VFD was installed and the supply fan was operated at 80% of full speed for eight days from February 11-19, 2001. The system was then controlled to maintain a constant static pressure set point (0.7 in. H2O) from February 19-27, 2001.

Figure 4-19 presents the measured hourly total air flow for the constant fan speed and the constant static pressure control modes. The total air flow decreased from 48,000 cfm to 41,000 cfm when the operation was switched from constant speed to constant static pressure control. This airflow reduction indicates that the terminal boxes are actually pressure dependent. However, the total airflow reduction did not affect thermal comfort.

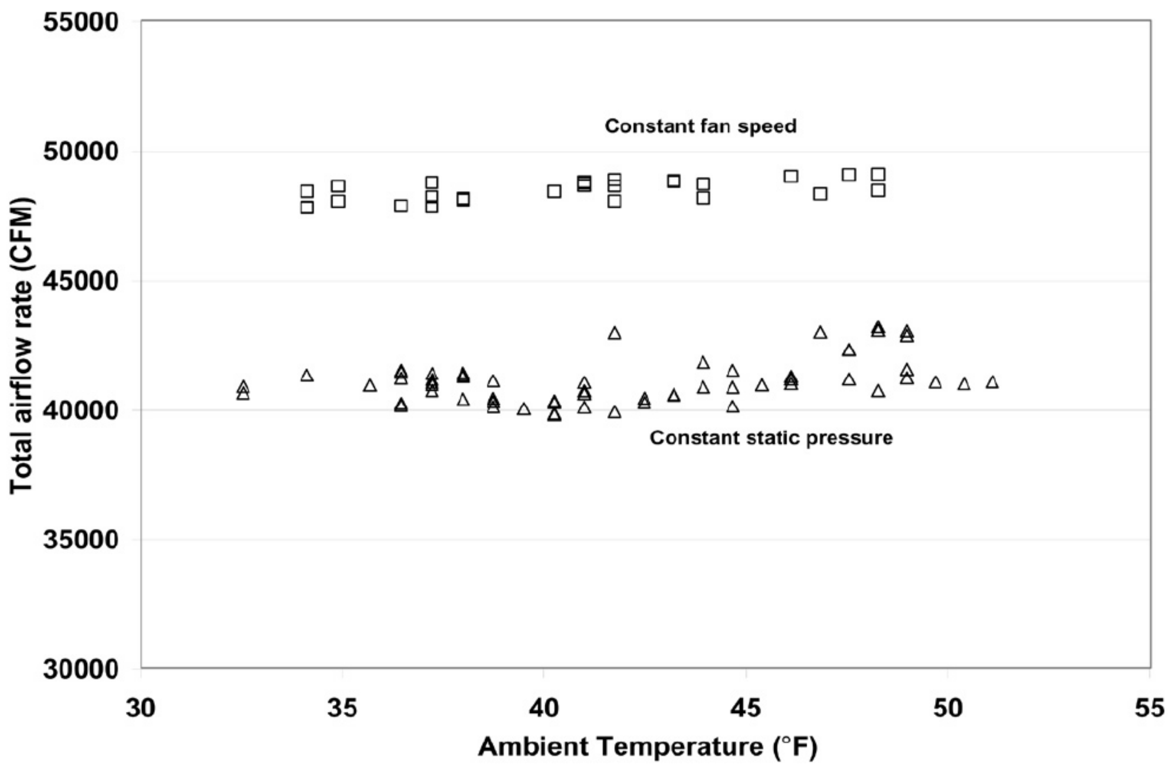

Fig. 19: Figure 4-19. Measured Hourly Total Air Flow for Both the CSFS and the VSFS Operations

Figure 4-20 compares the measured hourly supply fan power for the constant speed and constant static pressure operation. The average fan power was reduced from 35.8 kW for the CSFS operation to 23.1 kW for the VSFS operation. The average fan power savings of 12.7 kW corresponds to a 35% reduction in fan power. More details may be found in Joo et al. [2002].

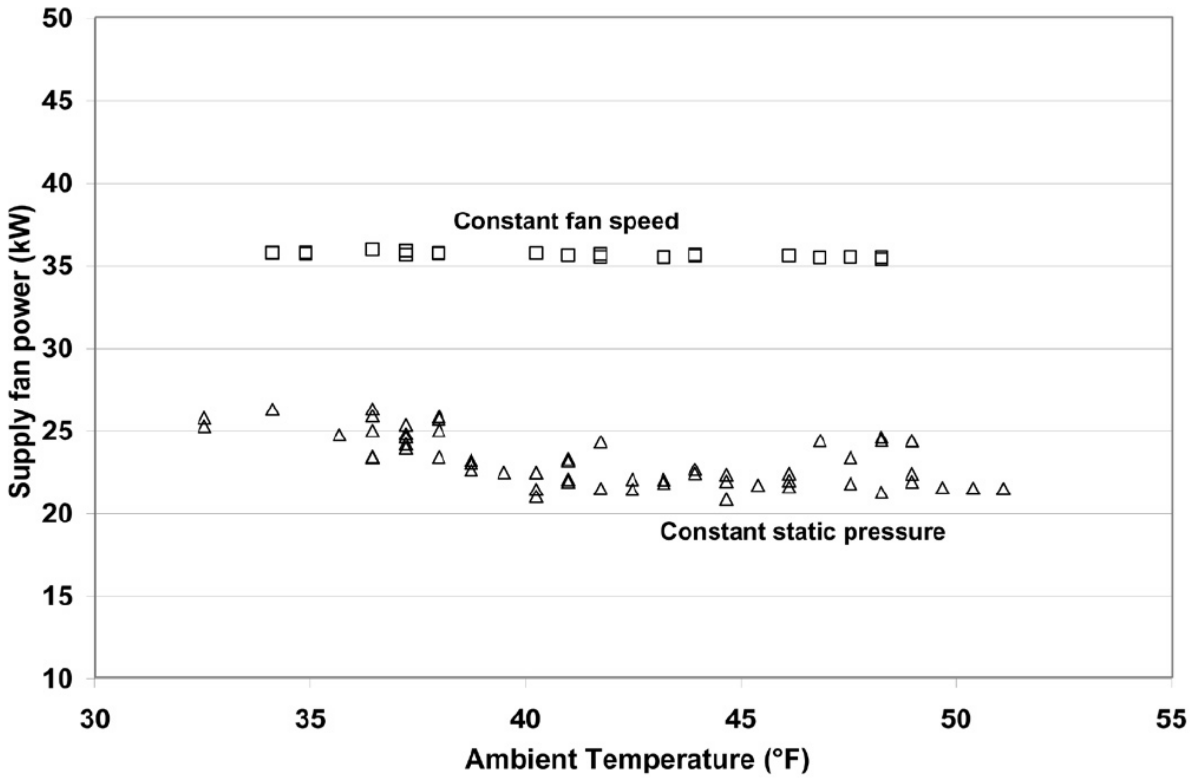

Fig. 20: Figure 4-20. Measured Hourly Supply Air Fan Power for Both the CSFS and VSFS Operations

4.9 4.9 Airflow Control for VAV Systems

Airflow control of VAV systems has been an important design and research subject since the VAV system was introduced. An airflow control method should: (1) ensure sufficient air flow to each space or zone, (2) control outside air intake properly, and (3) maintain a positive building pressure. These goals can be achieved using the variable speed drive volume tracking (VSDVT) method.

Figure 4-21 presents the airflow control schematic of the new VSDVT method. The physical (hard) input signals include the supply and return fan heads, the supply and return air static pressures, the return air temperature, the mixed air temperature, the outside air temperature and the return air or the critical zone CO2 concentration. The output signals include the supply fan VSD speed command, the return fan VSD speed command, the outside air damper command and the return and relief air damper command. The temperature signals are used for airflow control during economizer cycle operation.

The VSDVT has four control loops: the supply fan speed, the return fan speed, the return air damper position and the outside air damper position. For the supply fan speed loop, the controlled variable is the supply air static pressure. The controlled device is the VSD of the supply fan. This control loop maintains the set point of the supply air static pressure by modulating the supply fan VSD speed.

For the supply air control loop, the controlled variable can be the static pressure set point or the maximum air damper position. To minimize the supply fan energy, the optimal supply air static pressure should be developed based on guidelines in section 4.3 of this chapter. The supply air fan speed is modulated to maintain the static pressure set point or the damper position.

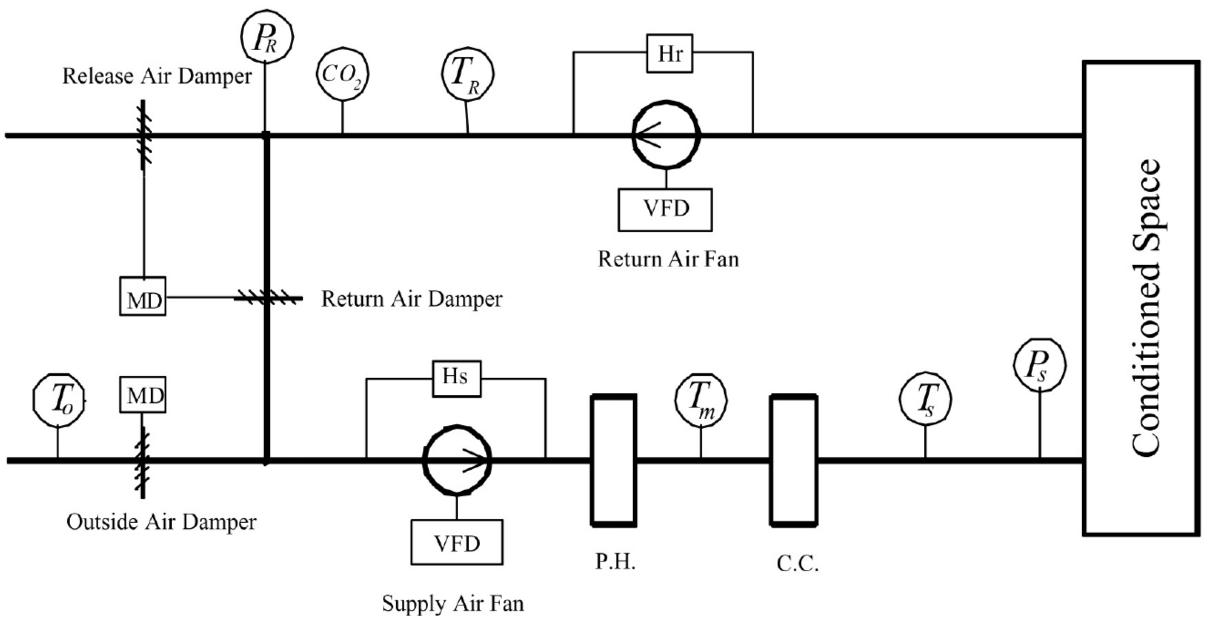

Fig. 21: Figure 4-21. Airflow Control Schematic of the VSDVT Method

For the return fan speed loop, the controlled variable is the return airflow rate. The controlled device is the return air VFD. The controlled loop output is the return fan VFD speed. The return airflow set point equals the difference between the supply air flow and the building exhaust and air exfiltration.

$$\dot{V_r} = \dot{V_s} - \dot{V_{ex}} - \dot{V_{exf}}$$

The supply air flow can be calculated using equation 4-13 based on the supply fan VSD speed (ωs) and the fan head (Hs). The coefficients a1, a2 and a3 are polynomial regression coefficients of fan head against fan air flow at the design fan speed. The fan curve can be obtained from the fan performance cut sheet.

$$\dot{V}_s = \frac{\left(-a_1 \pm \sqrt{a_1^2 - 4a_2\left(a_0 - \frac{H_s}{\omega_s^2}\right)}\right)\omega_s}{2a_2}$$

The sum of the exhaust air flow and exfiltration can be approximated as a constant as long as the building is maintained at a constant value of positive pressurization. This condition depends primarily on the building envelope. The tighter the building envelope, the smaller the value.

The return air flow is calculated using equation 4-14 according to the return fan VSD speed (ωr) and the return fan head (Hr). The coefficients b1, b2 and b3 are the polynomial regression coefficients of fan head against fan air flow at the design fan speed. The fan curve can be obtained from the manufacturer's cut sheet.

$$\dot{V}_r = \frac{\left(-b_1 \pm \sqrt{b_1^2 - 4b_2\left(b_0 - \frac{H_r}{\omega_r^2}\right)}\right)\omega_r}{2a_2}$$

The control loop modulates the return fan speed to maintain the return airflow set point.

For the outside air damper loop, the controlled variables are the return air static pressure and the return air or the critical zone CO2 concentration when the economizer is not activated, or the mixed air temperature when the economizer is activated. The controlled device is the outside air damper. The set point of the CO2 concentration should be predetermined using engineering principles. The set point of the return air static pressure is zero. The controller modulates the outside air damper to maintain both the CO2 and the return air pressure set points only when the return air damper is in its maximum open position. If the return air static pressure is lower than the set point, the controller opens the outside air damper farther regardless of the CO2 concentration. This prevents negative building pressure when the fresh air requirement of the occupants is less than the mechanical exhaust and the exfiltration. When the economizer is activated, the controller modulates the outside air damper to maintain the mixed air temperature set point.

For the return air damper loop, the controlled variable is the return air or the critical zone CO2 concentration when the economizer is not activated, or the mixed air temperature when the economizer is activated. The controlled devices are the return air and the relief air dampers. The relief and the return air dampers are interlinked. When the relief air damper is in the minimum position, the return air damper is in the maximum position. The return air damper loop is activated only when the outside air damper is in the fully open position. The controller decreases the return air damper opening if the CO2 concentration is higher than the set point, or if the mixed air temperature is higher than the cold deck set point during the economizer cycle. Conversely, the controller increases the return air damper opening if the CO2 concentration is higher than the set point or if the mixed air temperature is higher than the cold deck set point.

The VSDVT method reduces the fan energy by using the improved static pressure reset and decoupling the outside and return air dampers. It implements the volumetric tracking using the VSD speeds and the fan heads, and uses CO2 demand control to minimize outside air intake. The method can result in significant building pressurization control improvement and significant energy savings.

Figure 4-22 presents simulated building pressure, outside air flow, and fan power for the typical fan tracking (FT) control method. The damper positions are selected to provide the required minimum outside air flow when the supply fan provides 60% of the design air flow to the building. Outside air flow and building pressure are shown as ratios with respect to design flow and pressure. The outside air, the return and the relief air dampers are fixed at the initial condition regardless of the load conditions. A constant static pressure set point is used. The return air fan speed tracks the supply air fan speed.

The simulation results indicate that the outside air intake decreases as the total air flow decreases when the FT method is used. The AHU provides more than the design value of outside air to the space when the total air flow is higher than 60% of the design air flow. When the total air flow is at the design level, the outside air intake is 210% of the design flow. The building pressure decreases from the design value to negative values when the supply air flow is less than 54%. The typical control method used today is prone to IAQ problems, or high thermal energy consumption and building pressure control problems due to the introduction of inadequate or excessive amounts of outside air.

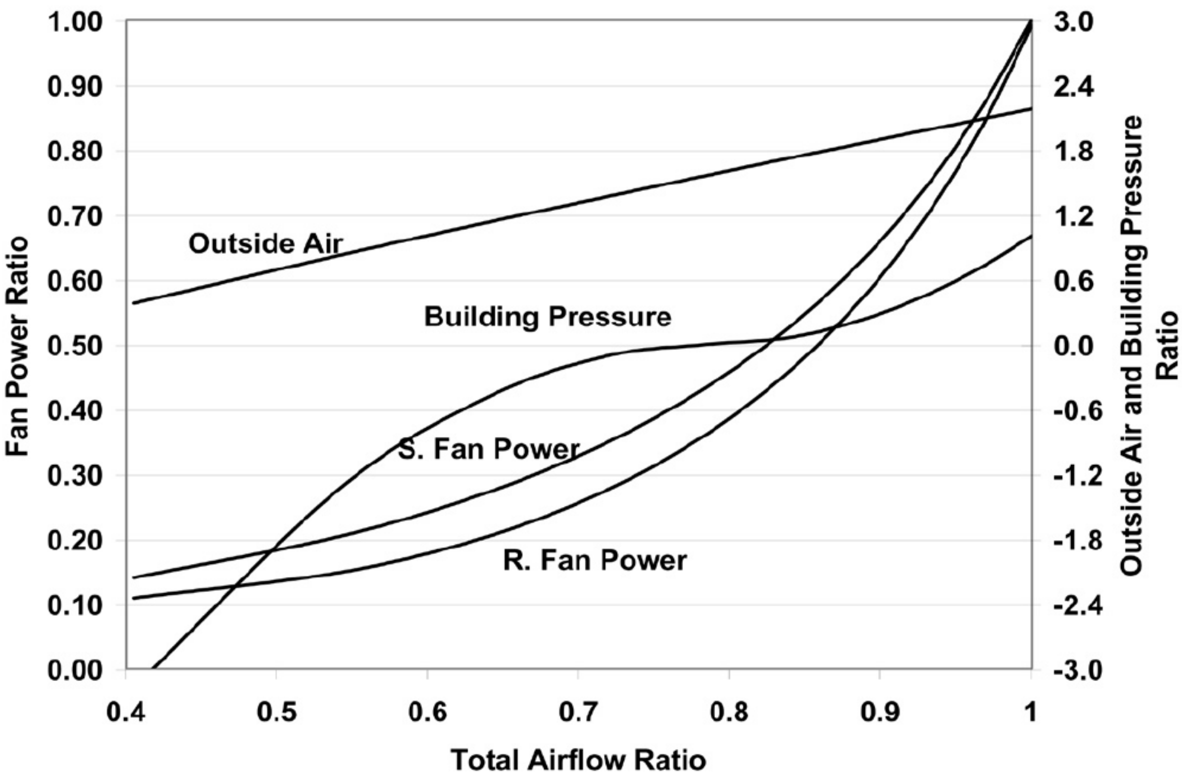

Fig. 22: Figure 4-22. Simulated Building Pressure, Outside Air Intake and Fan Power Using Typical Fan Tracking Control

Figure 4-23 presents the simulated results of the VSDVT method. The VSDVT method maintains constant building pressure and the required outside air intake. The fan power is significantly lower than the typical method used today.

More information can be found in "Variable Speed Drive Volumetric Tracking for Air Flow Control in Variable Air Volume Systems" [Liu 2002].

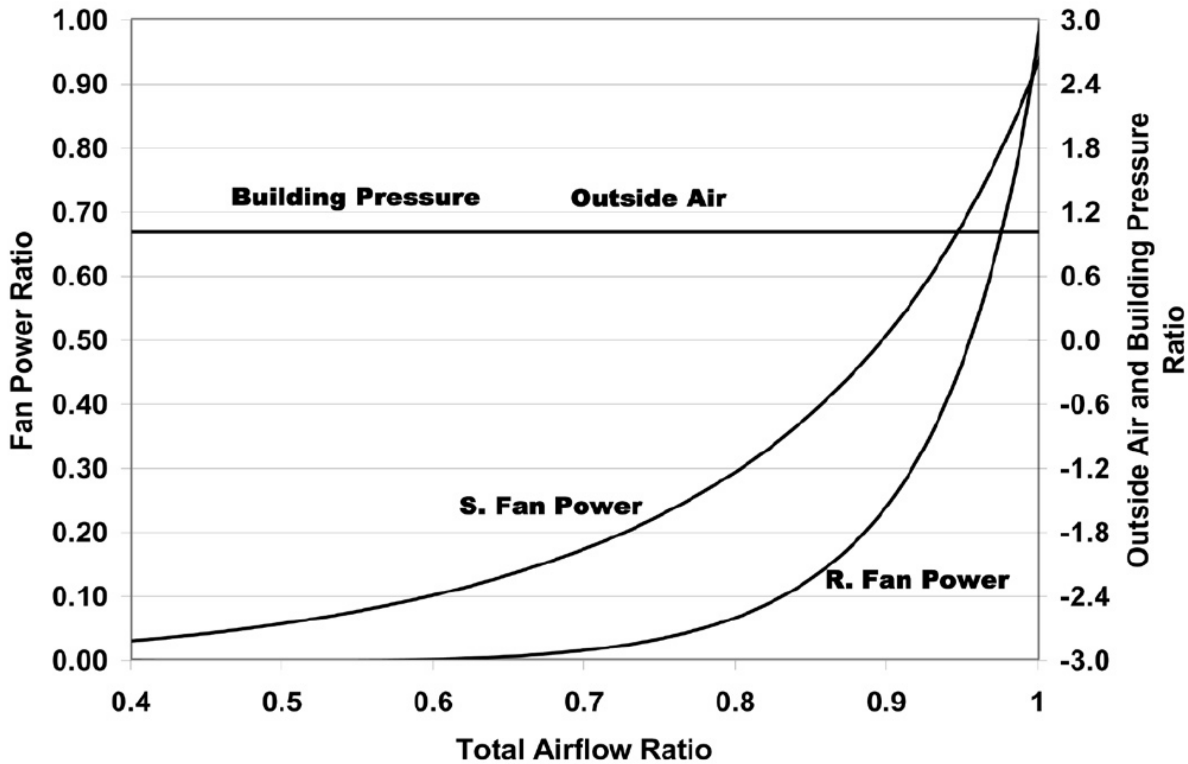

Fig. 23: Figure 4-23. Simulated Building Pressure, Outside Air Intake and Fan Power Using VSDVT Control

Figure 4-24 presents simulated VSDVT fan power savings compared with the typical method. The fan power savings are expressed as the ratio of the power savings to the design fan power. The maximum fan power savings are 37% for the return fan and 17% for the supply fan. Therefore, the VSDVT method can result in significant fan energy savings. The annual savings strongly depend on fan airflow distribution. Maximum fan energy savings would be achieved if the fan air flow is near 85% of the design value all of the time.

4.10 4.10 Improve Terminal Box Operation

The terminal box is the end device of the AHU system. It directly controls room temperature and air flow. Improving the set up and operation are critical for room comfort and energy efficiency. The following CCSM measures are suggested:

- Set minimum air damper position properly for pressure dependent terminal boxes. The minimum air damper position may be set based on ideal parallel damper characteristics. For example, if the minimum air flow is 30%, the minimum damper position is often set at 40% open. Under partial load conditions, the actual static pressure on the terminal box damper is higher than under full load conditions. Therefore, the actual minimum air flow can be 50% to 100% higher than the intended flow. The minimum damper position should be fine-tuned under partial load conditions.

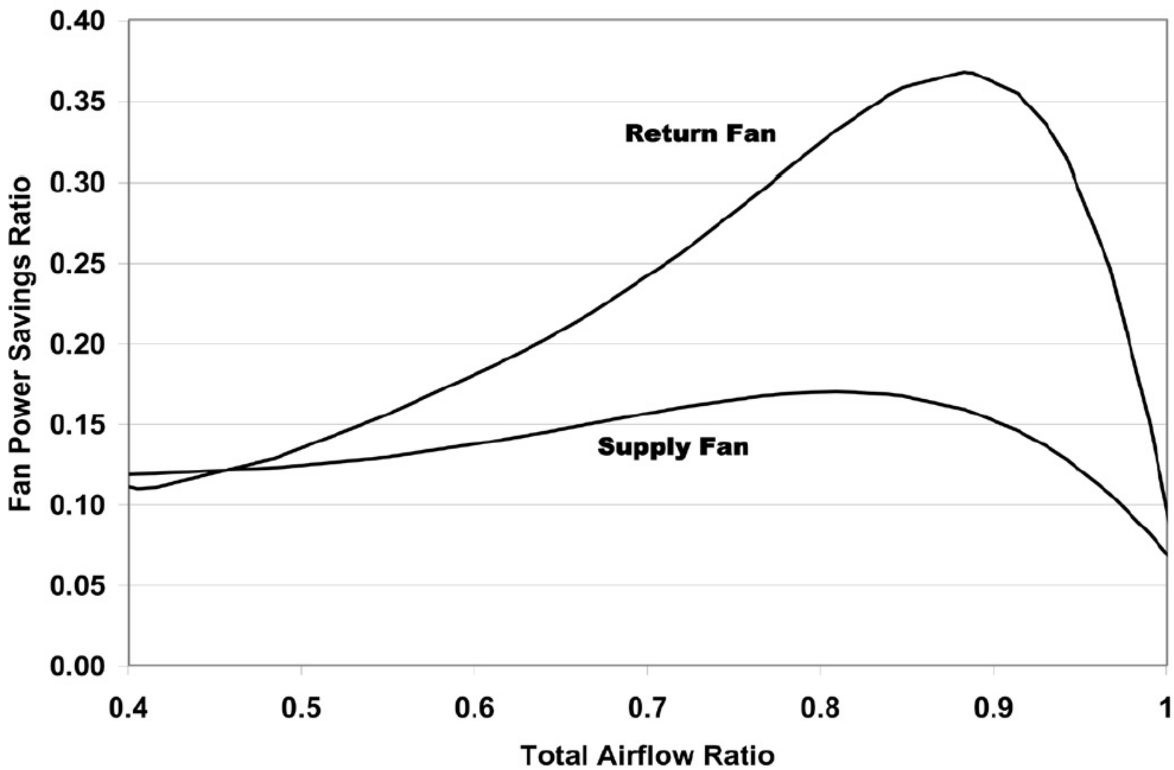

Fig. 24: Figure 4-24. Potential Fan Power Savings of the VSDVT Method

- Use a VAV control algorithm for constant air volume terminal boxes. Set the minimum air flow at the maximum for a constant airflow box during occupied hours. The terminal box will behave the same as a constant air volume terminal box. Set minimum air flow to zero or almost zero during unoccupied hours in order to significantly improve energy performance.
- Use airflow reset. Reset the minimum air flow to a lower value, possibly zero, during unoccupied hours and lightly occupied hours.
- Integrate lighting and terminal box control. Occupancy sensors are increasingly used to turn lights off when a space is unoccupied. Reset the minimum airflow lower, or to zero, and turn off lights when the occupants are not present. Note that this signal should not be used to change the room temperature set point.
- Integrate airflow and temperature reset. The potential energy savings of room temperature reset are relatively small for the following reasons:
  - The minimum airflow ratio (minimum airflow over the design airflow) is often higher than the load ratio during unoccupied hours. No savings will occur.
  - Commercial buildings have high thermal capacity. Heat stored during unoccupied hours is eventually removed by the AHU during occupied hours.
  - The chiller efficiency is higher at night than during the daytime. The electricity price may be also higher during the daytime. Temperature resets may actually increase cost.

In large commercial buildings, airflow reset may result in the same amount of savings produced by combined air flow and temperature reset.

- Improve terminal box control performance. The following tips can help significantly:
  - Set a minimum 2°F dead band between the heating and cooling set points. This will prevent frequent switching from heating to cooling or vice versa.
  - Set the terminal box maximum air flow to the highest possible value. If a box has a capacity of 500 cfm, the maximum air flow should be programmed to 500 cfm instead of the design value of 400 cfm. This may reduce the maintenance effort and decrease hot complaints. However, this may overload the fan during start up. Therefore, this can only be implemented in a limited number of terminal boxes. A fan speed limit should also be implemented for start up.
  - Set the minimum air flow based on actual building use rather than design values. For example, if the ventilation air was designed for 6,000 people and the actual census is 1,800 people, set outside air for 1,800 people.
  - Use different minimum airflow set points to minimize energy waste. Some perimeter reheat zones have their minimum flow set based on the heating load. The required minimum flow during the cooling season is often lower.
  - Verify flow sensor accuracy. If the inlet conditions are poor and not compensated in the flow sensor calibration, a calibration should be conducted.

More information can be found in “Terminal Box Airflow Reset: An Effective Operation and Control Strategy for Comfort Improvement and Energy Conservation” [Liu et al. 2002] and “Optimization Control Strategies for HVAC Terminal Boxes” [Zhu et al. 2000]

EXAMPLE:

Airflow reset was implemented in a dual duct variable air volume system in a medical building in Houston, Texas. The AHU had 45 terminal boxes with a 40 hp. supply air fan. The design air flow was 19,650 cfm. The occupied hours were from 8:00 a.m. to 7:00 p.m., Monday through Friday.

The design schedule required a minimum air flow from 40% to 70% with an average of 60%. This schedule was used during occupied and unoccupied hours. Airflow reset changed the minimum air flow to 20% for all boxes. Since the exhaust air fan could not be turned off, the minimum air flow from the AHU maintained the positive building pressure. To save fan power, the static pressure was at 0.5 in. H2O.

Airflow reset can be implemented using existing box control algorithms for some terminal boxes when the airflow reset is built into the box controller. Most terminal box controllers, however, do not have this option, which was the case for this building. Two box control schedules were programmed in the central control system. The daytime terminal box control sequence had a higher minimum airflow set point than the nighttime control sequence. At the beginning of the occupied period, the daytime control schedule was downloaded to each box. At the beginning of the nighttime period, the nighttime schedule was downloaded to each box from the central control system. The download was automatically performed using a schedule based on the time of the day and the day of the week.

Airflow reset was implemented on the last week of September 1997. The hourly ambient temperature and variable frequency drive (VFD) speed were recorded from August 1 to November 25. The airflow rate was calculated using VFD speed, design fan head and VFD speed.

Figure 4-25 presents the airflow ratio versus ambient temperature during unoccupied hours. Before implementation of airflow reset (August 1 to September 23), the nighttime air flow varied from 60% to 70% of the design value. After implementation (October 1 to November 25), the nighttime air flow varied from 20% to 30%. The airflow reset decreased air flow by 40% of design flow during unoccupied hours.

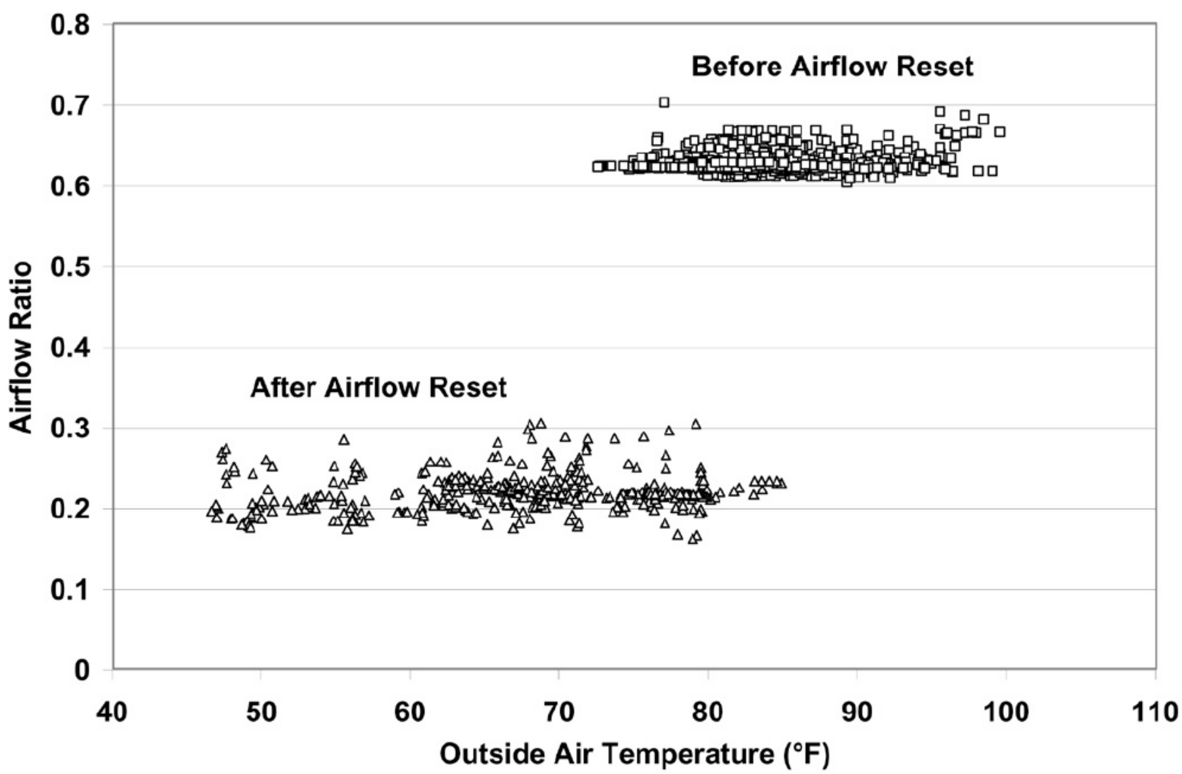

Fig. 25: Figure 4-25. Comparison of Air Flow Before and After Airflow Reset During Unoccupied Hours (Airflow Ratio is defined as the ratio of the air flow to the design air flow)

References

Hartman, T., 1989. "RAV – A new HVAC concept," Heating/Piping/Air Conditioning, July, pp. 69 –73.

Joo, I. and M. Liu, 2002. "Economizer Application in Dual-Duct Air Handling Units," Proceedings of Thirteenth Symposium on Improved Building Systems in Hot and Humid Climates, May 20-23, Houston, Texas, pp. 373-380.

Joo, I., M. Liu, K. Conger and G. Wang, 2002. "Variable Speed Drive (VSD) Application in Dual Duct Constant Volume Systems," Proceedings of Thirteenth Symposium on Improved Building Systems in Hot and Humid Climates, May 20-23, Houston, Texas, pp. 206-212.

Liu, M. A. Athar, A. Reddy, D. E. Claridge, J. S. Haberl and Ed. White, 1995. "Reducing Building Energy Costs Using Optimized Operation Strategies for Constant Volume Air Handling Systems" ASHRAE Transactions-Symposia. Volume 101, Part 2, pp. 688-700.

Liu, M., Y. Zhu, M. Abbas, R. D. L. Cruz, J. Perez, D. E. Claridge, D. Feary and J. Gains, 1996. "An O&M Story in An Old Building," Proceedings of Fourth National Conference on Building Commissioning, St. Pete Beach, FL, April 29 to May 1, pp. 14.3.1-14.3.14.

Liu, M., D. E. Claridge and B. Y. Park, 1997a. "An Advanced Economizer Controller for Dual Duct Air Handling Systems With a Case Study," ASHRAE Transactions-Research. Volume 103, Part 2, pp. 256-263.

Liu, M., Y. Zhu, D. E. Claridge and Ed. White, 1997b. "Impacts of Static Pressure Set Level on the HVAC Energy Consumption and Indoor Conditions," ASHRAE Transactions-Research. Volume 103, Part 2, pp. 221-228.

Liu, M., Y. Zhu, T. Powell and D. E. Claridge, 1998a. "System Optimization Saves $195,000/yr. in a New Medical Facility," Proceedings of the 6th National Conference on Building Commissioning, Lake Buena Vista, Fla., May 18-20, pp. 14.2.1-14.2.11.

Liu, M., M. Abbas, B. Veteto and D. E. Claridge, 1998b. "Impacts of Optimized Cold & Hot Deck Reset Schedules On Dual Duct VAV System-Application and Results," The Eleventh Symposium on Improving Building Systems in Hot and Humid Climates Proceedings, June 1-2, Fort Worth, Texas, pp. 153-160.

Liu, M. and D. E. Claridge, 1998. "Impacts of Optimized Cold & Hot Deck Reset Schedules on Dual Duct VAV Systems-Theory and Model Simulation," The Eleventh Symposium on Improving Building Systems in Hot and Humid Climates Proceedings, June 1-2, Fort Worth, Texas, pp. 146-152.

Liu, M., Y. Zhu, B. Y. Park, D. E. Claridge, D. Feary and J. Gain, 1999. "Air Flow Reduction To Improve Building Comfort and Reduce Building Energy Consumption," ASHRAE Transactions, Vol. 105, Part I, pp. 384-390.

Liu, M. and D. E. Claridge, 1999a. "Converting Dual Duct Constant Volume Systems to Variable Volume Systems Without Retrofitting the Terminal Boxes," ASHRAE Transactions, Vol. 105, Part I, pp. 66-70.

Liu, M. and D. E. Claridge, 1999b. "The Maximum Potential Energy Savings from Optimizing Cold and Hot Deck Reset Schedules for Dual Duct VAV Systems," Journal of Solar Energy Engineering, Vol. 121, pp. 171-175.

Liu M. and J. Wang, 2001a. "System Performance Analysis of Coupled Control Units With Both Thermostat and Humidistat," Proceedings of the ASME International Solar Energy Conference 2001, April 21-25, Washington, D. C., CD-ROM.

Liu M. and J. Wang, 2001b. "System Performance Analysis of Coupled Control Units Without Humidistat," Proceedings of the ASME International Solar Energy Conference 2001, April 21-25, Washington, D. C., CD-ROM.

Liu, M., M. Abbas, Y. Zhu and D. E. Claridge, 2002. "Terminal Box Air Flow Reset: An Effective Operation and Control Strategy for Comfort Improvement and Energy Conservation," Proceedings of Thirteenth Symposium on Improved Building Systems in Hot and Humid Climates, May 20-23, Houston, Texas, pp. 80-86.

Liu, M., 2002. "Variable Speed Drive Volumetric Tracking (VSDVT) for Airflow Control in Variable Air Volume (VAV) Systems," Proceedings of Thirteenth Symposium on Improving Building Systems in Hot and Humid Climates, May 20-23, Houston, Texas, pp. 197-205.

Wei G., D. E. Claridge and M. Liu, 2000a. "Optimize the Supply Air temperature Reset Schedule for Single-duct VAV Systems," Proceedings of Twelfth Symposium on Improving Building Systems in Hot and Humid Climates, May 15-16, San Antonio, Texas, pp. 154-157.

Wei, G., D. E. Claridge, Y. Sakurai and M. Liu, 2000b. "Improved Air Volume Control Logic, " Proceedings of Twelfth Symposium on Improving Building Systems in Hot and Humid Climates, May 15-16, San Antonio, Texas, pp. 195-198.

Wei, G., W. D. Turner, D. E. Claridge, M. Liu, M. J. Hewett and M. W. Hancock, 2001. “Continuous Commissioning of a Hospital Complex,” Proceedings of National Conference on Building Commissioning, pp. 19-2.1-19-2.10.

Zhu, Y., M. Liu, D. E. Claridge, W. D. Turner and T. Lewis, 1998. “A Novel Procedure to Determine Optimal Air Static Pressure Set-points and Reset Schedules in VAV Air Handling Units,” Proceedings of the Eleventh Symposium on Improving Building Systems in Hot and Humid Climates, June 1-2, Ft. Worth, Texas, pp. 294-301.

Zhu, Y, M. Liu, T. Batten, H. Noboa, D. E. Claridge and W. D. Turner, 2000. “Optimization Control Strategies for HVAC Terminal Boxes, “ Proccedings of the Twelfth Symposium on Improving Building Systems in Hot and Humid Climates Proceedings, May 15-16, San Antonio, Texas, pp. 278-284.

CHAPTER 5

Ch 5: CC Measures for Water/Steam Distribution Systems

Distribution systems include central chilled water and hot water/steam systems, that deliver thermal energy from central plants to buildings. In turn, the system distributes the chilled water and hot water or steam to AHU coils and terminal boxes. Distribution systems consist mainly of pumps, pipes, control valves and variable speed pumping devices.

> **Notes**
>
> Distribution systems consist mainly of pumps, pipes, control valves and variable speed pumping devices.

A central chiller plant could have primary and secondary loops. The primary pumps are only used to circulate water through the chillers. The secondary pumps are used to distribute water to the buildings. The distribution systems can be categorized as source-distributed and distributed systems. The source-distributed system has secondary pumps located only in the central plant. The distributed pumping system has pumps located in the buildings but with no secondary pumps. Most central plants are not pure source-distributed nor pure distributed systems. Most have all three types of pumps: primary, secondary, and building pumps.

This chapter focuses on the CCSM measures for optimal pressure control, water flow control, and general optimization. The supply temperature and steam optimization measures are covered in Chapters 6 and 7.

5.1 5.1 Improve Building Chilled Water Pump Operation

Most building chilled water pumping systems are equipped with variable speed devices (VSDs). If a VSD is not installed, retrofit of a VSD is generally recommended. The discussion here is limited to systems where a VSD is installed. The goal of pumping optimization is to avoid excessive differential pressures across the control valves while providing enough water to each building, coil, or other end use1. Optimal pump operation schedule should be developed using the following procedure:

- Inspect each heating and cooling coil. Identify all three-way valves. Convert three-way valves into two-way valves by closing the manual valves on the bypass line. If necessary, cut the bypass line. In rare cases, the three-way valves may hydraulically lock if the bypass is fully closed. In these cases, the three- way valves must be replaced.

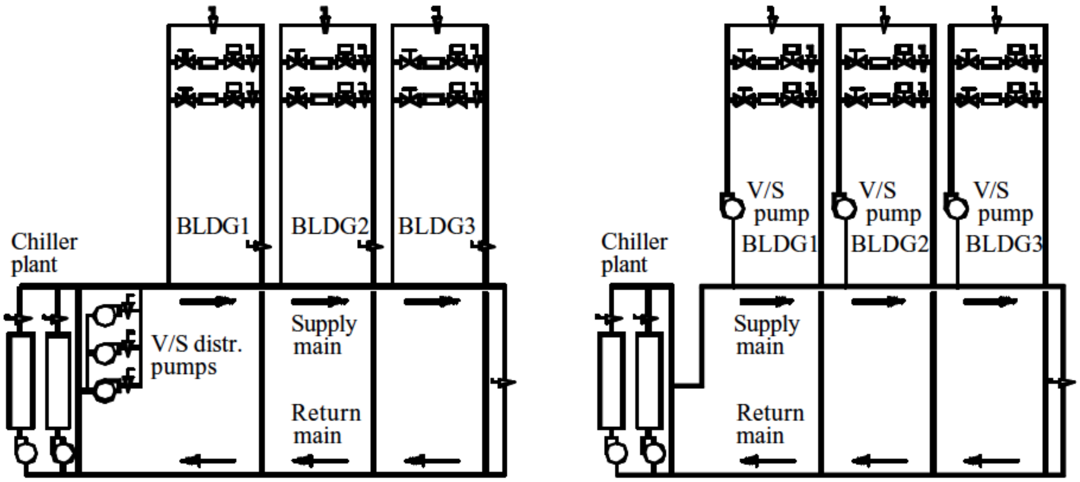

Fig. 1: Figure 5-1. Schematic Diagrams of Source Distributed Pumping (on left) and Distributed Pumping Configurations

- Inspect the building entrance. Disable any blending station that allows return water to blend with primary water from the central plant using manual valves or by closing off the bypass line. Note that if the chiller plant provides chilled water to only one building, the primary/secondary loops may be converted into a single loop. Single-loop operation or conversion is discussed later in this chapter.
- Identify the coil which calls for the highest differential pressure to deliver the required flow. Open all manual valves between this coil and the building pump. Check the coil supply air temperature set point. If the set point is lower than the design value, reset it back to the normal value.
- Slow the pump down until the control valve at the coil is 85% open. After the system is stabilized, measure the differential pressure (ΔP1) at the loop sensor position and the chilled water flow rate. The differential pressure (ΔP1) is the optimal set point (ΔPo) under the measured flow condition.
- If step 4 cannot be performed, measure the differential pressure across the coil (ΔP2), including the valve, the differential pressure (ΔP1) at the loop sensor, and the chilled water flow rate (Go). Determine the set point of the optimal loop pressure as:

  $$\Delta P_0 = \Delta P_1 - \Delta P_2 + C$$

  The constant C is the differential pressure required by the coil. It typically varies from 1 to 5 psi depending on the size of the coil. However, exceptions are possible. If a value above this range is detected, compare it with the design value. If it is higher than the design value, the coil may be blocked. Cleaning or repair may be required.
- If the differential pressure sensor is located at the most remote coil, control the VSD to maintain the set point
- If the differential pressure sensor is not located at the most remote coil, reset the differential pressure set point using measured chilled water flow rate. The reset schedule may not apply to extremely low flow conditions. A low limit is often necessary. Generally, the set point should not be lower than 5 psi. A high limit is also recommended in case the flow meter malfunctions. The high limit should be calculated by introducing the design flow for G in the equation for ΔP.

  $$\Delta P = \Delta P_0 \left(\frac{G}{G_0}\right)^2 + 2$$

  Turn off the building pump(s) if the primary/secondary loop provides enough differential pressure to the building.

- If the chilled water flow is not measured, the differential pressure can be reset based on the VSD speed as:

$$\Delta P = \Delta P_0 \left(\frac{VSD}{VSD_0} \right)^2 + 2$$

where VSD0 is the VSD speed when ΔP0 is determined.

- Implementat the optimal reset schedule using the BAS. After initial implementation, hot complaints may occur due to existing mechanical problems or incorrect manual valve positions. When a hot call is received, check the appropriate coil. Special attention should be given to balance valve positions and any other valve position in that branch. Take action to properly balance the system. If the problem persists, check the coil and its valves to identify any mechanical problems and repair. If this does not solve the problem, this coil should be used as the most remote coil.

More information can be found in "A Simple and Quick Chilled Water Loop Balancing for Variable Flow Systems" [Zhu et al. 2000], and "System Optimization Saves $195,000/yr in a New Medical Facility" [Liu et al. 1998].

EXAMPLE:

The G. R. White Annex is part of the G. R. White Coliseum on the Texas A&M University Campus in College Station. The total conditioned space is 177,838 square feet, which includes classrooms, offices and gym areas. Two chilled water pumps (40 hp. each) are installed to circulate water in the building. A VFD is installed on the lead pump with the other pump as a standby pump. The differential pressure sensor is installed at the entrance of the building loop.

The original differential pressure set point varied from 15 psi to 54 psi as the chilled water flow varied from 50% to 100% of design flow. During most of the year, this schedule caused the pump to run at full speed.

A test was conducted. The measured optimal set point was 10 psi when the chilled water flow was 275 gpm. The optimal reset schedule was then constructed as:

$$\Delta P = 10 \left(\frac{G}{275} \right) + 2$$

The minimum limit was set as 5 psi. The high limit was set at 15 psi. Figure 5-2 compares the measured chilled water flow rate before and after the implementation of the optimal reset schedule.

Under the initial schedule, the pump was running at full speed. Under the new schedule, all control valves can function properly. The chilled water flow was reduced by 50% and building comfort was still maintained. The optimal schedule decreased pump power by 20 kW based on a one-time measurement.

After implementing the optimal chilled water loop differential reset schedule, the supply air temperature was maintained at the set point. Cold complaints were significantly reduced and simultaneous heating and cooling decreased. Figure 5-3 compares the measured hourly cooling energy consumption under both the original and optimal chilled water differential reset schedules. Under the same temperature conditions, the chilled water consumption was approximately 250 kBtu/hr lower under the optimal schedule. This translated into 2,000 MMBtu/yr chilled water energy savings. If the building comfort is maintained at the same level, the same amount of heating energy savings should be obtained. The annual energy cost savings were estimated to be $28,760/yr. This includes $8,760 for pump power, $10,000 for chilled water, and $10,000 for hot water based on energy prices of $0.05/kWh, and $5/MMBtu for hot water and chilled water.

The pump control may also be improved by controlling the pressure so that at least one control valve is at least 85% open (adjustable). If none of the control valves are at least 85% open, slow down the pump and vice versa. This method requires special attention to all control valves. A single malfunctioning valve may cause the pump to run at full speed. Generally speaking, this method should be integrated with the pressure reset schedule. If the actual differential pressure approaches the set point and one control valve is still fully open, it often indicates a problem associated with the valve or the coil controlled by the valve.

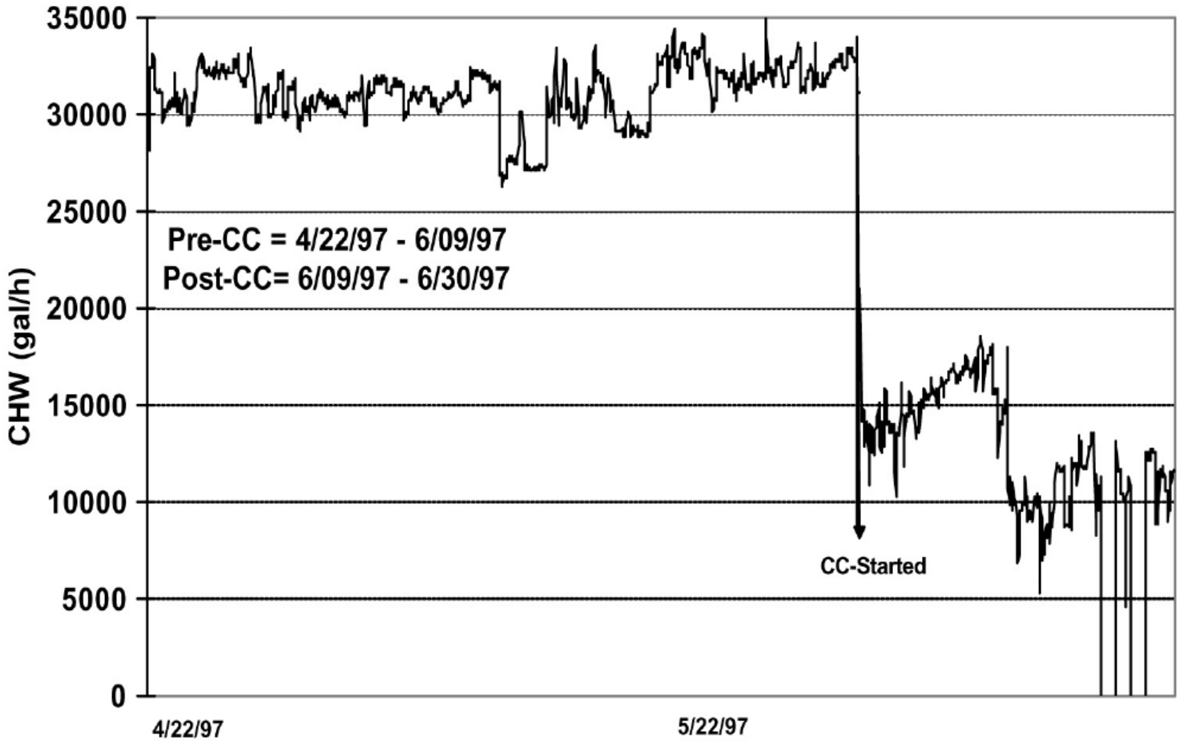

Fig. 2: Figure 5-2. Comparison of Measured Chilled Water Flow Under Both Initial and Optimal Differential Pressure Reset Schedules

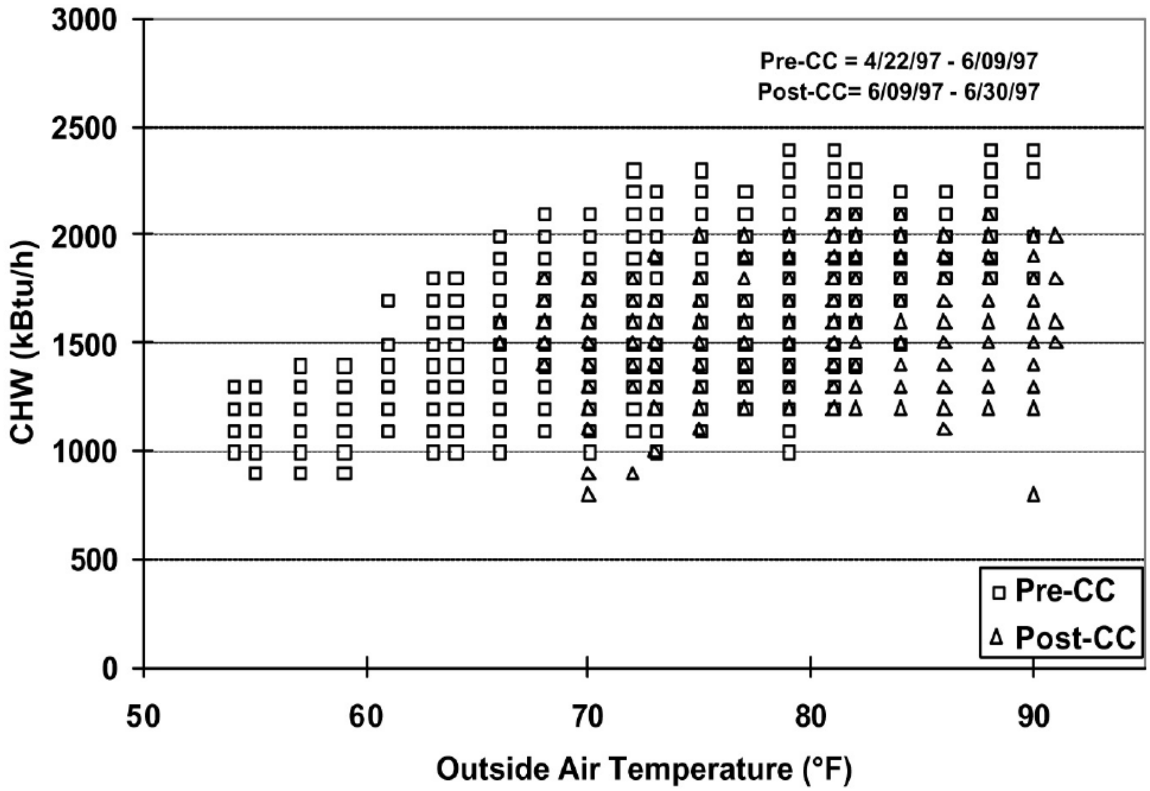

Fig. 3: Figure 5-3. Comparison of Chilled Water Energy Consumption Under Original and Optimal Chilled Water Differential Pressure Schedules

5.2 5.2 Improve Secondary Loop Operation

The building loop optimization should be performed before the secondary loop optimization.

5.2.1 5.2.1 Source Distributed Systems

If there are no building pumps, the secondary pumps must provide the pressure head required to overcome both the secondary loop and the building loop pressure losses. In this case, the secondary loop is called a source distributed system. The secondary loop pumps should be controlled to provide enough pressure head for the most remote coil. If VFDs are installed, the differential pressure can be controlled by modulating pump speed. Otherwise, the differential can be modulated by changing the number of pumps in operation.

Notes

Installing building pumps can decrease total pumping power by as much as 50%.

The source-distributed system is the least efficient distribution system. Installing building pumps can decrease total pumping power by as much as 50% when the pumps are controlled and operated properly. The source distributed system will often have water balance problems because it over-pressurizes the control valves of the buildings nearest to the central plant. Due to excessive water flow through these buildings, often the remote buildings do not receive enough water. Alternatively, the distribution pump at the central plant must pump extra water. It is recommended that building pumps be installed for relatively large complexes with several buildings.

5.2.2 5.2.2 Source Distributed Systems With Building Pumps

In most campus settings, both secondary distribution and building pumps are installed. The optimal differential pressure set point should be determined using the following procedure:

- Inspect major buildings and solve any major water balance problems.
- Identify the most remote building. Inspect the building and identify any control and mechanical problems in the major AHUs. Fix major mechanical and control problems for the cooling coils.
- Modulate the secondary loop pump until the building pump in the most remote building runs at full speed while maintaining building comfort. Record the secondary chilled water flow rate (Go), and loop differential pressures (ΔPpo) at the central plant (ΔP at the sensor location) and the remote user entrance (ΔPbo). Measure the pump head (Hp). If the secondary loop ΔP sensor is located at the entrance of the most remote building, ΔPp should be reset based on the measured loop flow rate using the following formula:

  $$\Delta P_p = (H_p + \Delta P_{b0}) \left(\frac{G}{G_0} \right)^2 - H_p + 2$$

  If the secondary loop ΔP sensor is located at the central plant, ΔPp should be reset based on the measured loop flow rate using the following formula:

  $$\Delta P_p = (H_p + \Delta P_{p0}) \left(\frac{G}{G_0} \right)^2 - H_p + 2$$

  A maximum limit should be imposed on the reset schedule.
- If secondary pumps cannot be adjusted during the test, follow the procedure below:
  - Select a manual valve in the main chilled water loop of the building

- Adjust the valve until the building pump runs at full speed and the maximum opened chilled water control valve is 85% open
- Record the secondary chilled water flow rate (Go), loop differential pressure (ΔPpo) at the central plant (ΔP at the sensor location) and the most remote building entrance (ΔPbo). Measure the pump head (Hp). Record the pressure loss across the manual valve (ΔPl).
- If the secondary loop ΔP sensor is located at the entrance of the most remote building, the ΔPp should be reset based on the measured loop flow rate using the following formula:

$$\Delta P_p = (H_p + \Delta P_{b0}) \left(\frac{G}{G_0}\right)^2 - H_p - \Delta P_l + 2$$

- If the secondary loop ΔP sensor is located at the central plant, the ΔPp should be reset based on measured loop flow rate using the following formula:

$$\Delta P_p = (H_p + \Delta P_{p0}) \left(\frac{G}{G_0}\right)^2 - H_p - \Delta P_l + 2$$

  A maximum limit should be imposed on the reset schedule.

- The optimal reset schedule can be easily implemented using the BAS. After initial implementation, hot calls may occur due to existing mechanical problems or incorrect manual valve positions. When a hot call is received, check the building controls and mechanical systems. Special attention should be given to balance valve positions and any other valve position in the branch. Take action promptly to balance the system. If the problem persists, check the building to identify mechanical problems and repair. If this does not solve the problem, this building should be treated as the most remote building.
- Turn off building pumps where the secondary pump provides enough pressure difference to the building loop

The advantages of optimizing the secondary loop control can be demonstrated best using the simplified example below. Consider the simplified loop shown in Figure 5-4. We assume that the secondary loop has a pressure drop of 5 psi between the plant and the first building, as well as between each successive building. The pressure drops on the return side of the loop are the same as in the supply side of the loop. A building pump is available in each building and flow through each building is 1000 gpm. Assume that each building loop has a 10 psi pressure drop. We want to find the loop pressure that meets these flow requirements and minimizes the combined building and loop pumping power.

We first consider source distributed pumping where the secondary loop pumps provide all the pumping power for the loop and the buildings as shown in Figure 5-5. This figure shows the differential pressure distribution throughout the loop. We see that 10 psi will be needed for the last building. We also assume 10 psi at the return to meet NPSH requirements for the secondary pump. In a real system, the static pressure will satisfy the positive pressure in the entire system. We note that the excess differential pressure for each of the first three buildings, 30 psi to 10 psi, must be dropped across valves in the buildings.

The pump power depends on flow rate, pump head and pump efficiency:

Ppump = VΔP/η

where V is the volumetric flow rate, ΔP is the differential pressure drop, and η is the pump efficiency. This equation gives the answer in ft-lbf/s, so we will use the equation

Ppump = 0.000597GΔP/η

where now G is in gpm, ΔP is in psi and the answer is in hp.

For source distributed pumping, if we have a pump efficiency of 0.83, we get:

Ppump = 0.000597*4*1000gpm*50 psi/0.83 = 144 hp.

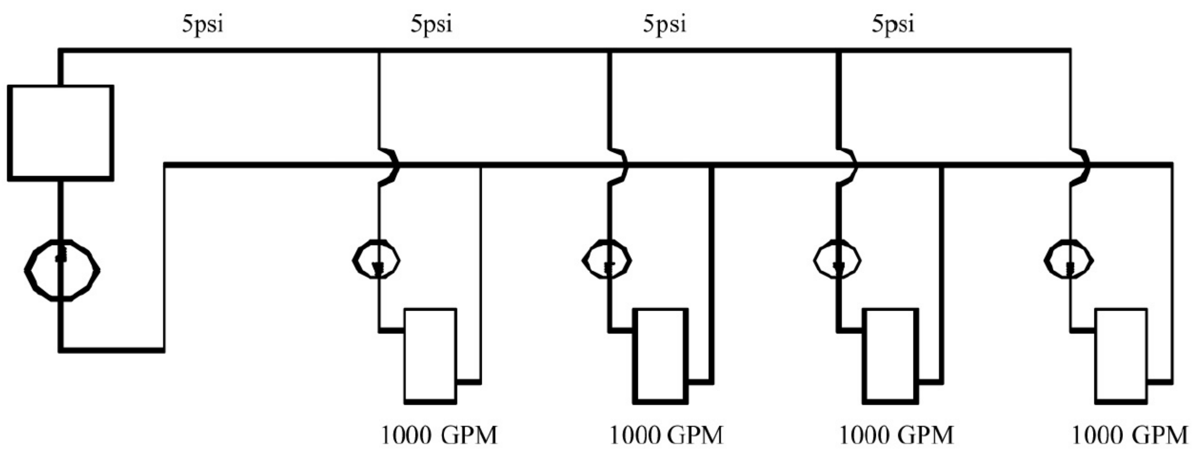

Fig. 4: Figure 5-4. Loop and Building Flows and Pressure Drops for Example Problem

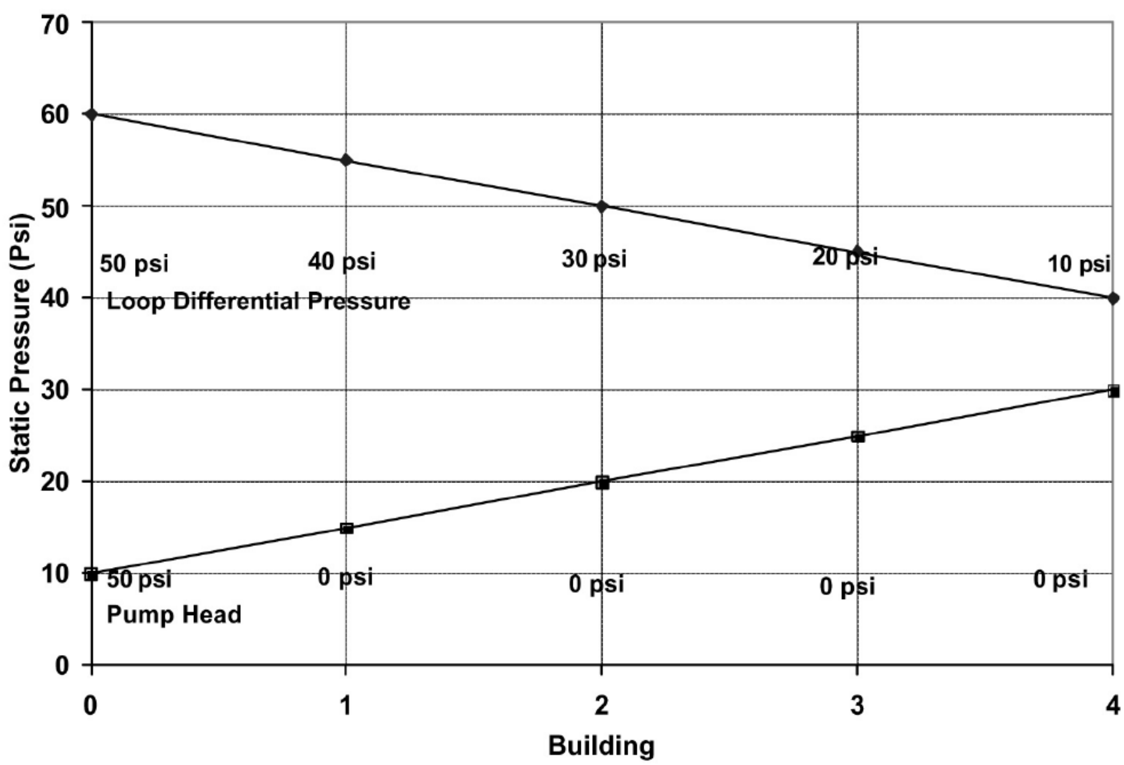

Fig. 5: Figure 5-5. Differential Pressure Distribution for Source Distributed Pumping

If we use the combination of source and distributed pumping shown in Figure 5-6, we note that we now have 20 psi differential pressure across the secondary pump that still pumps 4000 gpm. The first building obtains its pumping power from the secondary loop. Therefore, the building pump is off. The second building has a flow of 1000 gpm pumped across 10 psi, the third building has 1000 gpm pumped across 20 psi and the fourth building has 1000 gpm pumped across 30 psi. Hence we obtain:

Ppump = 0.000597*[4000gpm*20 psi+1000 gpm*(10 psi + 20 psi + 30 psi)]/0.83 = 101 hp.

If we evaluate the pumping power required for all possible combinations ranging from source pumping to distributed pumping for this system, we arrive at the plot shown in Figure 5-7.

Here we see that, for this system, pure distributed pumping or a combination of source and distributed pumping provides a loop pump head of up to 20 psi and building pumping in three or four of the buildings. All have the same pumping power requirements. The pumping power then increases as the loop pump head is increased to reach a maximum value for pure source pumping. The savings of the optimum combinations are (144-101)/144*100 = 30%.

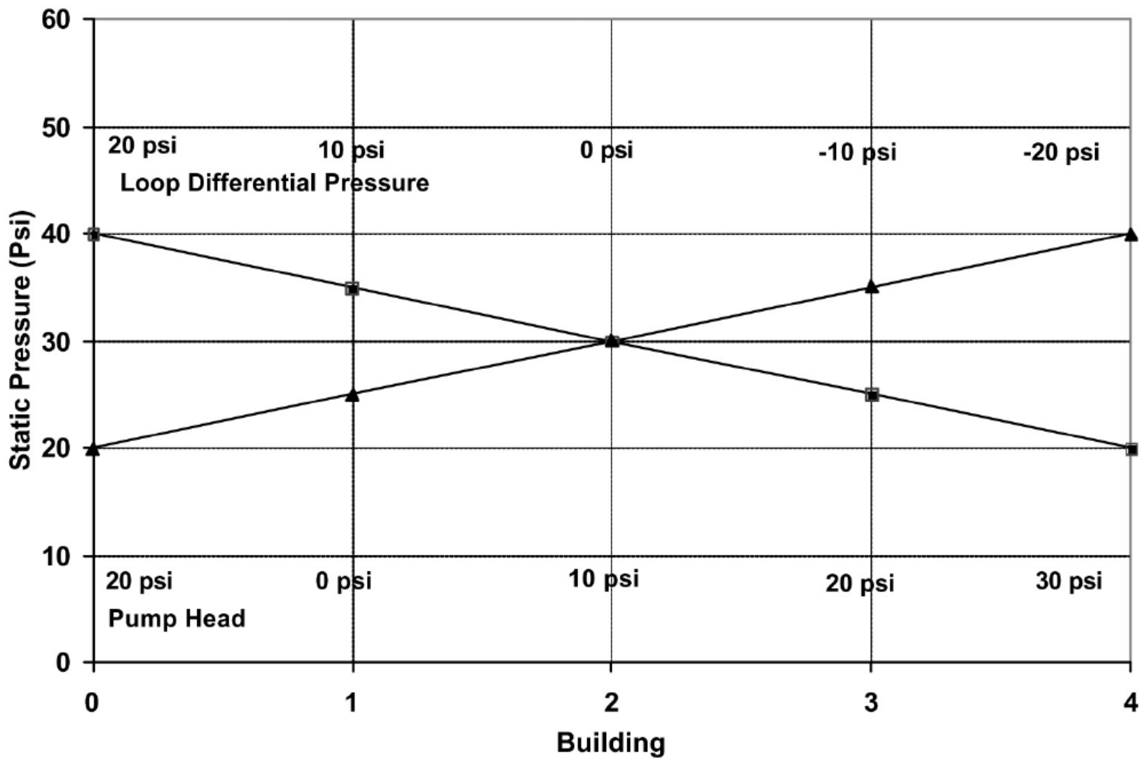

Fig. 6: Figure 5-6. Differential Pressure Distribution for the Evaluation of Combined Source and Distributed Pumping

EXAMPLE:

The main campus of Texas A&M has 107 buildings. The central heating plant, located at the north end of the main campus, distributes heating water through two loops to all buildings using three distribution pumps with a total capacity of 1025 hp. The longest branch of the loop is 0.53 miles (2,800 feet). Building pumps are installed in each building and most buildings have standby pumps. Excluding the standby pumps, the building pumps have a total capacity of 947.5 hp. VFDs are installed in 21 buildings that have a total heating pump capacity of 214.5 hp.

Before optimizing the heating loop operations, all building heating pumps operated continuously. The central plant maintained differential pressure within a range of 50 psi to 60 psi using manual control (changing the number of pumps in operation). The heating water temperature was set at 180 °F.

In the winter of 1996-1997, optimization of the heating loop was conducted. The main activities are summarized

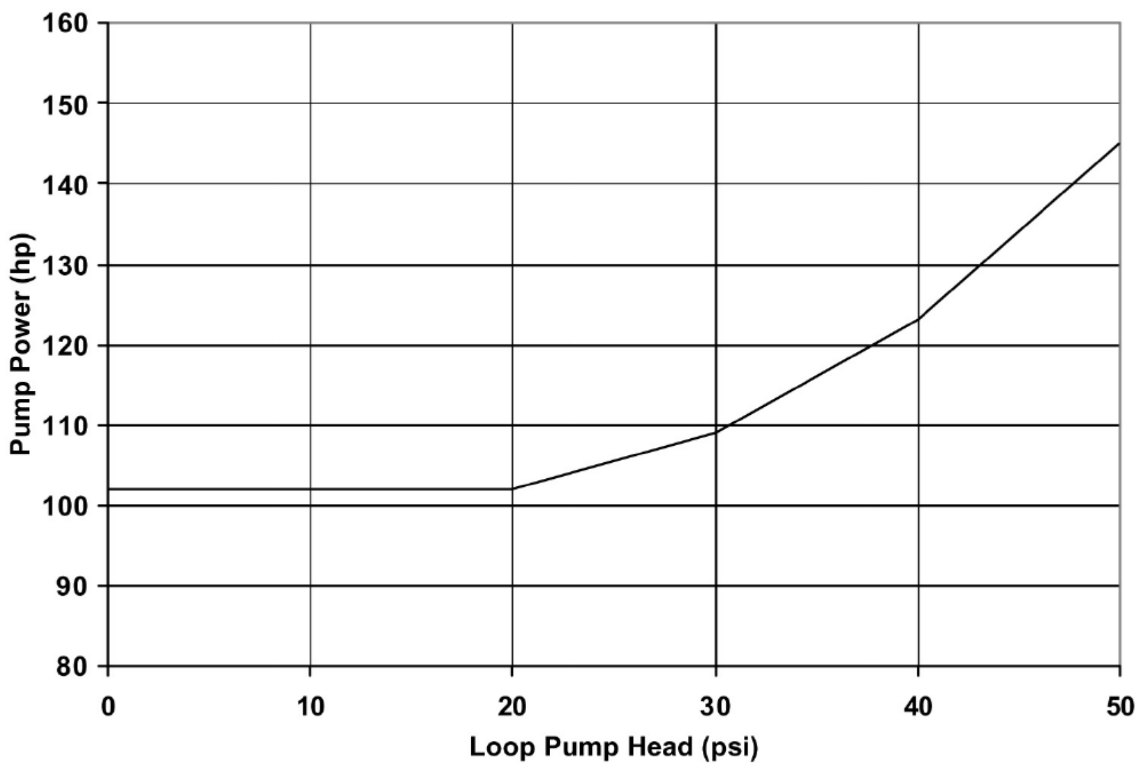

Fig. 7: Figure 5-7. Pumping Power as a Function of Loop Pump Head for the System Considered in the Example

below:

- All three-way valves and blending stations were closed. Blending stations were used in 106 buildings. 86 blending stations were disabled using manual valves. 20 blending stations were disabled by blocking the bypass lines. Most three-way valves at AHUs were converted into two-way valves by closing the manual valves on the bypass lines.
- A large pump was installed in the most remote building, the Commons Dining Hall. The Commons Dining Hall is located at the very end of the heating water loop. The building used a pump with a 30 hp. motor. However, its pump head is less than 10 psi since its impeller was trimmed some years earlier. No records were found as to when or why the impeller was trimmed. Engineering calculations showed that a 30 psi pump head for this building would align it with adjacent buildings and provide the best overall loop performance. To correct the problems, a new 10 hp. pump was installed.
- The loop differential pressure was reset from 30 psi to 40 psi as a linear function of the ambient temperature as it varies from 100°F to 30°F. The operating staff preferred to use ambient temperature instead of flow rate as the basis for resetting the hot water loop differential pressure because of control system limitations and other factors.
- The hot water temperature was reset from 180°F to 140°F. Since most buildings were using blending stations, 140°F temperatures can satisfy the requirement of most buildings. Minor system commissioning was conducted in a few buildings, such as the Chemistry building, to resolve existing building problems and reduce the required hot water temperature to 140°F.
- The building heating water pumps in 60 buildings were turned off, resulting in a total 400 hp. reduction in capacity

Figure 5-8 presents a schematic of the campus heating water distribution diagram. Pumps in buildings shaded with gray have been turned off permanently. This results in pump power savings of 400 hp. Due to the reduced differential pressure set point, the central plant runs one pump less than under the original schedule. This results in additional pump power savings of 300 hp. The total pump power savings are approximately 700 hp., or 35%, of the original total pumping power (1972 hp.). Since the heating systems run 24 hours per day, the annualized energy savings are estimated to be 4,500 MWh. The potential cost savings are $225,000/yr at an average electricity cost of $0.05/kWh.

The reduced loop differential pressure reduced hot calls significantly in the buildings nearest the central plant. Under the original schedules, the building heating water loop experienced excessive pressure differences and the heating water control valves were not able to function properly. Consequently, the room temperatures were not properly maintained and excessive thermal energy was used. More information about this case study can be found in "Reducing Pump Power Consumption by (1000 kW) 40% Through Improved Pump Management in a Central Plant" [Deng et al. 1998].

5.3 5.3 Improving Central Plant Water Loop Operation

The central plant loop optimization should be performed after secondary loop optimization.

5.3.1 5.3.1 Single Loop Systems

For most heating distribution systems and some chilled water systems, a single loop is used instead of primary and secondary systems. Under partial load conditions, fewer pumps can be used for both chillers and heat exchangers. This can result in less pump power consumption. The following procedures should be followed to optimize the system operation:

- Balance chiller or heat exchanger loops to maintain the same ratio of flow through each chiller or heat exchanger. Ensure that all manual valves are fully open in at least one branch. This minimizes the loop pressure loss to save pump power.

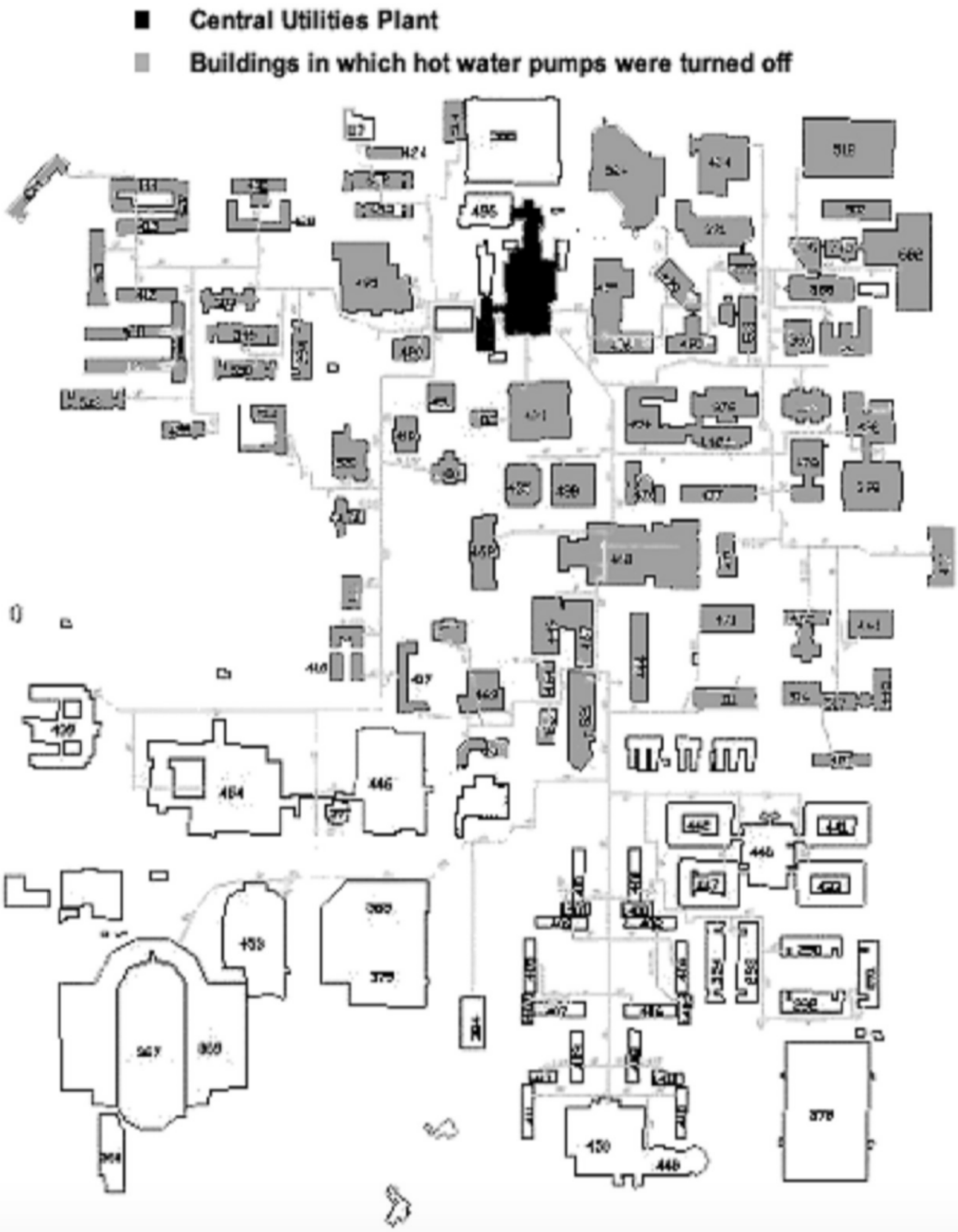

Fig. 8: Figure 5-8. Schematic Diagram of Central Heating System on the Texas A&M University Main Campus in College Station

- Adjust the flow switch on the chiller. Ensure that the switch sends a flow signal to the chiller control if the flow is higher than the minimum flow. Typically, the flow switch is set at the design flow rate. This can be decreased to 30% of the design flow rate for most chillers without causing any damage. The chiller manufacturer should be consulted for verification of the low flow setting.
- Under partial load conditions, match the pump flow rate to the entire chiller plant load ratio. For example, assume a central chiller plant has four chillers with a total capacity of 4,000 tons. If the load is 2,000 tons, two pumps and three chillers should be turned on. Each chiller operates at its most efficient load ratio of 67%. Each pump has a load ratio of 100%.
- Adjust the automatic control valve (isolation or shut off valve) cycle time to at least 60 seconds if old chillers (5 years or older) are used
- Verify that the pump will not overload. Find the current working point on the pump curve and identify future working points or ranges. Ensure that the pump brake hp. will not exceed the pump motor capacity.

EXAMPLE:

The Materials Research Institute (MRI) building, located in State College, PA, has a total floor area of 50,000 square feet including offices, classrooms, laboratories and a clean room facility.

Figure 5-9 presents a schematic diagram of the central plant chiller system. Two 285-ton York chillers are installed in parallel. The design chilled water flow rate is 570 gpm. The chilled water pumps are 25 hp. each with a design pump head of 100 feet of water column and a flow of 695 gpm. This is 22% higher than the rated chiller flow.

An automatic control valve is installed for each chiller. The pressure loss across the chiller is monitored. If the pressure loss is higher than the set point, the automatic valve closes and vice versa. Another automatic control valve is installed on the building by-pass line. If the building loop differential pressure is higher than the set point (25 psi), the bypass valve opens to maintain the set point.

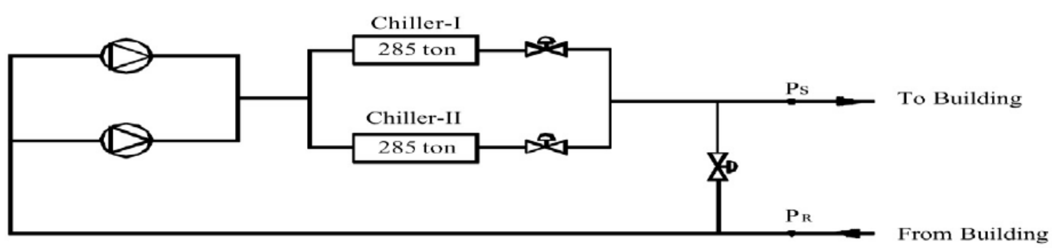

Fig. 9: Figure 5-9. Schematic Diagram of Chiller Plant at MRI Building

On September 10, 1998, a field inspection found that (1) the manual valve on the chiller exit was 50% closed and (2) the automatic valves were 50% closed when two chillers were on. The chilled water return temperature to the chillers was 47°F. Chiller supply temperature was 41°F. The building chilled water return temperature was 54°F. Approximately 54% of the chilled water bypassed the building loop.

A test measured a chilled water flow rate of 550 gpm when one chiller was on with one pump. When two chillers were on, the chilled water flow rate was measured to be 400 gpm for each chiller with one pump on.

To improve the chilled water loop operation, the following actions were taken:

- The building loop differential pressure set point was increased from 25 psi to 35 psi
- The chiller loop section was balanced and each chiller was determined to have the same flow rate. All manual valves were opened to 100%. Since it is a parallel loop, balancing only required opening the valves.
- York was contracted to verify that the chillers can be operated safely if the chilled water flow is at least 150 gpm, or 26% of the design flow rate

- Only one pump was controlled to run regardless of the number of chillers in operation. According to the pump curve shown in Figure 5-10, the pump power will be less than 25 hp. if the chilled water flow is less than 930 gpm. Since the maximum building chilled water flow is less than 700 gpm based on measured data, the chilled water pump will not be overloaded. Note that the building loop differential pressure set point was raised from 25 psi to 35 psi. The increased setpoint prevents excessive building bypass. Consequently, it prevents excessive pump flow and pump overloading.
- An operating technician implemented the procedure following the formal recommendations by the CCSM engineer.

The improved pump operating procedures turned off one pump for 4 months, according to the operating log. Consequently, the pump energy savings were estimated to be 53,640 kWh/yr, or $3,754/yr assuming an electricity price of $0.07/kWh.

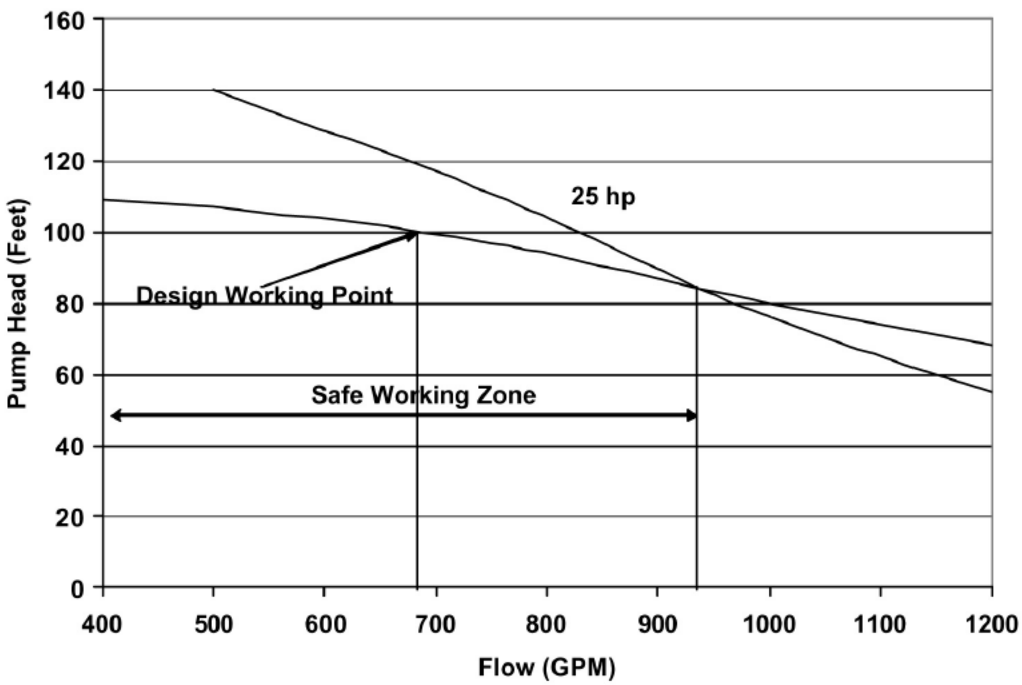

Fig. 10: Figure 5-10. Chilled Water Pump Curve (regenerated using pump curves from TACO, TA Series Model 1229, speed 1760 RPM, and 10.25” impeller)

5.3.2 5.3.2 Primary and Secondary Loop Systems

Primary and secondary systems are the most common chilled water distribution systems used with central chiller plants. This design is based on the assumption that the chilled water flow through the chiller must be maintained at the design level. This is seldom needed. Due to this incorrect assumption, a significant amount of pumping power is wasted in numerous central plants. Design engineers sometimes include an isolation valve on the bypass line of the primary loop. Sometimes, no valve is included. If no isolation valve is incorporated in the system, the following procedures should be followed to optimize system operation:

- Balance chiller loops to maintain the same flow ratio (chilled water flow over the design flow) through each chiller. Ensure that all manual valves are fully open in at least one branch.
- Adjust the flow switch on each chiller. Ensure that the switch sends a flow signal to the chiller control if the flow is higher than the minimum flow. Typically, the flow is set at the design flow rate. This can be decreased to 30% of the design flow rate.

- Match the pump flow rate to the entire load ratio under partial load conditions. For example, assume a central chiller plant has four chillers with a total capacity of 4,000 tons. If the load is 2,000 tons, two pumps and three chillers should be turned on. Each chiller then has a load ratio of 67% and each pump has a load ratio of 100%. It is recommended that chillers not run at load ratios higher than 80% or lower than 40%.
- Adjust automatic control valve cycle time to at least 60 seconds if older chillers (5 years or older) are used If an isolation valve is installed, the following procedures should be followed to optimize the pump operation.
- Balance chiller loops to maintain the same flow ratio through each chiller. Ensure that all manual valves are fully open in at least one branch.
- Adjust the flow switch on each chiller. Ensure that the switch sends a flow signal to the chiller control if the flow is higher than the minimum flow. Typically, the flow is set at the design flow rate. This can usually be decreased to 30% of the design flow rate.
- Close the isolation valve and turn off the primary pumps when the plant load is low enough that the secondary pumps can provide enough head for the entire loop. Typically, this is possible when the load is below 80% of the design load, but this point will vary from plant to plant.
- Run chillers with loads no higher than 80% and no lower than 40%. Use this guideline to determine the number of chillers in operation. For example, assume a central chiller plant has four chillers with a total capacity of 4,000 tons. If the load is 2,000 tons, three chillers should be turned on so each chiller has a load ratio of 67%. Pumps and cooling towers should use an optimal operating schedule as well.
- Use the secondary pump to circulate water through both chillers and buildings. No changes are required for the secondary pumping control.
- Adjust the automatic control valve cycle time to at least 60 seconds if older chillers (5 years or older) are used

To decrease operational mode changes, the daily maximum load should be estimated when high loads are expected. If the daily maximum load requires primary pump operation, it is recommended that the primary pump(s) be left on all day.

More information can be found in "Variable Water Flow Pumping for Central Chilled Water Systems" [Liu, 2002], "Continuous CommissioningSM of a Central Chilled Water and Heating Hot Water System" [Deng et al. 2000a] and "Retrocommissioning of Central Chilled/Hot Water Systems" [Deng et al. 2002].

EXAMPLE:

The South Satellite Utility Plant on the Texas A&M campus has three 1,100-ton chillers that operate in parallel as shown in Figure 5-11. Three 75 hp. chiller pumps are connected to a common header. The chillers are connected to the campus secondary loop through a decoupler, and three 250 hp. pumps connect to the campus secondary loop.

This is a typical primary and secondary loop system. Each primary pump is interlinked with an individual chiller. When the chiller is on, the pump is on. The secondary pumps are controlled to maintain a required differential pressure at the exit of the plant. However, operators found that a significant amount of water flowed through the bypass line. Sometimes, chilled water bypassed the chillers. The supply water temperature to the buildings was too high and building comfort was not maintained. When chilled water bypassed the buildings, a significant amount of pump energy was wasted. As a fix, the central plant had an operator manually adjust the differential pressure set point to minimize the bypass flow to less than 200 gpm. This became a boring and costly task.

After conducting an engineering analysis, it was concluded that these problems could be solved by using a single loop operation. The major actions and procedures are listed below:

- The flow meter on the main loop and the flow meters on each chiller branch were calibrated. These flow signals were sent to the Building Automation System (BAS).
- All three chiller loops were balanced. All chillers have the same flow when they are on. All manual valves were opened 100% on each chiller branch.
- All three primary pumps were turned off

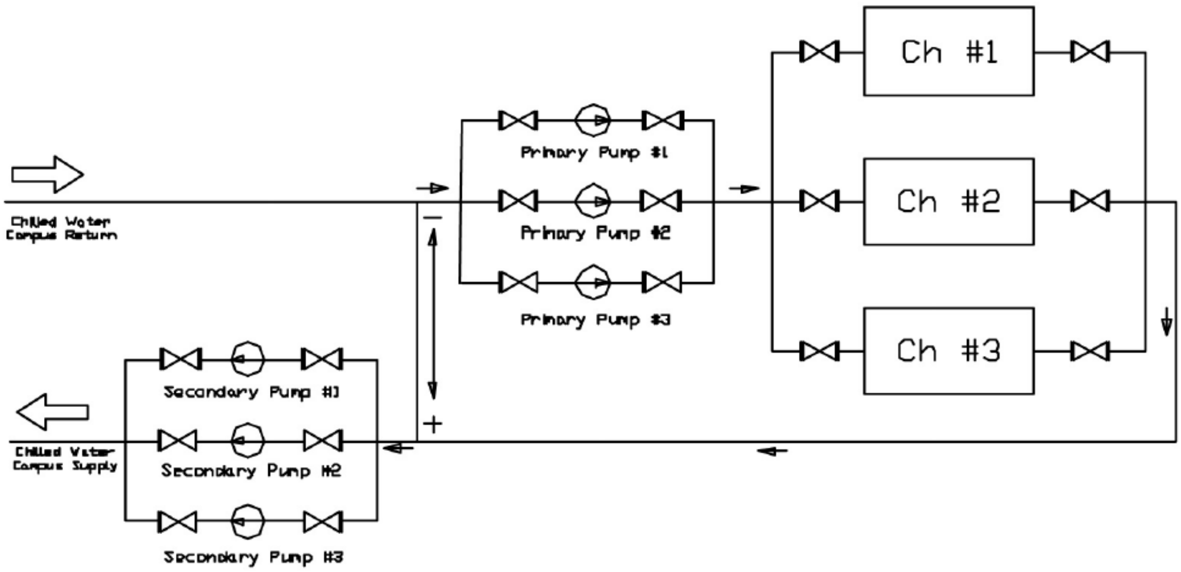

Fig. 11: Figure 5-11. Primary and Secondary Loop Configurations for the South Satellite Plant on the Texas A&M Campus

- The bypass valve was closed permanently
- The chilled water flow rate was maintained using the secondary loop pump.

When the loop differential pressure exceeds the set point by a certain value, one chiller is turned off. When the loop differential pressure drops below the set point by a certain value, one more chiller is turned on.

This change in loop operation maintained the plant supply water temperature at the required temperature (42°F). Hot calls decreased significantly. The new schedules were implemented in the existing BAS and the operators were liberated from a boring job of manual control. The pump power consumption was also reduced due to decreased chilled water bypass. The detailed calculation is summarized in Table 5-1.

Table 1: Table 5-1. Operational Characteristics of the South Plant with Primary-Secondary Pumping and with Secondary-Only Pumping

Chillers	Primary secondary Pumping			Secondary only Pumping			Savings
in Use	No. of Sp in Use	VFD Speed	Power Consumption	No. of SP in Use	VFD Speed	Power Consumption	
1	2	3	6X2+65=77	1	5	29	48
2	2	4	15X2+65X2=160	2	6	48X2=96	64
3	3	4	15X3+65X3=260	3	6.5	60X3=180	80

With the original operating scheme, measured pumping power consumption was as shown in the "power consumption" column under "Primary-secondary Pumping" in the table. It shows 65 hp. for each primary pump with smaller amounts for the secondary pumps. After conversion to secondary only pumping, the primary pump consumption was eliminated with secondary pump power shown in the "power consumption" column under secondary-only pumping. We see that pumping savings range from 48 hp. for single chiller operation to 80 hp. for three chiller operation. Removing the primary pumps can save more pump power. However, the flow resistance across the pump is very small when the pump is rotating freely.

For more details on this example, see "Installing Chiller Isolation Valves and Staging Chillers Under EMCS-A Case Study" [Deng et al. 2000b].

5.4 5.4 Other Tips

Check the expansion tank frequently and ensure it maintains a positive pressure for the entire system and does not over-pressurize the system.

Supply water temperature reset has a significant impact on the differential pressure set point. The differential pressure reset schedule should consider the impact of the temperature reset schedules. Typically, the temperature reset schedule should limit the chilled water flow below 60%. When the water flow is higher than 60% of the design value, the temperature reset significantly increases the pumping power.

Frequently check the make-up water to identify any leakage. Make-up water costs money but more importantly, it also causes corrosion and fouling in coils.

References

Deng, S., M. Liu and W.D. Turner, 1998. "Reducing Pump Power Consumption by (1000 kW) 40% Through Improved Pump Management in a Central Plant." 33rd Intersociety Engineering Conference on Energy Conversion, Colorado Springs, Colo., August 2-6, pp. 1428-1434.

Deng S., W. D. Turner, T. Batten, and M. Liu, 2000a. "Continuous CommissioningSM of a Central Chilled Water and Heating Hot Water System," Proc. Twelfth Symposium on Improving Building Systems in Hot and Humid Climates, May 15-16, San Antonio, Texas, pp. 199-206.

Deng S., W. D. Turner, and M. Liu, 2000b. "Commissioning of a Large Office Building in Texas - A Case Study," Proc. Twelfth Symposium on Improving Building Systems in Hot and Humid Climates, May 15- 16, San Antonio, Texas, pp. 146-153.

Deng S., W. D. Turner, D. E. Claridge, M. Liu, H. Bruner, H. Chen, and G. Wei, 2002, "Retrocommissioning of Central Chilled/Hot Water Systems," ASHRAE Transactions - Research, Vol. 108, Part II, pp. 75-81.

Liu, M., Zhu, Y., Powell, T., and Claridge, D.E., 1998 . "System Optimization Saves $195,000/yr. in a New Medical Facility," Proceedings of the 6th National Conference on Building Commissioning, Lake Buena Vista, FL, May 18-20, pp. 14.2.1-14.2.11.

Liu, M., 2002, "Variable Water Flow Pumping for Central Chilled Water Systems." ASME Journal of Solar Energy Engineering, Vol. 124, pp. 300-304.

Zhu Y, M. Liu, T. Batten, W. D. Turner, D. E. Claridge, B. Keeble, C. Cameron, and R. Hirchak, 2000. "A Simple and Quick Chilled Water Loop Balancing for Variable Flow Systems," Proc. Twelfth Symposium on Improving Building Systems in Hot and Humid Climates, May 15-16, San Antonio, Texas, pp. 158-162.

CHAPTER 6

Ch 6: CCSM Measures for Central Chiller Plants

The central chiller plant includes chillers, cooling towers, a primary water distribution system and the condenser water distribution system. Although a secondary pumping system may be physically located inside the central plant, commissioning issues dealing with secondary loops are discussed in Chapter 5. The central chiller plant produces chilled water using electricity, steam, hot water or gas. The detailed commissioning measures vary with the type of chiller. This chapter is designed to give the general commissioning measures that apply to a typical central cooling plant that can produce significant energy savings.

6.1 Use the Most Efficient Chillers

Most central chiller plants have several chillers with different performance factors or efficiencies. The differences in performance may be due to the design, performance degradation, age or operational problems. One chiller may have a higher efficiency at a high load ratio while another may have a higher efficiency at a lower load ratio. Running chillers with the highest performance can result in significant energy savings and will also reduce the number of complaints because you will be providing the greatest output for the least input.

The poorer-performing chillers are often older chillers that cannot produce rated capacity and often require more maintenance. However, an old chiller sometimes operates at the manufacturer's design efficiency, while others will be 20% lower. Measurement of actual chiller performance is very important.

Notes

Running chillers with the highest performance can result in significant energy savings … because you will be providing the greatest output for the least input.

The chiller performance measurement involves measuring chilled water production and energy input. The chilled water production can be determined from measured chilled water flow and supply, and return water temperatures. Flow measurement is extremely important. Do not assume the flow is the design flow if there is a constant speed pump in the chiller primary loop. This should be measured and can be measured accurately and non-intrusively using an ultrasonic meter. To reduce the measurement error, chilled water supply and return water temperatures should be measured using the same sensor. This is a place where a 1°F error in the temperature difference (typically around 10°F) makes an error of approximately 10% in the cooling output determined for the chiller. When existing meters are used, they should be calibrated before the measurement. The power consumption should be determined from a true power measurement instead of using percent of design current as a surrogate for power measurement. The power factor varies significantly as the chiller load changes. The true power fraction is often significantly smaller than the current fraction under partial load. The true power can be measured using an appropriate power meter. The chiller control panel number may be used if it measures true power.

Once the chilled water flow rate, supply, and return chilled water temperatures and chiller electricity consumption have been measured, the kW/ton can be determined using the following formula:

kW/ton = 24kW/[gpm(Treturn – Tsupply)]

Record all other important operating parameters such as condenser supply and return temperatures and flow. When you compare performance of multiple chillers, ensure that the condenser water and chilled water temperatures are similar for all chillers.

For turbine-driven and absorption chillers, the performance is evaluated using the ratio of the tonnage over the heat input. Both heat input and chilled water production should use the same units. The appropriate manual should be consulted for accurate techniques to use for measuring steam and gas energy.

6.1 6.2 Reset the Supply Water Temperature

Increasing the chilled water supply temperature can decrease chiller electricity consumption significantly. The general rule-of-thumb is that a one-degree Fahrenheit increase corresponds to a decrease in compressor electricity consumption of 1.7%. The chilled water supply temperature can be reset based on cooling load or ambient conditions.

- Increase the chilled water temperature linearly from the design value up to 5°F (adjustable) higher than the design value as the chiller plant load decreases from 100% to 40%. Or,
- Increase the chilled water temperature from the design value up to 5°F (adjustable) higher than the design value as the ambient temperature decreases from the design value to 60°F. Or,
- Adjust the chilled water temperature within a range of 45°F to 50°F using the valve position. If the maximum open valve in the primary chilled water loop is less than 90-95% open, increase the chilled water supply temperature. If more than one valve is 100% open, decrease the chilled water supply temperature.
- The supply water temperature reset can be implemented using the building or plant automation system (BAS). If it cannot be implemented using the BAS, the operator can reset the set point daily, based on the daily maximum temperature.

Increasing chilled water temperature may increase distribution pump (secondary pump) power consumption. The secondary chilled water flow should be less than 60% of the design flow rate before implementing the chilled water supply temperature reset. Supply temperature reset should not increase it above this level.

Notes

Most building chilled water pumping systems are equipped with variable speed devices (VSDs). If a VSD is not installed, retrofit of a VSD is generally recommended.

The chilled water supply temperature reset directly impacts the dehumidification capability of the coils. The chilled water supply temperature should not be reset to a higher value until the ambient humidity ratio is less than 0.009 or the ambient dew point temperature is less than 57°F for typical facilities.

6.2 6.3 Reset Condenser Return Water Temperature

Decreasing cooling tower return water temperature has the same effect as increasing the chilled water supply temperature. The cooling tower return temperature should be reset based on weather conditions. The following provides general guidelines:

- The cooling tower return water temperature set point should be at least 5°F (adjustable according to design) higher than the ambient wet bulb temperature. This prevents excessive cooling tower fan power consumption.
- The cooling tower water return temperature should not be lower than 65°F for chillers made before 1999 and should not be lower than 55°F for newer chillers. It is also recommended you consult the chiller manufacturer's manual for more information.

The cooling tower return water temperature reset can be implemented using the BAS. If it cannot be implemented using the BAS, operators can reset the set point daily using the daily maximum wet bulb or dry bulb temperature.

Notes

Decreasing the cooling tower return temperature may increase fan power consumption. However, fan power may not necessarily increase with lower cooling tower return water temperature.

Decreasing the cooling tower return temperature may increase fan power consumption. However, fan power may not necessarily increase with lower cooling tower return water temperature. The following tips can help.

- Use all towers. For example, use all three towers when one of the three chillers is used. This may eliminate fan power consumption entirely. The pump power may actually stay the same. Ensure the other two tower pumps are off.
- Never turn on the cooling tower fan before the by-pass valve is completely closed. If the bypass valve is not completely closed, the additional cooling provided by the fan is not needed and will not be used. Save the fan power!
- Balance the water distribution to the towers and within the towers. Water is often seen flowing down only one side of the tower, or one tower may have twice the flow of another. This significantly increases the water return temperature from the towers.

EXAMPLE:

The University of Texas Medical Branch at Galveston has a conditioned area of 3,500,000 ft2. The campus is cooled by a large central plant with seven chillers having a total capacity of 19,400 tons. The plant metering at this site permits trending of major chiller plant operating parameters as shown in Figure 6-1. The figure shows the chiller kW/ton (open rectangle symbols) as a function of the average value of the chilled water supply (open triangle symbols) and return temperatures . When the average chilled water temperature was increased from 42.5°F to 44.5°F, the average kW/ton decreased from approximately 1.02 to approximately 0.90. The average condenser temperature remained at 75°F.

When the average chilled water temperature was increased from 44.5°F to 46°F, the chiller kW/ton increased from 0.9 to approximately 0.95. This increase is due to the increase of the average condenser water temperature. The condenser water temperature increased from 75°F to approximately 89°F. Improving the chilled water and cooling tower water temperature set points can significantly decrease the central plant power consumption.

Figure 6-2 presents the measured cooling tower return water temperature, improved return water temperature and ambient wet bulb as a function of ambient wet bulb temperature. When the wet bulb temperature is below 60°F, the water is returned from the cooling tower at about 73°F. For higher wet bulb temperatures, the water is about 10°F above the wet bulb temperature. This provided a significant opportunity to improve chiller efficiency by lowering the temperature of the supply water to the condenser to 65°F when the wet bulb is 60°F or below and keeping it approximately 5°F above the wet bulb at higher temperatures.

The measured chilled water supply temperature is shown in Figure 6-3. The plant was operating with a constant supply temperature of about 39°F. The supply temperature schedule suggested for implementation is also shown. It ranges from a high of 45°F when the ambient is about 40°F to a low of 41°F when the ambient is above 85°F.

The projected savings from implementing the supply temperature reset schedule and changing the cooling tower control was a 22% reduction from 0.92 kW/ton to 0.72 kW/ton on average. The historical plant electrical consumption

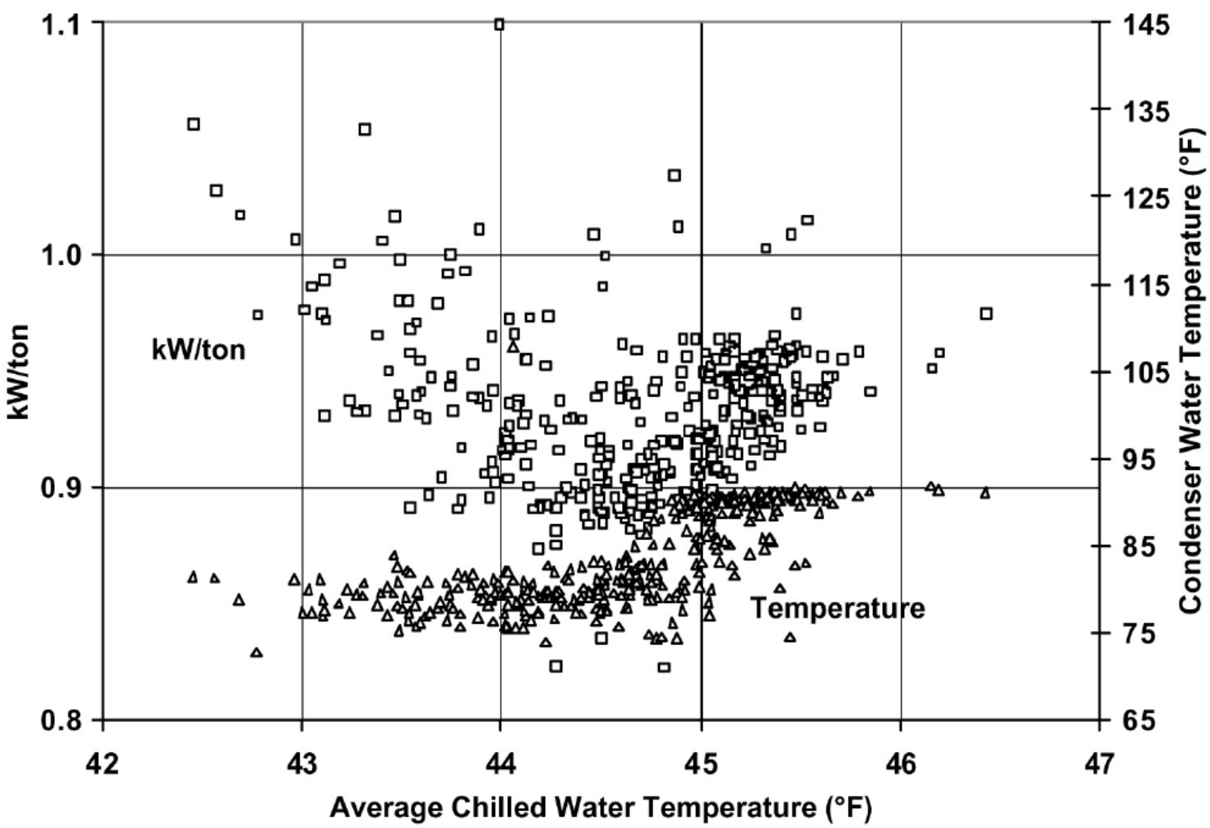

Fig. 1: Figure 6-1. Measured Chiller Plant Efficiency and Condenser Water Temperature Plotted as Functions of Average Chilled Water Temperature

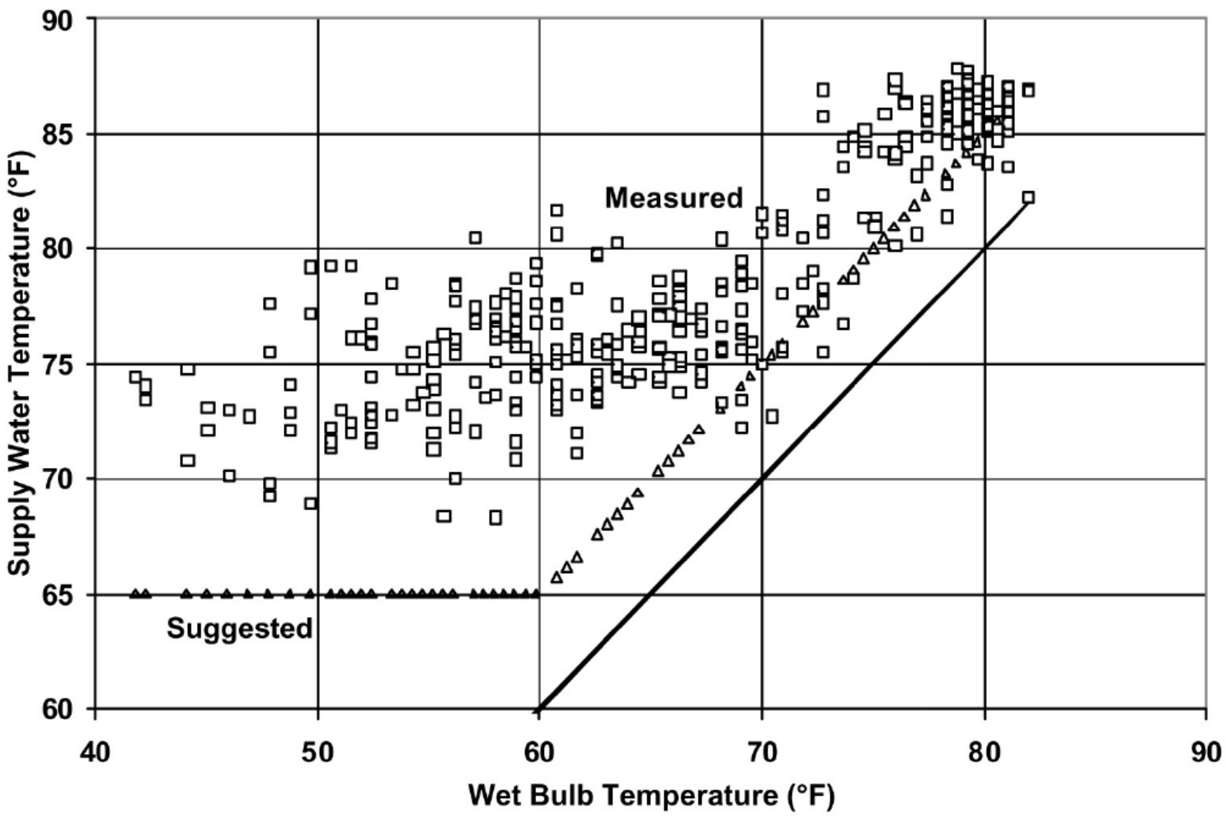

Fig. 2: Figure 6-2. Measured Condenser Supply Water Temperature at the UTMB Chiller Plant as a Function of Ambient Wet Bulb Temperature

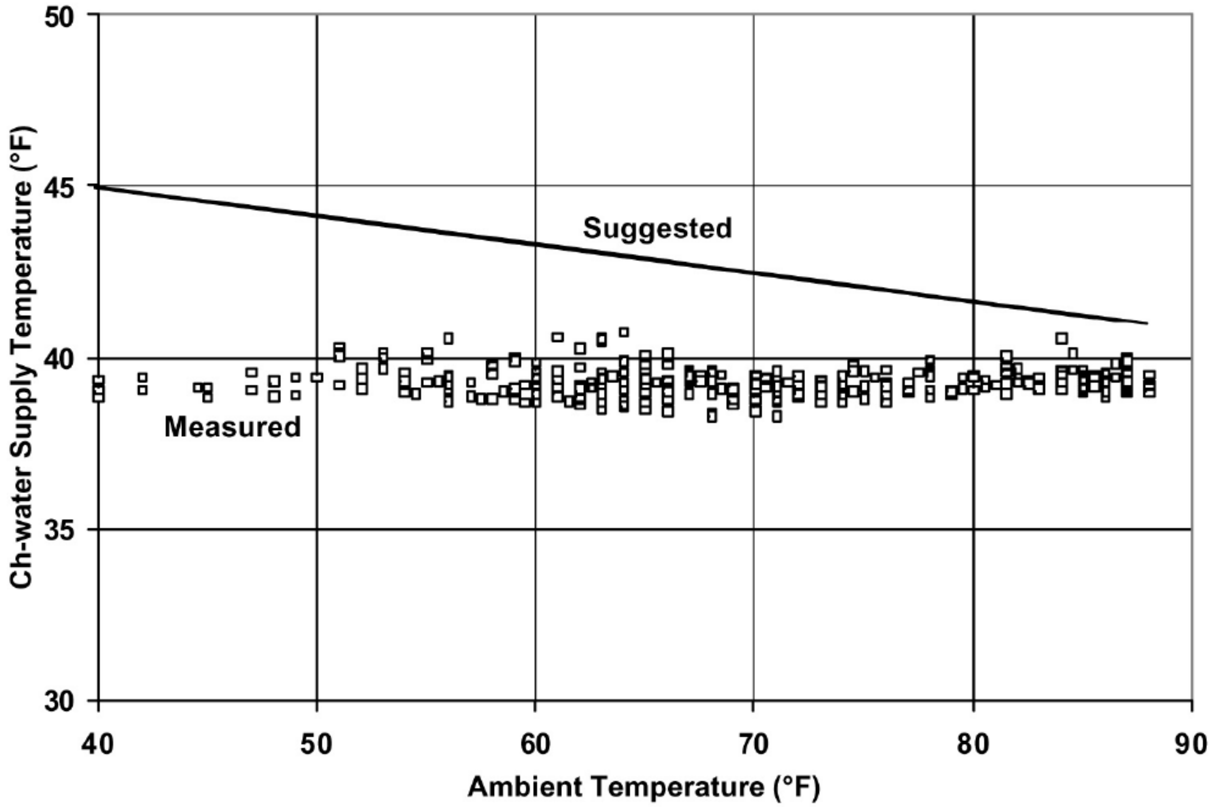

Fig. 3: Figure 6-3. Measured And Suggested Chilled Water Supply Temperature at UTMB

was 69,711 MkWh which was projected to be reduced to 54,489 MkWh for savings of 15,222 MkWh.

More information can be found in "Use of EMCS Recorded Data to Identify Potential Savings Due to Improved HVAC Operations and Maintenance," [Liu et al. 1997].

6.3 6.4 Increase Chilled Water Return Temperature

Increasing chilled water return temperature has the same effect as increasing chilled water supply temperature. It can also significantly decrease the secondary pump power because the higher the return water temperature (for a given supply temperature), the lower the chilled water flow. The following measures should be used to increase the chilled water return temperature.

- Maximize the chilled water return temperature by closing three-way valves. Three-way valves are often used in existing systems. Under partial load conditions, the chilled water flow rate can be higher than the design flow due to reduced resistance in the valve and coil sections. When a primary/secondary chilled water loop is used or a variable flow loop is used, these three-way valves should be closed. When a single chilled water loop is used, some of the three-way valves can be closed. The number of valves closed depends on the minimum allowable chilled water flow through the chiller.
- Solve existing water balance problems in the buildings to increase the chilled water return temperature
- Optimize the chilled water loop differential pressure set point. In most cases, the differential pressure set point is too high. The control valve often loses control and causes excessively low return water temperature. The optimal pressure set point is discussed in Chapter 5.

Maximizing chilled water return temperature is much more important than optimizing supply water temperature since it often provides much more savings potential. It is difficult to increase supply temperature 5°F above the design set point. It is often easy to increase the return water temperature as much as 7°F by conducting water balancing and shutting off by-pass and three-way valves.

EXAMPLE:

Mclnnis is a university campus building in central Texas. The design differential temperature is 12°F (42/54°F). The building experienced differential temperatures less than 10°F before the water loop balance and building commissioning. During the building commissioning, the chilled water bypass valves were completely closed, and the chilled water loop differential pressure was decreased from 30 psi to a range of 15 psi. Figure 6-4 presents the measured chilled water supply and return water temperatures from the building. The differential temperature was maintained above 15°F after commissioning.

6.4 6.5 Use Variable Flow under Partial Load Conditions

Typical central plants use primary and secondary loops. A constant speed primary pump is often dedicated to a particular chiller. When the chiller is turned on, the pump is on. Chilled water flow through each chiller is maintained at the design flow rate by this operating schedule. When the building-loop flow is less than the chiller loop flow, part of the chiller flow bypasses the building and returns to the chiller.

This practice causes excessive primary pump power consumption and low entering water temperature to the chiller which increases the compressor power consumption.

It is the general perception that the chilled water flows have to remain constant for chiller operational safety. Actually, most new chillers allow chilled water flow as low as 30% of the design value. The chilled water flow can be decreased as low as 50% for most existing chillers if the following procedures are followed:

- Adjust the flow switch first. The chiller will shut down if the flow switch sends a no-flow signal to the chiller controller. For existing chillers, the flow signal will not be generated until design flow is achieved. Adjust the flow switch and make it send a flow signal as soon as flow reaches 30% of design flow, or more if necessary

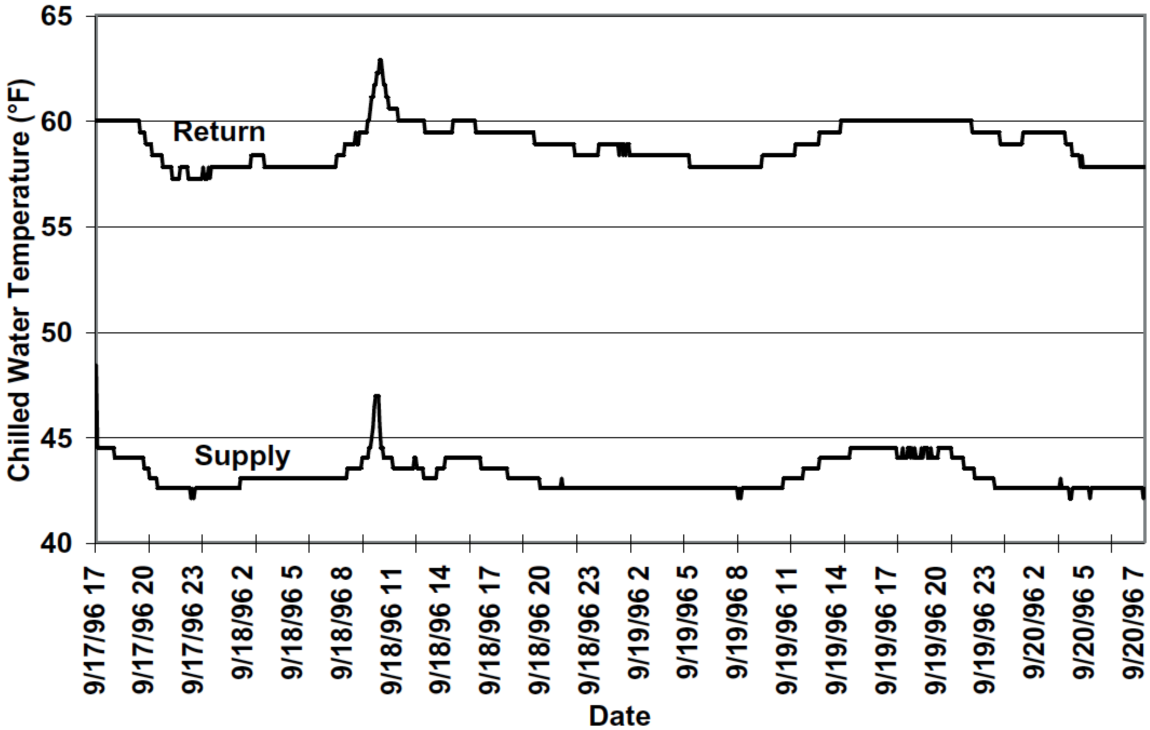

Fig. 4: Figure 6-4. Measured Chilled Water Supply and Return Water Temperature After Building Commissioning (the building differential temperature was less than 10°F before commissioning)

- Set a start-up and shut down cycle time of no less than 60 seconds for all pumps and valves. This will prevent sudden water flow changes. This is necessary for old chillers since most controls are very slow.

Varying chilled water flow can be implemented using the following procedures:

- Determine the minimum chilled water flow ratio for each chiller. Vary chilled water flow slowly through the chiller until the chiller shuts down or chilled water flow is reduced to 30% of the design rate. This flow is defined as the minimum flow rate. During the test, the chilled water return temperature should be maintained at the design level; 54°F, for example. The chilled water supply temperature should be set at the design level as well; 42°F, for example. The chilled water flow should be maintained at 30% or higher to prevent deposition of dirt and degradation of heat transfer.
- If the secondary loop flow rate is higher than the minimum flow rate of the chiller(s), close the building bypass valve. Keep the primary pumps on if a VSD is installed on the secondary loop pumps.
- If the secondary loop flow is less than the minimum flow rate of the chiller(s), modulate the bypass valve to maintain the minimum chilled water flow through chillers.

Notes

Varying chilled water flow through a chiller can result in significant pump power savings.

Varying chilled water flow through a chiller can result in significant pump power savings. Although the primary pumps are kept on all the time, the secondary pump power consumption is decreased significantly when compared to the conventional primary and secondary system operation. Figure 6-5 presents the potential pump power savings for a central plant with three identical chillers. If the central plant total load is 60% and the chilled water flow through each chiller is 60%, the primary pump power savings is 78%. If the design pump capacity is 100 kW for each pump, the total pump power savings would be 234 kW. If the central plant has a total load of 40%, two chillers operate at 30% load. The pump power savings ratio is 65%, or 195 kW. If the central plant has a total load of 20%, one chiller is operated at 60%. The pump power savings is 26%, or 78 kW.

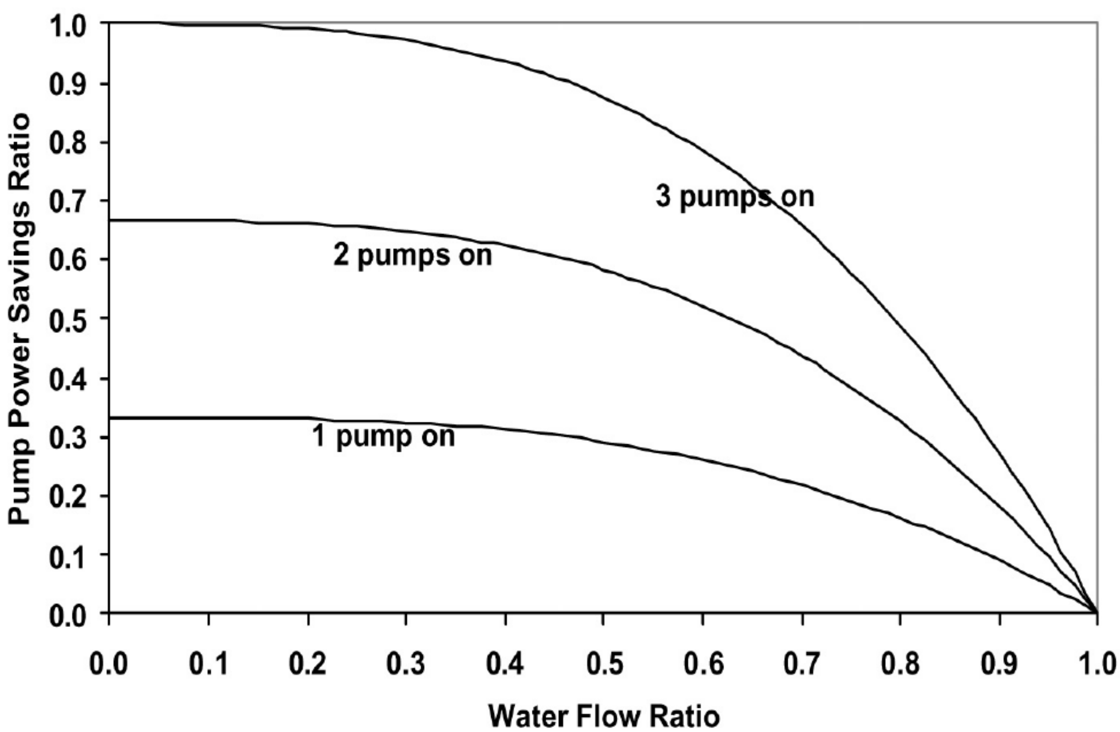

Fig. 5: Figure 6-5. Potential Primary Pump Power Savings Ratio Versus Chilled Water Flow Ratio through Each Chiller

Varying chilled water flow through the chillers will also increase the chiller efficiency when compared to constant water flow with chilled water bypass. More information can be found in "Variable Water Flow Pumping for Central Chilled Water Systems" [Liu 2002].

6.5 6.6 Optimize Chiller Staging

For most chillers, the kW/ton decreases (COP increases) as the load ratio increases from 40% to 80%. When the load ratio is too low, the capacity modulation device in the chiller lowers the chiller efficiency. When the chiller has a moderate load, the capacity modulation device has reasonable efficiency. The condenser and evaporator are oversized for the load under this condition so the chiller efficiency is higher.

When the chiller is at maximum load, the evaporator and condenser have a smaller load ratio, reducing the chiller efficiency below its maximum value. Running chillers in the high efficiency range can result in significant electrical energy savings and can improve the reliability of plant operation. The optimal chiller staging should be designed using the following procedures:

- Determine and understand the optimal load range for each chiller. This information should be available from the chiller manufacturer. For example, the kW/ton typically has a minimum value when the chiller load varies from 50% to 70% of the design value. However, the chiller system, which includes the chilled water pump and cooling tower fans, may not have the best efficiency when the pump and fans run at full speed.

- Turn on the most efficient chiller first. Optimize the pump and fan operation accordingly.
- Turn on more chillers to maintain the load ratio (chiller load over the design load) within the optimal efficiency range for each chiller. It is assumed that the building bypass is closed.

If the building bypass cannot be closed, the minimum chiller load ratio should be maintained at 50% or higher. In this case, the primary pump power consumption increases with the number of chillers in operation. Although the compressor power is decreased, the primary pump power increases significantly. The total power consumption is often higher if the chiller load is less than 50%.

A single loop may be used for some plants. In this case, a control schedule can be developed to share primary pumps under partial load conditions. For example, when the load is less than 50% for two chillers, a single pump can sometimes be used. If two pumps are used, the central plant may use approximately the same amount of energy as one chiller at peak load.

6.6 6.8 Maintain Good Operating Practices

It is important to follow the operating procedures recommended by the manufacturer. It is important to calibrate the temperature, pressure and current sensors and flow switches periodically. The temperature sensors are especially important for maintaining efficient operation. Control parameters must be set properly, particularly the time delay relay.

References

Liu, M., Y. Zhu and D. E. Claridge, 1997. "Use of EMCS Recorded Data to Identify Potential Savings Due to Improved HVAC Operations and Maintenance," ASHRAE Transactions-Research. Vol. 103, Part 2, pp. 122-129.

Liu, M., 2002. "Variable Water Flow Pumping for Central Chilled Water Systems." ASME Journal of Solar Energy Engineering, Vol. 124, pp. 300-304.

CHAPTER 7

Ch 7: CCSM Measures for Central Heating Plants

Central heating plants produce hot water, steam, or both, typically using either natural gas, coal or oil as fuel. Steam, hot water, or both are distributed to buildings for HVAC systems and other end uses, such as cooking, cleaning, sterilization and experiments. Figure 7-1 illustrates the major components in a steam boiler with a hot water converter and a hot water boiler. Boiler plant operation involves complex chemical, mechanical and control processes. Energy performance and operational reliability can be improved through numerous measures. However, the CCSM measures discussed in this chapter are limited to those that can be implemented by operating technicians, operating engineers and CCSM engineers.

7.1 7.1 Optimize Supply Water Temperature and Steam Pressure

Steam pressure and hot water temperature are the most important safety parameters for a central heating plant. Reducing the boiler steam pressure and hot water temperature have numerous benefits including:

- Improved plant safety
- Increased boiler efficiency and decreased source energy consumption
- Increased condensate return from buildings and improved building automation system performance. Most condensate tanks are open to mechanical rooms. When the steam pressure is decreased, secondary evaporation is significantly decreased and mechanical room relative humidity level is decreased. This also improves the humidity level of the compressed air provided to the pneumatic systems.
- Reduced hot water and steam leakage through malfunctioning valves. For example, 5% hot water leakage at 180°F carries five times more energy into the space than the same amount of water at 90°F.

To identify and implement the optimal steam or hot water temperature set point, the following procedures should be followed:

- Interview personnel who use steam or hot water in applications such as cooking, cleaning and medical uses to identify the highest steam or hot water temperature requirement. For example, a typical medical building may have a steam pressure requirement of 40 psig. It is often supplied at 110 psig or higher. Pressure set point adjustment should be performed with care. For small boilers, adjustment can be performed by simply adjusting the thermostat. For large boilers, adjustment involves complex control system and hardware work. The equipment manual should be consulted.

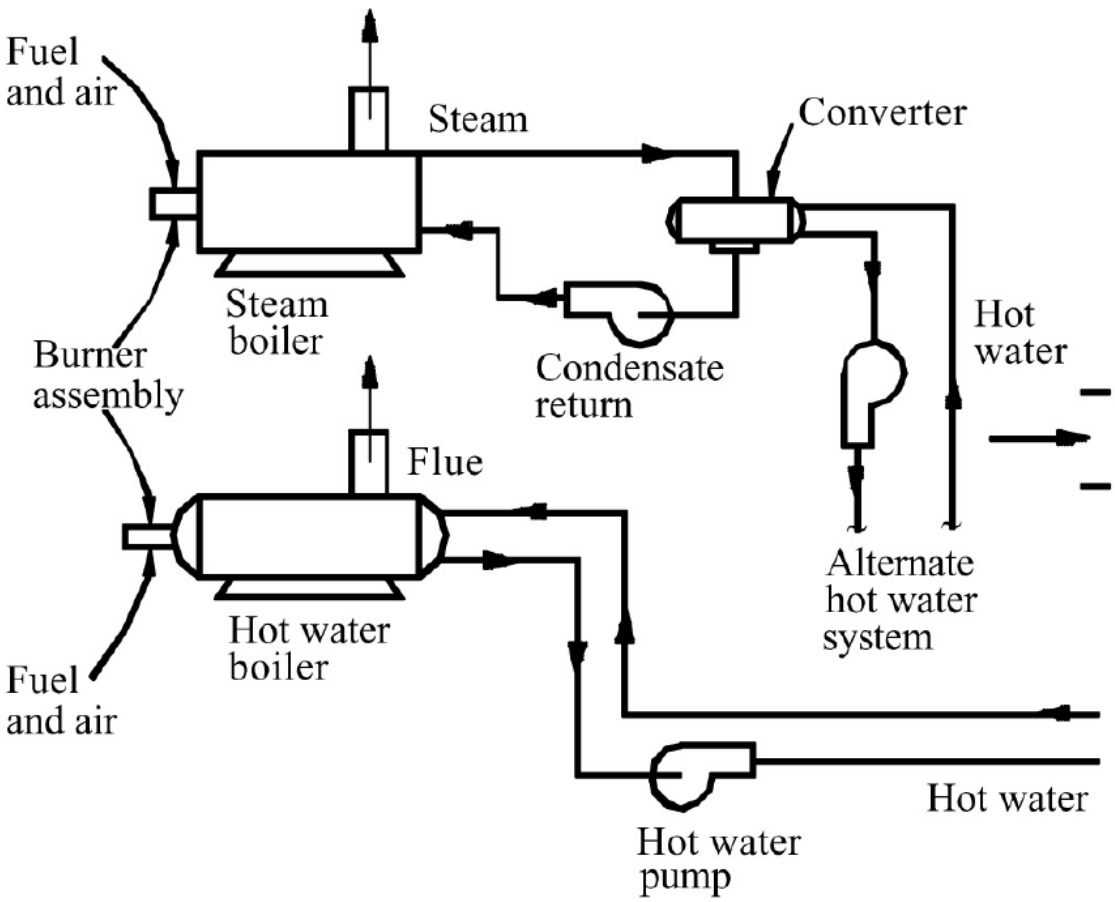

Fig. 1: Figure 7-1. Steam and Hot Water Boilers

- Identify the actual pressure loss through the pressure reducing valve (PRV). Some pressure loss is required to maintain stable operation. However, the pressure loss should be less than 10 psi.
- Ensure the plant steam pressure is the sum of the maximum required end use steam pressure and the steam loss of the distribution system. Likewise, the plant hot water temperature should be the sum of the maximum end use hot water temperature requirement and any temperature loss in the hot water loop.
- Implement the improved steam pressure and hot water temperature set point or reset schedules in the central heating plant
- Reset the hot water supply temperature and the steam pressure according to the ambient temperature, time of day, or other appropriate scheduling variable if building loads are the only requirement for the hot water or steam. The optimal temperature set point can be determined using a trial and error method. Reduce the hot water temperature until a cold call occurs or until the low temperature limit on the boiler is reached.

More information can be found in "Use of EMCS Recorded Data to Identify Potential Savings Due to Improved HVAC Operation and Maintenance" [Liu et al. 1997].

EXAMPLE:

A thermal energy central plant had two boilers that supply steam to the University of Texas Medical Center at Galveston (UTMB), a modern 3.5 million square-foot medical facility. The central plant provided steam at 150 psig during the summer and at 125 psig during the winter to the campus until CCSM engineers visited the campus in 1994. No one knew why the steam pressure was set at 150 psig during the summer and 125 psig during the winter. Questions about this practice received the response "This is the way we have always operated." In facility operations, practices often are simply the result of history and operating tradition.

A field survey determined that the maximum steam requirement was 45 psig. The steam pressure was reset to 125 psig right after the steam pressure problem was identified.

Figure 7-2 shows the boiler efficiency and steam pressure in a time series plot. It shows that the boiler efficiency varies from 0.55 to 0.75 and the steam pressure varies from 125 psi to 145 psi. When the steam pressure is approximately 125 psi, the boiler efficiency is approximately 0.70. When the steam pressure is near 145 psi, the boiler efficiency drops to values generally in the range 0.60-0.66.

Figure 7-3 shows the steam production and steam pressure as a time series. Relatively low steam production (about 32 MMBtu/hr) is observed in January 1994 and November and December 1993, when the boilers had relatively high efficiency. Clearly, the boilers can run at a high efficiency regardless of steam production if the steam pressure is at 125 psi.

Based on these observations, it is assumed that the boilers would have an average annual efficiency of 0.72 if operated at 125 psi which is the measured average efficiency in January 1994.

Table 7-1 presents actual annual steam production, gas consumption and boiler efficiency. The boilers produced 301,274 MMBtu/yr of steam with a total gas consumption of 437,563 MCF/yr from April 1, 1993, to March 30, 1994. The annual average boiler efficiency was 0.67. If the boiler efficiency is increased to 0.72 by setting steam pressure to 125 psi, the fuel required by the boilers would drop to 405,512 MCF/yr with the same steam production of 301,274 MMBtu/yr. Consequently, the potential annual gas savings were 32,051 MCF/yr, or 7% of the annual consumption. At a gas price of $2.57/MCF, the cost savings would be $82,000/yr.

Table 1: Table 7-1. Summary of Boiler Efficiency Analysis

–	Steam	gas	Efficiency	Savings
–	MMBtu/yr	MCF/yr		%
Current	301,274	437,563	0.67	
Improved	301,274	405,512	0.72	7

Note: Actual energy savings are higher than $82,000/yr since the steam pressure was reset to 100 psig instead of 125 psig.

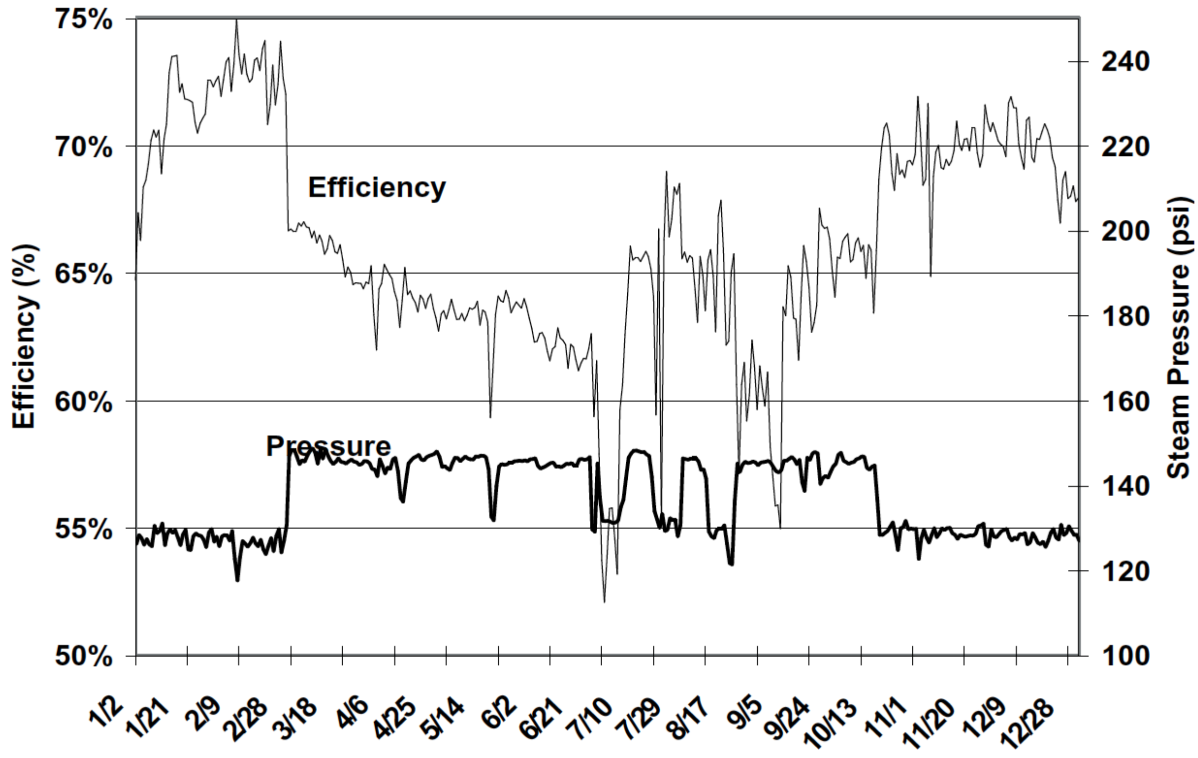

Fig. 2: Figure 7-2. Boiler Efficiency and Steam Pressure at the UTMB Galveston Boiler Plant

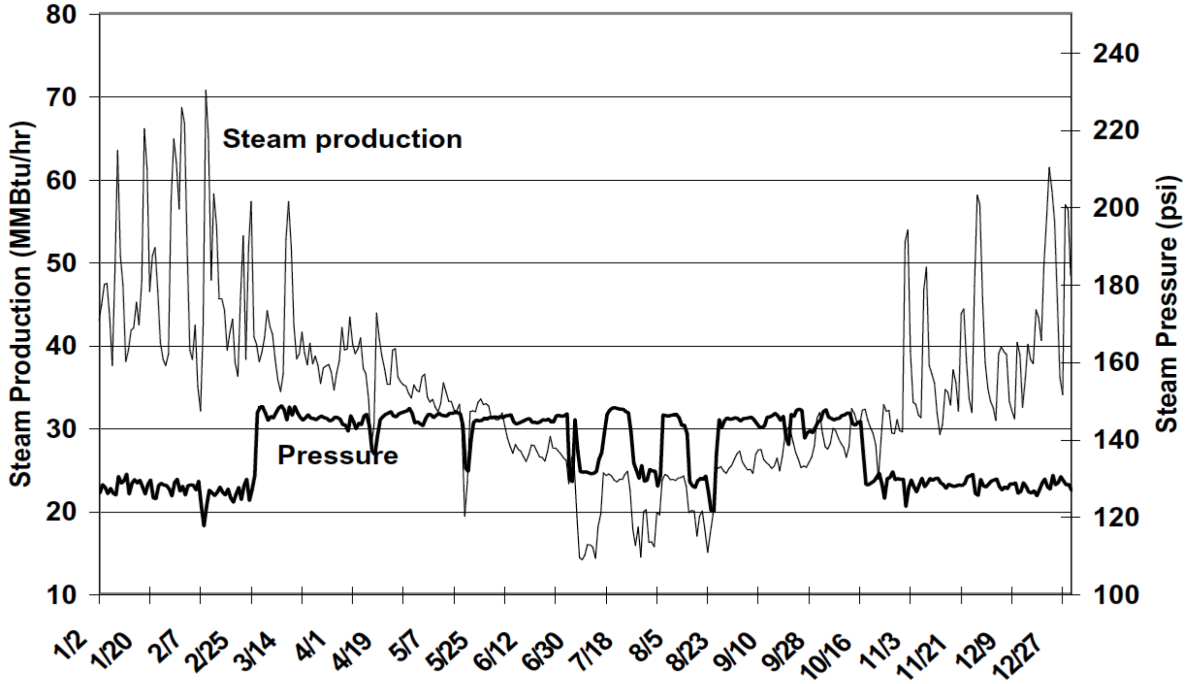

Fig. 3: Figure 7-3. Variation of Steam Consumption and Boiler Pressure with Time at UTMB

In 1995, the steam pressure was reset to 100 psig. In 1996, the operating staff tried to lower the steam pressure to 70 psig. However, the boiler shut down automatically because 70 psig was out of the controller range. Since 1996, the central plant has been providing steam pressure at 100 psig to the UTMB campus. The steam pressure reset improved plant safety and decreased gas consumption as well.

7.2 7.2 Optimize Feed Water Pump Operation

The feed water pump is sized based on boiler design pressure. Since most boilers operate below the design pressure, the feed water pump head is often significantly higher than required. This excessive pump head is often dropped across pressure reducing valves and manual valves. Installing a VSD on the feed water pump in these cases can decrease pump power consumption and improve control performance. The need for a VSD and the potential savings can be evaluated using the following procedures:

- Inspect the pressure-reducing valve (PRV) on the steam supply pipe. Measure the steam pressure loss across the PRV. The pressure loss should be limited to the manufacturer's recommended range. For a typical boiler, the pressure loss should be less than 20 psi. If the pressure is higher than the required value, adjust the PRV to reduce the pressure loss. Adjust boiler drum pressure set point.
- Identify the potential pressure head reduction. Open all partially open manual valves on the feed water line. Measure the pressure loss across the control valve and the pressure-reducing valve. The potential pump head reduction is the difference between the measured pressure loss across the PRV and the required PRV pressure loss (often less than 10 psi).
- Estimate the potential pump energy savings using annual steam/hot water production (G) and pump head reduction (Δh):

$$kWh = 0.0000512154 \frac{G(lbm/yr)\Delta h(psi)}{\eta}$$

- Install a VSD on the feed water pump and adjust VSD speed to maintain required boiler drum pressure.

Trimming the impeller or changing feed water pumps may also be feasible and the cost may be lower. However, the VSD provides more flexibility and can be adjusted to any level. Consequently, it maximizes the savings and can be adjusted to future changes as well.

EXAMPLE:

The thermal energy plant at UTMB Galveston, previously discussed, produced a total of 300,000,000 lbm of steam from April 1993 to March 1994. If the boiler steam pressure is decreased from 150 psi to 100 psi, the feedwater pump head reduction is 50 psi. The pump efficiency is approximately 0.8. If a VFD is installed on the feedwater pump, the annual pump power energy savings are 782 MWh. The savings are $39,000/yr at $0.05/kWh.

7.3 7.3 Optimize Airside Operation

The key issues are excessive air flow and flue gas temperature control. Some excess air flow is required to improve the combustion efficiency and avoid having insufficient combustion air during fluctuations in air flow. However, excessive air flow will consume more thermal energy since it must be heated from the outside air temperature to the flue gas temperature. The boiler efficiency decreases as excessive air flow increases. The flue gas temperature should be controlled properly. If the flue gas temperature is too low, acid condensation can occur in the flue. If the flue gas temperature is too high, it carries out too much thermal energy. The airside optimization starts with a combustion analysis, that determines the combustion efficiency based on the flue gas composition, flue gas temperature and fuel composition. The typical combustion efficiency should be higher than 80%. If the combustion efficiency is lower than this value, the following procedures can be used to determine the reasons:

- The flue gas temperature may be too high. Compare the measured flue gas temperature with the manufacturer's suggested flue gas temperature. If the flue gas temperature is higher than the suggested value, reduce the set point (for a large boiler system).
- Excess air is too high. If the volumetric oxygen content of the flue gas is higher than 3%, or the excess air flow is higher than 20%, more air than needed is being supplied to the boiler. Reduce the air flow to the boiler. Caution must be taken for small boilers without air modulation devices. In these cases, the measurement should be conducted under full load conditions. Otherwise, an engineer must conduct a calculation to convert to full load excess air flow.
- For small boilers, airflow modulation systems are seldom installed. The air flow is often set based on full load requirements. However, heating loads are often significantly overestimated due to the current design methods and practices. Few heating boilers ever operate at full load or nearly full load. To decrease energy consumption in these boilers, the air flow should be set based on the actual maximum boiler load. Set the boiler load limit at this value. This can significantly reduce the boiler loss associated with excessive air flow.
- Reduce/eliminate air leakage in air economizers. If a boiler has an economizer or heat recovery unit to preheat the combustion air, be sure there is no leakage between the two air streams. In an economizer, the outside air has a positive pressure (up to 3 in. H2O) while the flue gas is under negative pressure. Outside air can easily leak through any physical cracks or holes as a result of the high pressure difference. When air bypasses the boiler directly to the flue gas, it causes high flue gas oxygen levels and low flue gas temperatures. Consequently, the control system may reduce air flow lower than required for complete combustion and produce dangerous CO. The combustion efficiency also decreases significantly.

More information can be found in "In-situ Calibration of Boiler Instrumentation Using Analytical Redundancy" [Wei et al. 2001].

EXAMPLE:

Boiler 9 in the central utility plant on the Texas A&M campus was installed in 1962 with a rated capacity of 175,000 lbs/hr. The electrically driven feed water pump has a capacity of 468 gpm (234,300 lbs/hr). The supply air fan has a capacity of 63,750 cfm. A VFD is used to control the fan speed to accommodate the load changes. The boiler is equipped with an air preheater to warm the combustion air using the flue gas. A superheater heats the steam after the steam leaves the drum (see Figure 7-4 for details).

Boiler 9 is equipped with extensive metering to measure the boiler production and evaluate the performance. The key parameters measured are boiler feedwater flow and temperature, steam pressure and temperature, steam production, natural gas consumption, air flow, flue gas temperature and O2 level. These data are recorded by a Westinghouse control system.

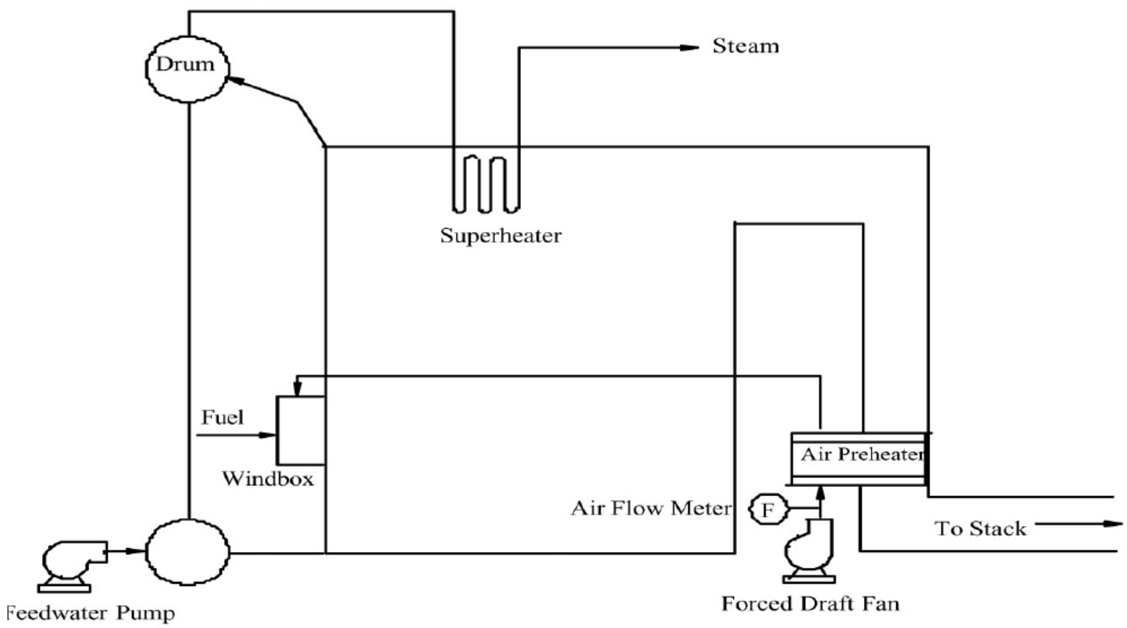

Fig. 4: Figure 7-4. Schematic Diagram of Boiler 9

The preheater for the incoming combustion air is a Ljungstrom-type air preheater shown schematically in Figure 7-5. A heavy metal wheel rotates slowly to transfer heat from the flue gas to the incoming air. The static pressure on the incoming air side is much higher than that on the flue side. Unfortunately, this leads to significant air bypass and consumes unnecessary fan power and causes other problems. When the incoming air bypasses to the flue gas side, the flue gas temperature decreases. The lower stack temperature, due to air leakage, gives a false indication that the boiler is operating efficiently.

The air leakage was determined using both O2 and CO2 measurements before and after the pre-heater. Three flue gas analyses were taken: one before the air preheater, one at the stack outlet and one right after the air preheater. The oxygen concentration was much higher following the preheater than at the stack outlet, indicating the gas sample following the preheater was not thoroughly mixed. Thus, the gas composition at the stack outlet was used to calculate the air leakage rate. The leakage rate was found to be 27% using an oxygen balance method and 29% using a carbon dioxide balance method. Table 7-2 is a summary of the gas analysis before and after the air preheater.

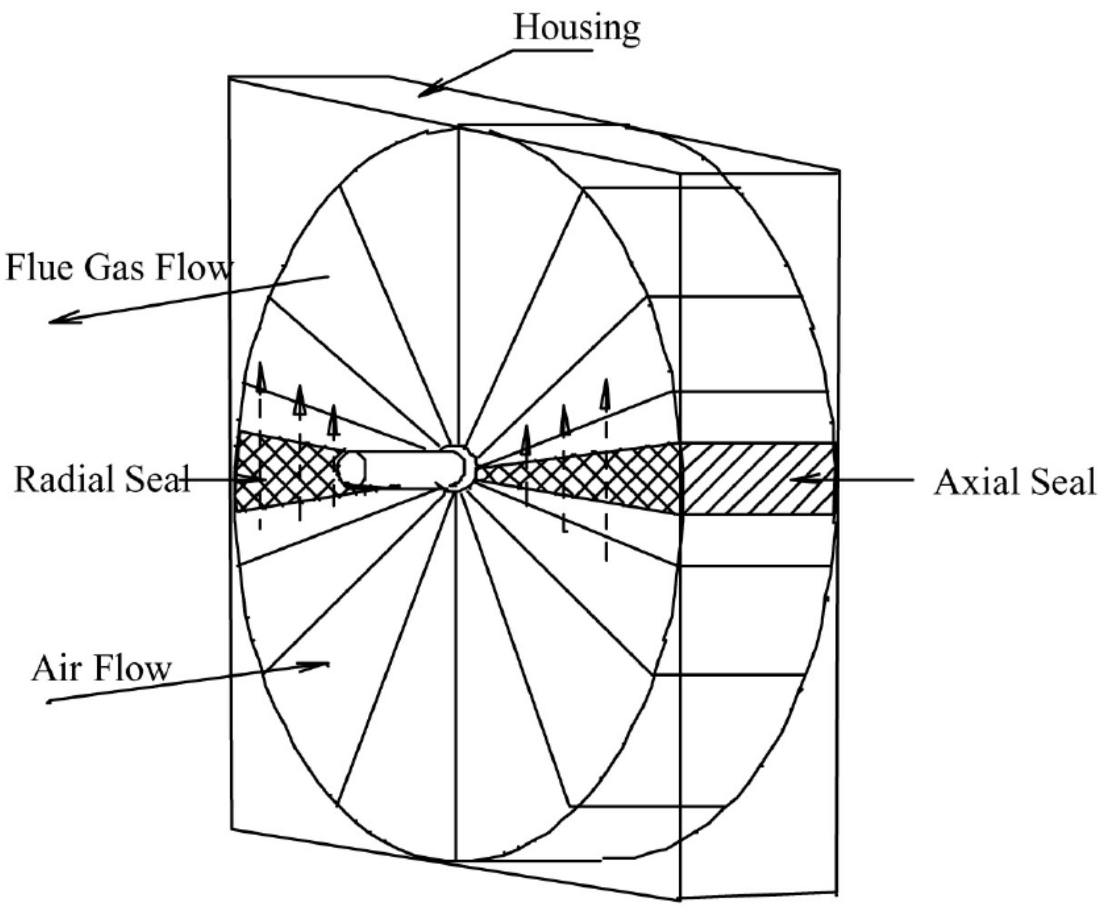

Fig. 5: Figure 7-5. Schematic of Regenerative Air Preheater

Table 2: Table 7-2. Summary of Gas Analysis Before and After the Air Preheater

–	Before air preheater	After air preheater	Ambient air	Leakage rate
O2 concentration	4.6%	8.9%	21%	27%
CO2 concentration	9.1%	6.5%	0%	29%

If the air leakage rate was reduced from 27% to 10%, it would result in the following savings, assuming an average load ratio of 70%:

- Fan power savings: 579,000 kWh/yr or $29,000/yr at $0.05/kWh
- Thermal energy savings due to reduced air leakage: 80,700 MMBtu/yr or $202,000/yr at $2.5/MMBtu.
- Gas savings due to increased efficiency (2%): 21,000 MMBtu/yr or $52,600/yr at $2.5/MMBtu.

The total annual cost savings is $282,600/yr.

7.4 7.4 Optimize Boiler Staging

Most central plants have more than one boiler. Using optimal staging can improve plant energy efficiency and reduce maintenance cost. The optimal staging should be developed using the following guidelines:

- Measure boiler efficiency. The boiler efficiency should be determined using thermal energy production, fuel consumption and fuel higher heating value (HHV). The following parameters should be measured:
  - Hot water or steam production (lbm/hr), m
  - Water enthalpy at entrance of the boiler system (Btu/lbm), h0
  - Water/steam enthalpy at the exit of the boiler system (Btu/lbm), he
  - Fuel consumption (lbm/hr), mf

$$\eta = \frac{m(h_0 - h_e)}{m_f HHV}$$

If the boiler efficiency cannot be determined using this method, combustion efficiency may be used. Combustion efficiency can be determined easily using a combustion analyzer.

- Run the higher efficiency boiler as the primary system and run the lower- efficiency boiler as the backup system.
- Avoid running any boiler at a load ratio less than 40% or higher than 90%
- If two boilers are running at average load ratios less than 60%, no standby boiler is necessary. If three boilers are running at loads of less than 80%, no standby boiler is necessary.

Boiler staging involves boiler shut-off, start-up and standby. Because of the large thermal inertial and temperature changes between shut-off, standby and normal operation, precautions must be taken to prevent corrosion and expansion damage. Generally speaking, short-term (monthly) turn on/off should be avoided for steam boilers. Hot water boilers are sometimes operated to provide water temperatures as low as 80°F. This improves distribution efficiency, but may lead to acid condensate in the flue. The hot water temperature must be kept high enough to prevent this condensation.

7.5 7.5 Improve Multiple Heat Exchanger Operation

Heat exchangers are often used in central plants or buildings to convert steam to hot water or high temperature hot water to lower temperature hot water. If more than one heat exchanger is installed, use as many heat exchangers as possible provided the average load ratio is 30% or higher. This approach provides the following benefits:

- Lower pumping power. For example, if two heat exchangers are used instead of one under 100% load, the pressure loss through the heat exchanger system will be decreased by 75%. The pumping power will also be decreased by 75%.
- Lower leaving temperature on the heat source. The condensate should be super-cooled when the heat exchangers are operated at low load ratio. The exit hot water temperature will be lower than the design value under the partial load condition. This will result in less water or steam flow and more energy extracted from each pound of water or steam. For example, the condensate water may be sub-cooled from 215°F to 150°F under a low heat exchanger load. Compared with leaving the heat exchanger at 215°F, each pound of steam delivers 65 Btu more thermal energy to the heat exchanger.

Using more heat exchangers will result in more heat loss. If the load ratio is higher than 30%, the benefits mentioned above normally outweigh the heat loss. More information can be found in "System Optimization Saves $195,000/yr in a New Facility" [Liu et al. 1998].

7.6 7.6 Maintain Good Operating Practices

Central plant operation involves energy efficiency and safety issues. Proper safety and maintenance guidelines should be followed. The following maintenance issues should be carefully addressed:

Notes

Central plant operation involves energy efficiency and safety issues. Proper safety and maintenance guidelines should be followed.

- Blowdown: Check blowdown setup if a boiler is operating at partial load most of the time. The purpose of blowdown is to remove the mineral deposits in the drum. The mineral deposit is proportional to the make-up water which is then proportional to the steam or hot water production. The blowdown can often be set back significantly. If the load ratio is 40% or higher, the blowdown can be reset proportional to the load ratio. If the load ratio is less than 40%, keep the blowdown rate at 40% of the design blowdown rate.
- Steam traps: Check steam traps frequently. Steam traps still have a tendency to fail, and leakage costs can be significant. A steam trap maintenance program is recommended. Consult the manufacturer and other manuals for proper procedures and methods.
- Condensate return: Inspect the condensate return frequently. Ensure as much condensate is returned as possible. This is very expensive water. It has high energy content and is treated water. When condensate is lost, make-up water, chemicals, fuel, and in some cases sewage costs, must be paid.

References

Liu, M., Y. Zhu and D. E. Claridge, 1997. "Use of EMCS Recorded Data to Identify Potential Savings Due to Improved HVAC Operations and Maintenance," ASHRAE Transactions-Research, Volume 103, Part 2, pp. 122-129.

Liu, M., Y. Zhu, T. Powell and D.E. Claridge, 1998. "System Optimization Saves $195,000/yr. in a New Medical Facility," Proceedings of the 6th National Conference on Building Commissioning, Lake Buena Vista, Fla., May 18-20, pp. 14.2.1-14.2.11.

Wei, G., M. Liu and D. E. Claridge, 2001. "In-situ Calibration of Boiler Instrumentation Using Analytical Redundancy," International Journal of Energy Research, Vol. 25, pp. 375-387.

CHAPTER 8

Ch 8: CCSM Measures for Thermal Storage Systems

Thermal storage systems are intended to meet part or all of the cooling needs of a building during specified periods of peak demand. It is essential they have sufficient capacity to provide the cooling during these periods. Figure 8-1 presents the schematic diagram of a thermal storage system. During off-peak hours, part of the cooling energy produced by the chillers is used for charging the storage systems that generally store cold water or ice. During peak hours, buildings receive cooling energy from the tank and/or chillers.

Thermal storage systems save costs by decreasing peak electrical demand during the electric utility's peak demand window. However, the billed demand may also depend on off-peak demand. Therefore, the cost savings strongly depend on the utility rate structure. For example, if there is a ratchet on the demand, a single system failure during the summer peak can largely eliminate the savings produced by the system for the entire year. Although optimizing thermal storage operation often requires special considerations based on each case, several CCSM measures apply to any type of thermal storage system. This chapter only discusses these measures.

8.1 8.1 Maximize Building Return Water Temperature

Low return water temperature is the primary problem with thermal storage systems. A low return water temperature prevents the chiller from operating at full load. Consequently, the chiller may not be able to charge the tank during off-peak hours. A low return water temperature can also significantly reduce the storage capacity of the system. For example, if the building return temperature is 50°F instead of 54°F, the thermal storage capacity will be decreased by 33% if the design temperature difference is 12°F. To increase the return water temperature, the following procedures should be followed:

- Implement building loop and secondary loop CCSM measures. The key measures are repeated below:
  - Close three-way valves on the cooling coils
  - Close building by-pass to prevent supply water from flowing directly back to the plant
  - Optimize building loop and secondary loop differential pressure set points
  - Ensure supply air temperatures are set properly and control valves are functioning properly. Any excessively low supply air temperature set points can cause low return water temperatures. For example, setting the air temperature at 50°F instead of 55°F will cause the control valve to open fully if the chilled water supply temperature is 42°F. The return temperature will be well below 54°F.

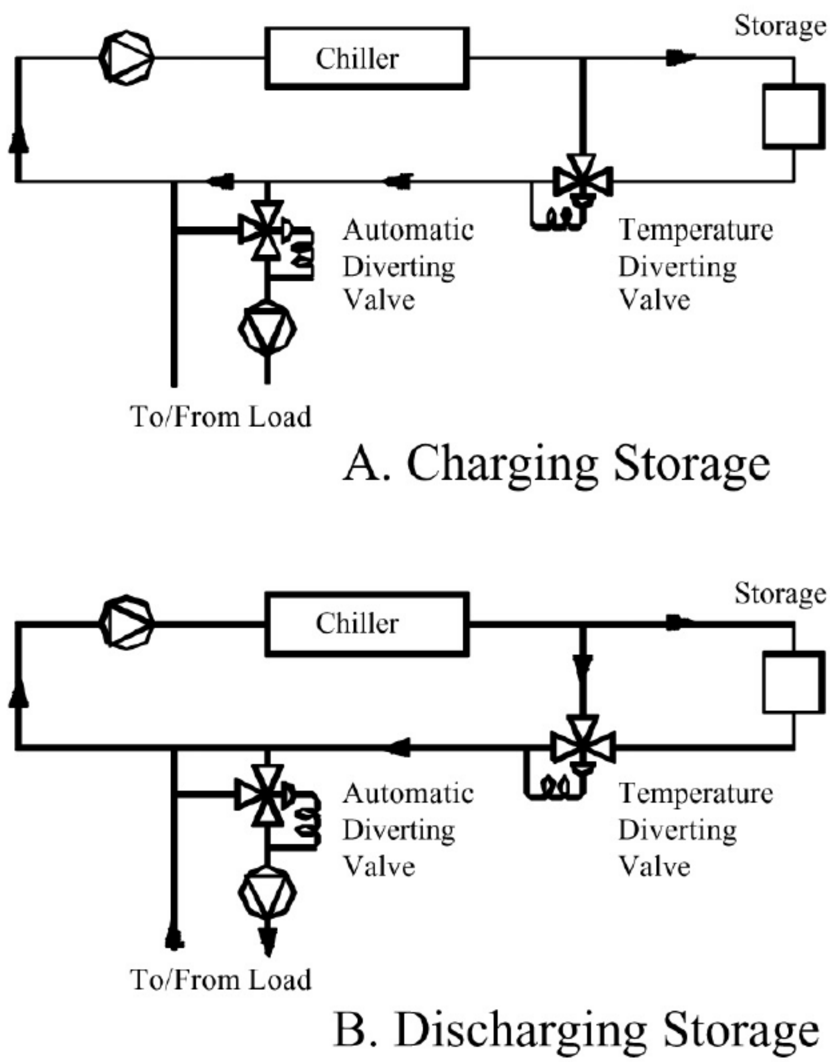

Fig. 1: Figure 8-1. Thermal Storage System in Charging Mode and in Discharge Mode

- Use a blending station to control return water temperature if the building water balance problems cannot be solved in a timely manner. This blends part of the building return water with primary cooling water before sending it to the building. Using the blending station causes extra pump power consumption and may also cause humidity control problems. As soon as the water balance is improved, the blending station should be disabled.

8.2 8.2 Improve Chilled Water Flow Control Through Chillers

It is important to control the chilled water flow at the design value. If the chilled water flow is lower than design flow, the chiller will not be able to produce full capacity. The tank may not be fully charged as a result. If the chilled water flow is higher than the design flow value, the chilled water supply temperature will not control at the required value. The tank capacity can be decreased significantly. For example, if the chilled water supply temperature is 44°F instead of 42°F, the tank capacity is decreased by 16% based on a design temperature difference of 12°F. To improve the chilled water flow control through chillers, the following guidelines are given:

Notes

It is important to control the chilled water flow at the design value. If the chilled water flow is lower than design flow, the chiller will not be able to produce full capacity.

- If cooling is not provided to the building at night, the designated loop config- uration should be determined. Balance the system to allow design chilled water flow through chillers and tanks at night.
- If cooling is required at night, use a lower loop differential pressure set point for the building loop at night. Reset cold air supply temperature to a higher value if that is acceptable.
- Modulate tank charging pump to maintain a constant chilled water flow through the chiller. This requires special handling on a case by case basis.

EXAMPLE:

A thermal storage system was installed in a 37,000 sq.ft. county hospital in Monahans, Texas, in the early 1990s. The hospital has a 120-ton chiller and the thermal storage system included a 80,675 gallon above ground chilled water storage tank shown in Figure 8-2.

The waterside of the chiller/thermal storage system is shown in Figure 8-3. The figure shows the system in the tank discharge mode with the chiller isolated from the loop. The tank pump shown has a 125 gpm capacity, while the design flow rate in the building loop is 180 gpm. The chiller pump capacity is 228 gpm. The building has 12 small roof top units to serve the administrative areas. Two control valves are used for the cooling coil. The patient rooms were conditioned by 32 fan-coil units connected to the chilled water loop through three way valves.

The thermal storage system had problems from the beginning and appeared to have inadequate storage capacity, and/or chiller capacity, to handle the cooling requirements of the hospital during the noon to 8 p.m. peak window. The chiller had to be brought on during some of the hottest days. Figure 8-4 presents the hourly whole building electricity consumption of the hospital plotted from midnight on the left through the 24 hours of one day on the right. The first line visible at the front of the figure is the data for July 31. The second line the data for July 30, etc., with two months of data shown in the figure. The figure shows that on July 31, consumption through the early hours of the morning was somewhat above 200 kW, rising above 250 kW by noon, and dropping to levels near 100 kW until around 8:00 p.m. when it increased to about 200 kW. Hence, the thermal storage system had operated properly and the chiller had not been operated during the peak demand window from noon to 8 p.m. However, many days are visible when the chiller operated during this window. The days when the chiller operated continuously were generally weekend days when there was no penalty for operation during this period. The operators were trying to get the storage fully charged so they would be able to handle the loads during the week. However, several lines are visible where it appears the chiller went off, then came back on in the late afternoon. In these cases the system was not operating as designed.

Field measurements identified the following problems:

Fig. 2: Figure 8-2. Chilled Water Storage Tank at the Hospital in Monahans, Texas

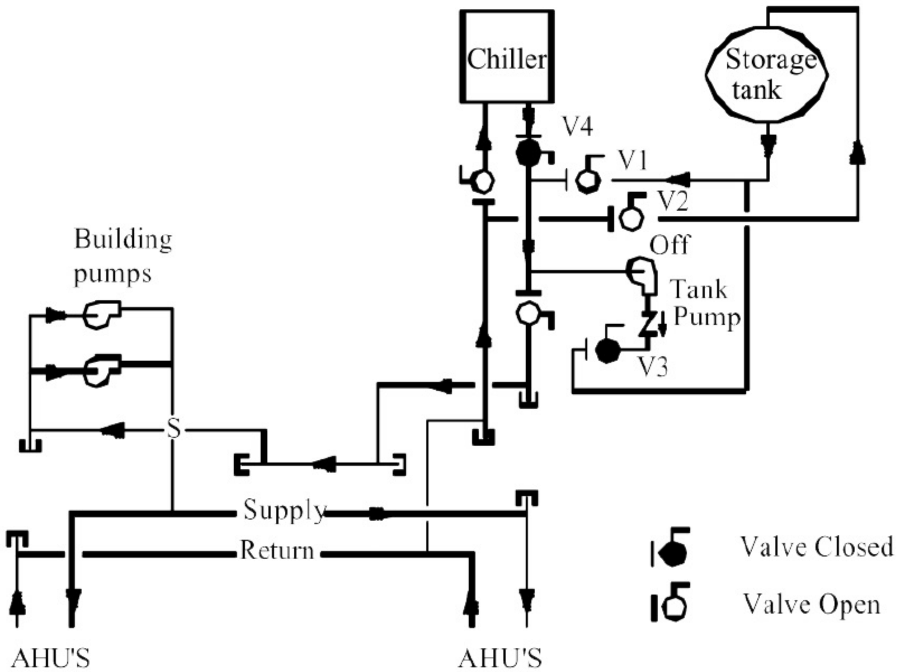

Fig. 3: Figure 8-3. Waterside Schematic of the Chiller/Thermal Storage System at the Hospital in Monahans, Texas

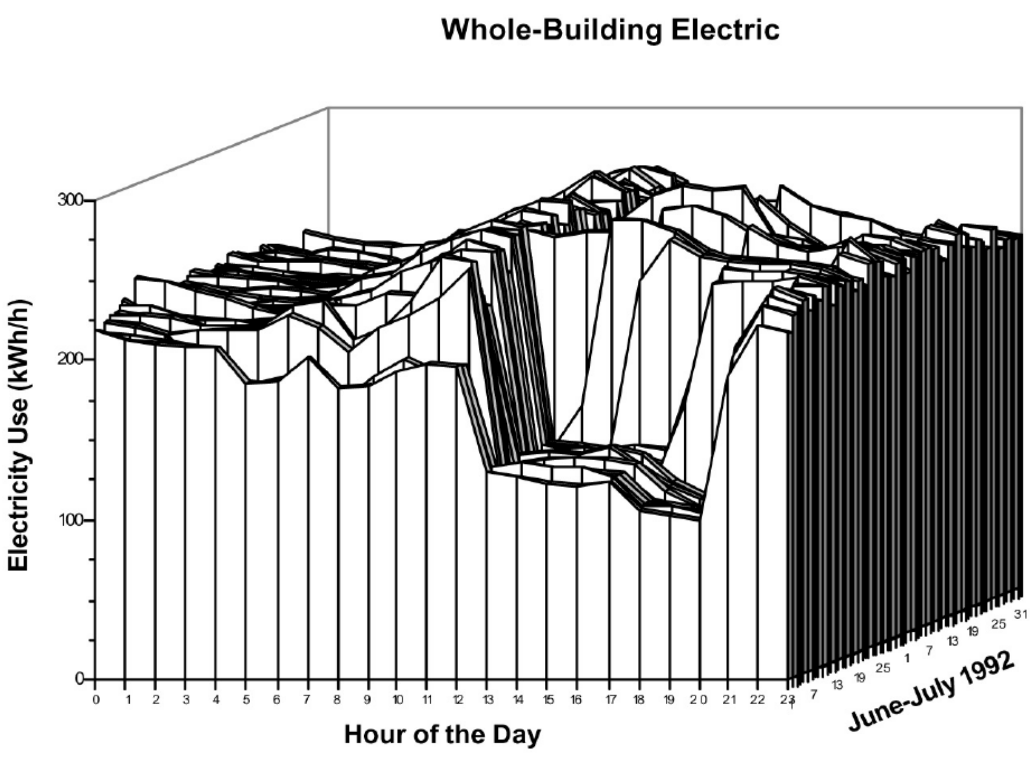

Fig. 4: Figure 8-4. Hourly Electricity Consumption for the Hospital in Monahans, Texas, for June and July, 1993

- The highest differential temperature in the building loop was 7°F (42/49°F), which indicates the building loop flow was at least 70% higher than required. The low differential temperature was largely due to chilled water bypass of the fan coils and a few coils in the roof top units, where excessive outside air was used.
- Measurements showed the chilled water flow to the building reached 200 gpm. Since the chiller rated flow rate is 228 gpm, the chilled water flow to the tank was much lower than the tank pump capacity of 128 gpm and the tank could not be fully charged during the off-peak hours.

To solve the problem, the following actions were taken:

- Conducted AHU system commissioning, which included the following major items:
  - Decreased outside air intake by up to 50% in five major roof top units. Since no return air fan was installed, the mixed air chamber had a negative pressure of –1.0 in. H2O or lower. The negative pressure sucked excessive outside air into the AHU. However, the cooling coil did not have the capacity to handle the outside air. This caused the cooling control valve to fully open. Reducing the outside air flow to the required level allows the chilled water control valve to function properly. Eliminating the three-way valves was not immediately implemented due to lack of funds for that purpose.
  - Calibrated the cold air temperature sensor and set the supply air temperature at a minimum of 55°F
- Installed a VFD on the building pump and a temperature sensor on both the return and supply.
  - Modulated VFD speed to control the differential temperature at a minimum of 10°F. The differential pressure sensor was not used since the system did not include a differential pressure sensor and funds were not available for installation.
  - Set the maximum VFD speed to 60%. There was no manual valve in the bypass line of the fan coil unit. It was also impossible to cut off the bypass line when the system was commissioned. However, engineering calculations showed that the three-way valve would be 90% open to the coil if the VFD was set at 60% under maximum building load conditions. Although this does not maximize the pump power savings, it would provide reliable system operation. This was the top priority of the project.

Figure 8-5 shows the building electricity consumption during June and July the summer after these measures were implemented. It can be seen that it was never necessary to operate the chiller during the peak demand window. The system has operated successfully through some of the hottest summers in Texas history since then.

The pump power before and after the CC process is shown in Figure 8-6. The pumping power has typically been reduced by 50% or more.

The savings from restoring the thermal storage system to operation were $14,190/year, with additional savings of $5,540/year resulting from the improved pump and AHU operation, for total savings of $19,730/yr. More information can be found in “Rehabilitating a Thermal Storage System through Commissioning” [Liu et al. 1999].

8.3 8.3 Minimize the Off-Peak Demand

Some utility rate schedules also use off-peak demand to determine the billing demand. In these cases, decreasing off-peak demand can also result in significant cost savings. To decrease the off-peak demand, the following procedures should be followed:

- Turn off the chiller earlier. If the peak period starts at noon, the off-peak demand is often set between 10:00 a.m. and noon. Turning off one or two chillers during this short period can result in significant off-peak demand reduction.
- Turn on chiller earlier. If the peak period ends after 6:00 p.m., one or more chillers can often be turned on after 5:00 p.m. since office lights and equipment are gradually turned off beginning at 5:00 p.m. or even earlier. Turn on one or more chillers to keep a constant peak demand over the off-peak period.

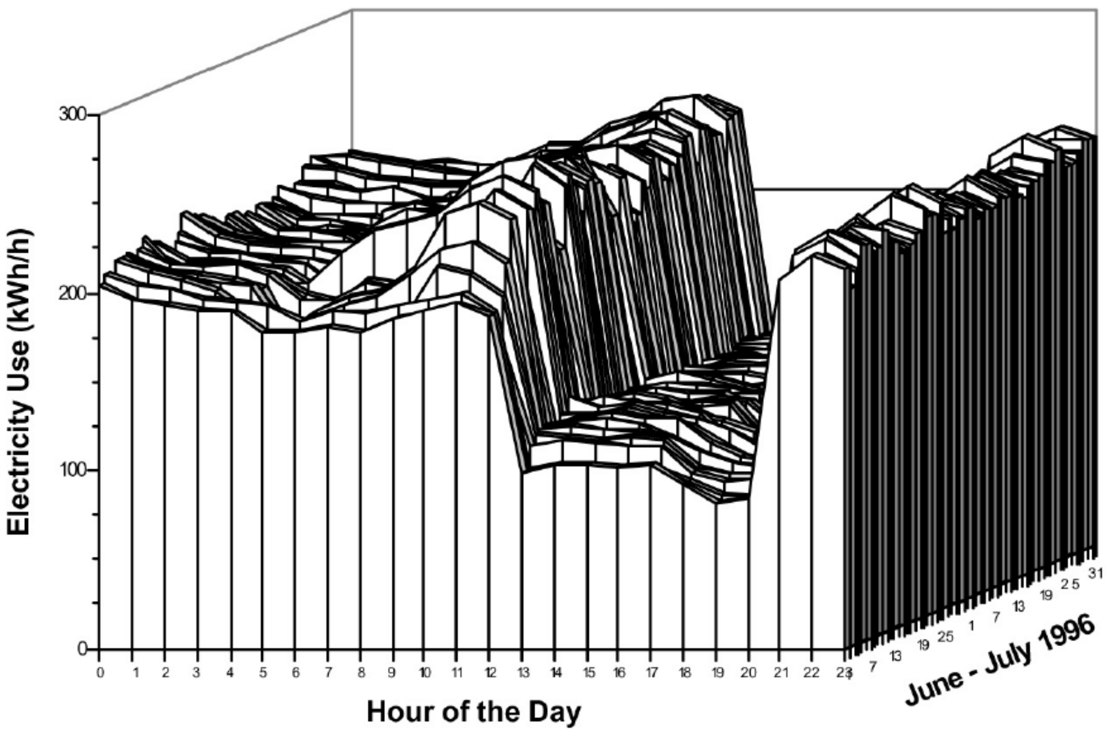

Fig. 5: Figure 8-5. Electricity Consumption for Hospital in Monahans Texas Following Implementation of CCSM Measures

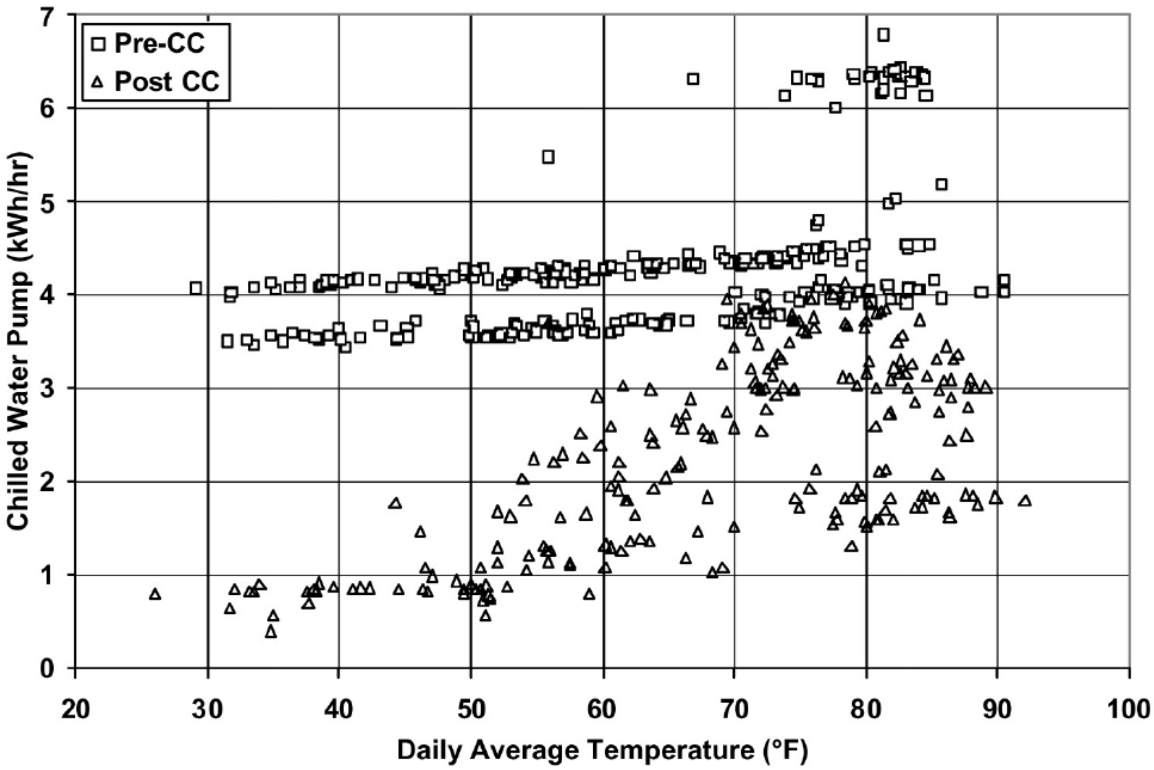

Fig. 6: Figure 8-6. Pumping Power at the Monahans, Texas, Hospital Before and After CCSM Measures Were Implemented

- Do not entirely charge the tank in a single day when cooling requirements are low. During mild days or during the winter, a small amount of cooling may be required. The tank should be partially charged to minimize the thermal losses. The tank should not be fully charged in a single day since it may require turning on all chillers for a short period and could set up a very high off-peak demand for that month.

EXAMPLE:

Terrell State Hospital, located in Terrell, Texas, is a mental health campus with more than 600,000 square feet of conditioned space in more than a dozen buildings. A chilled water thermal storage system with 7,000 ton-hours capacity was constructed in 1995 to provide cooling to the campus. There are four chillers in three different chiller plants, with total cooling capacity of 1,325 tons (two 425-ton, one 275-ton and one 200-ton).

On-peak hours were from noon to 8:00 p.m. weekdays, June through September. The thermal storage system was intended to decrease on-peak demand by 700 kW. Figure 8-7 presents the procedures for determining billed demand. First, demand candidate 1 was determined as the lesser of the current month's peak demand (off-peak or on-peak) and as a factor including ratchet demand and current off-peak demand (25% current off-peak demand plus 75% of the highest on-peak demand in the last 12 months). Secondly, demand candidate 2 was determined as the largest of the following: (1) a ratchet factor, 80% of the annual on-peak demand, (2) 50% of the contract demand and (3) 50% of the annual peak demand, including off-peak demand. The actual monthly demand charge was based on the larger of the two candidate values.

The hospital had a contract demand of 2,800 kW meaning it had a 1,400 kW minimum monthly demand charge. The peak demand hours were from noon to 8:00 p.m., Monday through Friday during the months of June, July, August and September. The historical on-peak demand varied from 1,475 kW to 1,731 kW during the on-peak months. The off-peak demand varied from 1,678 kW to 2,736 kW. Figure 8-8 presents the simulated demand penalties with hypothetical on-peak demand and off-peak demand values. The demand penalty is defined as the amount that will be added to 1,400 kW to get the billed demand. The current month demand values will be the off-peak demand whenever it is higher than the annual on-peak demand. The parametric values shown (1400 – 1700) are annual on-peak demand.

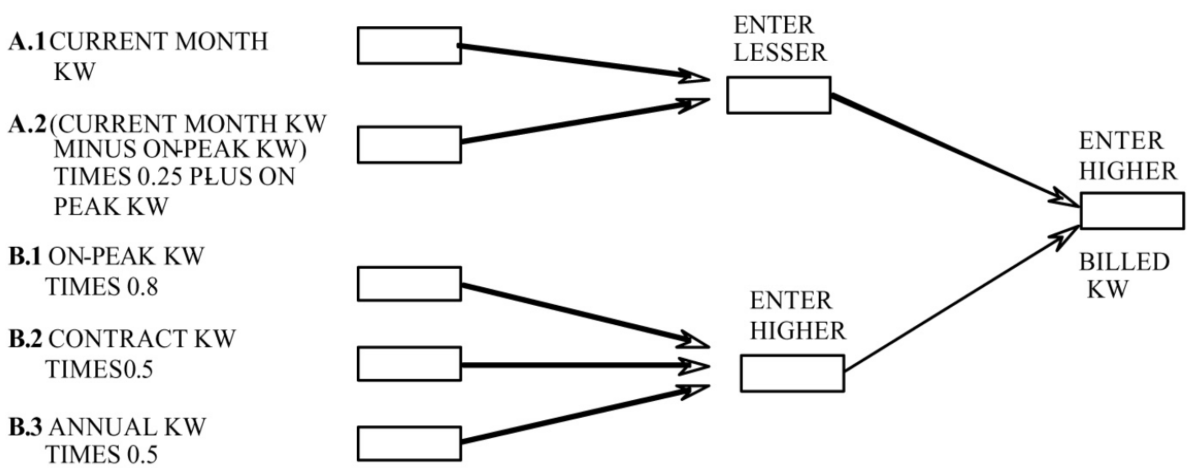

Fig. 7: Figure 8-7. Flowchart for the Calculation of Monthly Billing Demand

Both on-peak and off-peak demand controls are important in minimizing the demand penalty. When the off-peak demand is less than the annual on-peak demand, the demand penalty varies from zero to the difference between on-peak demand and 1,400 kW as the off-peak demand increases to the on-peak value. When the off-peak demand is higher than the annual on-peak demand, the demand penalty is the difference between the on-peak demand and 1,400 kW plus 25% of the off-peak demand increase. For example, if the off-peak demand decreases from 1,700 kW to 1,400 kW when the annual on-peak demand is 1,700 kW, the demand charge will decrease by 300 kW to 1,400 kW. If the annual on-peak demand is 1,400 kW, the demand charge will decrease by 75 kW from 1,475 kW. The demand

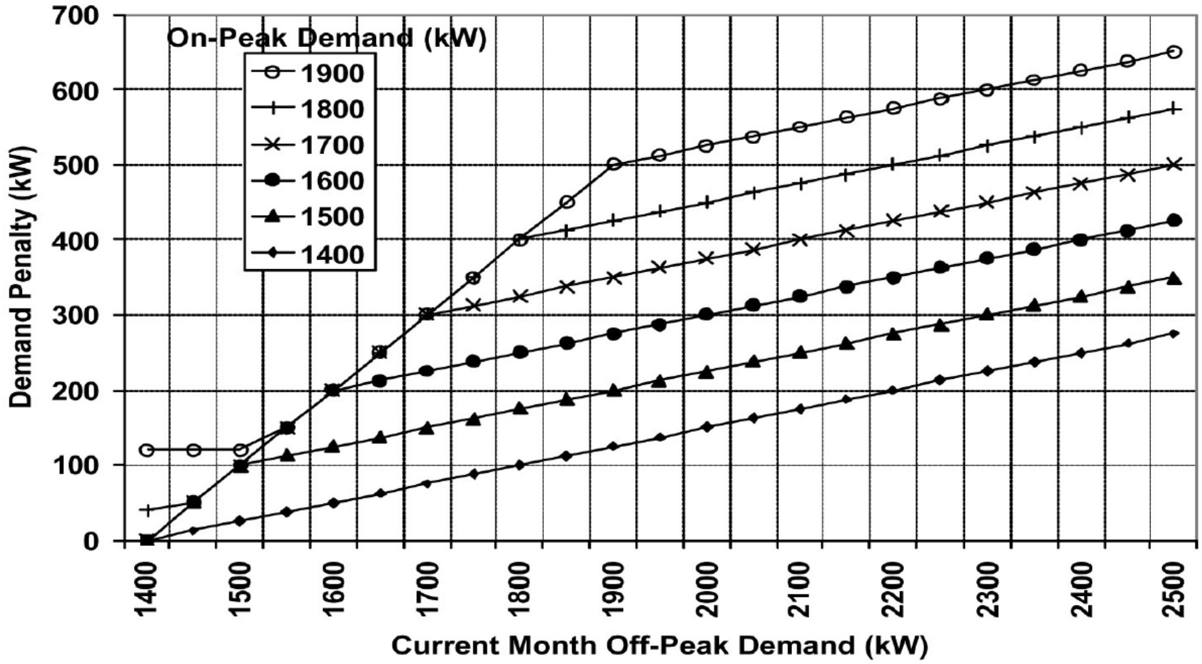

Fig. 8: Figure 8-8. Demand Penalties Under Different Current Month Demand Values When On-Peak Demand Has Been Set at Different Levels

penalty increases by 75% of the annual on-peak demand increase when the annual on-peak demand is higher than the current off-peak demand.

Figure 8-9 presents a typical storage tank inventory profile, base building electrical load profile (without chillers) and facility electrical load profile during on-peak months. The daily facility electrical load includes the base building electrical power and chiller power. The graphs show a general trend of increased demand from 4:00 a.m. to 2:00 p.m. The base load is 800 kW higher at 2:00 p.m. than at 4:00 a.m. Also note that the demand drops below 1,200 kW after 5:00 p.m. when most staff start to leave the hospital. The base load electrical demand for this hospital is less than 1,400 kW throughout the year, making it possible to set 1,400 kW as the target during off-peak periods.

The chart shows that peak demand is controlled below 1,400 kW. The electrical power is maintained below 2,000 kW until 6:00 am. The off-peak demand is 2,356 kW directly before the on-peak hours.

The inventory continuously increases after 9:00 a.m. until noon. Obviously, the chiller is charging the tank until noon. If the tank can be fully charged before 9:00 a.m., the off-peak demand can be decreased from 2,350 kW to 2,200 kW. The demand charge will be decreased from 1,650 kW to 1,600 kW with 50 kW demand savings. If the discharging mode could start as early as 7:00 a.m., the off-peak demand can be limited to 2,000 kW. The monthly demand charge can be further decreased from 1,600 kW to 1,550 kW. It appears that the off-peak demand reduction can potentially decrease the demand charge by 100 kW during the summer months.

During winter months, the cooling energy consumption is very low. Most of these buildings were built before 1950. Each room has exterior walls and windows. If the chillers are kept off, the off-peak demand will be below 1,400 kW. Then, 600 kW off-peak demand reduction can be achieved during the winter months since the current control sequence often runs chillers during the daytime. The potential demand charge reduction will be 200 kW for winter months. The demand charge savings varies between 100 kW and 200 kW from summer months to winter months. This is 14% to 28% of design peak load reduction for the thermal storage system.

The peak demand can be controlled at 1,400 kW if the chillers are kept off during the on-peak hours. However, the chillers had to be turned on before the project, which created a peak demand of 1,731 kW. Keeping stable operation means another 250 kW demand charge reduction. It appears that optimizing on-peak and off-peak demand control

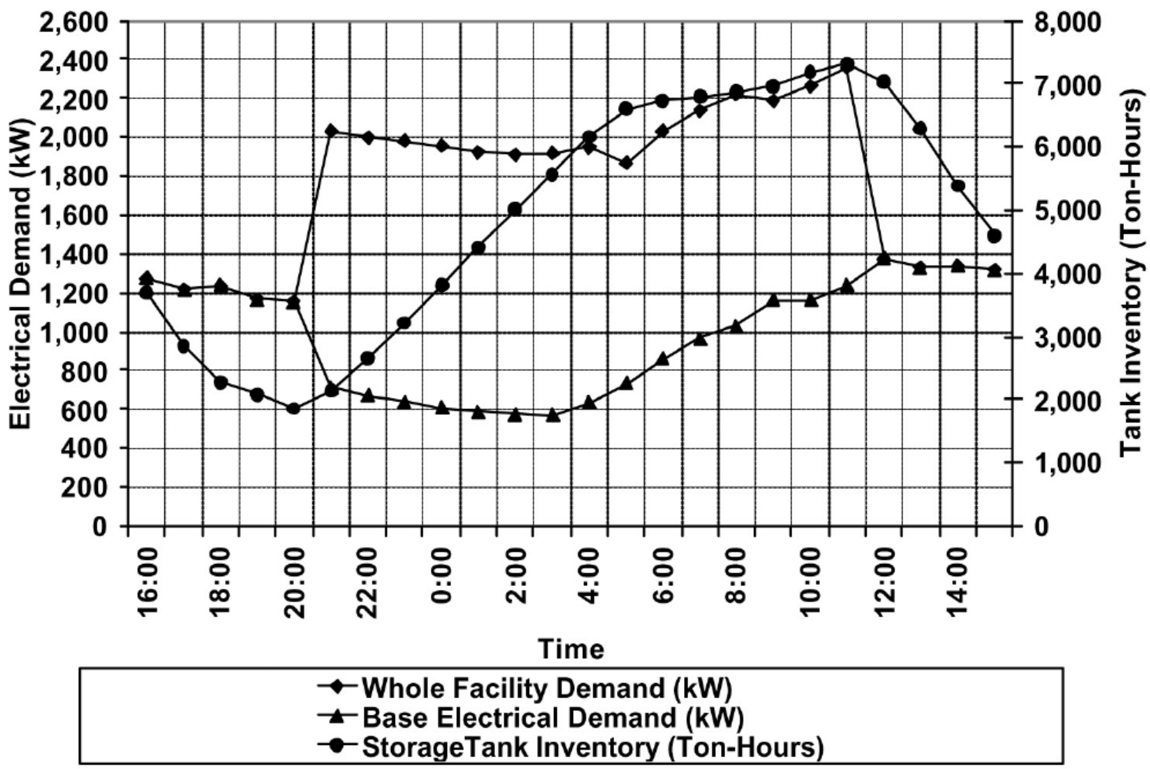

Fig. 9: Figure 8-9. Typical On-Peak Months Whole Facility Demand, Base Electrical Demand and Storage Tank Inventory Profiles

can decrease the monthly demand charge by 350 kW to 450 kW, which is 50% to 64% of the initial design demand reduction expected from the thermal storage system.

Comprehensive building commissioning was conducted prior to developing the optimal control sequences. During the building commissioning, the AHU operation was optimized. This included static pressure reset, supply air temperature reset, outside air adjustment and other measures. The building chilled water loop was optimized using a loop differential pressure reset. A loop water balance was also conducted in a number of buildings.

The optimized control sequences are discussed separately for on-peak months and off-peak months. A number of factors are incorporated into the sequence to increase the savings and simplify the operation.

On-Peak Months Optimal Control Sequences:

- Turn off one 450-ton chiller at 8:00 a.m.
- Start discharge mode at 9:00 a.m. or later if necessary
- Start the 200-ton chiller at 5:00 p.m. if the inventory is inadequate and the facility load is below 1,200 kW

These sequences are easy to understand and easy to implement. Figure 8-10 compares the measured facility electrical load profiles before and after implementation of the optimized control sequences. The improved control sequences limited the off-peak demand to 2,000 kW by turning off the 450-ton chiller after 8:00 am. Figure 8-10 also shows that commissioning also decreased the facility electrical load significantly.

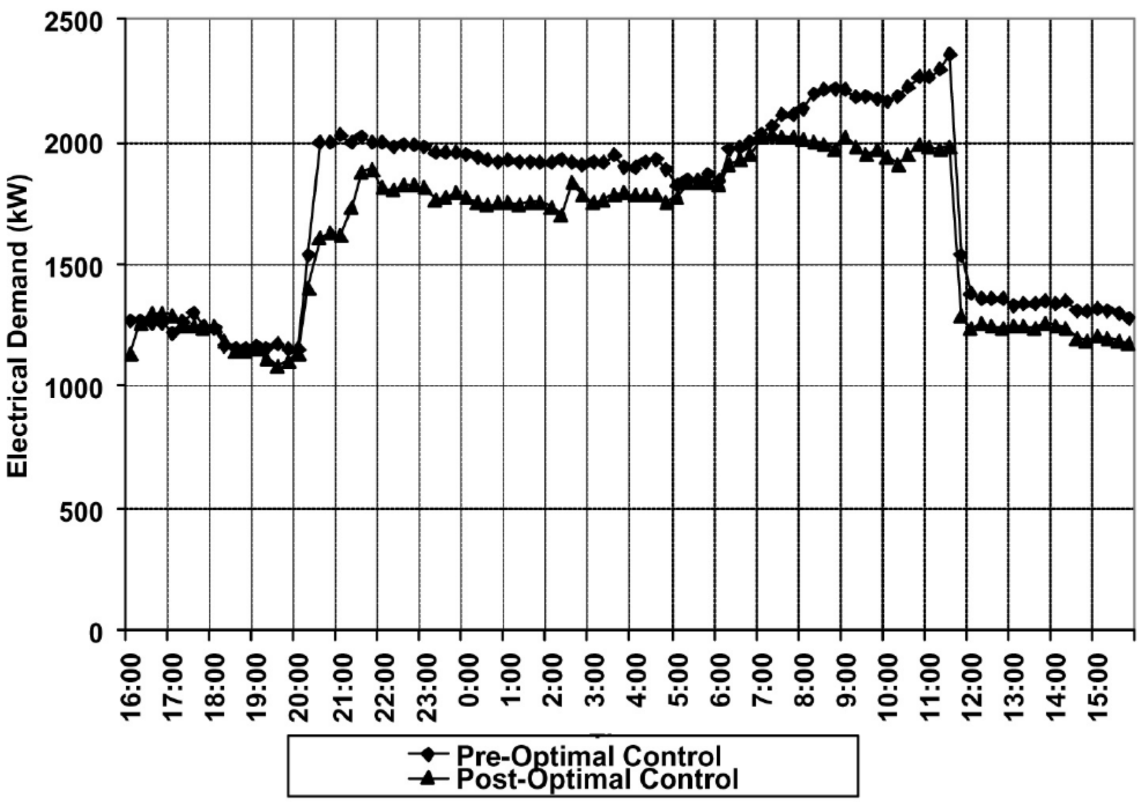

Fig. 10: Figure 8-10. Comparison of Typical Facility Electrical Demand Profile Before and After Implementation of Optimal Control Sequences

Optimal Control Sequences for Off-Peak Months

Load projection is critical for the off-peak months optimal control sequence. Cooling load models were developed using measured data as:

$$E_{c,0} = 9.83T_{oa} - 466.37 \tag{8.1}$$

$$E_{c,u} = 7.54T_{oa} - 467.50 \tag{8.2}$$

The hourly temperature is projected for each day using standard daily profiles combined with forecasted high and low temperatures of the next day. Whole campus cooling load is predicted 24 hours ahead of time to determine the cooling tonnage to be charged. Hourly cooling load up to 5 p.m. of the next day is calculated and totaled every hour after 5 p.m. It is then compared with the cooling capacity remaining in the storage tank.

At 5:00 p.m., the high and low temperature forecasts for the next day are manually entered into the control system. Then the chiller status is determined using the following rule: if the forecast daily cooling is 1,000 ton-hours less than the tank inventory, no chiller will be turned on. Otherwise turn on the chiller and charge the tank until the inventory is 1,500 ton-hours higher than the forecasted cooling load. If the chillers are to be operated, the electrical demand is monitored to make sure that it does not surpass 1,400 kW. If demand drops below 1,000 kW while the 200-ton chiller is in operation and the tank capacity is more than 1,000 ton-hours short, turn on the 425-ton chiller to serve the buildings and charge the tank. If the demand approaches 1,400 kW, turn off the smaller chiller.

The optimal control sequences were implemented in two phases. The first phase, implemented in 1998, focused on controlling the on-peak demand below 1,400 kW. The second phase, implemented in 1999, concentrated on managing the off-peak demand.

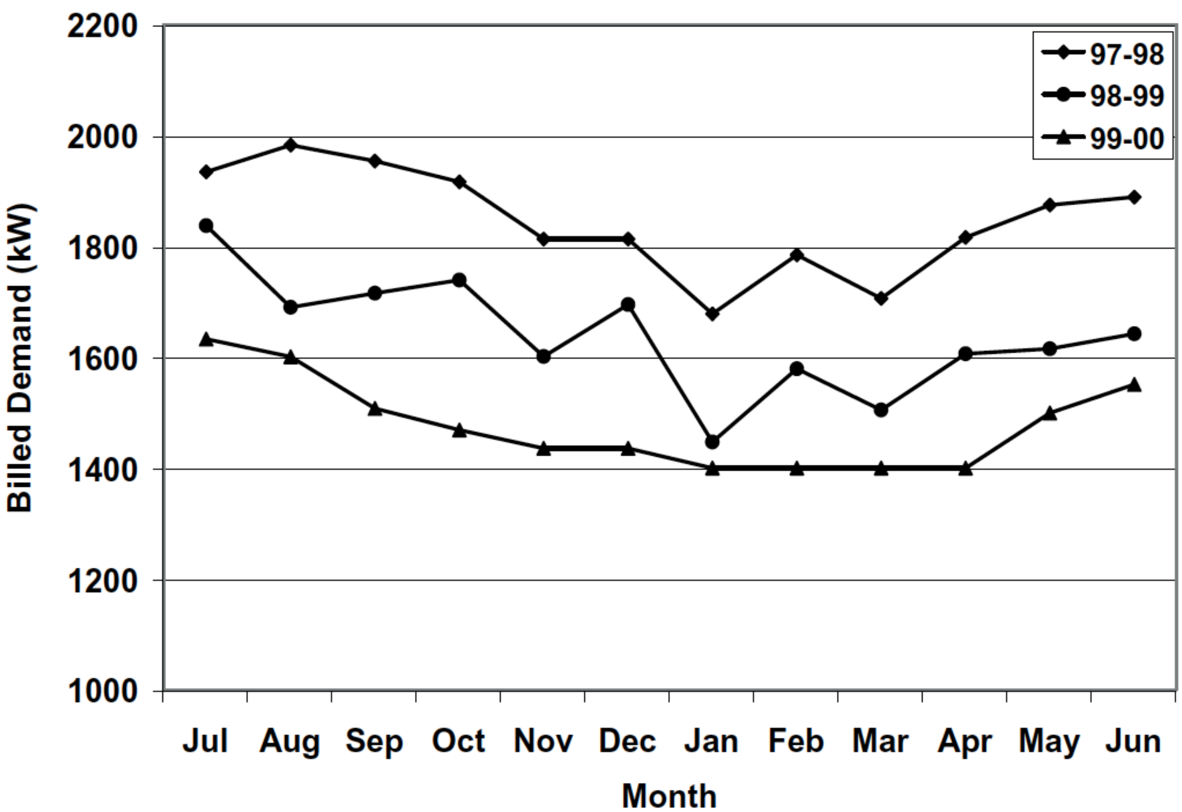

Fig. 11: Figure 8-11. Comparison of Billed Electrical Demand Before and After Commissioning

Figure 8-11 compares the demand charges for the base year (1997-1998), after the first phase (1998-1999) and after the second phase (1999-2000). In the base year, the on-peak demand was 1,731 kW and the off-peak demand was 2,736 kW. After the first phase, the on-peak demand was supposed to decrease to 1,400 kW. However, unexpected system failures resulted in an on-peak demand of 1,559 kW. Still, the annual demand charge reduction was 2,215 kW, or 10.0% of the base year demand charge. After the second phase, the on-peak demand was decreased to 1,332 kW. The off-peak demand was controlled between 1,879 kW and 2,207 kW during on-peak months and between 1,318 kW and 1,746 kW during off-peak months. The annual demand charge decreased from 22,158 kW/yr to 18,075 kW/yr for an 18.4% reduction. If the facility were to reduce its contract demand from 2,800 kW to 2,500 kW, it would result in 612 kW of additional demand charge savings. The total potential demand charge savings are 6,910 kW, or 28% of the base year demand charge. More information regarding this case study can be found in "Practical Optimization of Full Thermal Storage Systems Operation" [Wei et al. 2002].

8.4 8.4 Set Up an Alarm System

An alarm system should be in place that will immediately alert the operator if a chiller is accidentally turned off. This will enable the operator to take any actions needed to get the chiller back on line so the tank will be fully charged during the charging period.

References

Liu, M., B. Veteto and D. E. Claridge, 1999. "Rehabilitating A Thermal Storage System Through Commissioning." ASHRAE Transactions, Vol. 105, Part II, pp. 1134-1139. Wei, G., M. Liu, Y. Sakurai, D.E. Claridge and W.D. Turner, 2002, "Practical Optimization of Full Thermal Storage Systems Operation." ASHRAE Transactions-Research, Vol. 108, Part II, pp. 360-368.

CHAPTER 9

Ch 9: Ensuring Optimum Building Performance

CCSM activities are essential to optimize building system operation and reduce energy consumption. To ensure excellent long-term performance and maintain the CCSM performance, the following activities should be conducted:

- Document CCSM activities
- Measure energy and maintenance cost savings
- Train operating and maintenance staff
- Measure energy data and continuously measure energy performance
- Obtain on-going assistance from CCSM engineers

This chapter discusses guidelines to perform these tasks.

9.1 9.1 Document CCSM Project

The documentation should be brief and accurate. The operating sequences should be documented accurately and carefully. This documentation should not repeat the existing building documentation. It should describe the procedures implemented, including control algorithms and briefly give the reasons behind these procedures. The emphasis is on accurate and usable documentation. The documentation should be easily used by operating staff. For example, operating staff should be able to create operating manuals and procedures from the document.

The CCSM project report should include the accurate documentation of current energy performance, building data, AHUs and terminal boxes, water loops and pumps, control system, and performance improvements.

9.1.1 9.1.1 Current energy performance

Before commissioning a building, the engineer should investigate the energy performance of the facility and major systems in the facility by examining the measured energy data or the utility bills. The following information should be collected and examined:

- Whole facility heating, cooling and electricity consumption data (measured hourly data or utility bills)

- The quality of data (bad, out of scale, or good). Ensure these data can be used to document the improvements of the CCSM effort.
- Briefly analyze the potential savings from CCSM efforts

9.1.2 9.1.2 Building

The following information should be obtained and documented:

- Building name and location
- Building floor plan for each floor
- Special areas, such as a computer facility, operating rooms, etc., marked on floor plan with a brief description
- Name of AHU which serves each area marked on floor plan
- Building envelope area. Collect wall and window data and calculate heat transfer values
- Building comfort level. Conduct a brief walkthrough of the building and measure room relative humidity, CO2 and room temperature conditions.
- Comfort problems in each area
- Comfort improvements after commissioning

9.1.3 AHUs

The following information should be documented for each AHU:

- Single line diagram (fill in standard forms)
- Fan: hp., VFD, inlet guide vane, eddy switch or other
- VFD: hp., brand, working condition (% speed, hunting?)
- Automatic valve: type (normally open or closed), range (3-8 psi or 0 - 13 psi), working condition, and control (by EMCS or stand alone controller, DDC or pneumatic)
- Coils: inlet and outlet temperature (design and measured), differential pressure
- Dampers: working condition (adjustable or not); actuator condition
- Temperature sensors: EMCS readings and hand meter readings
- Controller: working or disabled
- Air flow: outside air flow, return air flow, maximum total flow and minimum total flow
- Air condition: temperature and CO2 level for outside air, return air and supply air
- Control sequence: cold and hot deck set points, economizer control sequence and static pressure control sequence
- Optimal control sequence and implementation

9.1.4 Water Loop

The following information should be collected for the water loop and pumps:

- Water loop riser diagram: differential pressure sensor position, temperature sensor position, automatic valve position, building bypass, coil bypass
- Pump: single line diagram of pump and pipe line connection, hp., VFD, differential pressure across pump
- VFD: condition (working, manual, bypass, damaged), % of speed or Hz, control logic

- Automatic valves: condition (working, bypassed), type (normally open or closed), operation range, location, function, position
- Control: loop control logic, differential pressure reset schedule, return temperature reset schedule, automatic control valve control schedule
- Water condition: building supply and return temperature, coil supply and return temperatures, differential pressure across building and each coil

9.1.3 9.1.5 Terminal Box

The following information should be collected for the terminal boxes:

- Type: CV, VAV, fan-powered, DDC or pneumatic
- Design specification: minimum and maximum flow, temperature setpoint, thermostat location, zone floor area (sq.ft.)
- Reset schedules

9.1.4 9.1.6 Controller or Control System

The following information should be collected for the control system:

- Control system brand and model
- Input and output for each mechanical sub-system
- Control system logic for each mechanical sub-system
- Component conditions, such as power line, pneumatic line and p/e or e/p switches.

9.1.5 9.1.7 Potential Impacts

- Indoor condition improvement
- Energy consumption improvement
- Mechanical and electrical system operation improvement
- Other improvements

9.2 9.2 Measure Energy Savings

Most building owners expect the CCSM project to pay for itself through energy savings. Measurement of energy savings is one of the most important issues for CCSM projects. The measurement should follow the International Performance Measurement and Verification Protocol [2001 IPMVP], or other widely accepted standard methods. The process for determining savings as adopted in the IPMVP [2001] defines energy savings, Esave, as:

ESave = Ebase – Epost

where Ebase is the "baseline" energy consumption before the CCSM measures were implemented and Epost is the measured consumption following implementation of the CCSM measures.

Figure 9.1 shows the daily electricity consumption of the air handlers in a large building in which the HVAC systems were converted from constant volume systems to VAV systems using variable frequency drives. Consumption is shown for slightly longer than a year before the VFDs were installed (pre-retrofit), for about three months of construction and for about two years after installation (post-retrofit). In this case, the base daily electricity consumption is approximately 8,300 kWh/day. The post-retrofit electricity consumption is approximately 4,000 kWh/day. The electricity savings are approximately 4,300 kwh/day. During the construction period, the savings were slightly lower.

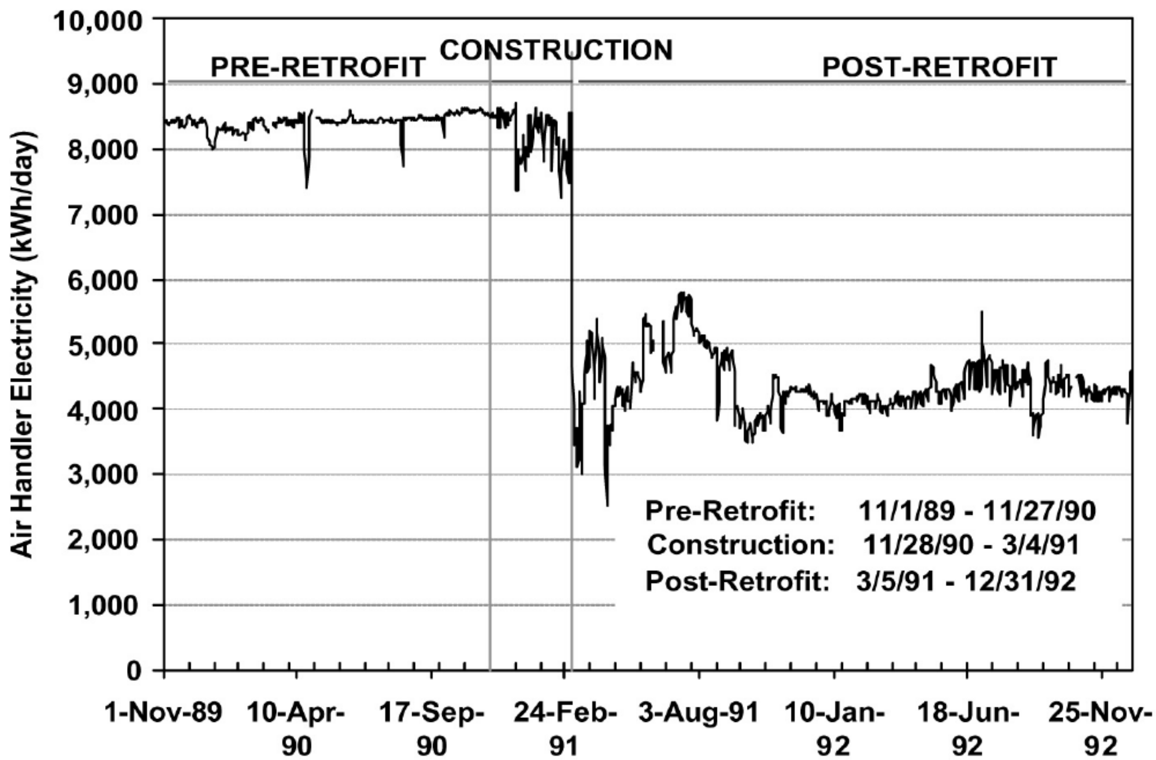

Fig. 1: Figure 9-1. Daily Electricity Consumption for Approximately One Year Before a Retrofit and Two Years After the Retrofit

However, in most cases, consumption shows more variation from day to day and month to month than shown by the fan power for these constant speed fans. Hence, determination of the baseline must consider a number of factors including weather changes, changes in occupancy schedule, changes in number of occupants, remodeling of the spaces and equipment changes.

In the IPMVP, the baseline energy use, Ebase, is determined from a model of the building operation before the retrofit (or commissioning) that uses post-installation operating conditions (e.g. weather, occupancy, etc.). The post-installation energy use is generally the measured energy use, but it may be determined from a model if measured data are not available.

The IPMVP includes four different M&V techniques or options. These options, may be summarized as Option A: some measurements, but mostly stipulated savings; Option B: measurement at the system or device level; Option C: measurement at the whole-building or facility level; and Option D: determination from calibrated simulation. Each option has advantages for certain applications.

The cost savings must also consider changes in utility rates. Since savings projections are made based on the rates in effect before the retrofit or CCSM measures are implemented, it is recommended those rates be used for any savings projections.

9.2.1 9.2.1 Option A - Stipulated Savings (Involving some measurements)

The stipulated option estimates savings by measuring the capacity or the efficiency of a system before and after retrofit or commissioning and multiplying the difference by an agreed upon or "stipulated" factor such as the hours of operation, or the load on the system. This option focuses on a physical determination of equipment changes to ensure that the installation meets contract specifications. Key performance factors (e.g. lighting wattage) are determined with spot or short-term measurements. Operational factors (e.g. lighting operating hours) are stipulated based on historical data or spot measurement. Performance factors are measured or checked yearly. This method provides reliable savings estimates for applications where the energy savings are independent of weather and occupancy conditions (for most loads that are constant).

For example, during the CCSM process, the fan pulley was decreased from 18" to 16" for a constant volume AHU. The fan power savings can be determined using the following method:

- Measure the fan power consumption before changing the pulley and the power consumption after changing the pulley
- Determine the number of hours the fan operates
- Determine the fan power savings as the product of the hourly fan power energy savings and the number of hours

If the energy consumption varies with occupancy and weather conditions, this option should not be used. For example, the minimum air flow was adjusted from 50% to 0% for 100 VAV terminal boxes at night and during weekends. Since the air flow depends on both internal and external loads, the air flow may not be 0% even if the minimum flow setting is 0%. This method cannot be used to determine savings.

If the goal of the measurement is for savings determination, option A should be considered first. If it can provide the required accuracy, option A should be used.

9.2.2 9.2.2 Option B - Device/System Level Measurement

Within Option B, savings are determined by continuous measurements taken throughout the project term at the device or system level. Individual loads or end-uses are monitored continuously to determine performance and long-term persistence of the measures installed. The base line model can be developed using the measured energy consumption and other parameters. The energy savings can be determined as the difference of base model energy consumption and the measured energy consumption. This method provides the best savings estimation for the device or system.

The data collected can also be used to improve or optimize the system operation and are particularly valuable for Continuous CommissioningSM projects. Since measurements are taken throughout the project term, the cost is higher than option A.

9.2.3 9.2.3 Option C - Whole Building Level Measurement

Option C determines savings by analyzing "whole-building" or facility level data measured during the baseline period and the post-installation period. This option is required when it is desired to measure interaction effects, e.g. the impact of a lighting retrofit on the cooling consumption as well as savings in lighting energy. The data used may be utility data, or sub-metered data.

The minimum number of measurement channels recommended for performance assurance or savings measurement will be the number needed to separate heating, cooling and other electric uses. The actual number of channels will vary, depending on whether pulses are taken from utility meters, or if two or three current transformers are installed to measure the three phase power going into a chiller. Other channels may be needed, depending on the specific measures that are being evaluated.

Option C requires that installation of the proper systems/equipment and proper operating practices be confirmed. It determines savings from metered data taken throughout the project term. The major limitation in the use of Option C for savings determination is that the size of the savings must be larger than the error in the baseline model. The major challenge is accounting for changes other than those associated with the ECMs, or commissioning changes implemented.

Accurate determination of savings using Option C normally requires at least 9 months of continuous data [Reddy et al, 1992, Ruch et al, 1992, and Katipamula et al., 1995] before a retrofit, and continuous data after retrofit. However, for commissioning applications, a shorter period of data, during which daily average ambient conditions cover a large fraction of normal yearly variation, is generally adequate.

Note that monthly bills may be used to estimate the energy savings. This method is one version of Option C described above. It is typically the least expensive method of verification. It will yield reliable results under the following conditions:

- Significant savings are expected at the utility meter level
- Savings are too small to cost-justify metered data
- There will be no changes in:
  - Equipment
  - Schedules
  - Occupancy
  - Space utilization

The case shown in Figure 9.2 is an example where monthly bills clearly show the savings.

Notes

Accurate determination of savings normally requires 12 months . . . However, for commissioning applications, a shorter period of data during which daily average ambient conditions cover a large fraction of normal yearly variation is generally adequate.

The savings were large and consistent following the retrofit until June. At this point, a major deviation occurred. The presence of other metering at this site showed that the utility bill was incorrect. Further investigation showed that the utility meter had been changed and was not considered in the bill sent. The consumption included in this bill was greater than if the site had used the peak demand recorded on the utility meter for every hour of the billing period!

9.2.4 9.2.4 Option D - Calibrated Simulation

Savings are determined through simulation of the facility components and/or the whole facility. The most detailed application of this approach calibrates a simulation model to baseline consumption data. For commissioning applications, it is recommended that calibration be to daily or hourly data. This type of calibration may be carried out most rapidly if simulated data are compared to measured data as a function of ambient temperature. Wei et al. [1998] have developed "energy signatures" which greatly aid this process. More information can be found from "Use of Calibrated HVAC Models to Optimize System Operation" [Liu and Claridge 1998].

Similar to the other options, the implementation of proper operating practices should be confirmed. It is particularly important that personnel, experienced in the use of the particular simulation tool, conduct the analysis. The simulation analysis must be well documented, with electronic and hard copies of the simulation program input and output archived.

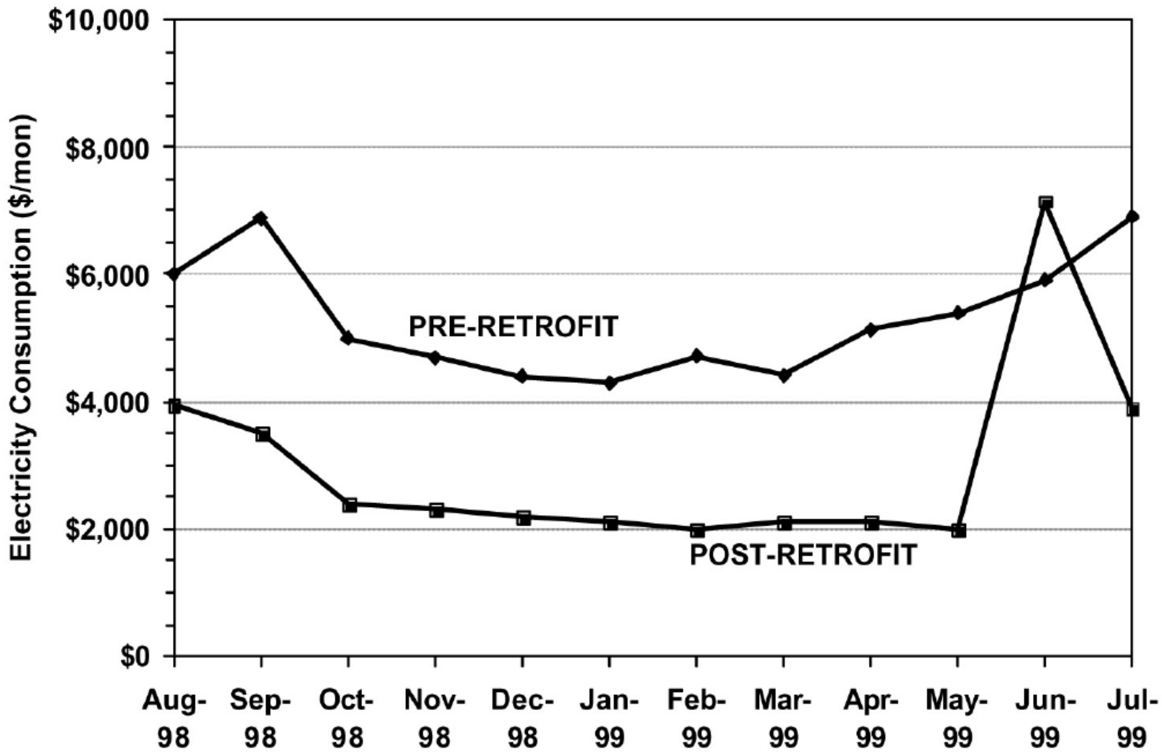

Fig. 2: Figure 9-2. Comparison of Monthly Utility Bills Before (Top Line) and After (Bottom Line) a Retrofit

9.3 9.3 Trained Operating and Maintenance Staff

Efficient building operation begins with a qualified and committed staff. Since the CCSM process generally makes changes in the way a building is operated in order to improve comfort and efficiency, it is essential that the operators be a part of the commissioning team. They must work with the CCSM engineers, propose CCSM measures and implement or help implement them. In addition to actively participating in the CCSM process, formal technical training should be provided to ensure that the operating staff understands the procedures implemented so they can perform trouble-shooting properly.

9.4 9.4 Continuously Measure Energy Performance

The measurement of energy consumption data is very important to maintain building performance and maintain CCSM savings. The metered data can be used to:

- Identify and solve problems. Metered consumption data is needed to ensure the building is still operating properly. If there is a component failure or an operating change that makes such a small change in comfort or operating efficiency that it is not visible in metered constumption data, it generally is not worth worrying about. If it does show up as even a marginal increase in consumption, trouble-shooting should be initiated.
- Trend/measure energy consumption data. This continuing acivity is the first line of defense against declining performance. The same procedures used to establish a pre-CCSM baseline can be used to establish a baseline for post-CCSM performance. This post-CCSM baseline can be used as a standard to which future performance is compared. Consumption that exceeds this baseline for a few days or even a month may not be significant, but if it persists much more than a month, trouble-shooting should be used to discover what led to the increase. If it is the result of a malfunctioning valve, it can be fixed. If it is the result of 100 new computers added to the building, adjust the base line accordingly.
- Trend and check major operating parameters. Parameters such as cold-deck temperatures, zone supply temperatures, etc. should be trended periodically for comparison with historic levels. This can be extremely valuable when trouble- shooting and investigating consumption above the post-CCSM baseline. * Find the real problems when the system needs to be repaired or fixed. It is essential that the same fundamental approach, used to find and fix problems while the CCSM process is initiated, be used whenever new hot calls or cold calls are received.

9.5 9.5 Utilize Expert Support as Needed

It is inevitable that a problem will arrive, which after careful trouble-shooting will point toward a problem with one or more of the CCSM measures which have been implemented. Ask the CCSM providers for help in solving such problems before undoing an implemented CCSM measure. Sometimes it will be necessary to modify a measure that has been implemented. The CCSM engineers will often be able to help find the most efficient solution, or find another explanation so the problem can be remedied without changing the measure.

Ask help from the CCSM providers when a new problem or situation is encountered. Problems occasionally crop up that defy logical explanation. These problems are generally resolved by trying one of several things that seem like possible solutions and playing with system settings until the problem goes away. This is one of the most important situations in which expert help is needed. These are precisely the kind of problems, and the trial and error solutions, which often lead to major operating cost increases.

References

Katipamula, S., Reddy, T.A., and Claridge, D.E., 1995. "Effect of Time Resolution on Statistical Modeling of Cooling Energy Use in Large Commercial Buildings," ASHRAE Transactions-Research. Volume 101, Part 2, pp. 321-334.

Liu, M., D.E. Claridge, 1998. “Use of Calibrated HVAC Models to Optimize System Operation,” ASME Journal of Solar Energy Engineering, Vol. 120, pp. 131-138.

Reddy, T. A., Kissock, K. and Claridge, D. E., 1992. “Uncertainty Analysis in Estimating Building Energy Retrofit Savings in the LoanSTAR Program,” Proceedings of the ACEEE 1992 Summer Study on Energy Efficiency in Buildings, Volume 3, American Council for an Energy Efficient Economy, Washington, D.C., pp. 225-237.

Ruch, D. K. and Claridge, D. E., 1992. “NAC for Linear and Change-Point Building Energy Models,” Proceedings of the ACEEE 1992 Summer Study on Energy Efficiency in Buildings, Volume 3, American Council for an Energy Efficient Economy, Washington, D.C., pp. 263-274.

Wei, G., M. Liu and D.E. Claridge, 1998, “Signatures of Heating and Cooling Energy Consumption for Typical AHUs,” The Eleventh Symposium on Improving Building Systems in Hot and Humid Climates Proceedings, June 1-2, 1998, Ft. Worth, Texas, pp. 387-402.

CHAPTER 10

Case Studies

Two detailed CCSM projects are presented to demonstrate the process and the energy efficiency and comfort improvements. Although numerous case studies were presented throughout the guidelines, the complete presentations of the case studies provide opportunities for readers to understand and learn the detailed techniques. The first case study is the Zachry Engineering Center (ZEC) located on the Texas A&M University Campus, College Station, Texas. The CCSM was conducted after a successful energy retrofit. The second example is the Brooke Army Medical Center (BAMC) located in San Antonio, Texas. The CCSM was conducted three years after completion of the building. This is considered as a CCSM project for a new building.

10.1 Project 1: Zachry Engineering Center Continuous CommissioningSM

Zachry Engineering Center (ZEC) is a 340,000 sq.ft. teaching and research building, located on the Texas A&M University campus. The building has four floors and a heated-only parking garage in the basement. It was constructed in the early 1970s and has heavy concrete floors and insulated exterior walls made of pre-cast concrete and porcelain-plated steel panels. Approximately 12% of the exterior wall area is covered with single-pane, bronze-tinted glazing. The windows are recessed approximately four feet from the exterior walls, providing some shading. Approximately 3,100 sq.ft. of northeast-facing clerestory windows admit daylight into the core of the building.

The ZEC includes offices, classrooms, laboratories and computer rooms and is open 24 hours per day, 365 days per year with heaviest occupancy during normal working hours between 8 a.m. and 6 p.m. on weekdays. Occupancy, electrical consumption and chilled/hot water consumption show marked weekday/weekend differences with peak weekend electrical consumption less than 10% above the nightly minimum. Weekday holiday occupancy is similar to weekend usage with intermediate usage on weekdays between semesters when class rooms are not in use, but laboratories and offices are occupied.

Original HVAC Systems

Twelve identical dual-duct constant volume systems with 40 hp. fans rated at 35,000 cfm and eight smaller air handlers (3 hp. average) supply air to the zones in the building. Supply and return air ducts are located around the perimeter of the building. These were operated with a constant outdoor air intake at a nominal value of 10% of design flow. Additional information about the building can be found in Katipamula and Claridge [1992a, 1992b], Bronson [1992], Bronson, et al. [1992], and Haberl, et al. [1993, 1995].

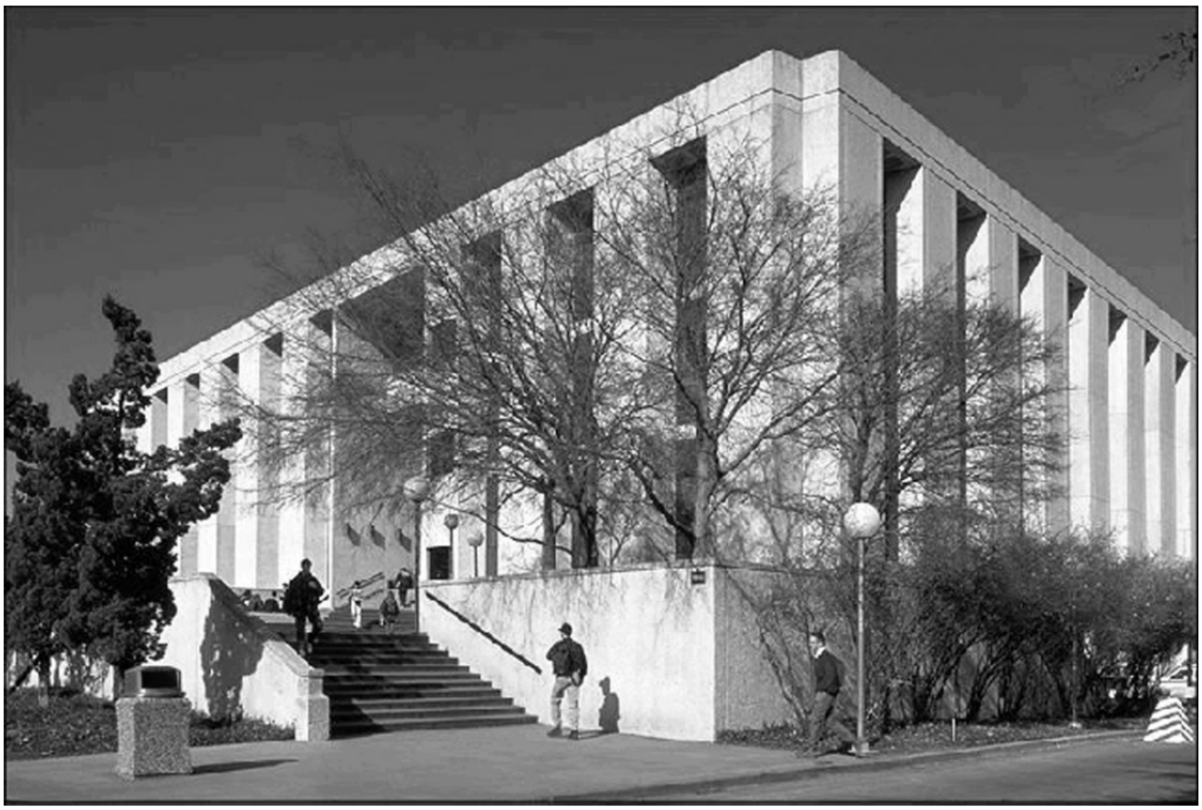

Fig. 1: Figure A-1. The Zachry Engineering Center (ZEC) on the Texas A&M Campus

Monitoring of energy use

Approximately 50 channels of hourly data have been collected and recorded each week for the Zachry Engineering Center since May 1989. The sensors are scanned every 4 seconds and the values are integrated to give hourly totals or averages as appropriate. The important channels for savings measurement are those for air handler electricity consumption and whole-building heating and cooling energy use. Air handler electricity consumption is measured at the building's motor control center (MCC) and represents all of the air-handling units and most of the heating, ventilating, and air-conditioning-related HVAC pumps in the building. Cooling and heating energy use are determined by a Btu meter which integrates the monitored fluid flow rate and temperature difference across the supply and return lines of the chilled and hot-water supply to the building. Most of the 50 channels of monitored information come from one air handler that is highly instrumented [Katipamula and Claridge 1992a].

HVAC System Retrofit

An energy audit of the Zachry Engineering Center was conducted in 1986 [TECCP, 1986]. This audit recommended a lighting retrofit to convert the four-lamp fixtures to three-lamp fixtures with reflectors, conversion of the large dual-duct constant air volume (DDCAV) systems to dual-duct variable-air volume (DDVAV) systems, and a connection to the campus energy management and control system (EMCS). The lighting retrofit was projected to save 975,600 kWh/yr of electricity, 2,500 MMBtu/yr of chilled water and increase hot water use by 832 MMBtu/yr with a payback of 4.4 years. The variable air volume (VAV) conversion, with controls improvements, was projected to save 1,952,776 kWh/yr on fan power, 115 MMBtu/yr of hot water and 265 MMBtu/yr of chilled water. The simple payback is 3.3 years. The university chose not to implement the lighting retrofit, but had variable speed drives installed on the twelve large AHUs, replaced the constant volume dual-duct terminal boxes with VAV dual-duct terminal boxes and connected the building to the campus automated control system.

The audit proposed that the DDCV be converted to DDVAV systems before installing variable speed drives on 24 fans with associated static pressure sensors and controls. The existing constant volume terminal boxes which typically operated at static pressures above 0.5 in. H2O, were to be replaced by DDVAV boxes with independent controls on the hot decks and cold decks to provide constant minimum flow rates of 0.65 cfm/sq.ft. The new boxes and controls were to be tied to the campus EMCS. This was intended to permit shutdown of heating and cooling to non-critical areas such as classrooms during unoccupied hours by closing the dampers in the VAV boxes while the fans continued to supply heating and cooling to laboratory areas. The repair and upgrade of the existing EMCS in the building was intended to control time-of-day settings, DDVAV box load group override, hot and cold deck reset, optimal start-stop of a load group DDVAV box to ensure space comfort, and space temperature reset.

After the building was retrofitted in 1991, fan power, chilled water consumption, and hot water consumption all dropped substantially. The fan power dropped from virtually constant consumption, approximately 350 kW before the retrofit as shown in Figure A-2, to consumption that varied from approximately 180 kW below 50°F to approximately 200 kW between 68°F and 86°F.

Chilled water use dropped substantially as well. Figure A-3 shows the daily chilled water consumption (MMBtu/day) in 1990 and 1994 as a function of daily average ambient temperature. Before the retrofit, the consumption depended almost linearly on the ambient temperature, increasing from approximately 95 MMBtu/day at 40°F to approximately 160 MMBtu/day at 86°F. The consumption following the retrofit was approximately the same at high temperatures, but dropped quite rapidly to about 85 MMBtu/day at 70°F and then dropped more slowly to about 43 MMBtu/day at 31°F. There is, of course, considerable scatter due to variation of ambient humidity conditions and certain operating practices. The general linear and piece-wise linear behavior of the chilled water consumption of the DDCV and DDVAV systems respectively, are consistent with the theoretically expected system behavior [Kissock et al. 1998].

The measured daily hot water consumption values for 1990 and 1994 are shown in Figure A-4. Summer use of hot water was essentially eliminated and peak winter consumption cut in half.

The annualized savings estimated in the audit and the savings determined from the measured consumption data, before and after the retrofit, are shown in Table A-1. The "measured" savings correspond to those for 1994, which was chosen as a typical year after the retrofit and before the Continuous CommissioningSM process was applied to the building. The fan power savings were determined by subtracting 1994 hourly consumption values from the average hourly fan power consumption before the retrofit. The fractional reduction in fan consumption is very close to that projected in

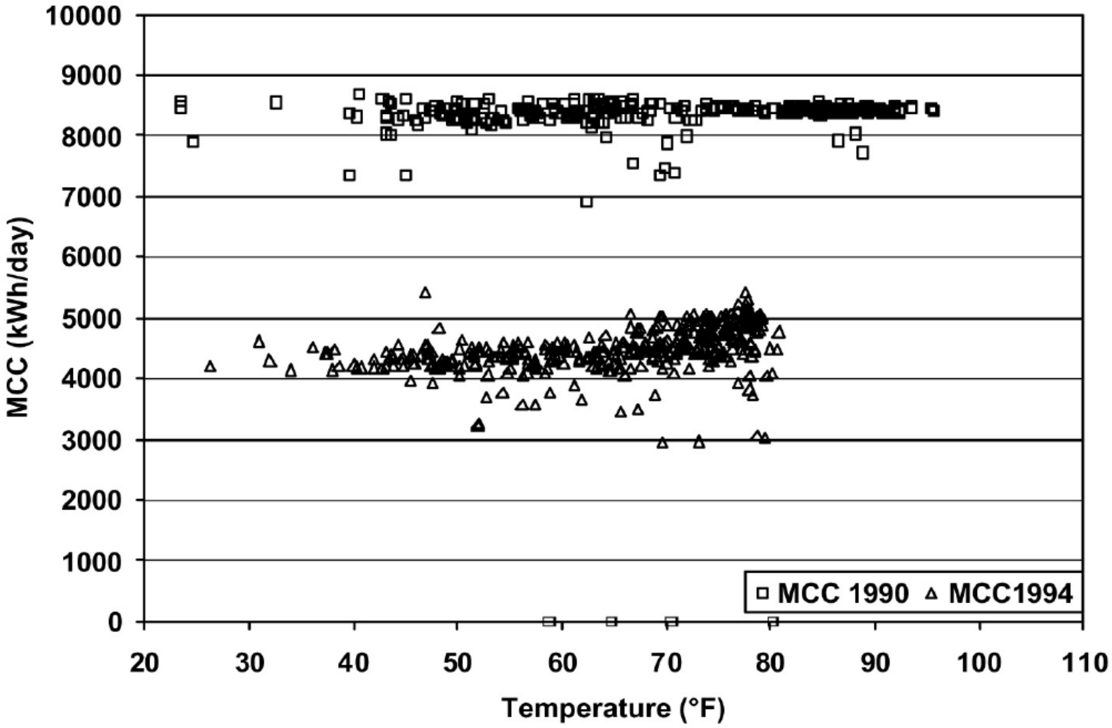

Fig. 2: Figure A-2. ZEC Daily MCC Consumption in 1990 Before the Retrofit, in 1994 After the Retrofit

the audit, but the consumption reduction is substantially smaller than expected since the audit engineers overestimated the original fan power.

The annual chilled water and hot water savings have been determined using a process that normalizes the savings from the retrofit and the subsequent CCSM process to the same weather year. The process calibrated a simulation program (AirModel) [Liu and Claridge 1998] to the measured consumption for one year prior to the retrofit (1989-90), one year after the retrofit (1994) and one year after the CCSM process was applied (1997). Each of the three calibrated simulations were then run using 1994 weather data and compared to determine the annualized savings.

Table 1: Table A-1. Retrofit Savings for the Zachry Building

—	–	Savings	–	–
—	Audit	Estimated	–	Measured
Fan power	40%	1,952,764kWh/yr	44%	1,300,000kWh/yr
Chilled Water	37%	26,600MMBtu/yr	23%	10,500MMBtu/yr
Heating Water	49%	11,500MMBtu/yr	84%	15,900MMBtu/yr

This process resulted in the weather-normalized "measured" chilled water and heating water savings shown in Table A-1. Both the fractional chilled water savings and the reduction in chilled water consumption were substantially smaller than projected in the audit. The audit engineers again overestimated pre-retrofit consumption – in this case by more than 50%. On the other hand, both the fractional hot water savings and the consumption reduction were larger than expected.

Continuous CommissioningSM of the Zachry Engineering Center

The Continuous CommissioningSM process described earlier was applied after the retrofit of this building. In 1996 and early 1997, it was applied to the Zachry Engineering Center as part of the campus-wide implementation. Metering was specified and installed in most campus buildings as described elsewhere [Claridge et al. 2000]. Metering had

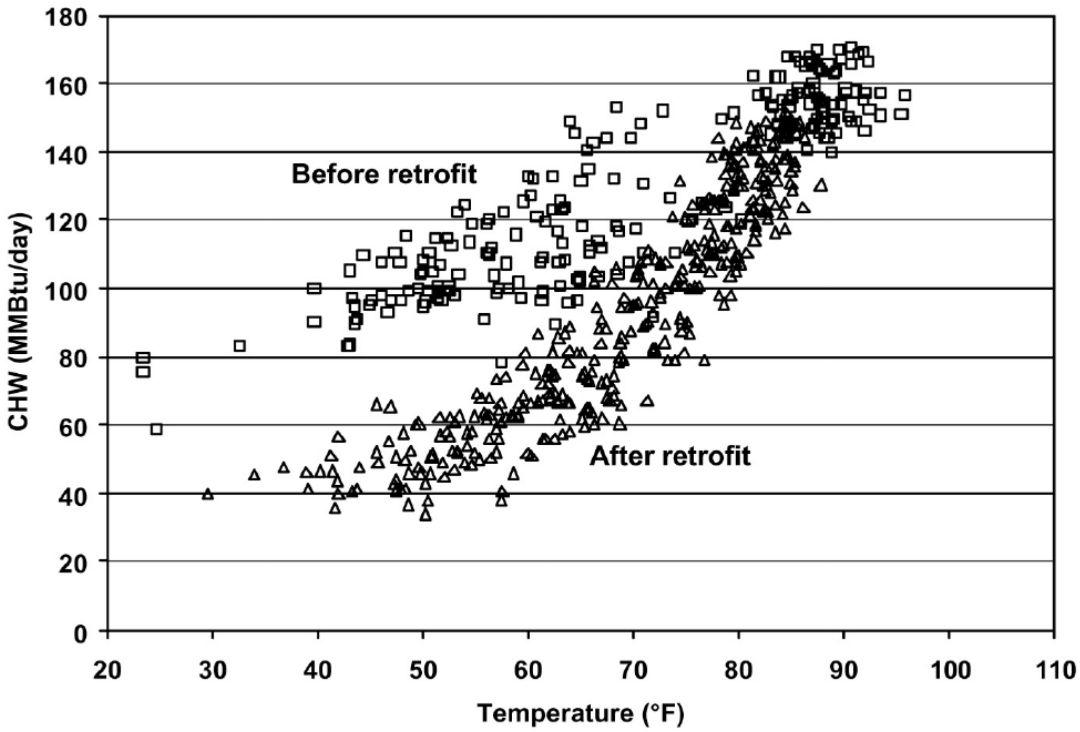

Fig. 3: Figure A-3. ZEC Daily Chilled Water Consumption in 1990 Before the Retrofit and in 1994 After the Retrofit

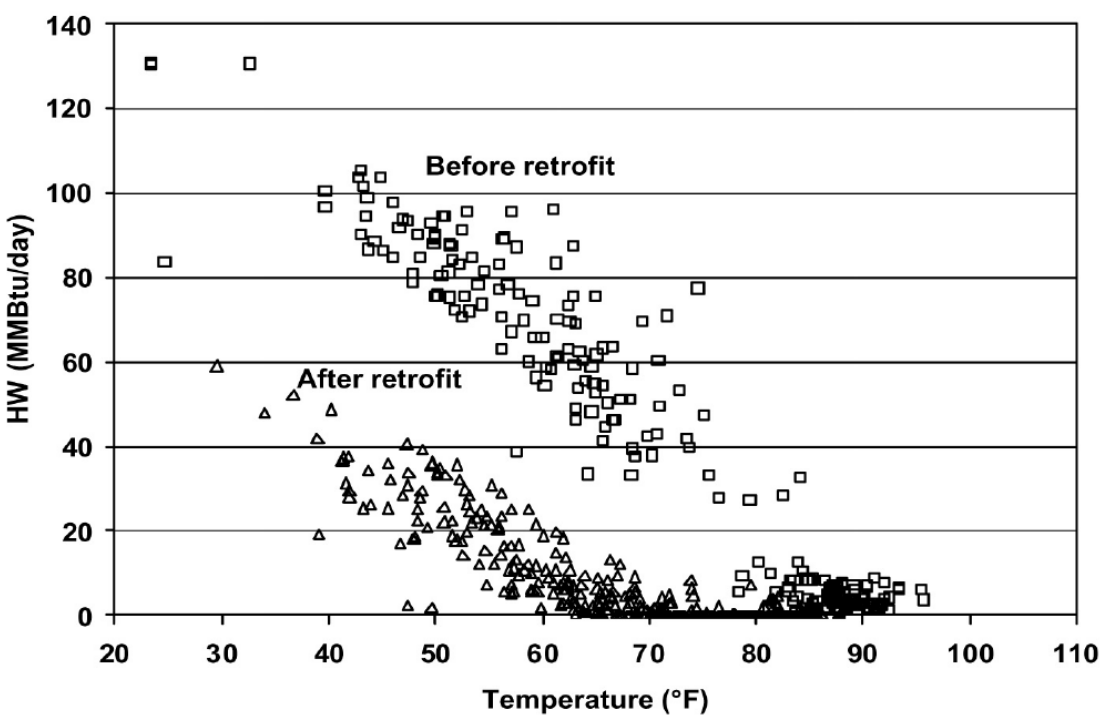

Fig. 4: Figure A-4. ZEC Daily Hot Water Consumption in 1990 Before the Retrofit and in 1994 After the Retrofit

been installed much earlier in the Zachry Engineering Center as part of the retrofit process. Therefore, no additional metering was installed.

The CCSM facility survey found that the building control system set-up was far from optimum and found numerous other problems in the building as well. The basic control strategies found in the building are summarized in Table A-2. The ranges shown for constant parameters reflect different constant values for different individual air handlers.

The control practices shown in the table are widely used in Texas, but none are optimal for this building. The campus controls engineer worked closely with the CCSM engineers during the survey. The items shown in Table A-2 could all be determined by examining the control system in the building. However, the facility survey also examined a great deal of the equipment throughout the building and found numerous cases of hot and chilled water valves that were leaking, control settings that caused continuous motion and unnecessary wear on valves, air ducts that had blown off of the terminal boxes, kinks in air ducts that led to rooms that could not be properly heated or cooled, etc.

Table 2: Table A-2. Major Control Settings Found in the Zachry Engineering Center During the CCSM Survey

Parameter	Control Practice
Pressure in air ducts	constant at 2.5-3.5 in.H2O
Cold air temperautre	Constant at 50°F-55°F
Hot air temperature	Constant at 110°F-120°F
Air flow to rooms	Variable - but inefficient
Heating pump control	Operated continuously
Cooling pump control	Variable speed with shut-off

Following the survey, the building performance was analyzed and optimum control schedules were developed for the building in cooperation with the campus controls engineer. Major control parameters for the air handlers, pumps and terminal boxes were changed to values shown in the "Post-CCSM" column of Table A-3.

In addition to optimizing the control settings for the building's heating and cooling systems, numerous problems specific to individual rooms, ducts or terminal boxes were diagnosed and resolved. These included: damper motors that were disconnected, bent air ducts that could not supply enough air to properly control room temperature, leaking air dampers, and dampers that indicated open when only partly open.

Problems of this sort had often led to occupant complaints that were partially resolved without fixing the real problem. For example, if a duct was constricted so that inadequate air flow reached a room, the pressure in the air handler might be increased to get additional flow into the room. "Fixes" like this typically improve room comfort, but lead to additional heating and cooling consumption in every other room on the same air handler. As has been discussed previously, an important part of the CCSM process is to find and fix problems that lead to energy inefficiency or to comfort problems. Too often "band-aids" are applied, and problems are not actually solved.

Table 3: Table A-3. Major Control Settings in the Zachry Engineering Center Before and After Implementing CCSM

Parameter	Pre-CC Control Practice	Post-CC Control Practice
Pressure in air ducts	Constant at 2.5-3.5 in. H2O	1.0-2.0 in.H2O as Toa increases
Cold air temperature	Constant at 50°F-55°F	60°F-55°F as Toa increases
Hot air temperature	Constant at 110°F-120°F	90°F-70°F as Toa increases
Air flow to rooms	Variable - but inefficient	Optimized min/max flow and damper operation
Heating pump control	Operated continuously	Off when Toa>55°F
Cooling pump control	Variable speed with shut-off	Pressure depends on flow

Most of the control parameters were optimized to vary as a function of outside air temperature, Toa, as indicated.

Implementing these measures resulted in significant additional savings beyond the original savings from the VAV retrofit and controls upgrade as shown in Figures A-5, A-6 and A-7. Figure A-5 shows the motor control center power consumption as a function of ambient temperature for 1990, 1994 and 1997. It is evident that the minimum fan power has been cut in half and there has been some reduction even at summer design conditions. Figure A-7 shows the hot water consumption for 1990, 1994 and 1997, again as a function of daily average temperature. The retrofit reduced the annual hot water (HW) consumption for heating to only 16% of the baseline, so there was little room for further reduction. However, it can be seen that the CCSM measures further reduced HW consumption, particularly at low temperatures. The largest savings from the CCSM measures are seen in the chilled water consumption as shown in Figure A-6. The largest fractional savings occur at low ambient temperatures, but the largest absolute savings occur at the highest ambient temperatures. The annualized consumption values for the baseline, post-retrofit and post-CCSM conditions are shown in Table A-4. The MCC consumption for 1997 was 1,209,918 kWh, 74% of the 1994 consumption and only 41% of the 1990 consumption. On an annual basis, the post-CCSM HW consumption normalized to 1994 weather was 1,940 MMBtu/yr, a reduction to only 10% of baseline consumption and a reduction of 34% from the 1994 consumption. The CCSM measures reduced the post-CCSM chilled water (CHW) consumption to 17,440 MMBtu, a reduction of 17,820 MMBtu/yr which is noticeably larger than the 13,930 MMBtu/yr savings produced by the retrofit. The CHW savings accounted for the largest portion of the CCSM savings in this cooling dominated climate.

Table 4: Table A-4. Consumption at the Zachry Engineering Center Before and After Retrofit and After Implementing CCSM Measures

–	Baseline Consumption	Post-retrofit	Post-Retrofit %	Post-CC
Fan Power	2,950,000 kWh	1,640,000 kWh	56%	1,210,000 kWh
Chilled Water	45,779 MMBtu	35,258 MMBtu	77%	17,440 MMBtu
Heating Water	18,766 MMBtu	2,938 MMBtu	16%	1,943 MMBtu

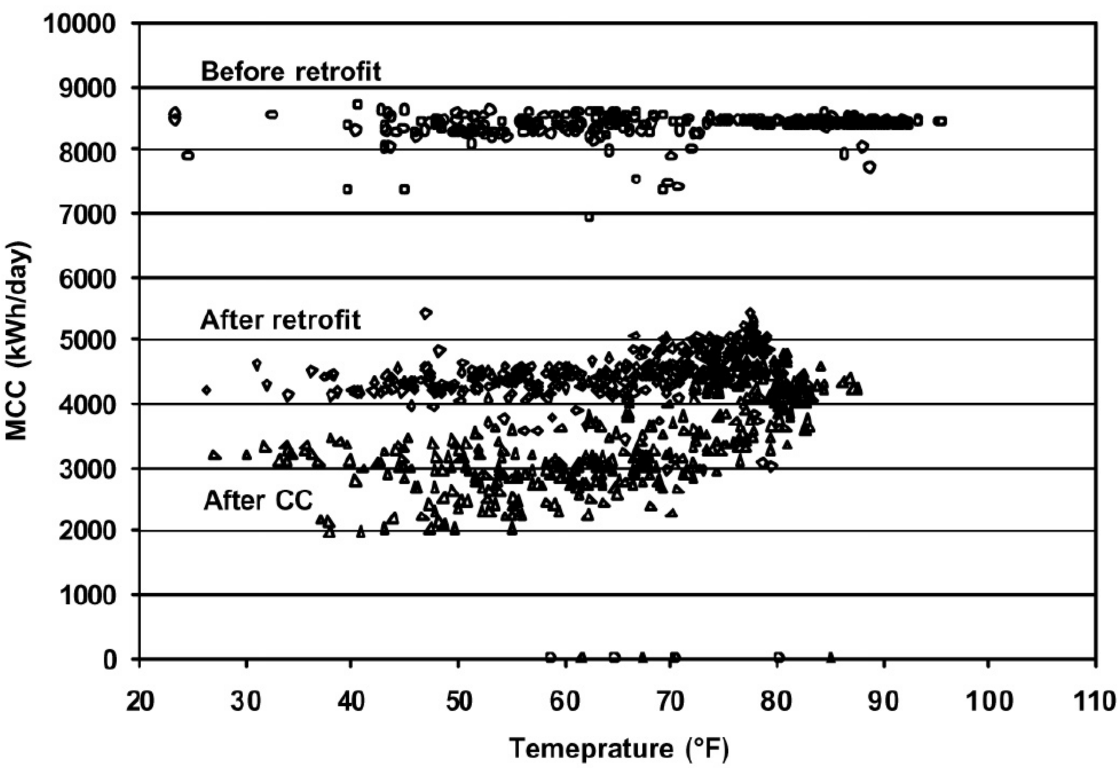

Fig. 5: Figure A-5. ZEC Daily Motor Control Center Electricity Consumption for 1990 Before the Retrofit, 1994 After the Retrofit and 1997 After CCSM

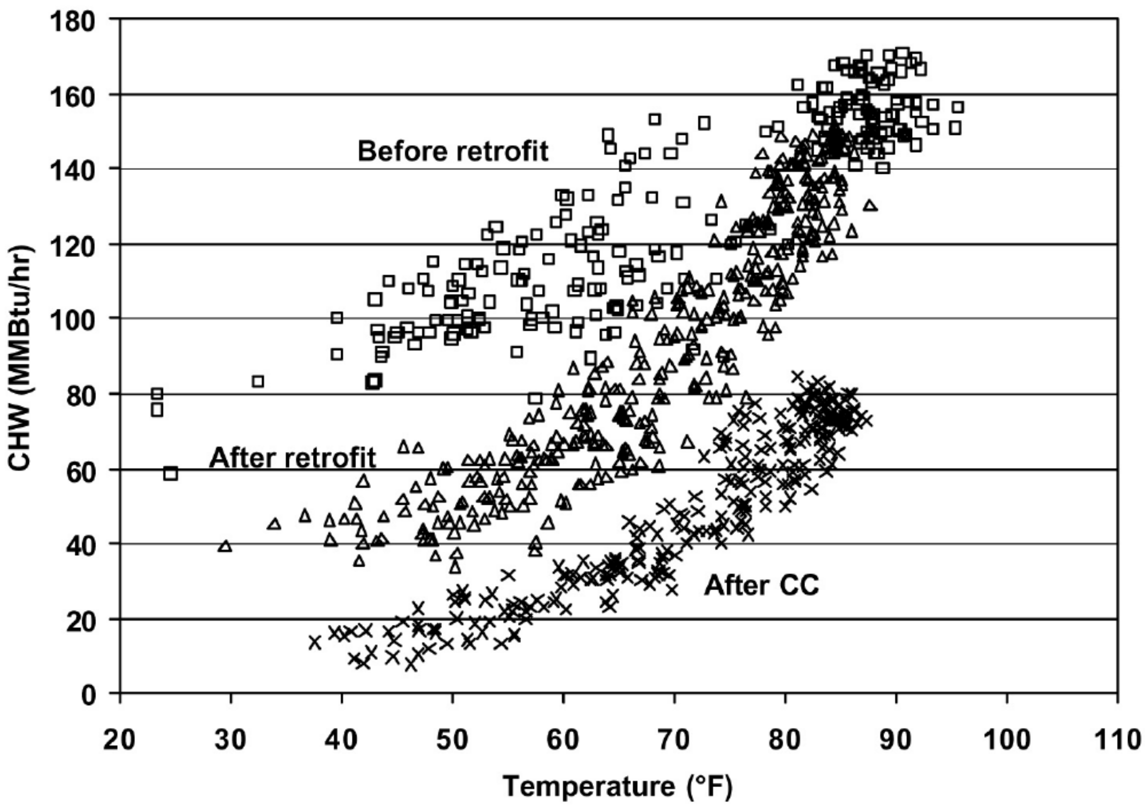

Fig. 6: Figure A-6. ZEC Daily Chilled Water Consumption for 1990 Before the Retrofit, 1994 After the Retrofit and 1997 After CCSM

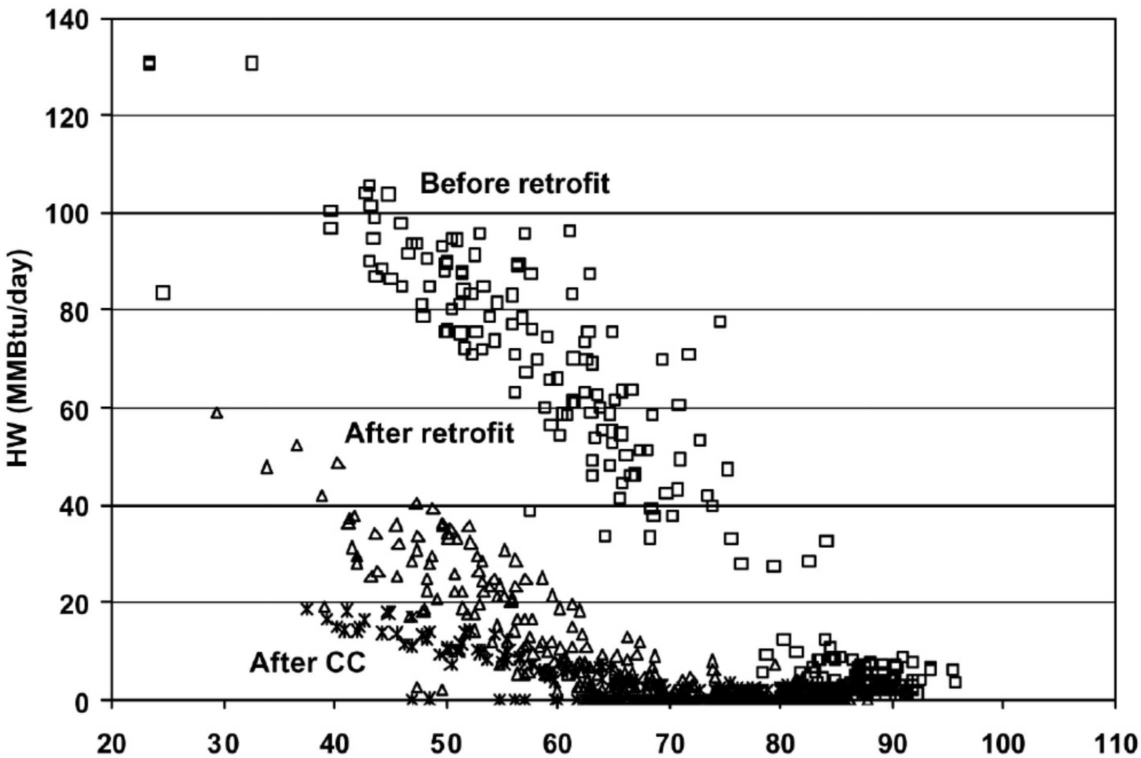

Fig. 7: Figure A-7. ZEC Daily Hot Water Consumption for 1990 Before the Retrofit, 1994 After the Retrofit and 1997 After CCSM

10.2 Project 2: Brooke Army Medical Center Continuous CommissioningSM

The Brooke Army Medical Center (BAMC) was a relatively new facility when the Energy Systems Laboratory (ESL) was hired to commission it. The facility was operated for the Army by a third-party company, and was operated in accordance with the original design intent. As has been discussed throughout this guidebook, the CCSM process looks at the facility as it is being operated and attempts to optimize the energy-using systems. This case study illustrates a wide range of CCSM opportunities.

Fig. 8: Figure A-8. The Brooke Army Medical Center (BAMC) in San Antonio, Texas

Building and HVAC Systems

The Brooke Army Medical Complex is a large, multi-functional medical facility. It consists of a medical center (main hospital), a research building (R) and a central energy plant (CEP). The medical center consists of four interconnected buildings: the C building (4 stories), M building (6 stories), A building (6 stories) and B building (8 stories) with a total floor area of 1,349,707 sq.ft. The research building is a three-story building with a floor area of 118,886 sq.ft. Figure A-9 illustrates the layout of typical floors in the complex.

The complex includes outpatient clinics, a nuclear medicine area, pharmacy areas, ICUs, CCUs, surgical areas, in-patient beds, emergency rooms, diagnostic areas, research labs, offices, animal holding areas, a cafeteria, computer rooms, training classrooms and an auditorium.

The complex is equipped with a central energy plant that has four 1,200-ton water-cooled electric chillers. Four primary pumps (75 hp. each) are used to pump water through the chillers. Two secondary pumps (200 hp. each), equipped with VFDs, supply chilled water from the plant to the building entrances. Fourteen chilled water risers equipped with 28 pumps totaling 557 hp. are used to pump chilled water to all of the AHUs and small FCUs. All of the chilled water riser pumps are equipped with VFDs.

This plant includes four natural gas-fired steam boilers. The maximum output for each boiler is 20 MMBtu/hr. Steam is supplied to each building where heating water is generated. The steam pressure set point for the boilers, prior to CCSM, was 125 psi.

Ninety major AHUs with a total of 2,570 hp. serve the complex. VFDs are installed in 65 AHUs. The others are constant volume systems. The complex contains 2,700 terminal boxes, of which 27% are dual duct variable volume (DDVAV) boxes, 71% are dual duct constant volume (DDCV) boxes, and 2% are single duct variable volume (SDVAV) boxes. Neither the warehouse nor the auditorium has terminal boxes.

The HVAC systems (chillers, boilers, AHUs, pumps, terminal boxes and room conditions) are controlled by a York DDC control system. Individual controller-field panels are used for the AHUs and water loops located in the mechanical rooms, which are also accessed by the central control system through an interface-ProComm plus and Facility Manager. The program and parameters can be changed by the central computers or by the field panels.

Energy Baseline and Metering

Once the contract was signed for the commissioning, metering was installed on the whole facility and the central plant to monitor whole facility energy consumption. Because the metering was not installed in time to obtain the pre-CCSM baseline, monthly utility bills were used for the pre-CCSM energy consumption baseline. The hourly data were used to determine the post-CCSM consumption. Some short-term loggers were also installed.

Building Conditions Before CCSM

A survey of every room was conducted to determine its function and operating hours as part of the effort to document details of system operation prior to developing and implementing CCSM measures. The EMCS was used to implement numerous control measures that increased system efficiency, including the following:

- Hot deck reset control for the AHUs
- Cold deck reset during unoccupied periods for some units
- Static pressure reset between high and low limits for VAV units
- Hot water supply temperature control with reset schedule
- VFD control of chilled water pumps with ΔP set point (no reset schedule)
- Box level control and monitoring

The facility was well maintained by the facility operator in accordance with the original design intent and the complex exhibited above average energy efficiency for a large hospital.

Commissioning Activities

Because the case study hospital is such a multi-functional complex, the commissioning activities were performed at the terminal box level, AHU level, loop level and central plant level. Several different types of improved operating measures and energy solutions were implemented in different HVAC systems. Each measure will be discussed, starting with the air handling units.

Optimizing AHU Operation

A total of 54 VAV AHUs serve the clinic areas, diagnostic areas, pharmacy, research laboratory, food services areas, offices, lobby areas, maintenance areas and storage rooms. The cold deck (CD) temperature set points were operated with a constant set point, ranging from 55 to 57F, and were occasionally adjusted by the operator during occupied periods. The hot deck (HD) temperature set points were modulated between a low limit and a high limit, according to the box requirements. If there is no call for heat from the box, the hot deck is maintained at the low limit. If there is a call for heat from a box, the hot deck will be increased up to the high limit until there is no longer a call for heating.

The actual measured results from the site measurements and short-term data loggers showed that HD temperatures ranged from 75F to 93F during the hot summer. The static pressure set points were modulated between the low and high limits according to the box conditions. The supply fan ranged from 55 Hz to 60 Hz for different AHUs during the site visit period. Table A-5 shows the control schedules before commissioning.

Table 5: Table A-5. Typical Set Points for the HD and Static Pressure Before Commissioning

Limit	HD Set Point	Static Pressure Set Point
High limit	110F if Toa<30F, 90F if Toa>70	1.8” H20 (fixed value)
Low limit	90F if Toa<30F, 70F if Toa>70F	1.4” H20 (fixed value)

The CCSM investigation found that the high hot deck temperature caused excessive heating and cooling consumption and hot calls during the summer. The cold deck set points, with a reset schedule, can better satisfy the building load under different weather conditions and reduce simultaneous cooling and heating consumption during winter conditions. Measurements of the static pressures indicated the static pressures could be much lower than the original values. The higher static pressures caused more fan electricity consumption and excess cold and hot air to the building. Based on measurements and calculations, the improved commissioning control schedules are presented in Tables A-6, A-7, and A-8.

Table 6: Table A-6. Post-CCSM HD Temperature Reset Schedules for DDVAV AHUs

Limit	HD Set Point
High limit	94-98F if Toa<30F, 75F if Toa>78F
Low limit	74-76F if Toa<30F, 70F if Toa>78F

Table 7: Table A-7. Post-CCSM CD Temperature Reset Schedules for DDVAV AHUs

Time	HD Set Point
Occupied period	94-98F if Toa<30F, 75F if Toa>78F
Unoccupied period	74-76F if Toa<30F, 70F if Toa>78F

Table 8: Table A-8. Post-CCSM Static Pressure Set Point Limits for DDVAV AHUs

Limit	C building	M building	A building	B building	R building
–	Inch H20	Inch H20	Inch H20	Inch H20	Inch H20
High limit	0.9-1.3	1.0-1.8	1.0-1.8	1.0-1.8	1.5-1.8
Low limit	0.5	0.5-0.7	0.5-0.7	0.5	0.5

Note: The static pressure set point limits are determined based on the duct condition, box condition, damper condition, and special resistance factors such as HEPA filters.

Prior to CCSM, the preheat set points were 2-5 degrees below the cold deck set points, causing simultaneous heating and cooling if the temperature sensors had an error. The new preheat set points after CCSM are 40F for all the AHUs.

Before commissioning, 12 AHUs in the C building had 2 hours of shutdown at night. The control philosophy allowed the indoor temperature to swing from 65F to 85F. In order to reduce the swing in the indoor air conditions, the AHUs are now run 24 hours a day after the commissioning. The continuous (24 hours) operation provides a constant room condition. The actual energy consumption did not rise significantly since the envelope loads were small for this section of building.

As one of the commissioning measures during unoccupied periods, in order to reduce the outside air intake after hours, the relief air dampers were closed for all the units. To maintain the building pressure, some of the exhaust fans were shut down as well.

The optimized operation schedules were tested and implemented in all 54 VAV AHUs. The commissioning team also performed troubleshooting and fine-tuning on the AHUs and associated terminal boxes. The indoor conditions were improved and the fan power consumption, as well as thermal energy consumption, was reduced significantly.

A total of nine constant-volume AHUs serve the diagnostic areas, pharmacy, offices, classroom areas and health promotion center. Seven constant-volume AHUs serve inpatient areas, which have special requirements such as indoor temperatures for burn patients.

The cold deck temperature set point was operated with a constant set point, ranging from 55F to 57F, and was occasionally adjusted by the operator during occupied periods. The hot deck (HD) temperature set points were modulated between the low limit and the high limit according to the terminal box requirements. The limits were almost the same for the AHUs and can be seen in Table A-9.

Table 9: Table A-9. Typical Set Points for HD Before Commissioning

Limit	HD Set Point
High limit	110F if Toa<30F, 90F if Toa>70F
Low limit	90F if Toa<30F, 70F if Toa>70F

The outside air intakes were constant day and night. All dampers stayed in the same position day and night. The relief and return air dampers were 100% open for all units. The post-commissioning control schedules are presented in Tables A-10 and A-11.

Table 10: Table A-10. Post-CCSM HD Temperature Reset Schedules for DDCV AHUs

Limit	HD Set Point
High limit	94-98F if Toa<30F, 75F if Toa>78F
Low limit	74F if Toa<30F, 70F if Toa>78F

Table 11: Table A-11. Post-CCSM CD Temperature Reset Schedules for DDCV AHUs

Limit	CD Set Point
High limit	60F if Toa<50F, 56F if Toa>80F
Low limit	Keep existing unoccupied schedule 80F if Tret<55F, 55F if Tret>80F

The new preheat set point after commissioning was 40F for all the AHUs, and the relief air dampers were closed for all the units during unoccupied periods. Some of the exhaust fans were shut down to maintain positive building pressure.

Due to special inpatient requirements, there was no modification for seven AHUs.

Two unique and large VAV multi-zone AHUs serve large storage areas and newly renovated office areas in the C building. The operation schedules before the commissioning can be seen in Table A-12. The outside air and relief dampers were always open, and the preheat set point was 2-5F lower than the CD temperature set points. There are no terminal boxes for the system, only supply ducts.

Table 12: Table A-12. Detailed Schedules for HD, CD and Static Pressure Before Commissioning

Item	HD Set Point	CD Set Point	Static Pressure Set Point
Occupied	90F to 100F if Toa<30F, 70F if Toa>70F	56 to 57F	0.9" H2O
Unoccupied	90F to 100F if Toa<30F, 70F if Toa>70F	80F if Tret<55F, 57F if Tret>80F	0.9" H2O

Through field measurements and analysis, the following opportunities to improve the operation of the two VAV multi-zone AHUs were identified:

- Balance zone air and determine new static pressure set points for VFDs
- Optimize the cold deck temperature set points with reset schedules
- Optimize the hot deck temperature reset schedules
- Control outside air intake, relief damper during unoccupied periods
- Optimize time schedule for fans to improve room conditions
- Improve the preheat temperature set point to avoid unnecessary preheating

The new operation schedules are presented in Tables A-13, A-14, and A-15.

Optimization of Terminal Box Operation

A total of 2,700 terminal boxes supplied conditioned air to the rooms. 27% are DDVAV boxes, 71% are DDCV boxes, and 2% are SDVAV boxes. The original control logic of the DDVAV boxes was the same as the constant volume terminal box operation, but different cfm settings were used at minimum and maximum conditions. The minimum air flow cfm settings for the VAV boxes were the same both day and night and ranged from 30% to 90% with an average of 60% of maximum air flow for the box. This schedule consumes excess heating and cooling air under normal room load conditions. Also, the heating capacity was limited for some boxes due to the existing box design setting. In some cases, the boxes supplied a limited amount of hot air, even in the full heating mode, even though the hot duct of the boxes could allow more flow through. To meet the minimum air flow requirements, the boxes had to use more cold air than was necessary.

Table 13: Table A-13. Post-CCSM HD Reset Schedules

Time	HD Set Point
Occupied periods (5:00 to 18:00)	76F if Min(damper%z1;damper%z2)<10%, 70F if Min(damper%z1;damper%2)>50%
Unoccupied periods (18:00 to 5:00)	72F if Min(damper%z1;damper%z2)<10%, 68F if Min(damper%z1;damper%z2)>50%

Note: damper%Z1:zone 1 damper control output. 0% means full open for the hot damper and full closed for the cold damper; 100% means full open for cold damper and full closed for hot damper.

Table 14: Table A-14. Post-CCSM CD Set Points

Time	HD Set Point
Occupied periods (5:00 to 18:00)	57F if Max(damper%z1;damper%z2)>90%, 60F if Max(damper%z1;damper%z2)<50%
Unoccupied periods (18:00 to 5:00)	Keep existing reset schedule 80F if Tret<55F, 55F if Tret>80F

Note: damper%z1:zone 1 damper control output. 0% means full open for the hot damper and full closed for the cold damper; 100% means full open for cold damper and full closed for hot damper

Table 15: Table A-15. Post-CCSM State Pressure Set Points

Time	LC11	LC12
Occupied periods (5:00 to 18:00)	0.8	0.9
Unoccupied periods (18:00 to 5:00)	0.5	0.5

The preheat set point changed to 40°F for the two units. The OA and relief dampers will be closed during unoccupied periods.

New control logic was developed that made the box run on a VAV operation schedule. As a result, simultaneous cooling and heating were reduced significantly during normal load conditions for the box. The hot air capacity was increased by an average of 30% in the full heating mode. The minimum supply air flow requirement was also satisfied.

Unoccupied Period Setback for the VAV Terminal Boxes

The setback control is as follows:

- Keep the room temperature set points the same as occupied periods (BAMC request)
- Reduce total flow minimum value to 0. The box will provide enough air when the load increases.

Unoccupied Period Setback for the CV Terminal Boxes

The setback control is as follows:

- Keep the room temperature set points the same as occupied periods based on existing set points (BAMC request)
- Reduce total flow to a percentage of the design flow. The percentage is determined based on the building pressure analysis. Generally, the percentage is from 30% to 70% for different AHUs.

Troubleshooting

During the commissioning period, it was found that some terminal boxes could not provide the required air flow before or after the control program modification. The major reason was higher flow resistance from the flexible and kinked ducts to the terminal boxes. The CCSM engineers performed detailed checks for every box on the computer first, then conducted field measurements for all the troublesome boxes. Specific problems were identified for approximately 200 boxes. The operation and maintenance personnel fixed the problems following the recommendations.

Loop Commissioning

Fourteen chilled water risers equipped with 28 pumps provided chilled water to the entire complex. One pump per riser was needed; the second pump was standby. During the commissioning audit phase, the following were observed:

- All the riser pumps were equipped with VFDs and were running from 41 Hz to 60 Hz
- All the manual balancing valves on the risers were balanced 30% to 60% open
- The ΔP sensor for each riser was located 10 to 20 feet from the far-end coil of the AHU on the top floor
- Differential pressure set points for each riser ranged from 13 psi to 26 psi
- The return loop has no control valve
- Although most of the cold deck temperatures were holding well, the cooling coils on 13 AHUs were 100% open but could not maintain cold deck temperature set points

Commissioning measures

Since the risers were equipped with VFDs, traditional manual balancing techniques were not appropriate. All the risers were rebalanced initially by opening all of the manual balancing valves. The actual pressure requirements for each riser were measured. It was determined that the ΔP for each riser could be reduced significantly.

Table A-16 summarizes the riser conditions before and after the commissioning and the horsepower savings for each riser pump.

After balancing, there was better cold deck temperature control of the AHUs, as well as a significant reduction in pumping requirements.

Table 16: Table A-16. Summary of Chilled Water System Condition Before And After Reset And New Set Points

Riser	Before Reset				After DP Reset				% of rated HP	Rated HP
Name	DP set point	Time	DP meas.	Pump Hz	DP set point	Time	DP meas.	Pump Hz	Savings	—
C11/C12	13	14:00	13.1	41.3	6	10:20	6.2	27.7	23%	15
C21/C22	16	14:20	16.1	49.1	7	15:50	7	33.8	37%	15
M1	15.2	11:00	15	45	7	13:40	7	29.1	31%	15
M2	25	11:10	25	60	8	13:45	8	37	77%	20
M3	25	15:00	24.5	60	8	16:00	7.8	35.4	79%	20
M4	26	15:10	24	59.8	8	16:06	8	40	69%	20
A1	20	9:00	20.5	50.5	8	10:50	8	33	43%	20
A2	16	13:15	16.2	40.3	13	16:19	13.1	36.7	7%	40
A3	15.2	11:20	15	43.8	8	15:00	8	32	24%	20
A4	16.2	13:50	16.3	53	12	13:16	12	45.3	26%	15
B1	14	14:00	13.6	50.7	8	14:30	8	42.7	24%	15
B2	17	14:00	17	55.7	9	13:42	8.7	42.2	45%	15
R1/R2	14	11:40	14	54.7	8	15:50	8	44.1	36%	15
R3/R4	16	11:45	15	50.4	8	15:54	7.1	36	38%	25
Ave/Tot									40%	270
Total Savings									108hp (80kw)	

Central Plant and Distribution Loops

Boiler steam pressure and boiler operation

The original steam pressure was set to 125 psi. However, the actual required pressure for BAMC was less than 125 psi. The recommendation was made to reduce the boiler pressure to 100 psi, thus reducing losses and gas consumption. The operations staff was not comfortable with 100 psi, but agreed to drop the pressure to 110 psi. The boiler efficiency increased after implementing the new steam pressure set point. Also, the practice at BAMC was to run two boilers year round. As part of commissioning, one boiler was shut down during the summer and swing seasons.

Chilled water loop

Before the commissioning, the blending valve separating the primary and secondary loops at the plant was 100% open. The primary and secondary pumps were both running. The manual valves were partially open for the secondary loop, although the secondary loop pumps were equipped with VFDs. After the commissioning audit and investigations, the following was implemented.

- Open the manual valves for the secondary loop
- Close the blending stations
- Shut down the secondary loop pumps

As a result, the primary loop pumps provided required chilled water flow and pressure to the building entrance for most of the year, and the secondary pumps stay offline for the majority of the time. The operator drops the online chiller numbers according to the load conditions and the minimum chilled water flow can be maintained to the chillers. At the same time, the chiller efficiency is also increased.

Results

Major commissioning activities were completed in April 1999 and the Energy Systems Laboratory continued to monitor BAMC through June 2000. We worked with the operations staff to assist with troubleshooting and continue to fine-tune the operation. The commissioning started in October 1998 and the 1997-1998 utility bills were used to establish the energy baseline.

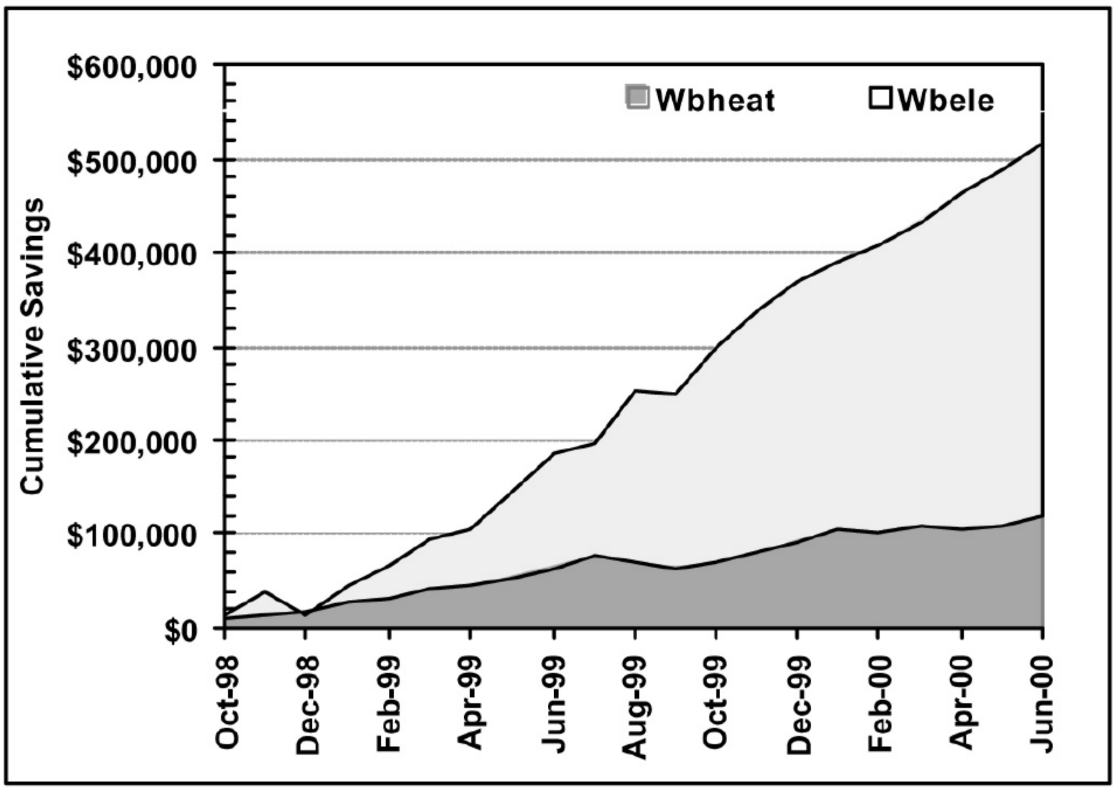

Fig. 9: Figure A-10. Accumulated CCSM Energy Cost Savings in Brooke Army Hospital

Figure A-10 presents the accumulated energy cost savings. Since the commissioning process extended over a seven-month period, savings were calculated to be approximately $105,000 during the implementation phase. For the 14-month period (May 1999-June 2000) savings were measured to be nearly $410,000, or approximately $30,000/month, based on 1997-1998 energy prices. Total savings from the commissioning process (October 1998 to June 2000) were approximately $515,000. The ESL cost to meter, monitor, commission and provide a year's follow-up services was less than $350,000. This cost does not include time for the facilities operations staff who repaired kinked flex ducts, replaced failed sensors, implemented some of the controls and subroutines, and participated in the CCSM process. More technical information regarding this case study can be found in the references.

Acknowledgements

We would like to acknowledge the assistance and cooperation of Chuck Cameron, lead operator, and Ben Keeble, project manager, of Johnson Controls, Inc.; and Roy Hirchak, chief, BAMC Facility Management, for their contributions to the BAMC Continuous CommissioningSM project.

References

Bronson, D., 1992. "Calibrated computer simulations for the analysis of retrofit energy savings. Master's Thesis. Energy Systems Laboratory Report Nos. ESL-TH-92/04-02 and ESL-TH-92/04-01. College Station: Texas A&M University System.

Bronson, D., S. Hinchey, J.S. Haberl and D. O'Neal, 1992. "A Procedure for calibrating the DOE-2 simulation program to non-weather-dependent loads." ASHRAE Transactions Vol. 98, Part I, pp. 636-652.

Claridge, D.E., M. Liu, Y. Zhu, M. Abbas, A. Athar and J.S. Haberl, 1996. "Implementation of Continuous Commissioning in the Texas LoanSTAR Program: Can You Achieve 150% of Estimated Retrofit Savings Revisited," Proc. ACEEE 1996 Summer Study on Energy Efficiency In Buildings,American Council for an Energy Efficient Economy, Washington, D.C., pp. 4.59-4.67.

Claridge, D.E., C.H. Culp, M. Liu, S. Deng, W.D. Turner and J.S. Haberl, 2002. "Campus-Wide Continuous CommissioningSM of University Buildings," Proc. of ACEEE 2000 Summer Study on Energy Efficiency in Buildings, Pacific Grove, CA, Aug. 20-25, pp. 3.101-3.112.

Claridge, D.E. and M. Liu, 2002. "HVAC System Commissioning," in Handbook of Heating, Ventilation, and Air Conditioning, Jan F. Kreider, ed., CRC Press, Boca Raton, Fla., pp. 7-1 through 7-25.

Haberl, J.S., D. Bronson, S. Hinchey, and D. O'Neal, 1993. "Graphical Tools to Help Calibrate the DOE-2 Simulation Program to Non-Weather-Dependent Measured Loads. ASHRAE Journal, Vol. 35, Part I, pp. 27-32.

Haberl, J., D. Bronson and D. O'Neal, 1995. "An Evaluation of the Impact of Using Measured Weather Data Versus TMY Weather Data in a DOE-2 Simulation of an Existing Building in Central Texas," ASHRAE Transactions Vol 101, Part II, pp. 558-576.

Katipamula, S., and C. Claridge, 1992a. "Monitored Air Handler Performance and Comparison with a Simplified System Model," ASHRAE Transactions, Vol. 98, Part I, pp. 341-351.

Katipamula, S. and D.E. Claridge, 1992b, "Importance of Monitoring Air Handler Performance," Proceedings of the ACEEE 1992 Summer Study on Energy Efficiency in Buildings, Volume 3, American Council for an Energy Efficient Economy, Washington, D.C., pp. 161-164.

Kissock, J.K., T.A. Reddy D.E. Claridge, 1998. "Ambient-Temperature Regression Analysis for Estimating Retrofit Savings In Commercial Buildings," ASME Journal of Solar Energy Engineering, Vol. 120, pp. 168-176.

Liu, M., D.E. Claridge, J.S. Haberl, W.D. Turner, 1997. "Improving Building Energy System Performance by Continuous Commissioning," Proceedings of the Thirty Second Intersociety Energy Conversion Engineering Conference, Vol. 3., July 27 - August 1, 1997, Honolulu, Hawaii, 9 pp.

Liu, M. and D.E. Claridge, 1998. "Use of Calibrated HVAC System Models to Optimize System Operation," ASME Journal of Solar Energy Engineering, Vol. 120, pp. 131-138.

Texas Energy Cost Containment Program, 1986. "Energy Cost Reduction Analysis of Texas A&M University (7110) Oceanography and Meteorology Bldg, Soil & Crop Sciences Bldg, Zachry Engineering Center," Texas Energy Engineers, Inc., 155 pp.

Zhu Y, M. Liu, T. Batten, W.D. Turner, D.E. Claridge, H. Noboa and J. Zhou, 2000. "Integrated Commissioning for a Large Medical Facility," The Twelfth Symposium on Improving Building Systems in Hot and Humid Climates Proceedings, May 15-16, San Antonio, Texas, pp. 62-69.

Zhu Y, M. Liu, T Batten, W. D. Turner and D. E. Claridge, 2000. "A Simple and Quick Chilled Water Loop Balancing for Variable Flow Systems," The 12th Symposium on Improving Building Systems in Hot and Humid Climates Proceedings, May 15-16, San Antonio, Texas, pp. 158-162.

Zhu, Y, M. Liu, T. Batten, H. Noboa, D.E. Claridge, and W.D. Turner, 2000. "Optimization Control Strategies for HVAC terminal Boxes," The 12th Symposium on Improving Building Systems in Hot and Humid Climates Proceedings, May 15-16, San Antonio, Texas, pp. 278-284.

CHAPTER 11

Indices and tables

- genindex
- modindex
- search