PNNL-29078

Pacific Northwest
NATIONAL LABORATORY

Army Re-tuning™ Implementation Guides

August 2019

RM Underhill
DJ Taasevigen

Prepared for the Deputy Assistant Secretary of the Army for Energy and Sustainability under a Government Order with the U.S. Department of Energy
Contract DE-AC05-76RL01830, Related Services

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Pacific Northwest National Laboratory
Richland, Washington 99352

Summary

The Army is a large, geographically dispersed organization with 156 installations and over 980 million square feet of building space to operate and maintain. The building stock spans a wide range of vintages and missions, with many serving cross-cutting functions and purposes. To meet Federal mandates and reduction goals organizationally, in recent years the Army has aggressively pursued policy and energy efficiency projects and programs resulting in a 9.6% reduction in energy use intensity (EUI) from fiscal year (FY) 2015 to FY2017. Despite these successes, the Army remains the largest consumer of electricity in the Federal government and spent approximately $1.1 billion in energy-related costs in FY2017

The Pacific Northwest National Laboratory (PNNL), in support of the Assistant Secretary of the Army (Installations, Energy and Environment) (ASA (IE&E)), was tasked with a multi-year study and pilot demonstration to develop a business case for the potential energy and cost reduction benefits from re-tuning™ efforts for the Army. Re-tuning is a systematic process aimed at minimizing building energy consumption by identifying and correcting operational problems that plague buildings. The methodology was developed by PNNL research staff in an effort to improve building operational efficiency at no- or low-cost, primarily through building automation system (BAS) controls. The methodology is based on two basic principles: If the equipment is not needed, turn it off; and if the equipment is not needed at full power, turn it down. Implementation of identified low-cost/no-cost operational improvements through the re-tuning process results in increased building energy efficiency, reduced operating costs, and improvement to occupant comfort.

Purpose of the Building Re-tuning Implementation Guides

The purpose of these implementation guides is to help building operations staff operate buildings more efficiently through improved building controls sequences, resulting in reduced operating costs and fewer occupant complaints. Decision makers at Army leadership and installation Department of Public Works (DPW) should direct these guides to their energy management and controls teams (including their O&M staff) to implement and monitor re-tuning measures through the means of these implementation guides.

Intended Audience and Building Type for the Building Re-tuning Implementation Guides

The intended audience for the building re-tuning implementation guides includes the following:

- Onsite employees responsible for day-to-day building operations (O&M staff, building engineers, energy management and controls staff).
- Offsite contractors (retro-commissioning agents or control vendors) hired to improve a building's energy performance.

The building re-tuning implementation guides are intended to be applied to the five building types that were included in the Army business case for re-tuning[1], which include:

- All large office (LO) buildings greater than 70,000 square feet in floor area
- All brigade headquarters (BdeHQ) buildings greater than 30,000 square feet in floor area
- All company operations facility (COF) buildings greater than 60,000 square feet in floor area

[1] Taasevigen et al. "Business Case for Re-tuning™ for the Army." Pacific Northwest National Laboratory (PNNL-28529), March 2019

- All tactical equipment maintenance facility (TEMF) buildings greater than 200,000 square feet in floor area
- All unaccompanied enlisted personnel housing (UEPH, barracks) buildings greater than 100,000 square feet in floor area.

However, the concepts and techniques presented in the guides can also be applied to any type and size of facility that has a BAS.

Energy Saving Potential from Implementation of Different Re-tuning Measures

Figure S.1 shows the energy savings potential of each individual re-tuning measure, ranked from highest to lowest among all of the buildings evaluated during the Army business case modeling and simulation effort. The following three re-tuning measures have savings potential between 8% and 10% annual energy savings:

- EEM06 [discharge-air temperature (DAT) reset],
- EEM03 (reduce VAV minimum airflow setpoints), and
- EEM02 (widen deadbands and night setback),

Two other re-tuning measures—EEM01c (shorten HVAC schedules) and EEM01b (optimal start)—produces about 5% savings. Six other re-tuning measures produce savings between 1% and 3.3%, while the remaining re-tuning measures each produce less than 1% savings.

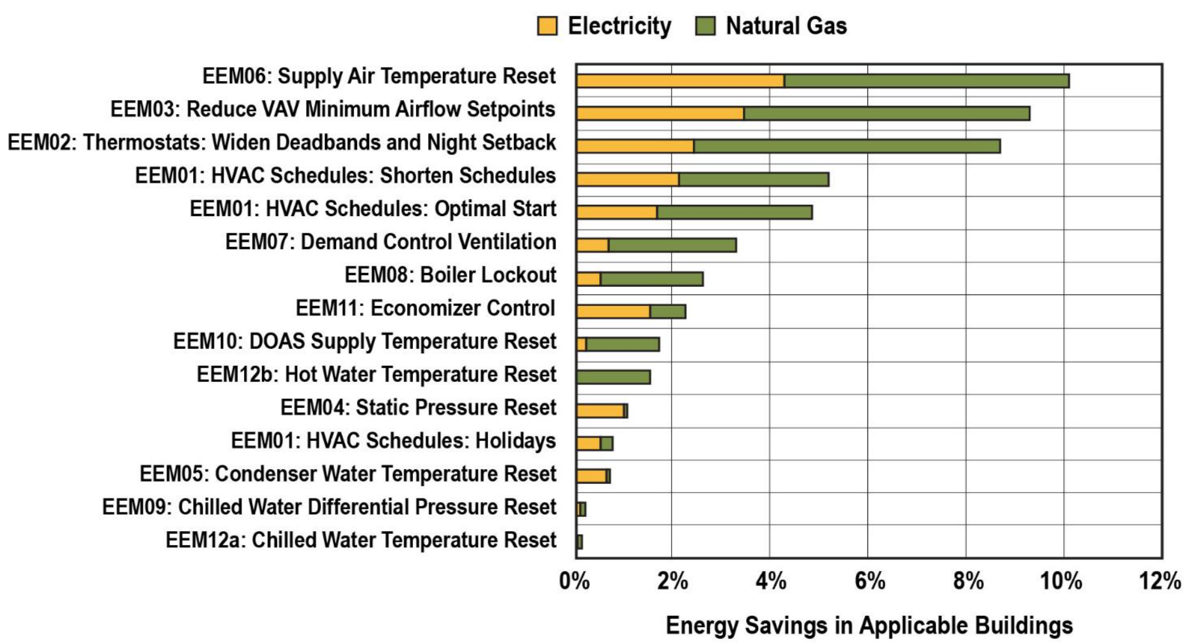

Figure S.1. Individual Measure Potential Energy Savings when Implemented in Isolation

Positive impacts to sustained persistence of implemented re-tuning measures will include operations staff who make sure the implemented measures remain operational (BAS overrides are limited and short-lived) while also making quick repairs to failed components and systems.

Negative impacts to the sustained persistence of implemented re-tuning measures include failure to maintain the BAS infrastructure (failed sensors, failed controllers and failed communications) or operational strategies that rely on long-term BAS overrides for operational performance.

Army Building Types with Most Commonly Implemented Building Re-tuning Measures

A variety of re-tuning measures were implemented during a pilot demonstration in the five building types with the priority being the highest impact measures on building efficiency. These measures varied by building type, complexity, and configuration of the HVAC systems, building mission, occupant feedback, and building occupancy schedule. Overall, PNNL was able to successfully implement 21 unique re-tuning measures across 19 buildings (covering over 2.6 million square feet) during the pilot. At any given building type between 4 and 11 re-tuning measures were implemented, and among the set of buildings at each installation, 12 to 14 total measures were identified and implemented. Following a review of the most common and impactful re-tuning energy efficiency measures (EEMs) applied to Army installations as part of the pilot, a set of 13 EEMs (some including sub-measures) were selected for inclusion in the energy modeling and simulation. These EEMs are summarized by building type in Table S.1 (are described in more detail below), and should be used by decision makers to determine which measures are applicable to the buildings selected for re-tuning at their installation.

Table S.1. Summary of Applicability of Individual Re-tuning EEMs to Specific Building Types

Measure	BdeHQ	UEPH	TEMF	COF	LO
EEM01_Holidays	*		*	*	*
EEM01_OptimalStart	*		*	*	*
EEM01_ShortenSchedules	*	*	*	*	*
EEM02_Thermostat	*	*	*	*	*
EEM03_Minimum Airflow	*			*	*
EEM04_SP_Reset	*			*	*
EEM05_CondWReset	*			*	*
EEM06_DAT_Reset	*			*	*
EEM07_DCV	*		*	*	*
EEM08_BoilerLockout	*	*	*	*	*
EEM09_CHWDPReset	*	*		*	*
EEM10_DOAS_DAT_Reset		*			*
EEM11_Economizer	*		*	*	*
EEM12_ChWTReset	*	*		*	*
EEM13_HWTReset	*	*	*	*	*

BdeHQ = brigade headquarters; UEPH = unaccompanied enlisted personnel housing; TEMF = tactical equipment maintenance facility; COF = company operations facility; LO = large office; SP = static pressure; CondW = condenser water; DAT = discharge-air temperature; DCV = demand control ventilation; CHWDP = chilled water differential pressure; DOAS = dedicated outdoor-air system; ChWT = chilled water temperature; HWT = hot water temperature.

Re-tuning Measure Descriptions

The 10 most common re-tuning measures implemented at Army installations, and included in this implementation guide, are defined below and discussed in greater detail in Sections 2.0 through 11.0). This does not include EEM02 (widen thermostat deadbands and expand heating night setback), EEM07 (demand control ventilation), and EEM09 (chilled water differential pressure reset) because they were not implemented at enough buildings and installations to fully write up implementation guides.

- EEM01: HVAC Scheduling and Optimal Start

  A common observation in Army buildings is that the HVAC schedules are set much wider than occupancy would require. The goal with scheduling is to shut off as many systems as possible whenever they are not needed, and refrain from starting up systems for occasional night-time or weekend use. From a true cost perspective, unoccupied periods can be the most expensive service, when less than 5-10% of occupants are in the building while the lights are on and the building's HVAC systems are operating. The goal for any building with properly implemented scheduling strategies is to see significant energy consumption reductions during unoccupied periods. Optimal start utilizes machine learning to optimize the daily start times of HVAC systems based on internal temperatures, occupied heating and cooling setpoints, and outdoor conditions.

- EEM03: Variable-Air-Volume (VAV) Minimum Airflow Setpoint Reductions

  VAV terminal boxes typically have minimum airflow requirements that are set during commissioning as a conservative measure to guarantee zone ventilation requirements are met at all times based on design occupancy. For many zones, this design occupancy is rarely, if ever achieved and when it is achieved, internal loads tend to drive the zone into cooling mode, which increases airflow to the zone anyway. Consequently, high minimum airflow setpoints tend to be unnecessary and are counterproductive to energy performance. High minimum airflow rates force the zone to accept too much relatively cool supply air from the AHU, forcing the zone into a heating mode. Reducing the minimum airflow rates reduces the aggregate airflow demands of the VAV system, saving fan and cooling energy, and saving significantly on zone-level reheat.

- EEM04: Static Pressure Reset

  The static pressure downstream of the supply fan is typically controlled to a fixed setpoint in VAV systems. This ensures that there is always adequate air pressure to every VAV box, even if all VAV boxes are calling for maximum airflow rates. During most operating conditions, however, reduced overall airflow demands mean that the static pressure setpoint can be reduced without compromising airflow for any of the VAV boxes. AHU fans that are equipped with variable-frequency drives (VFDs) can take advantage of fan affinity laws during low load conditions. These laws reveal that reductions in fan speed have a two to three times effect on reductions in fan power consumption. The reduction in supply fan speed translates directly to energy and cost savings without sacrificing comfort or ventilation requirements in the spaces served by the AHU(s). A static pressure reset control sequence automatically changes the static pressure setpoint in response to continuous feedback from downstream terminal boxes.

- EEM05: Condenser Water Temperature Reset

  Most water-cooled chillers have a fixed condenser water temperature setpoint. This setpoint pertains to the inlet condenser water temperature to the chiller, and is the outlet temperature maintained by the cooling tower. Cooling towers may target the condenser water temperature setpoint through a combination of changing the number of towers that are active (if they are not dedicated to individual chillers), and by ramping up and down the cooling tower's fan speed (or switching between fan speeds in the case of two-speed towers). The condenser water temperature setpoint, from an energy perspective, is a tradeoff between reducing power input to the chiller's compressor by reducing the condensing temperature and pressure, and reducing power to the cooling tower fans by increasing the condenser water temperature. Often, there is an optimal setpoint that balances these considerations to minimize total system electricity consumption.

- EEM06: Discharge-Air Temperature (DAT) Reset

For VAV systems, a reset of the supply (or discharge) -air temperature (SAT, DAT), when applied appropriately, can save on heating and electricity. During times when there is significant use of zone-level reheat, resetting the DAT higher can help to reduce simultaneous heating (at the VAV box reheat coils) and cooling (at the AHU's cooling coil). During times when there is little to no use of zone-level reheat, a reduction in DAT can provide the same amount of cooling energy, but with less airflow, and thus with less fan power required. An effective DAT reset strategy is able to strategically increase and decrease the DAT to take these factors into account.

- EEM08: Boiler Lockout

  Use of boiler lockouts based on an outdoor-air temperature (OAT) threshold is a common practice for managing inappropriate use of heating; however, lockouts are only present in a subset of buildings. A common scenario preventing effective boiler lockouts (or in other words driving summer space heating) is that building HVAC systems are often designed to provide excessive (or at least very conservative) guarantees of outdoor-air delivery at the zone level. When certain zones have very low internal loads (e.g., when they are unoccupied) but are forced to receive high supply-air fractions to guarantee ventilation, they become overcooled and require the use of reheat to maintain comfort. The use of reheat as a solution is in effect masking the problem of overventilation. Forcing the boilers to remain off during the summer (in real buildings) can force building operators to address the overventilation (or other component faults) that may be driving the need for heating. Boiler lockouts can also address day-night switching between heating and cooling. It is often beneficial during mild weather to restrict the use of heating at night so that the building can coast longer before requiring cooling.

- EEM10: Dedicated Outdoor Air System (DOAS) DAT Reset

  For buildings with DOASs that provide pre-conditioned outdoor air, a typical approach is to target a fixed conditioned-air (or discharge-air) setpoint. This DOAS DAT setpoint, however, can be crafted more strategically to avoid using supplemental heating and cooling, both in the DOAS system itself, or downstream in the zone or AHU. Because there are so many configurations of DOASs, the coils and heat exchangers present, and variation in the downstream use of heating and cooling, each system requires a unique analysis to anticipate ideal setpoints. Changes to the DOAS DAT setpoint should synchronize the DOAS heating and cooling coil control-valves, as well as the heat-recovery system (heat wheel, heat pipes or run-around loop). When these three sub-system components are synchronized and working correctly to maintain the desired DAT setpoint, optimal energy efficiency is achieved.

- EEM11: Economizer Control Improvements

  An air-side economizer that is properly configured will enable an AHU to use outdoor air to reduce or eliminate the need for mechanical cooling. When there is a need for cooling and the outdoor-air conditions are favorable for economizing, unconditioned outdoor air can be used to meet all of the cooling energy needs or supplement mechanical cooling. In a properly configured economizer control sequence, the outdoor, return and exhaust dampers sequence together to mix and balance the airflow streams (outdoor air, return air and exhaust air) to meet the AHU DAT setpoint or to reduce the total mechanical cooling energy required to meet the AHU DAT setpoint.

- EEM12: Chilled Water Temperature Reset

  Chilled water temperature reset attempts to save energy by increasing the chilled water temperature during times of low chilled water load (or conceptually, during times when the building's chilled water coils can make due with warmer chilled water temperatures). This can save electricity at the chiller because the warmer chilled water temperatures raise the evaporating pressure and thereby lower the pressure lift that the compressor must act against.

- EEM13: Hot Water Temperature Reset

  Hot water temperature reset attempts to save energy through one of two mechanisms. First, with lower temperature hot water flowing throughout the building, there is less heat loss in pipes, valves, and fittings between the hot water plant and the end-use coils. Because there are typically so many hot water coils throughout a building (when there is zone-level heating), there ends up being a lot of piping and a lot of opportunities for heat loss. When this heat loss occurs in unheated spaces (pipe chases, unconditioned zones, zones that are in cooling mode), the heat is at best wasted, and at worst, fighting active cooling. The second mechanism of savings applies to buildings with condensing boilers. Condensing boilers are able to recover heat that would otherwise be lost out of the boiler stack, meaning that the colder the (return) hot water temperature, the higher the fraction of combustion heat is delivered to the hot water. Conventional boilers on the other hand do not have any significant relationship between boiler efficiency and hot water temperature.

Abbreviations and Acronyms

°C	degree(s) Celsius
°F	degree(s) Fahrenheit
AHU	air-handling unit
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BAS	building automation system
BCU	blower coil unit
CAV	constant-air volume
CCV	cooling-coil-valve
CO_2	carbon dioxide
DAT	discharge-air temperature
DCV	demand control ventilation
DOAS	Dedicated Outdoor-Air System
ERV	energy recovery ventilation
FCU	fan-coil units
HDTL	high discharge-air temperature limit
hOA	enthalpy outdoor-air
hRA	enthalpy return-air
HSPL	high static pressure limit
HVAC	heating, ventilation, and air-conditioning
IAQ	indoor-air quality
LDTL	low discharge-air temperature limit
LSPL	low static pressure limit
MAT	mixed-air temperature
O&M	operations and maintenance
OAD	outdoor-air damper
OAF	outdoor-air fraction
OAT	outdoor-air temperature
OAWBT	outdoor-air wet-bulb temperature
RAT	return-air temperature
RH	relative humidity
RTU	rooftop unit
VAV	variable-air-volume
VFD	variable frequency drive
VRF	variable refrigerant flow

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1.0 Re-tuning Prerequisites

To help installations prioritize buildings for re-tuning, PNNL established the following high level prerequisites to increase the probability for successful re-tuning outcomes. These prerequisites should include the following minimum attributes at any installation:

- Building controls at the individual building-level are connected to a central building automation system (BAS)
- Reliable BAS communications
  - between all network-connected supervisory controllers and the network
  - between all field controllers and their supervisory controllers
  - between all integrated systems (boilers, chillers, variable frequency drives (VFDs), lighting controllers, etc.) and their supervisory controllers
  - between all field controllers and their connected sensors and control devices
- Reliable whole building interval-metered data that is accessible through the Metered Data Management System (MDMS), BAS, Enterprise Energy Data Reporting System (EEDRS), or other means. The installation should have at least 6 months of historical metered data available for the individual buildings that will be selected for the re-tuning effort.
- BAS and equipment failures should be identified and repaired, prior to initiating re-tuning efforts. Some common failures are described below.

1.1 List of Common Failures to Correct

The following list of common failures should be considered by operations and maintenance (O&M) staff and corrected (at the building level) prior to re-tuning or as a first step in the re-tuning process. Buildings that have gone through proper commissioning (new construction or existing buildings) exercises prior to initiating the re-tuning process are likely to address many of the common failures identified below.

- Failed temperature sensors (out of calibration, improperly wired or improperly located)
- Failed humidity sensors (out of calibration, improperly wired or improperly located)
- Failed static pressure sensors (out of calibration, improperly wired or improperly located)
- Failed airflow sensors (out of calibration, improperly wired or improperly located)
- Failed carbon dioxide (CO_2) sensors (out of calibration, improperly wired or improperly located)
- Leaking hot water valves (broken actuators, loose linkages or improperly wired)
- Leaking chilled water valves (broken actuators, loose linkages or improperly wired)
- Failed economizer dampers and variable-air-volume (VAV) box dampers
- VFD communication failures (VFD-driven fans and pumps)
- BAS alarms on HVAC equipment that are not resolved
- Equipment operating in “HAND” or manual mode
- Controllers that are offline and not communicating with the central BAS
- Operator overrides in place on several components of different HVAC systems
- Control parameters not communicating (not exposed) to the BAS
  - Impacts the ability of re-tuning improvements
  - Impacts O&M staff when they cannot adjust parameters for equipment operations
  - Impacts equipment operations which may result in 24/7 operation of equipment

- Inability to remotely control chilled water temperature or hot water temperature setpoints due to equipment integration issues
- Advanced meters that are not communicating
- Heat recovery and other HVAC systems that are not fully instrumented
- Broken fan belts (belt-driven equipment)
- Dirty or plugged filters
- Dirty or plugged coils

1.2 Screening Criteria used for Building Re-tuning Selection

Because installations may have several hundred to several thousand buildings connected to a central BAS, criteria were developed to screen installations to optimally identify the best candidates for re-tuning. 12 primary screening criteria are used, and are described below. It should be noted that criteria 8 through 12 highlight issues that, if present, should be reasons to not undertake re-tuning at a given building.

1. **Diversity in Building Type**. While re-tuning, in general, does not require diversity in building type, the overall objective of the project was to create a business case for the potential energy and cost reduction benefits derived from re-tuning for the Army. To effectively capture the energy and cost reduction potential of re-tuning for different building types common to army installations, implementation of re-tuning measures on multiple building types is needed. The goal is to identify and implement re-tuning measures in up to four different building types at each installation.
2. **Installation Flexibility**. Successful and sustainable re-tuning in any building requires the willingness of site staff to change operational strategies in the building, and for occupants to be patient as the strategies are adjusted and tuned as needed. While re-tuning has been shown overall to improve occupant comfort, implementation of re-tuning measures can be an iterative process that require adjustments based on the building's response. Patience and flexibility by site staff and building occupants is critical for successfully implementing, tuning, and maintaining re-tuning measures.
3. **BAS and Direct-Digital Control Integration**. Re-tuning from a central BAS is advantageous because multiple buildings can be controlled and monitored from one common location. To maximize the benefits of re-tuning at army installations, implementation of re-tuning measures is performed on as many candidate buildings as possible. Re-tuning of buildings with standalone control systems are more time-consuming to implement and monitor, limiting the number of buildings where re-tuning can be implemented for a site. Buildings with standalone control systems can still be great candidates for re-tuning if they are the only focus. Buildings with pneumatically actuated dampers or valves or buildings with equipment that is not connected to the central BAS, are not great candidates for re-tuning.
4. **Single-Duct Variable-Air-Volume (SDVAV) Air-Handling Units (AHUs)**. While re-tuning can be applied to a variety of AHU configurations, SDVAV systems are the most conducive configuration for the application of re-tuning principles. The Army building stock includes many building types that does not have SDVAV designs (e.g., warehouses, barracks, dining facilities).
5. **Variable-Frequency Drives (VFDs)**. One of two main principles of re-tuning is if equipment is not needed at full power, turn it down. Major fan and pumping systems that have VFDs can be leveraged in re-tuning by slowing down motors when they are not required to operate at full speed. Full-speed requirements generally make up less than 5% of the annual operating hours.
6. **Availability of BAS Trend Data**. BAS trend data is used to identify re-tuning measures prior to the re-tuning site visits. 2 to 4 weeks of sub-hourly BAS data is collected on major fan and pumping systems in each of the selected buildings and analyzed prior to the initial site visit.

7. **Availability of Whole Building Interval-Metered Data**. The whole building interval-metered data is used for the identification of re-tuning measures and the measurement and verification of energy savings derived from re-tuning. To quantify the energy savings from the effort, both baseline (before re-tuning) and post baseline (after re-tuning) data are collected for each building.
8. **Major Renovation or Occupancy Changes**. Buildings that have plans for major renovations or occupancy changes (i.e., drastic increases or reductions) are not good candidates for re-tuning because these changes strongly affect building energy consumption and make it difficult to evaluate the energy savings attributed to the re-tuning efforts. Renovation or occupancy changes may also create a subsequent need to re-evaluate some of the re-tuning measures if the loads and requirements of the internal spaces change.
9. **Plans for a BAS Upgrade**. Buildings that are on the list for BAS controller or software upgrades within 2 years of the re-tuning project are not good candidates for re-tuning. Controller and/or software upgrades can affect the re-tuning measures and overall savings derived from re-tuning.
10. **Restricted Access**. To effectively re-tune a building, a thorough building walk-through is required to evaluate the HVAC systems and the spaces served by those systems. Buildings with restricted access are not good candidates because of the inability to conduct thorough building walk-throughs.
11. **Performance Based Contracts**. Buildings that are part of or will become part of an upcoming performance-based contract (i.e., Energy Savings Performance Contract [ESPC]; Utility Energy Service Contract [UESC]) are not good candidates for re-tuning because of potential overlap in measures and difficulties related to validating energy savings derived from re-tuning versus those derived from the performance-based contract.
12. **Significant Equipment Failures or Replacements**. While re-tuning principles can be applied to older equipment and in many cases will prolong equipment operating life, equipment and systems that are scheduled to be replaced or under significant repair are not ideal for re-tuning.

1.3 How to Use the Implementation Guides

Once the installations and buildings are selected for re-tuning, the remainder of these guides (Sections 2.0 through 11.0 should be used as references to help implement and monitor specific measures in buildings. The guides are structured to answer the following questions:

- Why should I consider implementing the re-tuning measure? While the re-tuning measures might be pre-selected based on energy savings potential (shown in Figure S.1) and building-type applicability (as outlined in Table S.1), each guide provides more information to help the installation justify implementing the specific measure. For example, air-side economizing might be a much higher priority in a mild climate zone versus a hot and humid climate zone, while boiler lockouts might be much more useful in a hot climate zone versus a cold climate zone.
- Which systems should I select to implement the re-tuning measure in a given building? It is assumed up front that no two buildings are the same, and many of the systems are different as well. Because of that, each guide provides the reader with specific HVAC systems to target when implementing the re-tuning measure.
- What methodology should I use to implement the re-tuning measure? While the re-tuning methodology was developed based on engineering principles and system dynamics, the application of the re-tuning measure often times looks slightly different in each building. This part of each guide offers up different options for implementing the re-tuning measure in an attempt to cover the range of possible equipment and building configurations.
- How should I monitor the re-tuning measure during and after implementation? Re-tuning is an iterative process that requires monitoring and adjusting as feedback is collected from the building systems and the building occupants. This section of each guide aims at identifying trends to setup and monitor in the BAS and ways to identify potential issues with the re-tuning measure that require adjustment.

2.0 Scheduling/Optimal Start

The purpose of this scheduling/optimal start implementation guide is to show, through examples of good and bad operations, how the scheduling of various heating, ventilation and air-conditioning (HVAC) systems and equipment can be efficiently controlled and what the indicators are for both good and bad operations. This guide also assists in identifying HVAC systems that are good candidates for implementing scheduling/optimal start, and offers some common implementation strategies.

2.1 Why Consider Implementing Scheduling/Optimal Start?

Scheduling is very simple to implement, track, and administer. The goal of scheduling is to shut off as many systems as possible, whenever they are not needed and refrain from starting up systems for an occasional nighttime user or weekend user. From a true cost perspective, nighttime operations can be the most expensive service, when less than 5–10% of staff are working, yet most HVAC equipment is running while the lights are on and the building is being ventilated with fresh air.

The goal for any building with properly implemented scheduling strategies is for that building to see significant energy consumption reductions during night and weekend periods. The difference in consumption between the base load and the peak load will often be dependent on the climate zone and the number of process loads that exist in a building. Process loads usually run continuously (e.g., data centers, security lighting, refrigerated food storage, etc.). A valid target for setback reductions between typical occupancy periods and unoccupied periods for a typical building would be at least 50% and as much as 80% (see Figure 2.1) with aggressive setbacks on nights and weekends.

Air-handling units (AHUs), rooftop units (RTUs), fan-coil units (FCUs), variable-air-volume (VAV) boxes and other HVAC systems can be analyzed by reviewing graphs of their trend data that represent operating conditions (fan status, duct static pressure, etc.) as shown in Figure 2.2. Failure to investigate or correct/mitigate this situation, in all likelihood, will lead to increased fan, heating, and cooling energy consumption. HVAC and related electrical systems can account for a significant percentage of a building's (total) energy consumption. If HVAC systems are operating continuously (24/7) or near continuously, the energy-savings opportunities related to the scheduling/optimal start function of the HVAC system(s) is high.

Often overlooked in Building Automation System (BAS) controls is the importance of scheduling the minimum ventilation rates to the lowest value (zero) during unoccupied hours, while leaving nonessential exhaust systems (bathrooms, etc.) off. This can and should be performed by the BAS such that morning warm-up or morning cool-down functions occur without having to temper outdoor air, while the warm-up or cool-down functions are occurring (see Figure 2.3).

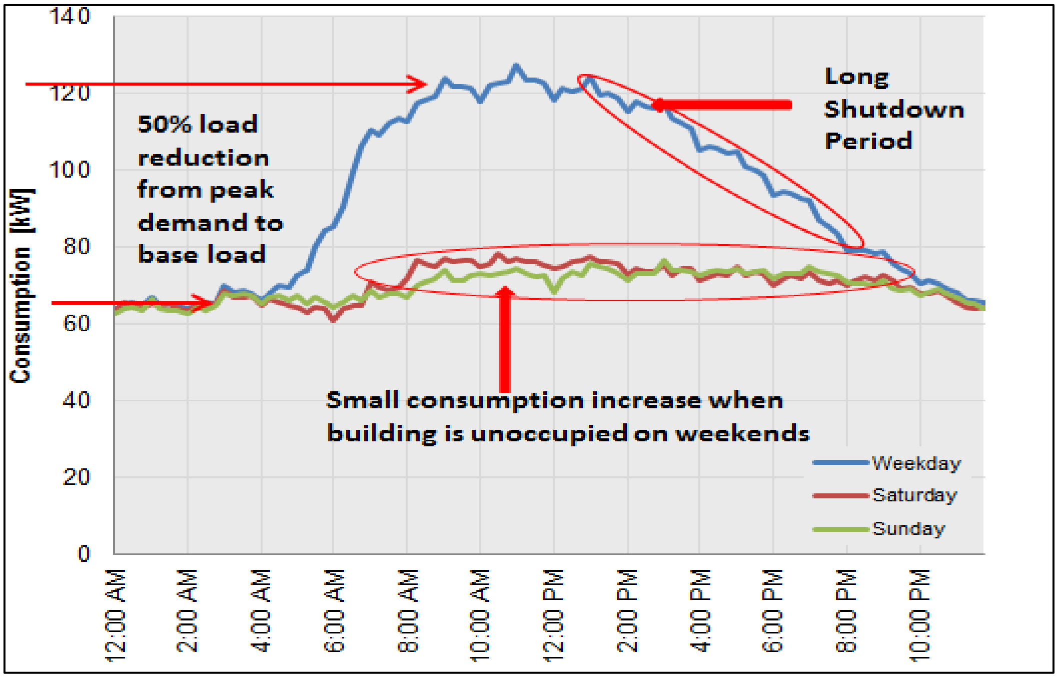

Figure 2.1. Whole-Building Consumption for Office Building with Night Setback Strategies

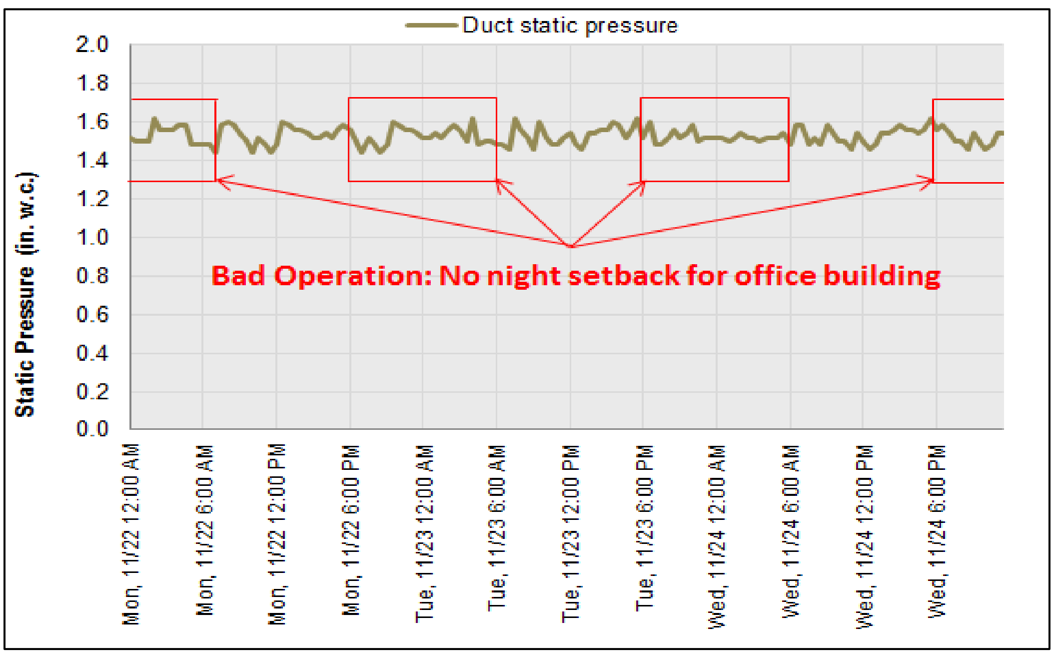

Figure 2.2. Static Pressure Does Not Reduce, Indicating Fan Is Running Continuously

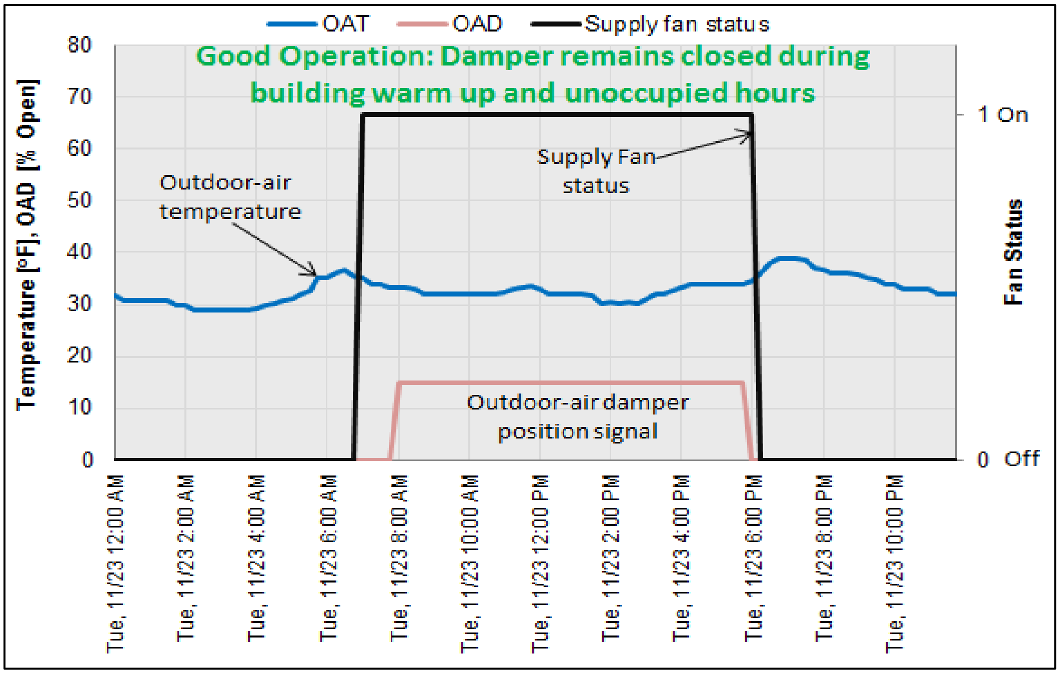

Figure 2.3. Outdoor-Air Damper Control during Building Warm-Up, Cool-Down or Occupied Periods

In some instances, HVAC systems may be required to operate continuously for one to two occupants or one to two small loads/processes. The ability to schedule HVAC systems to operate in a "setback mode" may include operating an AHU at a reduced static pressure (provide ventilation and cooling, but at reduced fan capacity) or operating a hot water plant at lower setpoints (provide heated water, but at reduced boiler or heating system capacity). In all of these situations, the scheduling of less aggressive setpoints during typical unoccupied periods recognizes that the building has fewer HVAC demands that still must be satisfied, the response rate can be slower, and it is not as critical to ensure that every setpoint is met at all times. This relaxed approach can deliver energy savings during reduced occupancy and/or reduced load periods that occur outside of the typical occupancy period. One example is operating HVAC systems for janitorial or cleaning staff that work inside a building after 6:00 p.m. (when most of the occupants have left).

Optimizing the scheduling of HVAC and other electrical loads while including optimal start for systems that serve building needs, will translate into energy and cost savings without sacrificing comfort in the spaces served by the AHU(s).

Lighting systems are not discussed in this implementation guide, but are noted as loads that are often overlooked and yet, hold great promise for energy savings when they are integrated into the BAS (and scheduled correctly). It is not unusual to find lighting systems that are turned on several hours before occupants arrive at the building and remain on several hours after occupants leave the building (including weekends when no one is in the building).

Note: Scheduling of various loads is often the number 1 energy-saving measure for a building or campus. Schedules that are overridden during extremely cold or hot weather is often the number 1 challenge to effective long-term scheduling. During extreme weather, schedules may be removed, overridden, or widened—negating the energy savings, while reducing equipment life due to extended hours of operations.

2.2 What Systems Should Be Considered for Scheduling and Optimal Start Implementation?

Some of the typical HVAC systems that may be found in various buildings that should be considered for scheduling (assuming that they are integrated into the BAS), regardless of size, include the following:

- AHUs
- RTUs
- FCUs
- VAVs
- exhaust fans
- unit heaters
- perimeter heating systems (baseboard)
- dedicated outdoor air systems (DOASs)
- pumping systems (domestic hot water, irrigation water, etc.)
- chilled-water systems (including pumping)
- hot water systems (including pumping)

Lighting systems and a myriad of other electrical loads should also be evaluated for scheduling—especially if they are already integrated into the BAS.

All end-use loads that are equipped with reliable BAS controls are good candidates for implementing scheduling as well as optimal start, under the following assumptions:

- The BAS and field controllers are reliable (communication between end-use loads and the supervisory controllers is more than 95% reliable and any integration issues for specialty systems such as chillers, boilers, VFDs, and lighting systems are fully resolved).
- Adequate and accurate zone temperature sensing is provided (*minimal to no pneumatic controls*) in the zone HVAC equipment that supports reliable morning warm-up or morning cool-down functions that might be initiated by an optimal start algorithm.

2.2.1 Methodology for Implementing Scheduling and Optimal Start

This section of the implementation guide presents some common strategies for implementing scheduling. As part of the scheduling of equipment (especially HVAC systems that maintain occupant comfort), this section also touches on aspects related to scheduling that are often overlooked (but have profound impacts on success as well as sustaining the schedules). The following list of scheduling techniques is covered in this guide:

- night setback (definition and how does it work)
- optimal start and optimal stop (definition and how it works)
- extreme weather operations.

Note that this list is not intended to cover every implementation technique, but to present a few techniques in detail to provide some context for what a good scheduling implementation might look like. All final scheduling and related parameters should be thoroughly discussed with the owner/operator of the BAS and the building(s) where the scheduling and related strategies (night setback, optimal start, and extreme weather operations) may be applied. The implementation strategy may vary by building type, system type, or other variables that this guide does not directly highlight.

2.2.1.1 Scheduling of Typical Buildings or Building Types

Depending on the building's mission and use, the scheduled hours of occupancy may be Monday–Friday or the building may operate 7 days per week. Some buildings may be in use for only 1–2 days per week (worship centers or buildings used for weekend training exercises such as those used by the Army Reserve or other education-related buildings used primarily during weekends).

Typical schedules for office buildings are often configured for 6:00 a.m. to 6:00 p.m., Monday–Friday. These hours cover the majority of hours that employees and others typically work in office-type buildings. O&M staff responsible for HVAC system operations often find that these systems must be operated prior to occupants arriving to maintain the indoor environment at acceptable levels. An acceptable indoor environment may include temperature, humidity, and indoor-air quality (IAQ). All of these variables require accurate sensors that provide reliable measurements.

When the scheduling of HVAC systems fails to provide or sustain acceptable indoor environments, O&M staff may be tempted to widen schedules (in response to occupant complaints) or to override schedules, placing HVAC systems into 24/7 operations (common during extreme weather events). Overrides of schedules (when observed) are often symptomatic of deeper issues that may represent improperly configured schedules, improperly configured optimal start, and improperly configured night setbacks.

When schedules are effectively implemented, the majority of HVAC systems are shut down as are the ancillary systems that support them (chillers, boilers and pumps, exhaust fans, miscellaneous systems, etc.).

2.2.1.2 Implementing Night Setback – Why Is This Important?

When schedules are put into place, the immediate challenge is how the building responds during the vacancy period when most of the HVAC systems are turned off. At a minimum, the HVAC controls for the zones should be configured to automatically re-activate the required HVAC systems to allow the building to maintain the zone temperatures at their setback values.

Typical setback values for heating are between 55°F and 62°F. This means that the HVAC systems will remain off until the zone temperatures (lowest or calculated average) drop below the heating setback value. Typical setback values for cooling are between 78°F and 85°F. This means that the HVAC systems will remain off until the zone temperatures (highest or calculated average) rise above the cooling setback value. The challenge for setback and schedules is the recovery period that is required to recover the zone temperatures back to desired occupancy values (typically between 68°F and 76°F) by between 6:00 a.m. and 8:00 a.m.

If the BAS does not have setback values configured or does not have the control code in place that would automatically activate the required HVAC systems to maintain the building at the setback values, then it is likely that the building will overheat during warm/hot weather and/or over-cool during cool/cold weather. This can be disastrous, not only for occupant comfort but also for food that is stored in vending machines (during hot weather) and to mitigate against possible frozen pipes that would burst, creating unwanted water damage in the building (during cold weather).

It is critical that setback values be properly configured, and that the automated response(s) that should be occurring during night and weekend periods are actually occurring in the most energy-efficient way.

Many buildings are configured with zone heating capability (perimeter baseboard, perimeter fan coil, perimeter induction box, and perimeter fan-powered VAV box). These systems are already designed to heat the building perimeter, without assistance from the AHU. As long as the hot water heating system is active, the perimeter systems should be able to maintain the night setback conditions (minimum temperatures) and may even be able to perform the morning warm-up for the building (without the AHU being active). In these cases, the AHU could be scheduled to only come on just prior to occupancy to meet the minimum ventilation requirements.

While a lower heating setback temperature (winter) and a higher cooling setback temperature (summer) will model the greatest energy savings, building operations staff may find that HVAC systems are unable to recover the zone temperatures in time, resulting in complaints when occupants arrive in over-cooled (winter) or over-heated (summer) office spaces. This can result in overrides (by O&M staff), negating the benefit of a properly configured night setback.

2.2.1.3 Optimal Start and Optimal Stop

Optimal start is an automatic feature that is embedded in most of the BAS vendor systems. Optimal start (when properly configured) should automatically start the HVAC systems at the latest time required to recover the vast majority of zone temperatures to within 1–2°F of the occupied heating or cooling thermostat setpoints by the target time (between 0 and 30 minutes prior to the start of occupancy).

Most optimal start algorithms use the outdoor-air temperature (OAT) and indoor zone temperatures to calculate the optimal starting time required to recover the zone temperatures. Recovery of zone temperatures in an optimally short time relies on an algorithm that determines how far the zone temperatures have to rise (during cold weather) or fall (during hot weather) to achieve the occupied setpoints, and uses a history (learning algorithm) of previous start-up operations to determine the expected recovery rate of zone temperatures.

Most optimal start algorithms require the accurate configuration of the optimal start parameters to match the building's mechanical design. This includes accurately informing the algorithm what the building's design-day conditions (cold-weather minimum and hot weather maximum) are and what the maximum recovery time should be for those conditions (2 hours, 4 hours, 6 hours, etc.). Most optimal start algorithms require accurate configuration of the desired heating and cooling zone temperatures when occupancy begins. If these values are set too high (winter heating) or too low (summer cooling), the optimal start program will most likely always start at the earliest possible time.

Zone temperature feedback to the optimal start program (depending upon the control vendor's software) can be configured as one zone or as many zones. Improperly configured optimal start algorithms will look at one zone temperature (instead of many) or will require the optimal start routine to achieve zone temperatures that are unattainable (too warm during cold weather, too cool during hot weather). These issues often result in optimal start programs that either under-estimate or over-estimate the time required for the HVAC system(s) to pre-start to achieve the desired results for the different zones in the building.

Many optimal start programs also provide an early stop function. Optimal (or early) stop is also a learning algorithm based on the same zone temperature(s) that will stop the AHU or RTU by several minutes (up to a maximum, configured value) and allow the building spaces to "coast." There is no ventilation during this period, because the HVAC system has been stopped, so caution should be exercised if the user decides to apply this capability to their building. Optimal stop values of no greater than 30 minutes are recommended and only widened based on experience and judgment for the spaces. Lightly occupied or sporadically occupied spaces (libraries, worship centers, auditoriums, conference rooms, tall atriums or common lobby/hallways that are exit points from the building and naturally ventilated by virtue of people exiting/entering the building) may benefit the most from this feature.

2.2.1.4 What Actions Should Be Taken When Zone Temperatures Do Not Recover in Time?

The optimum configuration of starting the HVAC system using optimal start should include a maximum pre-start of 2 hours (120 minutes). With a scheduled start time of 6:00 a.m., this means that the optimal start algorithm could calculate an early start time of 4:00 a.m. (120 minutes prior to 6:00 a.m.). If the building is operating in such a way that a 4:00 a.m. start time (for the morning warm-up or cool-down period) does not result in zone temperatures that are adequately recovered, several actions should be taken, before widening the schedules or overriding schedules to be 24/7. These actions should be thoroughly explored, prior to implementing schedule responses (widened schedule actions or overrides).

Summer – Cooling Operations

1. Verify that auxiliary systems (chillers, cooling towers, pumps, etc.) are active when AHU and zone HVAC systems activate during night setback actions, as well as morning cool-down periods.
2. Verify that the chilled-water temperature being delivered during the night setback or morning cool-down periods is sufficient to satisfy the cooling loads (AHU discharge temperature setpoints, zone FCUs, chilled beams, etc.).
3. Passive chilled beam systems rely on radiative cooling (not convective energy via air movement). Pumping systems serving radiative cooling systems often have much lower energy costs (than convective systems), so radiative cooling systems may benefit from other control sequences (versus optimal start sequences for AHUs) that may include operating the radiative cooling systems at night (when the OAT is greater than 75°F) to help minimize the setback recovery time period.

Winter – Heating Operations

1. Verify that auxiliary systems (boilers, pumps, etc.) are active when AHU and zone HVAC systems activate during night setback actions, as well as morning warm-up periods.
2. Verify that the heating hot water being delivered during the night setback or morning warm-up periods is sufficient to satisfy the heating loads (AHU discharge temperature setpoints, FCUs, VAV zone reheat, perimeter heating, etc.). If the building has a perimeter hot water heating system, ensure that the hot water heating system activates early enough to maintain the perimeter spaces of the building.
3. Perimeter systems rely on radiative heating (not convective energy via air movement). Pumping systems serving radiative heating hot water systems often have much lower energy costs (than convective systems), so perimeter radiation systems may benefit from other control sequences (versus optimal start sequences for AHUs) that may include operating the radiation system at night (when the OAT is less than 45°F) to help minimize the setback recovery time period.

Both Seasons

1. Verify that the night setback setpoints are appropriate for the building. Buildings with high thermal mass or low plant-side heating/cooling capacity and wide night setback temperatures (e.g., 55°F or 85°F) may take an excessively long time to warm up or cool down in the morning, especially after a weekend shutdown. For these buildings, 62°F/80°F setback limits are usually more appropriate (but

each building's setback limits should be evaluated to match the building's individual design and operational constraints).

2. Make sure outdoor air dampers (OADs) that have a fixed minimum damper position are separately scheduled to not have any outdoor-air introduction (minimum OAD position should = 0% or the minimum outdoor airflow should = 0 cfm) after 6:00 p.m. and until 6:00 a.m. (Monday–Friday) to minimize the outdoor air being introduced into the AHU/building during the night setback and morning warm-up/cool-down periods. If OADs have economizer control, it may still be productive to allow the economizer to open the OADs further when appropriate (e.g., morning cool-down when the OAT is cooler than the AHU return-air temperature (RAT).
3. Make sure that exhaust fans serving bathrooms, general exhaust systems, and similar loads are not activated until the very latest occupancy time (6:00 a.m. or later) to reduce infiltration and the requirement for ventilation/makeup air.

Extreme Weather Operations

In some situations, where overrides are the default operational method of O&M staff (during extreme weather—either very cold or very hot), it may be worthwhile to consider an automatic override in the BAS control code (versus a manual override that can be forgotten and left in place for several weeks or months). The control code is very simple and would look like this: When the OAT is less than 10°F (Low OAT Override Setpoint) or when the OAT is greater than 90°F (High OAT Override Setpoint), override the occupancy mode.

The key is to make sure that the setpoints are not set too high (Low OAT Override Setpoint) or too low (High OAT Override Setpoint). If a site can move away from overrides during extreme weather, this will pay dividends by sustaining the energy savings by automatically returning the HVAC system to its normal operating schedule (once the extreme weather is gone).

Recommended values for the Low OAT Override Setpoint and High OAT Override Setpoint is 10°F and 90°F, respectively, but should be adjusted for each building's design, performance (age, envelope, mechanical systems and reliability) and climate zone (humid climate zones may have other concerns but they should be addressed in the night setback configurations).

Miscellaneous Actions

While often overlooked in many scheduling efforts, most systems should be configured to include holiday scheduling. Most Federal sites recognize at least 10 Federal holidays (New Year's Day, Martin Luther King Day, President's Day, Memorial Day, Independence Day, Labor Day, Columbus Day, Veteran's Day, Thanksgiving Day and Christmas Day). Scheduling these 10 days to be unoccupied often achieves about 1% annual energy savings (if not currently implemented). Because holidays change (which day they fall on) each year, it is important to confirm that holidays are correctly configured each year or the wrong day may be inadvertently selected as a holiday.

2.2.1.5 BAS Data Needed to Verify Scheduling, Night Setback, and Optimal Start

Analyzing and detecting scheduling problems and opportunities can be achieved by using trend capabilities derived from the BAS. The following parameters should be monitored using the trend capabilities of the BAS:

- discharge (duct) static pressure (if provided) or fan status
- equipment status or command for different loads

- whole-building electricity
- zone temperatures.

Data should be collected at recommended intervals of between 5 and 30 minutes for a minimum of 1–2 weeks. When analyzing the discharge (duct) static pressure or whole-building electricity, the data trends to look for include the following:

- Does the AHU discharge static pressure show a drop in static pressure (to zero inches w.c. or close to that value) during scheduled vacancy periods (see Figure 2.4)?
  - The static pressure may show periodic increases (nights and weekends) when the AHU is responding to night setback or morning warm-up/cool-down actions.

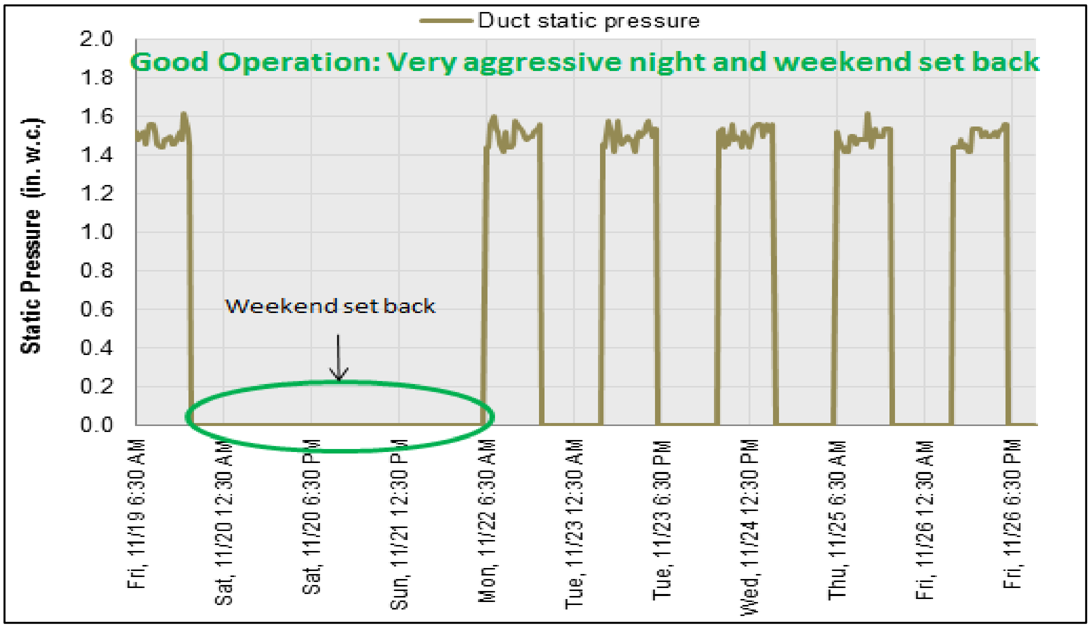

Figure 2.4. Duct Static Pressure – Aggressive Night and Weekend Scheduled Setback

- Does the whole-building electricity show aggressive or passive building scheduling activities (see Figure 2.5)? Aggressive scheduling results in greater savings, but may also result in greater challenges (longer recovery times or possible overrides).

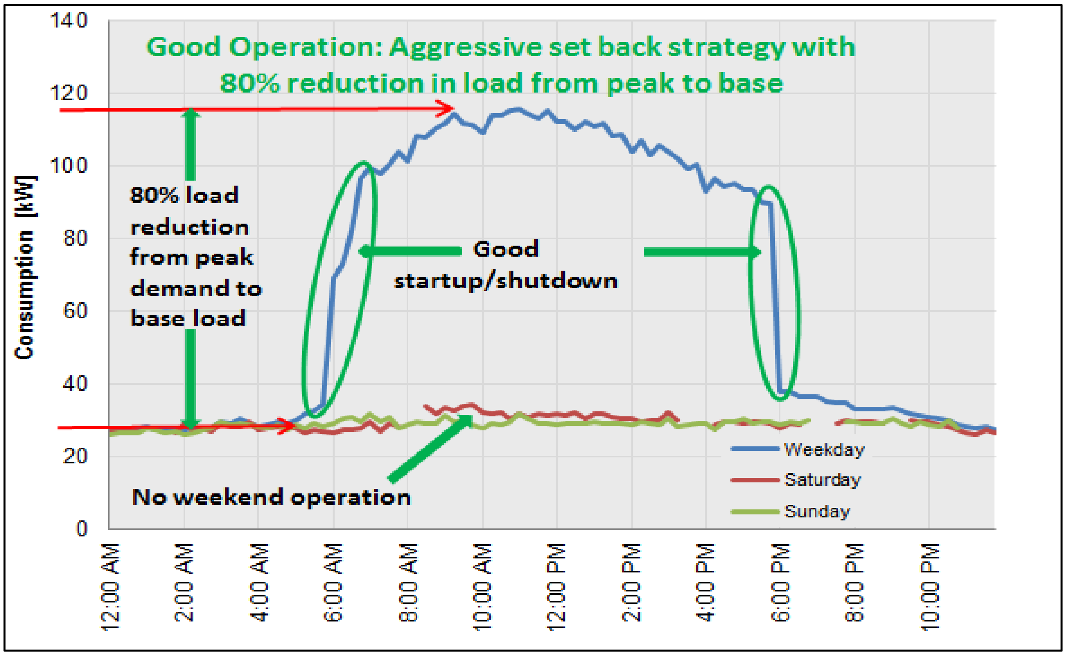

Figure 2.5. Whole-Building Consumption for an Office Building with Aggressive Setback Strategies

- Does the zone temperature trend data indicate that zone temperatures are dropping at night and during weekends (during cool-cold weather vacancy periods) to the configured heating setback values (55–62°F) or rising at night and during weekends (during warm-hot weather vacancy periods) to the configured cooling setback values (80–85°F)? When setback actions are occurring, the AHU, RTU, FCU, and other ancillary HVAC systems should also indicate temporary operations to maintain the setback values.
- Does the AHU, RTU, or other HVAC system configured with optimal start indicate morning start times that occur prior to the scheduled time? For instance, if the schedule occupancy is 7:00 a.m., but the BAS trend data indicates that the HVAC system started at 6:30 am, the optimal start program should be reviewed to verify that it calculated a 30-minute pre-start time. When this occurs, are the majority (not all) of the zones approaching the desired occupancy temperature (as configured in the optimal start program)? Remember, the intent of the optimal start program is to get the majority of the zones to within 1–2°F of their desired occupancy temperature setpoint values (versus a more precise value).

3.0 Zone Terminal Box Airflow Setpoint

The purpose of zone terminal box airflow setpoints reconfiguration and implementation guide is to show, through examples of good and bad operations, how the zone terminal box airflow setpoints can be efficiently reconfigured and what the indicators are for both good and bad operations. This guide is also intended to assist in identifying zone terminal boxes that are good candidates for reconfiguration, and offer some common implementation strategies and guidelines.

3.1 Why Consider Implementing Zone Terminal Box Airflow Setpoint Re-Configuration Improvements?

Air-handling unit (AHU) fan motors are configured to deliver air to downstream zone terminal boxes and can account for a significant percentage of a building's total energy consumption. AHU supply fans that are equipped with variable-frequency drives (VFDs) can take advantage of fan affinity laws during low load conditions. These laws reveal that reductions in fan speed have an exponential impact in decreasing fan power consumption. The reduction in supply fan speed translates directly to energy and cost savings without sacrificing comfort or ventilation requirements in the spaces served by the AHU(s).

Other improvements include reduced occupant comfort complaints (cold complaints is the primary culprit) and increased energy savings that comes from reduced fan energy, but may also include reduced chiller and hot water plant energy.

3.2 Factors Contributing to Zone Terminal Box Airflow Setpoint Configuration Problems

ASHRAE guidelines for variable-air-volume (VAV) zone terminal box minimum and maximum airflow rates may be derived from the estimated number of occupants and estimated cooling loads. Minimum airflow rates are intended to meet ventilation requirements for fresh air and maximum airflow rates are intended to meet cooling requirements (for comfort) during design-day conditions. Typical estimates for maximum airflow values are generally based on empirical data related to solar gain (window/envelop design), historical weather and other factors related to anticipated heat gain from electrical loads in the space (lights, appliances, etc.) – all of which can be accurately modeled by the mechanical design team.

Design estimates of zone occupancy are often an educated estimate and are not related to the actual number of occupants (measured or determined in real-time). The estimated number of occupants may be derived from a worst-case scenario for maximum occupants in certain spaces (conference rooms, lunch rooms, etc.). This can lead to minimum airflow rates that are excessive, resulting in carbon dioxide (CO_2) values that are significantly less than the maximum allowable threshold in the breathing zone (indicating over-ventilation of the spaces). Minimum ventilation rates that are too high can result in higher AHU fan horsepower and higher reheat energy requirements. They may also result in higher cooling energy requirements.

Reductions in occupancy from original design estimates is another factor that can result in over-ventilated spaces (therefore the actual space may be under-populated compared to the design estimate). There are many reasons why under-populated spaces may occur. This may be due to employee illness, holiday schedules where the workforce is significantly reduced (but still present), employees on business travel, employees taking vacations, employees working in a different part of the building or employees working outside of the building. Some employers allow employee teleworking, which also would result in decreased occupancy in the building. In some cases, it is not uncommon for certain parts of the country to find that weather-related disturbances (snow, fire, extreme rain/wind conditions, etc.) have resulted in buildings that are almost completely vacant – and this may occur many times during the year. Currently, there are not many intelligent methods to track occupancy in buildings, or to automatically adjust airflow rates for reduced occupancy conditions.

While it is difficult to determine the actual number of occupants in a space (conference rooms could have 1-2 or as many as 30-40), it may be possible to make educated assumptions, based on the building's use, mission and historical operations. In other cases, the building automation system (BAS) may have adequate instrumentation in the form of CO_2 sensors that provide feedback on the implied occupancy, versus the air quality (air quality, determined by CO_2 levels, can indicate ventilation rates that are excessive, adequate or inadequate). CO_2 sensors may be located in specific spaces or they may be located in the return-air plenum. When the CO_2 sensor is located in the return-air plenum, the reading from the sensor represents an average of all the spaces.

Historically, a building's air quality has been influenced by many operational factors, including the following:

- Poor control of the outdoor-air dampers (OADs), resulting in less-than-adequate outdoor ventilation rates:
  - If damper actuators are pneumatically driven in a building, the accuracy of the damper position is questionable
  - If the damper actuator has slipped on the mechanical linkage, the same (questionable accuracy) is also true
- Poor AHU maintenance and design:
  - Plugged filters and/or plugged coils
  - Condensate pans that are plugged and not adequately draining water (mold issues)
  - Outdoor-air intakes that are in close proximity to sources of fumes or obnoxious smells (garbage bins, sewer vent piping, building exhaust, vehicle parking lots or garages, etc.)
- Failed zone terminal boxes:
  - Failed VAV box dampers or actuators
  - Failed VAV box airflow sensors
  - Improperly configured VAV box control parameters

Historically, all of these factors may have contributed to questionable indoor-air quality, which may have influenced the configuration of the terminal box's minimum airflow rates to values higher than normally required.

3.3 What Zone Terminal Boxes Should Be Considered for Minimum Airflow Setpoint Improvements?

All VAV zone terminal boxes are good candidates for implementing reduced zone terminal box airflow setpoints, under the following assumptions:

- The BAS and field controllers are reliable (communications between VAV zone terminal boxes and the supervisory controllers are greater than 95% reliable)
- The targeted VAV zone terminal box controllers are working reliably. This includes the controller, the sensors (temperature, airflow) and the actuators (airflow damper actuator and reheat valve actuator)
- Mechanical aspects of the VAV zone terminal box are also working reliably. This includes the airflow damper (damper actuator and linkage are not broken and damper seals are good – air leakage

is minimal when the damper is closed) and the reheat valve (reheat valve actuator and linkage are not broken and reheat valve seals are good – water leakage is minimal when the reheat valve is closed)
- The current zone terminal box minimum airflow setpoint values are 30% or greater than the maximum airflow setpoint values
- Occasional or significant complaints from occupants due to cold drafts during the summer when the hot water boilers are turned off (if hot water systems are shut down during summer months).

3.3.1 Methodology for Implementing Zone Terminal Box Minimum Airflow Setpoint Improvements

ASHRAE recommendations for minimum outdoor-air ventilation rates are based on fixed airflow values per square foot (0.06 cfm per square foot) and fixed airflow values per person (depending upon the space use – 5 cfm for office, auditoriums, worship centers and hotels and up to 10 cfm for classroom spaces). The calculations are based on the design/maximum occupancy; however, the designed/maximum occupancy rarely occurs for many building types, resulting in HVAC systems that are always configured for higher than necessary ventilation rates.

Evaluation of over 100 Federal buildings has generally found that the cooling minimum airflow setpoints are often configured to be 30-50% of the maximum value (by design). The minimum airflow setpoints were found to exceed the ASHRAE guidelines for minimum outdoor airflow requirements due to calculated values that are found to be much greater than actual design and operations parameters. In some cases, minimum airflow setpoints have been increased to account for perceived air quality or temperature problems. The VAV box maximum airflow setpoints should already account for zones that are too warm (versus increasing the minimum airflow setpoint). ASHRAE guidelines allow for applying a diversity factor to the occupancy estimates, but diversity factor accounting may be overlooked, resulting in higher ventilation rates during low population count periods.

3.3.2 BAS Data Needed to Verify Zone Terminal Box Minimum Airflow Improvement Opportunities

Analyzing and detecting VAV zone terminal box airflow setpoint reconfiguration opportunities can be achieved by utilizing the individual BAS zone terminal box controller parameters associated with the minimum and maximum airflow setpoints. In addition, the following parameters should be monitored using the trend data capabilities of the BAS:

- Discharge (duct) static pressure
- Individual VAV box damper position (for zones served by each AHU).

The recommended frequency of data collection is between 5 and 30 minute intervals for a minimum of 1-2 weeks when most zones are in a heating mode (the zone temperature is lower than or close to the zone terminal box heating setpoint). When analyzing the data, the trends to look for include:

- If most VAV box dampers are less than 10-20% open, the static pressure may be too high, or the minimum airflow setpoint has been reduced, resulting in VAV box damper positions that are less than 10-20% open.
- If several VAV box dampers are more than 70-80% open, the static pressure may be too low, or the minimum airflow setpoint values may be too high, resulting in VAV box damper positions that are greater than 50-60% open.

- Review the zone terminal box minimum airflow setpoint values compared to the maximum airflow setpoint values. The ratio of the minimum to the maximum airflow setpoints should be 10-20% (varies based on ventilation requirements). Note VAV boxes with ratio values that are greater than 30% for possible implementation (reduction of the minimum airflow setpoint value to be no more than 10-20% of the maximum airflow setpoint value).

3.3.3 Is the Zone Terminal Box Minimum Airflow Setpoint Too High?

As an example of a real-life application of this measure, Table 3.1 represents an office building where the zone terminal box airflow minimum setpoints for approximately 30 zone terminal boxes serving one floor were reduced. The as-found value was approximately 30% of the zone terminal box airflow maximum setpoint and was decreased to approximately 10% of the zone terminal box airflow maximum setpoint.

The result was that the as-found zone terminal box cooling minimum airflow setpoint values were decreased by almost 70%. This reduction can result in significant fan power reductions, as well as better occupant comfort when heating systems are turned off or turned down during the summer and shoulder seasons.

Table 3.1. Approximately 30 VAV Zone Terminal Box Minimum Airflow Setpoints Reconfigured

Box ID	Baseline Cooling Minimum Airflow Setpoint (cfm)	Post-Re-tuning Cooling Minimum Airflow Setpoint (cfm)	Maximum Cooling Airflow Setpoint (cfm)
ATU-B2-25	180	52	552
ATU-B2-24	85	27	276
ATU-B2-23	300	90	900
ATU-B2-34	70	20	200
ATU-B2-21	350	105	1052
ATU-B2-20	90	26	260
ATU-B2-19	70	26	220
ATU-B2-18	100	29	292
ATU-B2-22	360	120	1200
ATU-B2-36	70	20	200
ATU-B2-35	500	150	1500
ATU-B2-26	500	150	1500
ATU-B2-27	400	120	1200
ATU-B2-28	70	20	200
ATU-B2-29	100	36	360
ATU-B2-30	100	36	360
ATU-B2-31	140	40	396
ATU-B2-32	100	28	276
ATU-B2-1	400	120	1200
ATU-B2-2	200	50	500
ATU-B2-14	500	150	1500
ATU-B2-15	250	70	700
ATU-B2-3	130	40	376
ATU-B2-4	200	50	500
ATU-B2-5	250	70	700
ATU-B2-6	70	20	200
ATU-B2-16	500	136	1500
ATU-B2-33	180	56	552
Total	**6,265**	**1,857**	**18,672**

3.3.4 Identifying and Correcting Problems That Interfere With Functional Zone Terminal Boxes

This section of the guide covers some common zone terminal box performance issues, provides guidance on how to identify these issues (through use of the BAS interface and trend data), and provides a list of possible problems that would cause the issue.

1. VAV box dampers that are not modulating with changing indoor and outdoor conditions, and VAV boxes that are not being controlled or not responding to control signals

    - VAV box damper commands that are always reading 100% open with very low airflow readings (close to 0 cfm) may indicate failed component(s)
        - VAV box damper has failed closed (actuator motor failure, loose wire, power failure or mechanical failure of damper-motor linkage)
        - Airflow sensor failure (loose sensing tube, plugged sensing line, sensing lines improperly connected or kinked)
        - Failed controller/electronics
        - This may alternatively be an issue of bad design (large loads served at the end of a duct; too much pressure drop in duct configuration)
        - Operational decisions that add high loads to a space designed for lower internal loads (e.g. more occupants and/or equipment than originally designed for)
    - VAV box damper commands that are always reading 0% open with airflow readings that are significantly higher than the airflow setpoint may indicate failed component(s)
        - VAV box damper has failed open (actuator motor failure or mechanical failure of damper-motor linkage)
        - Airflow sensor failure (sensor wired wrong, sensor improperly configured in the controller software)

2. Failed or improperly located duct static pressure sensors

    - Static pressure sensors that are always reading close to 0.0 in. w.c. or significantly lower than the discharge static pressure setpoint, may indicate failed component(s)
        - Duct static pressure sensor is wired incorrectly, has a loose wire or the sensor has failed
        - Duct static pressure sensor has a loose sensing line or plugged sensing line
        - Duct static pressure sensor is located downstream of fire dampers that have closed or ductwork that has a breach
        - The VFD has been overridden from the BAS or locally at the VFD (or has failed) to cause the VFD to operate at significantly reduced speed or not at all
    - Static pressure sensors that are always reading significantly higher than the discharge static pressure setpoint
        - Duct static pressure sensor is located immediately downstream of the primary fan (instead of 2/3 the distance of the longest duct from the primary fan)
        - The VFD has been overridden from the BAS or locally at the VFD to cause the VFD to operate at significantly increased speed (i.e., a lot of times this corresponds to 100% speed, or 60 Hertz)

4.0 AHU Static Pressure Control

The purpose of this AHU static pressure control implementation guide is to show, through examples of good and bad operations, how the AHU supply fan static pressure can be efficiently controlled and what the indicators are for both good and bad operations. This guide assists in identifying AHUs that are good candidates for implementing static pressure reset, and offers some common implementation strategies and guidelines.

4.1 Why Consider Implementing Static Pressure Reset?

Air-handling unit (AHU) fan motors (supply and return/relief) can account for a significant percentage of a building's total energy consumption. AHU supply fans that are equipped with variable-frequency drives (VFDs) can take advantage of fan affinity laws during low-load conditions. These laws reveal that reductions in fan speed have an exponential impact on decreasing fan power consumption. The reduction in supply fan speed translates directly to energy and cost savings without sacrificing comfort or ventilation requirements in the spaces served by the AHU(s).

A static pressure reset control sequence automatically changes the static pressure setpoint in response to continuous feedback regarding the ability of the downstream terminal boxes to meet individual airflow setpoints. A reduced static pressure setpoint will result in variable-air-volume (VAV) box dampers that open farther to maintain their individual airflow setpoints. A reduction in the static pressure setpoint will correspond to a reduction in supply fan speed. When the speed of the supply fan is reduced, the power required to operate the supply fan is reduced. If the static pressure setpoint is too low, individual VAV boxes that are not able to maintain their airflow setpoints will increase their VAV damper positions to increase airflow rates. This action (increasing VAV damper positions) will automatically direct the static pressure reset algorithm to increase the static pressure setpoint to a higher value. This will cause the variable speed fan motor to increase in speed, which will create enough static pressure to result in adequate airflow without starving zones. This is important, because when the static pressure setpoints are held constant (too high), excess fan energy is consumed with no commensurate benefit to the building occupants. In some cases, increased reheat energy at the zones (particularly during the shoulder and heating seasons) can occur if individual VAV boxes are not operating correctly (failed sensor, failed actuator, etc.). An automatic reduction in the static pressure setpoint follows a determination that most of the required zones are receiving adequate airflow. A drop in the static pressure setpoint means that the supply fan does not have to work as hard to deliver the required airflow. Optimal conditions for automatically changing the static pressure setpoint are detected through the building automation system (BAS) in near real time. Failure to identify and implement static pressure reset in all likelihood will lead to higher than required fan energy consumption.

4.2 What AHUs Should Be Considered for Static Pressure Reset Implementation?

Single-duct AHUs can be divided into two basic categories: constant-air volume (CAV) and variable-air-volume (VAV). CAV AHUs are generally configured with a constant speed motor and no means of varying the motor speed to vary the volume of air delivered downstream. These systems are designed to provide tempered air to one large zone and in some cases, multiple zones. In either case, the discharge-airflow that is served to the zone or multiple zones never varies in flow rate, only in temperature.

VAV AHUs are generally configured with a motor that has a VFD that can be configured to automatically vary the frequency of the drive, and in turn the operating speed, at which the fan motor rotates. The speed of the fan rotation can be changed directly through the VFD. This is typically accomplished through changes in the static pressure setpoint. Therefore, VAV AHUs that are equipped with VFDs and static pressure sensors are good candidates for implementing a static pressure reset, under the following assumptions:

- The BAS and field controllers are reliable (communication between VAV terminal boxes and the supervisory controllers is more than 95% reliable).
- VAV terminal box controllers are based on digital control (pneumatic signals and pneumatic actuators are not used at the VAV terminal boxes). This is important to the extent that the pneumatic controllers typically do not send reliable damper position feedback to the BAS. Damper position is often a critical control variable used in static pressure reset logic.
- VFDs are integrated to the BAS, allowing for command execution to the VFD from the BAS along with the ability to read data from the VFD (e.g., start, stop, and frequency or speed command and feedback).

4.3 Methodology for Implementing Static Pressure Reset

This section of the implementation guide presents some common strategies for implementing static pressure reset. Note that this list is not intended to cover every implementation technique, but to present a few techniques in detail to provide some context for what a good static pressure reset implementation might look like. All final reset parameters should be thoroughly discussed with the owner/operator of the BAS and the building(s) where the reset strategy will be applied. The implementation strategy may vary by building type, system type, or other variables that this guide does not directly highlight.

4.3.1 Approach 1: Average VAV Box Damper Position

Using the VAV box damper position (see Section 4.3.5, BAS Data Needed to Verify Static Pressure Reset below) for each VAV box served by the AHU, a calculated average of the VAV box damper positions served by a particular AHU can be determined and used as feedback for the static pressure reset algorithm. The average VAV box damper position should include a calculation of all VAV boxes served by that AHU, with exception of boxes serving non-occupied spaces (e.g., hallways, bathrooms, storage or other non-office spaces). Consideration of conference rooms for removal from the calculation may also have merit, but removal of any VAV box from the average calculation should be evaluated on case-by-case basis.

Typical algorithm reset strategies would use the design maximum static pressure setpoint value as the high static pressure limit (HSPL). The HSPL can be found in the control drawings for the AHU. If control drawings are not available, the HSPL may be the existing static pressure setpoint in the BAS, or maximum static pressure setpoint in the BAS if there is an existing reset strategy in place. The low static pressure limit (LSPL) is recommended to be 50% of the HSPL. For example, if the HSPL is 1.50 in. w.c., then the LSPL value would be 0.75 in. w.c.

If an average of the VAV box damper positions is used, the static pressure setpoint would linearly change between the HSPL and LSPL as the average VAV box damper position varies from 60% open to 40% open. For the HSPL and LSPL example above, the static pressure setpoint conditions can be summarized as follows:

- Maximum AHU static pressure setpoint (HSPL): 1.5 in. w.c. (anytime the average VAV box damper position is greater than or equal to 60% open)
- Minimum AHU static pressure setpoint (LSPL): 0.75 in. w.c. (anytime the average VAV box damper position is less than or equal to 40% open)
- The static pressure setpoint will vary linearly between the HSPL and LSPL as the average VAV box damper position varies between 60% open and 40% open.

Figure 4.1 shows this methodology using the example values above. The percentages used in the example above are user-adjustable, but it is recommended that the upper and lower limits (shown on the y-axis) have at least a 25–50% difference between them (to take full advantage of implementing a static pressure reset strategy).

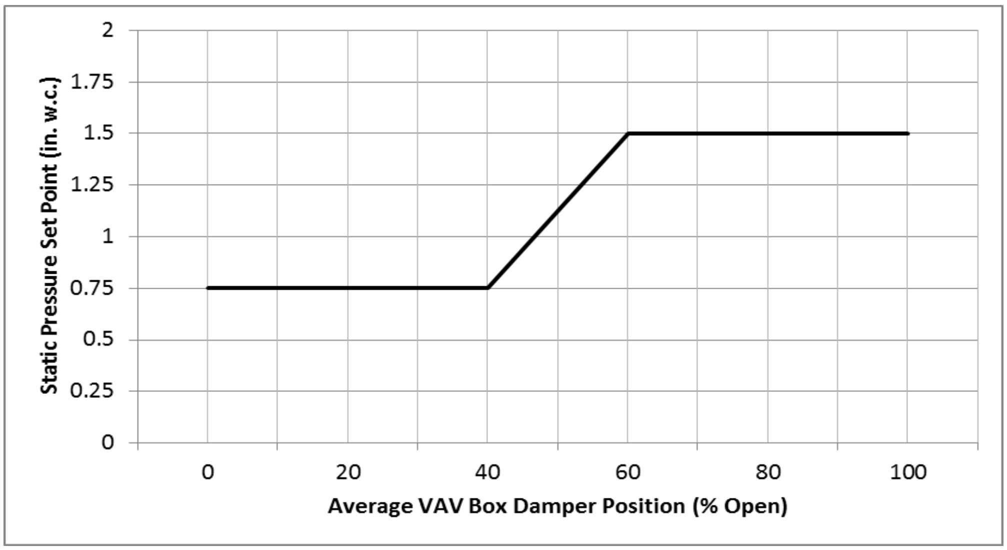

Figure 4.1. Example Static Pressure Reset Graph Based on Average VAV Box Damper Position

4.3.1.1 When Is It Appropriate to Use an Average VAV Box Damper Approach to Static Pressure Reset?

An averaging approach can yield strong energy savings, but it can be blind to the needs of starved or troubled zones. Here are some approaches to evaluating the likelihood of success for the averaging approach:

1. Evaluate the diversity of VAV box damper commands during a typical occupied day (12–24 hour period of trend data). A VAV system where the majority of the damper commands are somewhere near the average command (minimum to maximum – shown as 40%–60% in Figure 4.1) is favorable. If there are several damper commands below 20% and several damper commands above 80% (simultaneously), this approach can be problematic. In those cases, the averaging approach may be insufficient to satisfy the needs of starved boxes (when there are more than three starved boxes and when they sustain in a starved mode for more than 60 minutes). A starved box is defined as a VAV damper position (command) that is 100% open for more than 5 minutes. In cases where the VAV box dampers have such widely disparaging values, this may reflect poor design, poor air balance, poor sensor and actuator reliability or other anomalies that may or may not be easily corrected.
2. Evaluate the tenant and spaces served by the VAV network under the AHU to ensure that the averaging approach does not include “rogue” zones (VAV boxes that have failed or are misbehaving)

and does not include zones that serve hallways, bathrooms, lobby spaces, or other non-office spaces. This helps to eliminate noncritical spaces that could influence the automatic reset outcome.

Smart strategies for averaging can still be employed despite these concerns. For example, in a VAV network where it is known that critical VAV boxes are starved, the average could be set up to only include this set of starved boxes, until the root causes for the critically starved boxes are resolved (re-balance the ductwork, upsize the VAV box and/or the ductwork, fix failed VAV box sensors, etc.).

4.3.2 Approach 2: Trim and Respond based on the Maximum VAV Box Damper Position

Using the VAV box damper position (see Section 4.3.5, BAS Data Needed to Verify Static Pressure Reset below) for each VAV box served by the AHU, the calculated maximum VAV box damper position for all the VAV boxes served by a particular AHU can be determined and used as feedback for the static pressure reset algorithm. If the maximum VAV box damper position is inclusive of all VAV boxes, make sure to remove VAV boxes that serve non-occupied spaces (e.g., hallways, bathrooms, storage, or other non-office spaces).

Typical algorithm reset strategies would use the design maximum static pressure setpoint value as the HSPL. The LSPL is recommended to be 50% of the HSPL. For example, if the HSPL is 1.50 in. w.c., then the LSPL value would be 0.75 in. w.c.

If the maximum of the VAV box damper positions is used, the static pressure setpoint would increase between the LSPL and HSPL as the maximum VAV box damper position rises above 90–95% open. Once the maximum VAV box damper position drops below 70–75% open, the static pressure setpoint would decrease between HSPL and LSPL. The navigation between the HSPL and the LSPL is typically done in time intervals (e.g., once every 5 minutes), and increments or decrements the static pressure setpoint by a defined value (e.g., 0.1 in. w.c.). For the HSPL and LSPL example above, the static pressure setpoint conditions can be summarized as follows:

- Maximum AHU static pressure setpoint (HSPL): 1.5 in. w.c. (the calculated setpoint would approach this value over time, once the maximum VAV box damper position is greater than or equal to 90–95% open)
- Minimum AHU static pressure setpoint (LSPL): 0.75 in. w.c. (the calculated setpoint would approach this value over time, once the maximum VAV box damper position is less than or equal to 70–75% open)
- Some strategies will adjust the static pressure setpoint at a faster rate when the setpoint is increased and adjust the static pressure setpoint at a slower rate when the setpoint is decreased. This is done to respond to starved boxes in an appropriate manner (quickly), while slowly reducing the static pressure (to ensure that the static pressure does not cycle from low to high inadvertently or needlessly).

4.3.3 Approach 3: VAV Box Average Cooling Demand

Using the VAV box cooling demand signal (see Section 4.3.5, BAS Data Needed to Verify Static Pressure Reset below) for each VAV box served by the AHU, an average of the individual VAV box cooling demand signal served by a particular AHU can be determined and used as feedback for the static pressure reset algorithm. This guide assumes that the cooling demand signal for any given VAV box can have a value between 0 (no cooling required) and 100 (maximum cooling required). If the VAV box average cooling demand signal is inclusive of all VAV boxes, make sure to remove VAV boxes that serve non-occupied spaces (e.g., hallways, bathrooms, storage, or other non-office spaces).

Typical algorithm reset strategies would use the design maximum static pressure setpoint value as the HSPL. The LSPL is recommended to be 50% of the HSPL. For example, if the HSPL is 1.50 in. w.c., then the LSPL value would be 0.75 in. w.c.

For this strategy based on the average VAV box cooling demand signal, the static pressure setpoint would linearly change between the HSPL and LSPL as the average VAV box cooling demand signal varies from 60 to 40. For the HSPL and LSPL example above, the static pressure setpoint conditions can be summarized as follows:

- Maximum AHU static pressure setpoint (HSPL): 1.5 in. w.c. (anytime the VAV box average cooling demand signal is greater than or equal to 60)
- Minimum AHU static pressure setpoint (LSPL): 0.75 in. w.c. (anytime the VAV box average cooling demand signal is less than or equal to 40)
- The static pressure setpoint will linearly change between the HSPL and LSPL as the VAV box average cooling demand signal varies between 60 and 40.

Figure 4.2 shows this methodology using the example values above. The percentages used in the example above are user-adjustable, but it is recommended that the upper and lower limits have at least a 20% difference between them (to take full advantage of implementing a static pressure reset strategy). Also note that the VAV box may not have minimum and maximum cooling demand signal values of 0 and 100 (some VAV boxes may be configured for other values such as -100 to 100 and should be clearly understood).

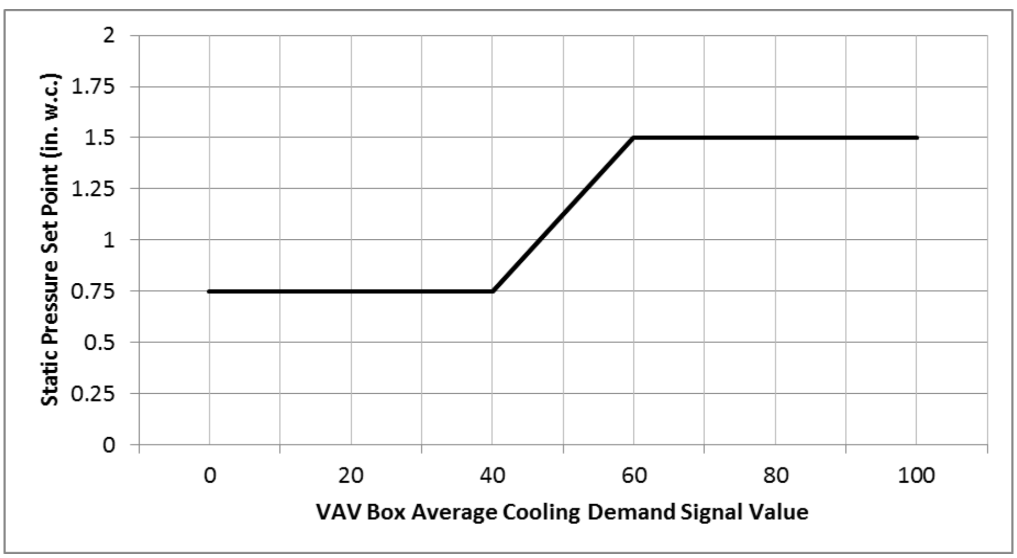

Figure 4.2. Example Static Pressure Reset Graph Based on a VAV Box Average Cooling Demand Signal Value

4.3.3.1 When Is It Appropriate to Use the Average Cooling Demand?

Note that this approach is similar to the average VAV box damper position approach already discussed. Because it is an averaging approach, it is prone to the same limitations surrounding starved and critical zones, so the same evaluation criteria can be applied.

The average VAV box cooling demand signal may provide convenient feedback to use in the event that it is already calculated for other purposes, or can be calculated at minimal additional effort. This may also be a viable alternative to damper position-based approaches when VAV damper position values are not reported or are known to be unreliable. This may be true when the VAV box uses a "floating" control damper actuator design (two binary signals) instead of a true analog signal and only provides a "cooling demand signal" output.

4.3.4 Approach 4: Number of VAV Boxes with Cooling Requests

Using the VAV box cooling request (see Section 4.3.5, BAS Data Needed to Verify Static Pressure Reset below) for each VAV box served by the AHU, a sum of the number of VAV boxes that have cooling requests served by a particular AHU can be determined and used as feedback for the static pressure reset algorithm. It is assumed that a VAV box cooling request takes a value of 1 in this implementation guide. If the sum of VAV boxes with cooling requests is inclusive of all VAV boxes, make sure to remove VAV boxes that serve non-occupied spaces (e.g., hallways, bathrooms, storage, or other non-office spaces).

Typical algorithm reset strategies would use the design maximum static pressure setpoint value as the HSPL. The LSPL is recommended to be 50% of the HSPL. For example, if the HSPL is 1.50 in. w.c., then the LSPL value would be 0.75 in. w.c.

For this strategy based on the percent of VAV boxes with cooling requests, the static pressure reset provides maximum airflow anytime 30% or more of the VAV boxes served by the AHU call for cooling, and minimum airflow anytime 10% or less of the VAV boxes served by the AHU call for cooling. As an example, for an AHU serving 100 VAV boxes, the static pressure setpoint would linearly change between the HSPL and LSPL as the number of VAV boxes with cooling requests varies from 30 to 10. This example set of conditions can be summarized as follows:

- Maximum AHU static pressure setpoint (HSPL): 1.5 in. w.c. (anytime the number of VAV box cooling requests is greater than or equal to 30)
- Minimum AHU static pressure setpoint (LSPL): 0.75 in. w.c. (anytime the number of VAV box cooling requests is less than or equal to 10)
- The static pressure setpoint will linearly change between the HSPL and LSPL as the number of VAV box cooling requests varies between 30 and 10.

Figure 4.3 shows this methodology using the example values above. The percentages (30% and 10%, respectively) used in the example above are user-adjustable, but it is recommended that the upper and lower limits have at least a 20% difference between them (to take full advantage of implementing a static pressure reset strategy).

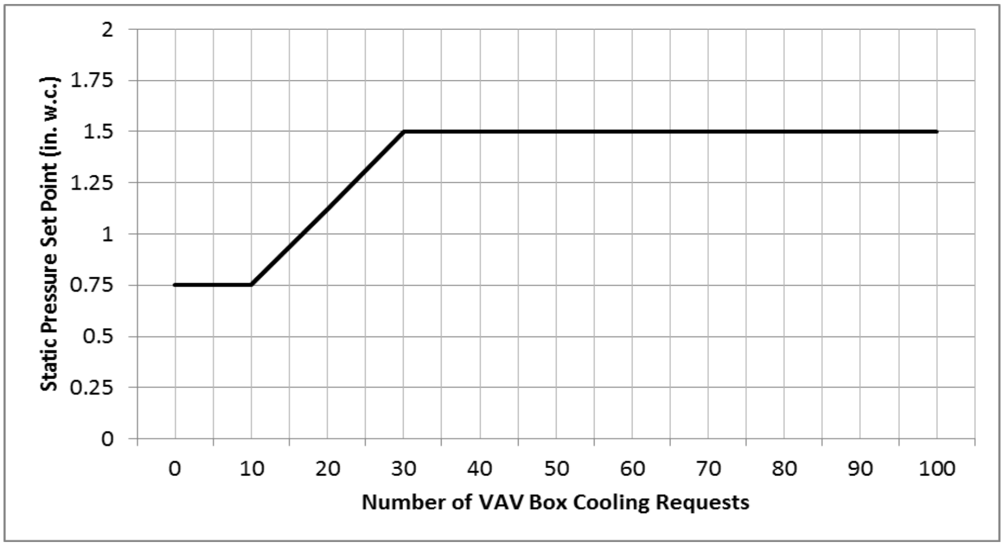

Figure 4.3. Example Static Pressure Reset Graph Based on the Number of VAV Boxes with Cooling Requests

4.3.4.1 When Is It Appropriate to Use the Number of VAV Box Cooling Requests?

This approach may be used if none of the other approaches are available due to lack of feedback from VAV box damper commands and VAV box cooling demand signals. This approach gives some insight into the potential need for greater airflow by using cooling requests as a proxy, but the other feedback variables (particularly VAV box damper command) give direct evidence of the ability of the VAV boxes to provide more airflow, which is the most desirable approach.

4.3.5 BAS Data Needed to Verify Static Pressure Reset

Analyzing and detecting AHU static pressure control problems and opportunities can be achieved by using trend capabilities through the BAS. The following parameters should be monitored using the trend capabilities of the BAS:

- discharge (duct) static pressure
- discharge (duct) static pressure setpoint
- individual VAV box damper position (for zones served by each AHU). This is needed only for Approaches 1 and 2 above.
  - Create a calculated average value of all VAV box damper positions served by each AHU for Approach 1 above. Ensure non-critical and/or rogue VAV boxes are removed from the calculation.
  - Create a calculated maximum value of all VAV box damper positions served by each AHU for Approach 2 above. Ensure non-critical and/or rogue VAV boxes are removed from the calculation.
- Individual VAV box cooling demand signal (for zones served by each AHU). Note that this data point may not be readily available in all VAV box vendor configurations. Create a calculated average of all VAV box cooling demand values served by each AHU. This is needed only for Approach 3 above.

- VAV box cooling request (for zones served by each AHU). Note that this data point may not be readily available in all VAV box vendor configurations. Create a calculated sum of all VAV box cooling requests served by each AHU. This is needed only for Approach 4 above.

Data should be collected at recommended intervals of between 5 and 30 minutes for a minimum of 1–2 weeks. When analyzing the static pressure reset, the data trends to look for include the following:

- Is there an existing reset for the duct static pressure?
- Is the static pressure setpoint too high or too low?
  - Review trends of damper position of VAV boxes vs. time:
    - If most VAV box dampers are nearly closed, the static pressure may be too high.
    - If several VAV box dampers are fully open (greater than 80%), the static pressure may be too low.

4.3.6 Alternatives to Automatic Resets

In Figure 4.4 below, the static pressure is reset twice per day during the 2 day sample period. The system turns off at roughly 7:00 p.m. on both Tuesday and Wednesday night, and turns on at 6:00 a.m. When the system turns on, the static pressure is scheduled to a setpoint value of 1 in. w.c., and as the building load increases, the static pressure is scheduled to a setpoint value of 1.5 in. w.c. Then, as the building load decreases in the afternoon, the system is scheduled back to a setpoint value of 1 in. w.c., until finally the system turns off.

The reset in Figure 4.4 is considered a scheduled reset, where the setpoint is reset as a function of the time of day. Application of this methodology is a good candidate for buildings that may operate 24/7 due to mission or programmatic requirements. However, the setpoint requirements during the day are often significantly greater than at night and on weekends. Rather than creating calculated values and additional control logic, this is a substitute for an automatic static pressure reset and it can be effective if configured correctly (based on time) and aggressively (based on the setpoint values).

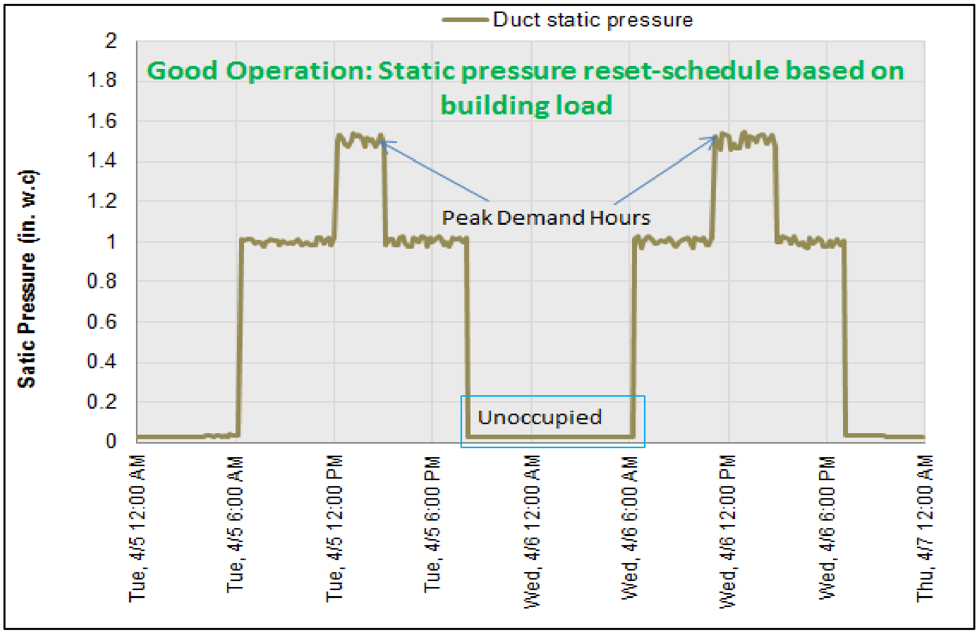

Figure 4.4. Scheduled Static Pressure Reset Based on Time of Day

4.3.7 Is There an Existing Reset for the Duct Static Pressure?

Duct static pressure control specifications are taken directly from design specifications. To maintain the static pressure setpoint, the supply fan VFD will vary the fan motor speed appropriately, but the setpoint is usually determined based on the equipment design and an air balance effort that typically targets design conditions. After construction, the static pressure setpoint is often adjusted as needed to satisfy the most demanding zone, which often is a problem area that sets a higher than required setpoint for the remainder of the system—and even for the critical zone itself under low-load conditions.

Figure 4.5 shows a system where it is unlikely that an automatic static pressure setpoint reset has been implemented. Here, the static pressure is set at a constant 2 in. w.c. during the day and is off only for a few hours at night.

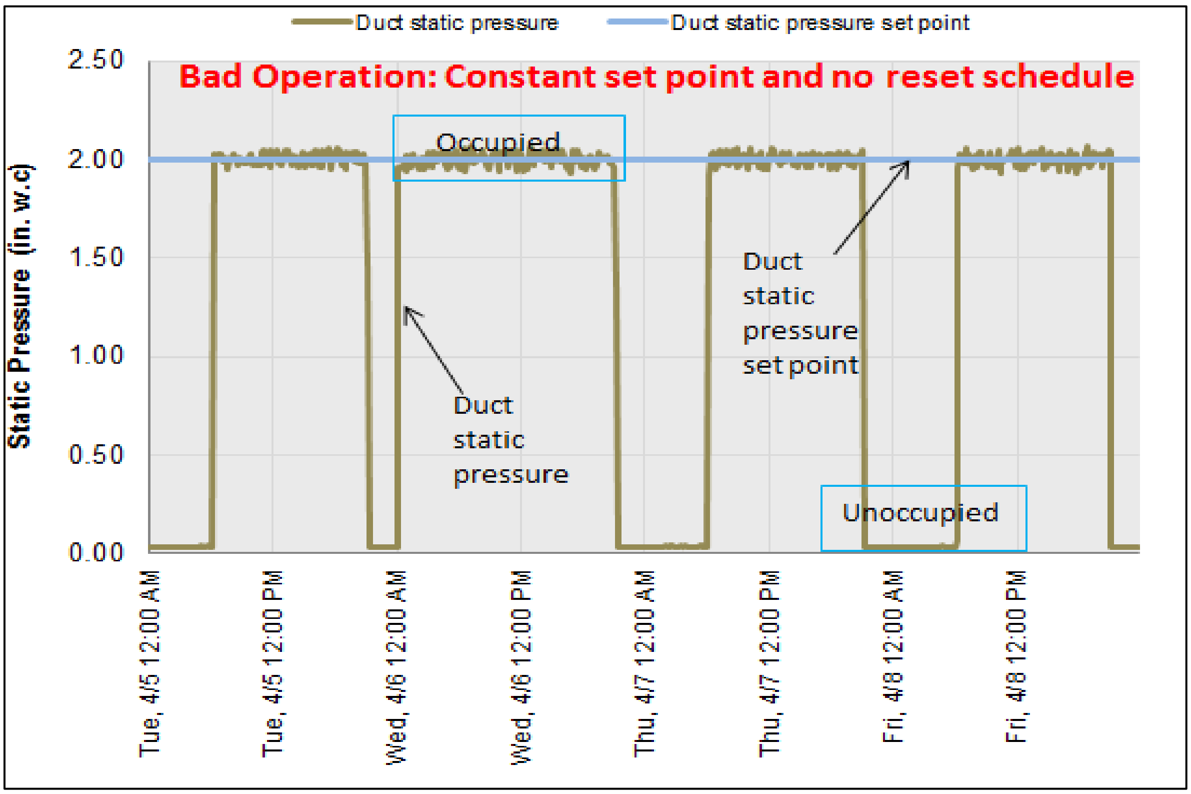

Figure 4.5. Constant Static Pressure Setpoint (No Reset)

Figure 4.6 provides an example of optimal operations, where the static pressure is *likely being continuously reset* based on zone feedback throughout the day (it could also be the case that this was a system that had a constant static pressure setpoint, but that the fan was unable to meet that setpoint). The morning start-up period (6:00 a.m.to 8:00 a.m.) often indicates a need for higher static pressure—especially during the cooling season when the building setback period results in a morning cool-down mode. In response to the setback event, the VAV boxes fully open, resulting in a need for higher static pressure from the AHU. Once the zones start to meet the occupied temperature setpoint, the VAV boxes will also start to “relax” by throttling their dampers, resulting in lower static pressure requirements from the AHU. This only occurs if an automatic static pressure reset is implemented.

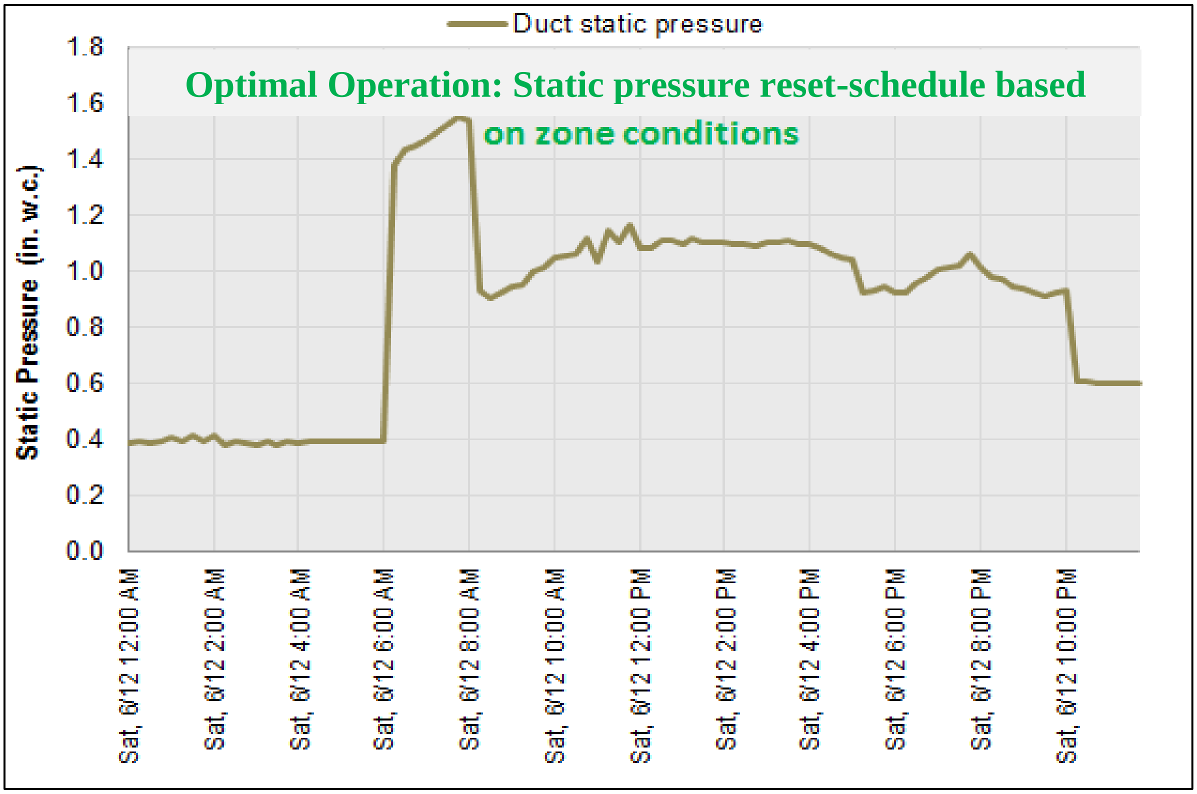

Figure 4.6. Static Pressure Reset Based on Zone Feedback

4.3.8 Is the Static Pressure Setpoint Too High or Too Low?

To determine whether the static pressure setpoint is too high or too low, review the VAV box damper positions versus time. Ideally, VAV box dampers should modulate between 50% and 75% open (when the system is operating at non-design conditions). If most of the VAV box dampers are closed during cooling, the static pressure setpoint is most likely too high. If most of the VAV dampers are greater than 75% open during cooling, the static pressure setpoint is most likely too low. Figure 4.5 shows a static pressure setpoint of 2 inches, which is constant. VAV box damper positions versus time for a typical AHU are shown in Figure 4.7.

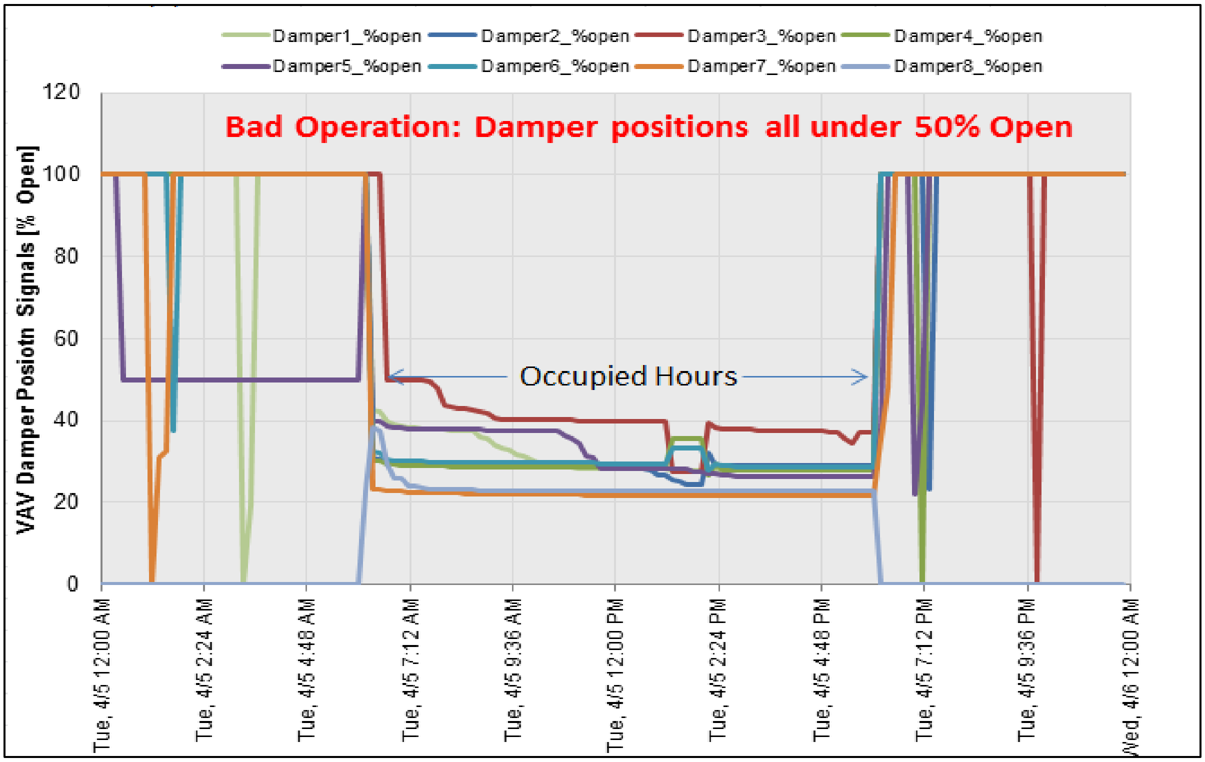

Figure 4.7. VAV Box Damper Positions vs. Time (Static Pressure Setpoint Is Too High)

Figure 4.7 shows the VAV boxes going to the occupied mode around 6:00 a.m. There are eight zones that this AHU serves, and all of the VAV box dampers are less than 50% open during the trend sample period. This is an indicator that the static pressure setpoint is too high (2 in. w.c. from Figure 4.5). Figure 4.8 shows another example of bad operation, this time for an AHU that has too low of a duct static pressure (during normal operation, not on a design day). In Figure 4.8, the majority of the VAV box damper positions are more than 80% open, which indicates that the boxes are starved for airflow. If the majority of the boxes have damper positions more than 80% open, they may be starved for airflow, and the duct static pressure setpoint should be increased because the setpoint is too low. Finally, Figure 4.9 offers an example of what the zone conditions should look like for an AHU operating with the proper duct static pressure setpoint.

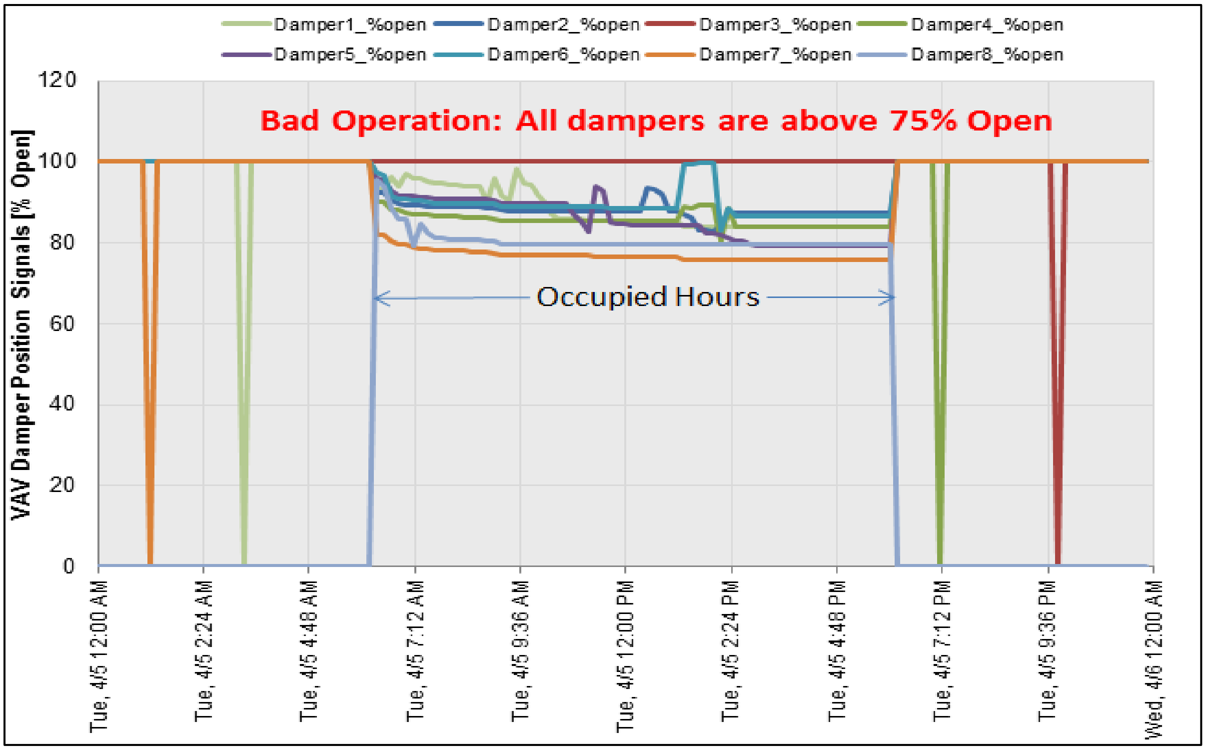

Figure 4.8. VAV Box Damper Positions vs. Time (Static Pressure Setpoint Is Too Low)

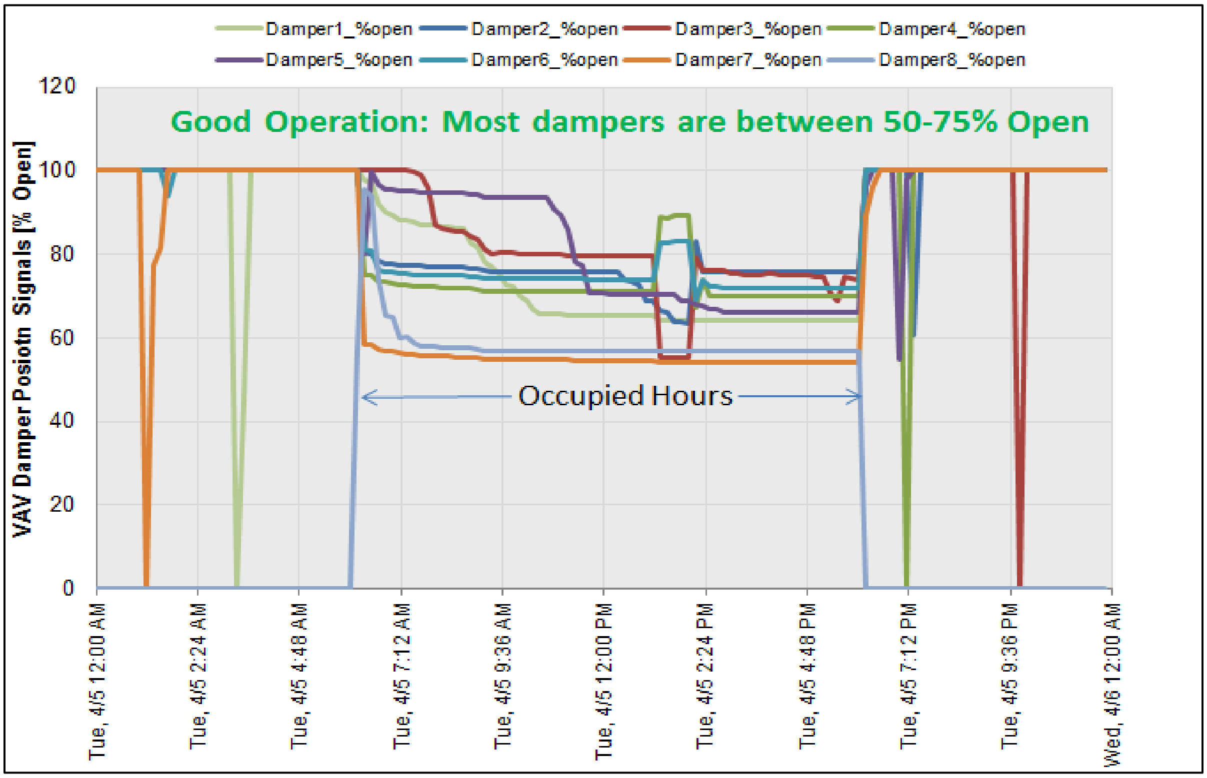

Figure 4.9. VAV Box Damper Positions vs. Time (Proper Duct Static Pressure Setpoint)

4.3.9 Identifying and Correcting Problems that May Interfere with Functional Static Pressure Reset

This section of the guide covers some common VAV box and AHU static pressure issues to look for (through use of the BAS interface and trend data), and a list of possible problems that would cause the issue.

1. Identify VAV box dampers that are not modulating with changing indoor and outdoor conditions, and VAV boxes that are not being controlled or responding to control signals.
   - VAV box damper commands that are always reading 100% open with very low airflow readings (close to 0 cfm) may indicate failed component(s), such as:
     - VAV box damper has failed closed (actuator motor failure, loose wire, power failure, or mechanical failure of damper-motor linkage).
     - Airflow sensor has failed (loose sensing tube, plugged sensing line, sensing lines improperly connected or kinked).
     - Controller/electronics have failed.
     - Poor design (large loads served at the end of a duct; too much pressure drop in duct configuration).
     - Operational decisions may add high loads to a space designed for lower internal loads (e.g. more occupants and/or equipment than originally designed for the space).
   - VAV box damper commands that are always reading 0% open with airflow readings that are significantly higher than the airflow setpoint may indicate failed component(s), such as:
     - VAV box damper has failed open (actuator motor failure or mechanical failure of damper-motor linkage).
     - Airflow sensor has failed (sensor wired wrong, sensor improperly configured in the controller software).
2. Look for failed discharge or improperly located duct static pressure sensors.
   - Static pressure sensors that are always reading close to 0.0 in. w.c. or significantly lower than the discharge static pressure setpoint may indicate failed component(s), as follows:
     - Duct static pressure sensor is wired incorrectly, has a loose wire, or the sensor has failed.
     - Duct static pressure sensor has a loose sensing line or plugged sensing line.
     - Duct static pressure sensor is located downstream of fire dampers that have closed or ductwork that has a breach.
     - The VFD has been overridden from the BAS or locally at the VFD to cause the VFD to operate at significantly reduced speed.
   - Static pressure sensors that are always reading significantly higher than the discharge static pressure setpoint.
     - Duct static pressure sensor is located immediately downstream of the primary fan (instead of two-thirds of the distance of the longest duct from the primary fan).
     - The VFD has been overridden from the BAS or locally at the VFD to cause the VFD to operate at significantly increased speed (i.e., a lot of times this corresponds to 100% speed).

5.0 Condenser Water Temperature Reset

The purpose of this condenser water temperature reset implementation guide is to show, through examples of good and bad operations, how a condenser water temperature reset can be efficiently controlled and what the indicators are for both good and bad operations. This guide also offers some common implementation strategies.

5.1 Why Consider Implementing a Condenser Water Temperature Reset?

Condenser water temperature resets are very simple to implement, track, and administer. The goal of condenser water temperature resets is to optimize the condenser water temperature that the cooling system is using for heat rejection in a water-cooled chiller application. The principles are the same: generate condenser water that is cool enough to help offset the compressor lift (load) without expending additional energy at the cooling towers than what provides commensurate value to the chiller (and avoid generating condenser water that is below the chiller's minimum required temperature). When a condenser water temperature setpoint is too low (not physically achievable due to ambient wet bulb conditions), no further reductions in the condenser water temperature are achieved while the cooling tower's energy consumption increases, negatively impacting the chilled water plant efficiency.

Most water-cooled chillers are designed with cooling towers that provide the means to remove waste heat from the chiller's compression cycle, which allows the refrigerant to condense from a vapor to a liquid. A chiller plant may have one or more cooling towers. Cooling towers may be designed with variable frequency drive (VFD)-connected fans or constant-speed fans. Multiple cooling towers may be operated in sequence (for staging) or they may operate in parallel, to take advantage of the fan affinity laws for cumulative-reduced speed operations when the tower fan motors have VFDs.

All chiller compressors perform work based on the lift that the compressor sees. Lift is defined as the difference between the condenser refrigerant pressure and evaporator refrigerant pressure. Using defined pressure-temperature relationships, lift can also be measured with the leaving chilled (evaporator) water temperature and the leaving condenser water temperature. The greater the difference, the greater the lift, which also translates into greater work required by the compressor. Compressor lift can be lessened by raising the chilled-water supply temperature or lowering the condenser water temperature (or both).

The goal for any building with properly implemented condenser water temperature reset strategies includes energy efficiency, which is often measured as kilowatts per ton. Water-cooled chillers rely upon condenser water pumping systems and cooling towers to aid in heat rejection, and this adds additional energy to the overall cooling process. Water-cooled chillers, with VFD-driven cooling tower fans, VFD-driven compressors, and VFD-driven pumps (condenser water) are often able to achieve lower total kilowatt per ton values, when these systems are properly controlled and sequenced.

While sequencing multiple chillers, pumps, and tower fans is the desired outcome for most chilled-water plants, this guide focuses on the optimum value to which the condenser water temperature setpoint should be reset. An optimized condenser water temperature setpoint can help decrease the chiller's energy consumption while also optimizing the cooling tower's fan energy consumption.

Water-cooled chillers rely upon cooling towers to cool the condenser water through evaporative cooling. The evaporative cooling process relies mostly on the ability to evaporate water in the tower, as air is forced or induced over water droplets (in an open tower) that are suspended in the air or clinging to the tower fill's surface. The tower fill is a large honeycomb surface (usually made out of plastic or similar material). In a closed tower, water is continually sprayed over coils that contain the condenser water. These coils continually evaporate water, resulting in a cooling effect that removes heat from the coil surface and this cooling effect continues to the condenser water in the coil. Open towers typically have a higher efficiency because the condenser water comes in direct contact with the ambient air, while closed towers do not expose the working fluid to dirt, dust, and biological debris that are found in the ambient air. Closed towers require an additional water loop that is exposed to ambient air, which is separate from the working fluid loop that runs through the chillers. Both tower designs require persistent maintenance and water treatment to ensure suspended solids, biological agents and other unwanted effects are minimized.

The evaporative cooling capacity of most cooling towers is dependent upon the outdoor wet-bulb temperature (i.e., the adiabatic saturation temperature). The wet-bulb temperature is the lowest temperature that can be reached under current ambient conditions, by evaporation of water in a cooling tower. As water evaporates out of the cooling tower, the water level will drop. As this occurs, the tower design will inject makeup water to maintain the design volume of water in the tower. Water makeup is generally managed by some type of float-valve arrangement, but may involve other technologies.

Cooling towers also have over-flow piping that allows excess water to flow out of the tower (in case the level controls fail or have not been configured correctly). Most cooling towers include automatic blow-down capability (to purge suspended solids that have accumulated in the tower water due to evaporation). As water in the cooling tower evaporates or is blown down or overflows, treated water is added to the tower. This water is generally cooler than the ambient air and (depending upon the volume of water) can also help lower the condenser water temperature (in spite of ambient wet-bulb temperature limitations). For cooling towers that are operating inefficiently, this generally is indicated by excess energy consumption and potentially excess water consumption. Excess energy consumption can be occurring at the cooling towers or at the chiller compressor(s).

If the condenser water temperature setpoint is configured at a setpoint value that is too high, this can result in reduced energy consumption at the cooling tower fan(s) and excess energy consumption at the chiller(s) because the compressor sees higher lift conditions than would be seen if the condenser water temperature setpoint is automatically configured to a lower (optimum) setpoint value (based on real-time ambient wet-bulb temperature conditions). It should be noted that the cooling tower is not capable of achieving the outdoor air wet-bulb temperature. Most cooling towers have an approach of between 5°F and 10°F above the outdoor air wet-bulb temperature. Optimal condenser water temperature setpoints should be automatically calculated, based on the current outdoor air wet-bulb temperature (OAWBT). The approach value should be added to the OAWBT and this would become the cooling tower condenser water temperature setpoint. If the approach is not known, it is recommended that 7°F be used as a starting point.

The other advantage of properly configured condenser water temperature setpoints is the ability to avoid generating condenser water temperatures that are excessively low. All chillers have minimum condenser water temperatures at which the chiller can reliably operate. In some cases, the minimum condenser water temperature for reliable chiller operations is 65-70°F. Some of the newer VFD-driven chillers are able to operate reliably with condenser water temperatures as low as 55°F. Each chiller's minimum condenser water temperature value should be verified with the chiller vendor, before configuring the minimum condenser water temperature value in the automatic condenser water temperature reset algorithm.

In some cases, cooling towers are configured to operate as a water-side economizer where the tower water is fed to a plate-and-frame heat exchanger. The tower water is allowed to operate at temperatures as low as 40–45°F. While this provides for a generous "chilled" water loop that is very cool without mechanical (compressor) cooling, the low temperatures create problems for the mechanical chiller when the loop switches from the water-side economizer operation back to the mechanical chiller operation. Supplying water to the condenser portion of the chiller while still at lower building chilled-water temperatures may result in low refrigerant pressure safety cutouts (the chiller is forced off and may go into an Alarm status). Before the water-side economizer switches over to mechanical cooling, the tower loop should be allowed to warm up to the minimum condenser water supply temperature of the chiller (typically 50°F to 55°F) in order to mitigate low refrigerant pressure safety cutout events caused by low condenser water temperatures. In addition, if the tower water temperatures are allowed to operate below 42°F, this may be low enough to automatically activate electric heaters (if provided) that are located in the tower basin and designed to mitigate ice from forming in the tower basin water. Therefore, caution should be exercised before configuring water-side economizer setpoints that are lower than 45°F.

Most cooling towers (when properly designed) will include a bypass valve. This allows the chiller's condenser water to bypass or mix a portion of the condenser water leaving the chiller with the water returning from the tower, to maintain minimum water temperatures. The bypass valve should be checked periodically for correct operation and the control sequences should be verified. Historically, maximum cooling tower water temperature requirements have been associated with hot outdoor-air dry-bulb temperature operations, but this is really a characteristic of the cooling tower design and the outdoor-air wet-bulb temperatures. Wet-bulb temperatures dictate the tower's ability to evaporate water to the ambient air. Lower wet-bulb temperatures enable more evaporation but if the bypass valve is leaking or intentionally open during warm weather, this decreases the cooling tower's effectiveness and causes the chillers to expend more energy at the compressor.

All chillers and cooling towers that are equipped with reliable controls are good candidates for implementing condenser water temperature resets, under the following assumptions:

- The BAS, chiller, pumps, and cooling tower controllers are reliable (communication between chillers, pumps, towers, and the supervisory controllers is more than 95% reliable) and any integration issues are totally resolved.

5.2 What Systems Should Be Considered for Condenser Water Temperature Reset Implementation?

This methodology for condenser water temperature reset only applies to water-cooled chillers or water-cooled chilled water plants.

5.2.1 Methodology for Condenser Water Temperature Reset

This section of the implementation guide goes through the most common strategy for implementing a condenser water temperature reset based on the outdoor air wet-bulb temperature. Note that this is not intended to cover every implementation technique, but to present a few techniques in detail to provide some context for what a good condenser water temperature reset looks like. All final condenser water temperature reset parameters should be thoroughly discussed with the owner/operator of the BAS, the building(s), and the chiller vendor(s). The implementation strategy may vary by building type, system type, or other variables that this guide does not directly highlight.

5.2.1.1 Condenser Water Temperature Reset – Outdoor-Air Wet-Bulb Temperature Reset

Figure 5.1 shows two examples of a condenser water temperature reset, using the OAWBT.

Warning: Confirmation of the minimum condenser water temperature is the responsibility of the engineering staff responsible for chiller operations.

The **red line** in Figure 10.1 shows the condenser water temperature reset that is between 65°F and 85°F, as the OAWBT rises from 60°F to 80°F (5°F approach value). The **black line** in Figure 5.1 shows the condenser water temperature reset that is between 65°F and 85°F, as the OAWBT rises from 55°F to 75°F (10°F approach value). The black line indicates a tower that requires lower OAWBT conditions to achieve the same condenser water temperatures as the red line.

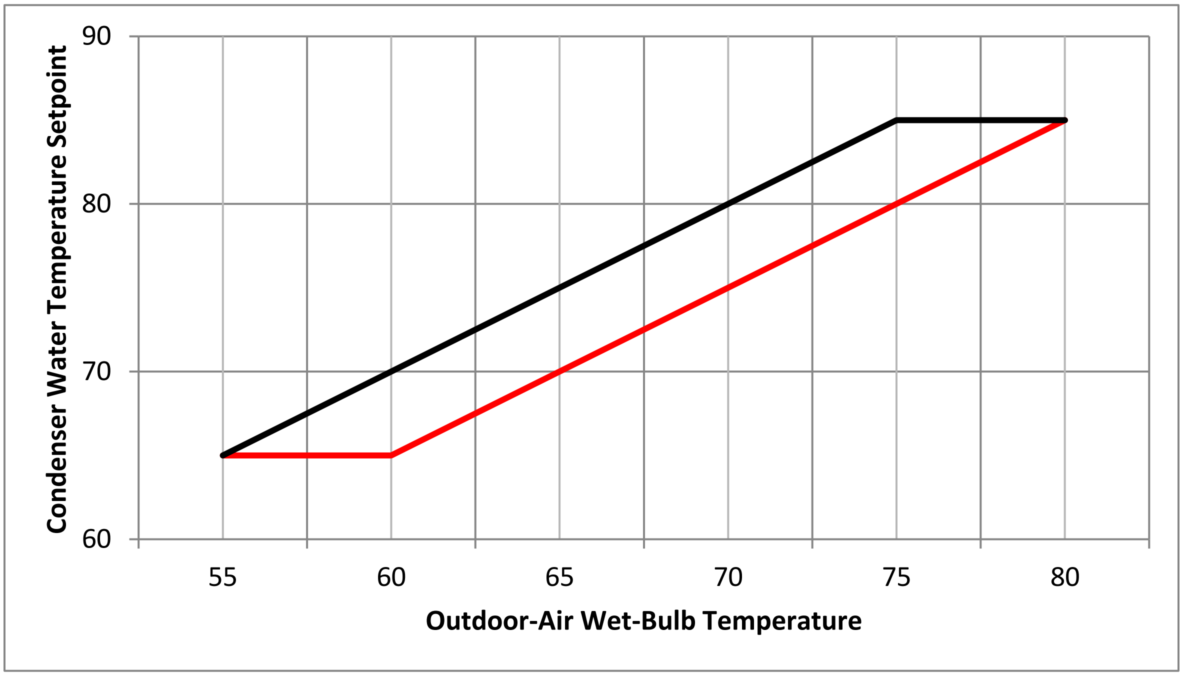

Figure 5.1. Condenser Water Temperature Reset Examples based on the Outdoor-Air Wet-Bulb Temperature

The approach value should be determined from the cooling tower vendor or other reliable engineering resources (if unknown, a good starting point is 7°F). Temperature (dry bulb/dew point) and humidity sensors should be calibrated periodically to ensure accuracy of the condenser water temperature reset strategy.

5.2.1.2 BAS Data Needed to Verify the Condenser Water Temperature Reset

Analyzing and detecting condenser water temperature reset problems and opportunities can be achieved by using trend capabilities derived from the BAS. The following parameters should be monitored using the trend capabilities of the BAS:

- condenser water supply temperature
- condenser water return temperature
- outdoor air wet-bulb temperature

Data should be collected at recommended intervals of between 5 and 30 minutes for a minimum of 1–2 weeks. When analyzing the condenser water temperature reset, the trend data to look for include the following:

- Is the condenser water supply temperature more than 2–4°F above the calculated setpoint for most of the day?
  - This may indicate that the calculated setpoint is too low.
  - This may indicate that the approach value is too small and needs to be widened.
  - This may indicate that the cooling tower cells are scaling up and in need of cleaning.
  - This may indicate that the cooling tower bypass valve (if part of the tower piping design) is open or has a leak, allowing warm condenser water to bypass the tower(s).
- Are the cooling tower fans (VFD-driven) operating at less than 100% fan speed while meeting the condenser water temperature setpoint, throughout most of the day?
  - This generally indicates a properly configured condenser water temperature setpoint, as long as the setpoint values are within the vendor-approved tolerances for the chiller.
- If the condenser water temperature setpoint is configured as a fixed value (by design or via override) at a setpoint value that is too low for the ambient conditions, this can result in the tower fan VFD(s) operating at 100% speed with no additional cooling benefit.
- Is the cooling tower bypass valve open while the tower fans are also operating? If true, are the tower fans operating at or near 100% fan speed?
  - This typically indicates a problem in the control code that sequences the bypass valve and the tower fans.
- Is the cooling tower fan off for long periods of time during lower OATs, while chiller part load efficiency does not improve (decrease in plant energy consumption)?
  - If so, this could indicate an opportunity to lower condenser water temperature setpoints during part load conditions (if the chiller vendor indicates lower temperatures can be supported by the chiller operation).
- Are the tower cells properly maintained (de-scaled and correct water-treatment occurring via chemical management system)?
  - If not, check the chemical storage tanks to ensure they are not empty – the automatic chemical injection system may have depleted all the chemical in the storage tanks.

6.0 AHU Discharge-Air Temperature Reset

The purpose of this AHU discharge-air temperature (DAT) implementation guide is to show, through examples of good and bad operations, how the DAT can be efficiently controlled and what the indicators are for both good and bad operations. This guide also assists in identifying AHUs that are good candidates for implementing DAT reset, and offers some common implementation strategies and guidelines.

6.1 Why Consider Implementing Discharge-Air Temperature Reset?

AHU heating and cooling energy can account for a significant percentage of a building's total energy consumption. If the AHU serves a large area, is sized for more than 10,000 cfm, operates 12 or more hours each day, or is located in severe climate zones (very cold or very hot/humid), the likelihood of energy improvement opportunities related to the DAT function of the AHU is high (but even mild climate zones should not be overlooked). In cold weather climate zones, AHUs are often equipped with dedicated heating coils for tempering the outdoor air (in addition to the standard AHU heating and cooling coils) and in warm/tropical climate zones, AHUs may be equipped with dedicated cooling coils for tempering the outdoor-air. AHUs may also be configured in humid climates for dehumidification (moisture removal), which often drives higher heating and/or reheating energy costs. AHUs may also be configured in dry climates with humidification capability. All of these subsystem components (outdoor air tempering, heating, cooling, dehumidification, and humidification) can result in complicated control sequences that may not be optimized. Optimizing the control sequence of AHU DAT will translate into energy and cost savings without sacrificing comfort in the spaces served by the AHU(s).

When the DAT setpoints are held constant (either too high or too low), this will often result in increased energy consumption in the building (at the AHU or at the zone variable-air volume [VAV] boxes and hot water heating plant). When the DAT is too cold, the result will often be increased reheat energy at the zones (and possibly increased energy at the chilled water plant as the chilled water is over-cooled to allow for AHU over-cooling). When the DAT is too warm, the result may be increased AHU fan energy (for VAV-designed AHU systems). Optimal conditions for automatically resetting the DAT setpoint can be detected through the BAS in near real time. Failure to identify and implement DAT reset in all likelihood will lead to increased heating and cooling energy consumption. It may also contribute to potential occupant discomfort.

The DAT reset automatically changes the DAT setpoint in response to continuous feedback from the building load conditions. The changes in the DAT setpoint should correspond to changes in the AHU heating and cooling coil control valves, as well as the outdoor economizer. It is important that these three subsystem components be synchronized and working correctly to maintain the desired DAT setpoint. When improper sequencing of the AHU heating and cooling coils and the outdoor air economizer occurs, it may go unnoticed for a long period of time—adding to the building's continued energy inefficiency woes.

6.2 What AHUs Should Be Considered for Discharge-Air Temperature Reset Implementation?

Single-duct AHUs can be divided into two basic categories: constant-air-volume (CAV) and VAV. CAV AHUs are generally configured with a constant-speed motor. These systems are designed to provide tempered air to one large zone and, in some cases, multiple zones. In either case, the discharge airflow that is served to the zone or multiple zones varies only in temperature.

VAV AHUs are generally configured with a motor that has a VFD that can be configured to automatically vary the frequency, and in turn the operating speed, at which the fan motor rotates. This allows the speed of the fan rotation to be changed directly through the VFD.

Both types of AHU configuration (VAV and CAV) may be configured to serve dual-duct AHUs. A dual-duct AHU has a heating coil and a cooling coil that are ducted in parallel. The parallel ducts leave the AHU and serve a section of the building. A dual-duct system has zone-mixing boxes located in the different zones where heated and cooled air is mixed at the zone to achieve the desired zone temperature. Single-duct AHUs have a cooling coil and may have a heating coil. Different design requirements will drive the coil operations (heating and/or cooling) in the AHU as well as the coil location in the AHU (dedicated outdoor-air intake, downstream of the mixing plenum, or after the fan).

Note: Where dehumidification (with the required cooling coil design and controls) is required, the reset strategies may be superseded by further manipulation to cause moisture removal (via over-cooling).

All AHUs that are equipped with reliable controls are good candidates for implementing DAT reset, under the following assumptions:

- The BAS and field controllers are reliable (communication between VAV or zone VAV boxes and the supervisory controllers is more than 95% reliable)
- Zone VAV box controllers are digital controls-based (pneumatic signals and pneumatic actuators are not used at the zone boxes). This is important to the extent that the pneumatic controllers typically do not send reliable reheat valve position feedback to the BAS. Reheat valve position is often a critical control variable used in DAT reset logic, when the valve position is used to determine the zone load and the optimal discharge temperature setpoint in response to that load.

6.2.1 Methodology for Implementing Discharge Temperature Reset

This section of the implementation guide goes through some common strategies for implementing DAT reset. Note that this list is not intended to cover every implementation technique, but to present a few techniques in detail to provide some context for what a good DAT reset implementation might look like. All final reset parameters should be thoroughly discussed with the owner/operator of the BAS and the building(s) in which the reset strategy will be applied. The implementation strategy may vary by building type, system type, or other variables that this guide does not directly highlight.

6.2.1.1 Approach 1: Average VAV Box Reheat Valve Position Reset

Using the VAV box reheat valve position (see Section 6.2.2., BAS Data Needed to Verify Discharge-Air Temperature Reset below) for each VAV box served by the AHU, an average of the reheat valve positions served by a particular AHU can be determined and used as feedback for the DAT reset algorithm. The average VAV box reheat valve position should include a calculation of all VAV boxes. If the average VAV box reheat valve position is inclusive of all VAV boxes, make sure to remove VAV boxes that serve non-occupied spaces (e.g., hallways, bathrooms, storage, or other non-office spaces). Consideration of conference rooms may also have merit, but removal of any VAV box from the average calculation should be evaluated on a case-by-case basis. Failed reheat valves that have been manually isolated by the O&M team should also be removed, as this will falsely bias the reset.

Typical algorithm reset strategies would use the design minimum DAT setpoint value as the low discharge-air temperature limit (LDTL). The LDTL can be found in the control drawings for the AHU. If control drawings are not available or do not specify this value, the LDTL should be the minimum DAT setpoint in the BAS if an existing reset strategy is in place. If an existing reset strategy is not in place, the LDTL is recommended to be 55°F and the high discharge-air limit (HDTL) is recommended to be 10°F higher than the LDTL. For example, if the LDTL is 55°F, then the HDTL value would be 65°F. In some cases, an LDTL value lower than 55°F may be required, but should never be lower than 52°F (possibly in hot, humid climates - during hot, humid weather).

If an average of the VAV box reheat valve positions is used, the DAT setpoint would linearly change between the HDTL and LDTL as the average zone VAV box reheat valve position varies from 60% open to 20% open. For the HDTL and LDTL example above, the DAT setpoint conditions can be summarized as follows:

- Maximum AHU DAT setpoint (HDTL): 65°F (anytime the average zone VAV box reheat valve position is more than or equal to 60% open)
- Minimum AHU DAT setpoint (LDTL): 55°F (anytime the average zone VAV box reheat valve position is less than or equal to 20% open)
- The DAT setpoint will automatically change between the HDTL and LDTL as the average zone VAV box reheat valve position varies between 60% open and 20% open.

Figure 6.1 shows this methodology using the example values above. The percentages used in the example above are user-adjustable. It is recommended that the upper and lower limits (shown on the y-axis) have at least a 15–20% difference between them (to take full advantage of implementing a DAT reset strategy).

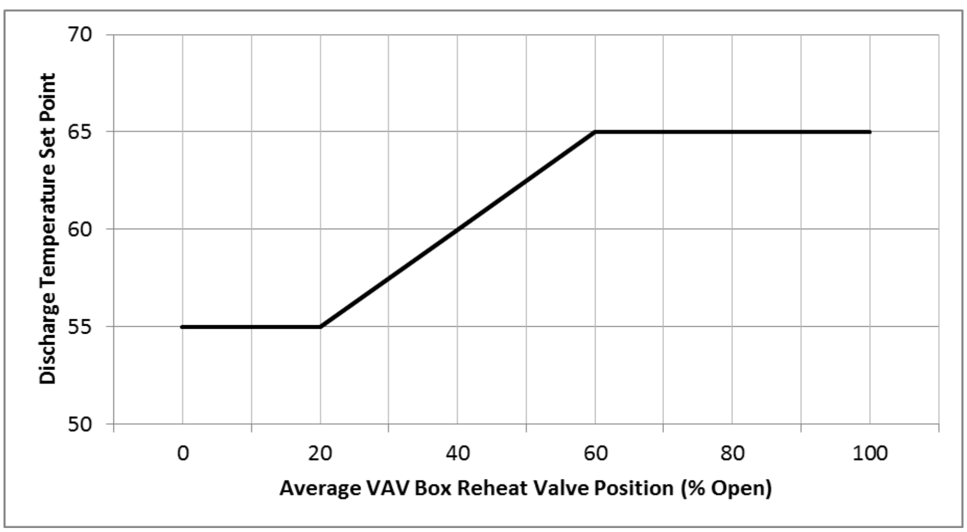

Figure 6.1. Example DAT Reset Graph Based on Average VAV Box Reheat Valve Position

When Is It Appropriate to Use an Average VAV Reheat Valve Approach to Discharge-Air Temperature Reset?

An averaging approach can yield strong energy savings, but it can be blind to the needs of over-cooled or over-heated zones. Here are some approaches to evaluating the likelihood of success for the averaging approach:

1. Evaluate the diversity of VAV box reheat valve commands during a typical occupied day (12–24 hour period of trend data). A VAV system where the majority of the reheat valve commands are somewhere near the average command (minimum to maximum – shown as 20%–60% in Figure 6.1) is favorable. If there are several reheat valve commands below 20% and several reheat valve commands above 80% (simultaneously), this approach can be problematic. In those cases, the averaging approach may be insufficient to satisfy the needs of under-performing VAV boxes (when there are more than three and when they persist in over-cooled or over-heated temperature mode for more than 60–120 minutes). In cases where the VAV box reheat valve commands have such widely disparaging values, this may reflect poor design, inconsistent zone temperature setpoints (for zones that are co-located), large window/glass (solar gains), poor water balance issues, leaking hot water reheat valves, AHU dehumidification (coupled with hot water system shutdown) or other anomalies that may or may not be easily corrected.

2. Evaluate the tenant and spaces served by the VAV network under the AHU to ensure that the averaging approach does not include "rogue" zones (VAV boxes that have failed or have a history of problems) and does not include zones that serve hallways, bathrooms, lobby spaces, or other non-office spaces. This helps to eliminate noncritical spaces that could influence the outcome.

Smart strategies for averaging can still be employed despite these concerns. For example, in a VAV network where it is known that critical VAV boxes are either over-cooled or over-heated, the average could be set up to only include these sets of temperature-challenged boxes, until the root causes are resolved.

6.2.1.2 Approach 2: Average Zone Temperature Setpoint Error Reset

Calculate the average zone temperature setpoint error for each zone VAV box served by the AHU (see Section 6.2.2, BAS Data Needed to Verify Discharge-Air Temperature Reset below) by calculating the ΔT of each VAV box (zone cooling temperature setpoint minus the respective zone temperature). Calculate the average value of all VAV box ΔTs. This value can be used as feedback for the DAT reset algorithm. If the calculated average of zone VAV box setpoints and their respective zone temperatures is inclusive of all zone VAV boxes, make sure to remove zone VAV boxes that serve non-occupied spaces (e.g., hallways, bathrooms, storage or other non-office spaces).

If an average zone temperature setpoint error is used, the DAT setpoint would linearly change between the HDTL and LDTL as the average zone temperature setpoint error varies from (+) 2.0°F to (-) 2.0°F. For the HDTL and LDTL example above, the DAT setpoint conditions can be summarized as follows:

- Maximum AHU DAT setpoint (HDTL): 65°F (anytime the average zone temperature setpoint error is greater than or equal to 2.0°F)
- Minimum AHU DAT setpoint (LDTL): 55°F (anytime the average zone temperature setpoint error is less than or equal to -2.0°F)
- The DAT setpoint will automatically change between the HDTL and LDTL as the average zone temperature setpoint error varies between 2.0°F and -2.0°F.

Figure 6.2 shows this methodology using the example values above. The values (2.0°F and -2.0°F, respectively) used in the example above are user-adjustable. It is recommended that the upper and lower limits have at least a 20% difference between them (to take full advantage of implementing a DAT reset strategy).

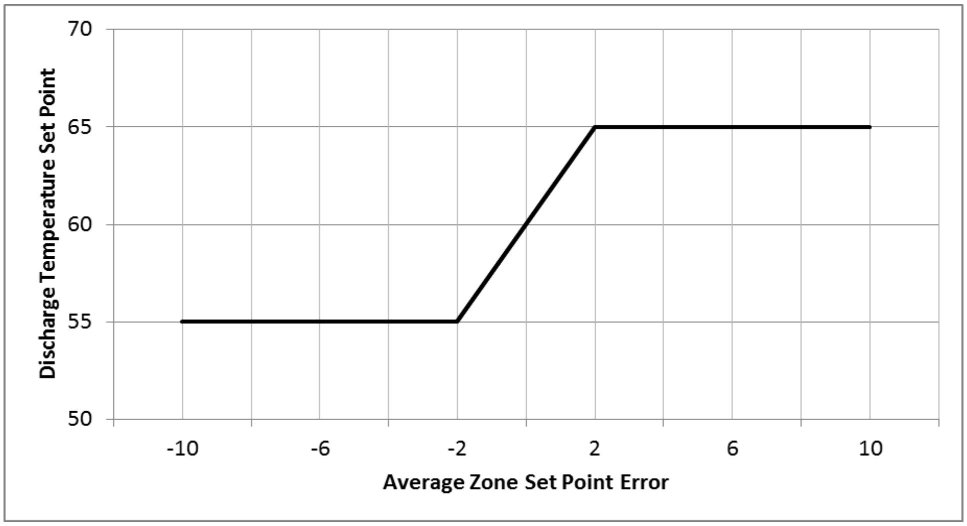

Figure 6.2. Example DAT Reset Graph Based on Average Zone Setpoint Error

Other methodologies for average zone setpoint error can include calculating the zone cooling error (zone temperature minus the zone cooling setpoint) and the zone heating error (zone heating setpoint minus the zone temperature). Each calculated value should be summed (all cooling errors and all heating errors). The highest value will determine if the system should reset up (heating) or down (cooling). This approach generally is code-intensive and not recommended unless the BAS is robust.

6.2.1.3 Approach 3: Outdoor-Air Temperature Reset

Using the OAT (see Section 6.2.2, BAS Data Needed to Verify Discharge-Air Temperature Reset below) the OAT sensor value (either associated with a particular AHU or a globally shared OAT value) can be determined and used as feedback for the DAT reset algorithm.

The DAT setpoint would linearly change between the HDTL and LDTL as OAT varies from 50°F to 75°F. For the HDTL and LDTL example above, the DAT setpoint conditions can be summarized as follows:

- Maximum AHU DAT setpoint (HDTL): 65°F (anytime the OAT is less than or equal to 50°F)
- Minimum AHU DAT setpoint (LDTL): 55°F (anytime the OAT is greater than or equal to 70°F)
- The DAT setpoint will automatically change between the HDTL and LDTL as the OAT varies between 50°F and 70°F.

The DAT setpoint will automatically change between the HDTL and LDTL as the OAT varies between 50°F and 70°F. Figure 6.3 shows this methodology using the example values above. The temperature values used in the example above are user-adjustable, but it is recommended that the upper and lower limits have at least a 20% difference between them (to take full advantage of implementing a DAT reset strategy).

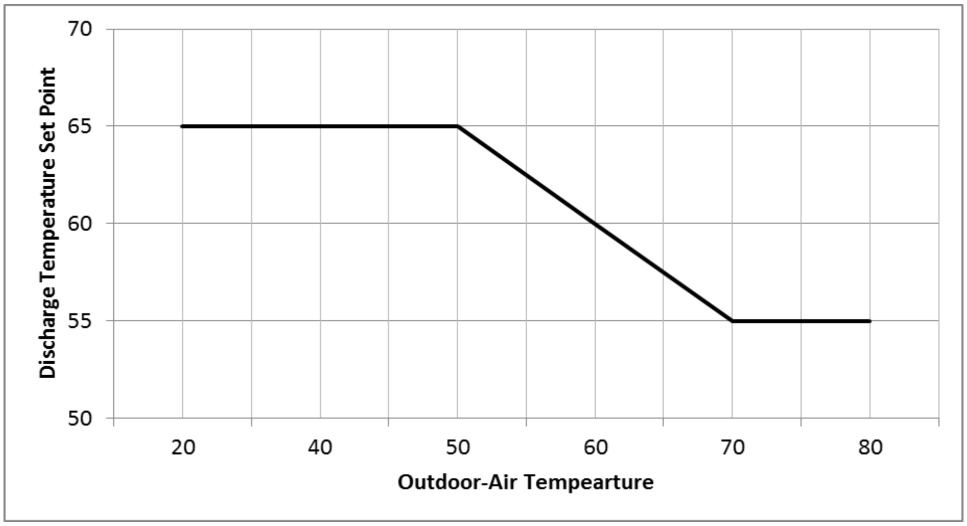

Figure 6.3. Example DAT Reset Graph Based on the OAT

When Is It Appropriate to Use the Outdoor-Air Temperature?

This approach should only be used if none of the other approaches are available due to lack of feedback from the average VAV box reheat valve commands or average VAV zone temperature versus zone setpoint error calculated values. The ability to calculate average values is more code-intensive in the local or supervisory controller and, if there are communication issues, may also create reliability issues. Using the OAT for the DAT reset is the simplest approach outlined, but the other feedback variables (particularly average VAV reheat valve command or average zone setpoint temperature versus zone setpoint error) will give direct evidence of the ability of the VAV boxes to meet the zone temperature requirements with the lowest energy input from both the AHU and the individual zone VAV boxes, which is the most desirable approach for a good DAT reset.

6.2.1.4 Approach 4: Return-Air Temperature Reset

Using the RAT (see Section 6.2.2, BAS Data Needed to Verify Discharge-Air Temperature Reset below) sensor value can be used as feedback for the DAT reset algorithm.

The DAT setpoint would linearly change between the HDTL and LDTL as RAT varies from 70°F to 74°F. For the HDTL and LDTL example above, the DAT setpoint conditions can be summarized as follows:

- Maximum AHU DAT setpoint (HDTL): 65°F (anytime the RAT is less than or equal to 70°F)
- Minimum AHU DAT setpoint (LDTL): 55°F (anytime the RAT is greater than or equal to 74°F)
- The DAT setpoint will automatically change between the HDTL and LDTL as the RAT varies between 70°F and 74°F.

The DAT setpoint will automatically change between the HDTL and LDTL as the RAT varies between 70°F and 74°F. Figure 6.4 shows this methodology using the example values above. The temperature values used in the example above are user-adjustable. It is recommended that the upper and lower limits have at least a 20% difference between them (to take full advantage of implementing a DAT reset strategy).

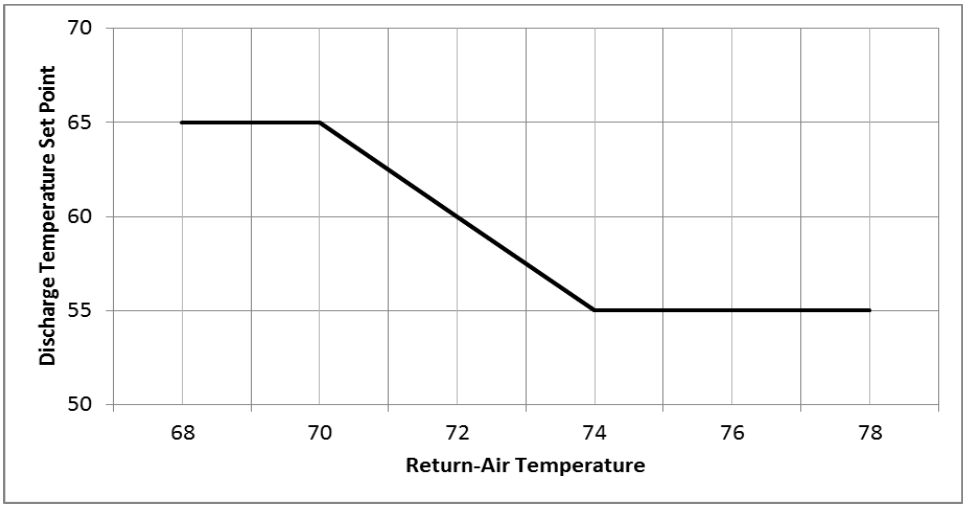

Figure 6.4. Example DAT Reset Graph Based on the RAT

When Is It Appropriate to Use the Return-Air Temperature?

This approach should only be used if none of the other approaches are available due to lack of feedback from the average VAV box reheat valve commands or average VAV zone temperature versus zone setpoint error calculated values. The ability to calculate average values is more code-intensive in the local or supervisory controller and, if there are communication issues, may also create reliability issues. Using the RAT for the DAT reset is the simplest approach outlined, but the other feedback variables (particularly average VAV reheat valve command or average zone setpoint temperature versus zone setpoint error) will give direct evidence of the ability of the VAV boxes to meet the zone temperature requirements with the lowest energy input from both the AHU and the individual zone VAV boxes, which is the most desirable approach for a good DAT reset.

6.2.2 BAS Data Needed to Verify the Discharge-Air Temperature Reset

Analyzing and detecting AHU DAT control problems and opportunities can be achieved by using trend capabilities through the BAS. The following parameters should be monitored using the trend capabilities of the BAS:

- discharge (duct) DAT
- discharge (duct) DAT setpoint
- individual zone VAV box reheat valve positions (for zones served by each AHU). This is needed for Approach 1 above.
  - Create a calculated average of all zone VAV box reheat valve positions served by each AHU for Approach 1 above. Ensure noncritical and/or rogue VAV boxes are removed from the calculation.
- Individual VAV box zone setpoint minus zone temperature (ΔT) values, which are all averaged. This is needed for Approach 2 above (calculated as the average of each zone's setpoint minus the same zone's temperature sensor value). This value generally ranges from (-) 2.0 up to (+) 2.0 as the zone varies from warmer to cooler (with respect to the zone setpoint)
  - Create a calculated average of all VAV box zone temperature setpoint versus VAV box zone temperature (delta) served by each AHU for Approach 2 above. Ensure noncritical and/or rogue VAV boxes are removed from the calculation.

- OAT – either the temperature sensor used globally or locally (choose the most accurate sensor). This is needed for Approach 3 above.
- RAT – the AHUs RAT sensor. This is needed for Approach 4 above.

Data should be collected at recommended intervals of between 5 and 30 minutes for a minimum of 1–2 weeks. When analyzing the DAT reset, the trends to look for include the following:

Is there an existing reset for the DAT?

- Is the DAT setpoint too high or too low?
  - Review the trends of reheat valve position of zone VAV boxes vs. time
    - If most reheat valves are nearly closed, the DAT may be too high
    - If several reheat valves are fully open (greater than 75%), the DAT may be too low.

6.2.3 Is There an Existing Reset for the Discharge-Air Temperature?

DAT control specifications are taken directly from design specifications. To maintain the DAT setpoint, the controls will modulate the heating coil, economizer, and cooling coil in sequence. After construction, the DAT setpoint is often adjusted as needed to satisfy the most demanding zone. This is generally a problem area that may need a lower (or sometimes higher) than required setpoint for the remainder of the system.

An example of bad operation can be seen in Figure 6.5 where the system operates without a DAT reset. Here, the DAT is set at a constant 52°F during the day and off only for 8 hours at night. The setpoint is also very low (52°F) for this time of year (February). Figure 6.6 shows an example of good operation, where the AHU has a DAT reset strategy in place.

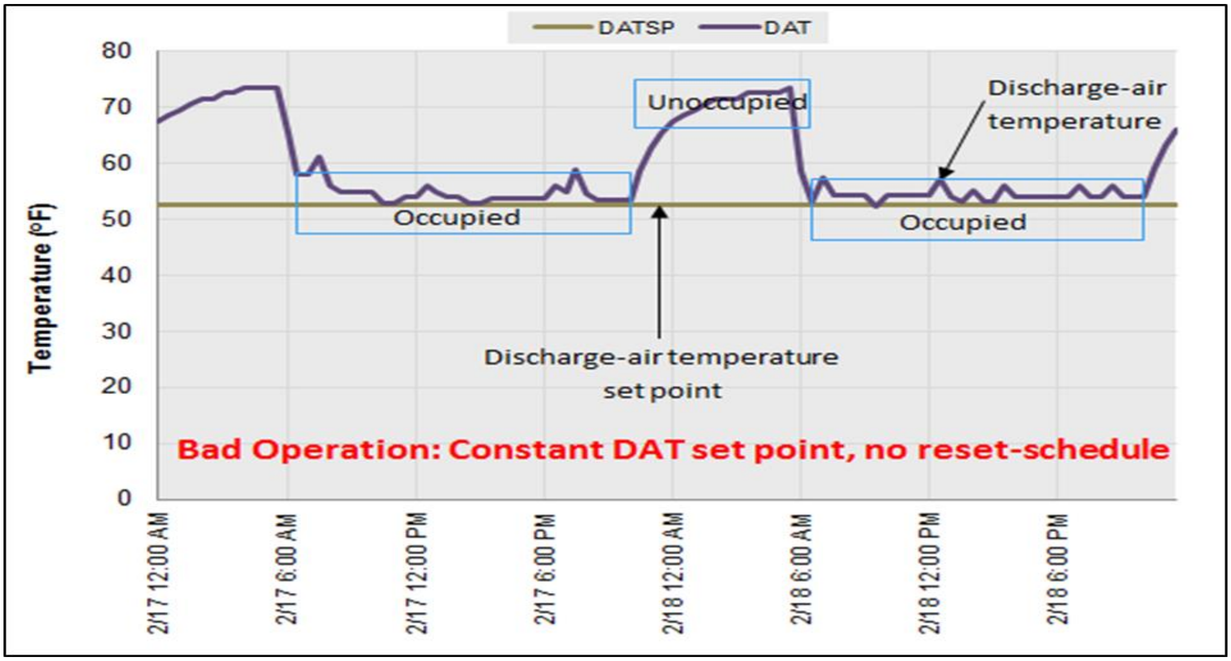

Figure 6.5. Constant DAT Setpoint (No Reset)

In Figure 6.6 the DAT is reset continuously during the 2 day sample period. The DAT setpoint appears to track with the time of day as the minimum DAT setpoint is occurring from approximately 12:00 p.m. until 6:00 p.m. Figure 6.7 provides an example of optimal operations, where the DAT setpoint is continuously reset based on zone feedback throughout the day and where the DAT and setpoint are almost perfectly matched (very stable control).

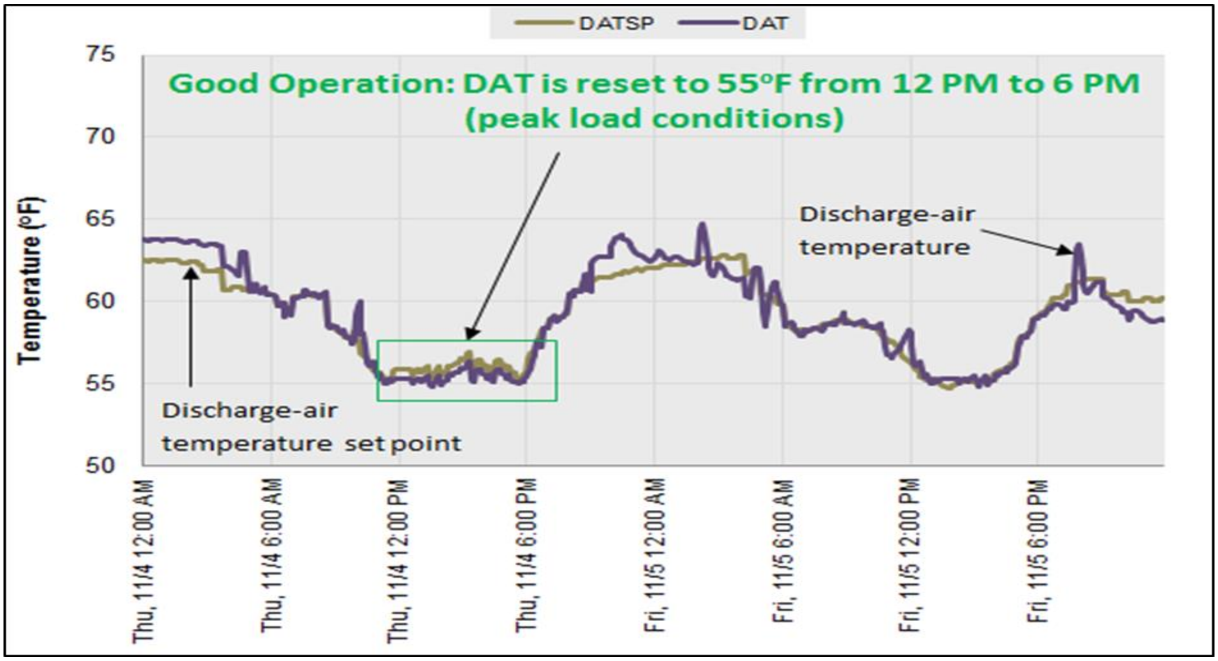

Figure 6.6. DAT Reset based on (Time of Day) or OAT

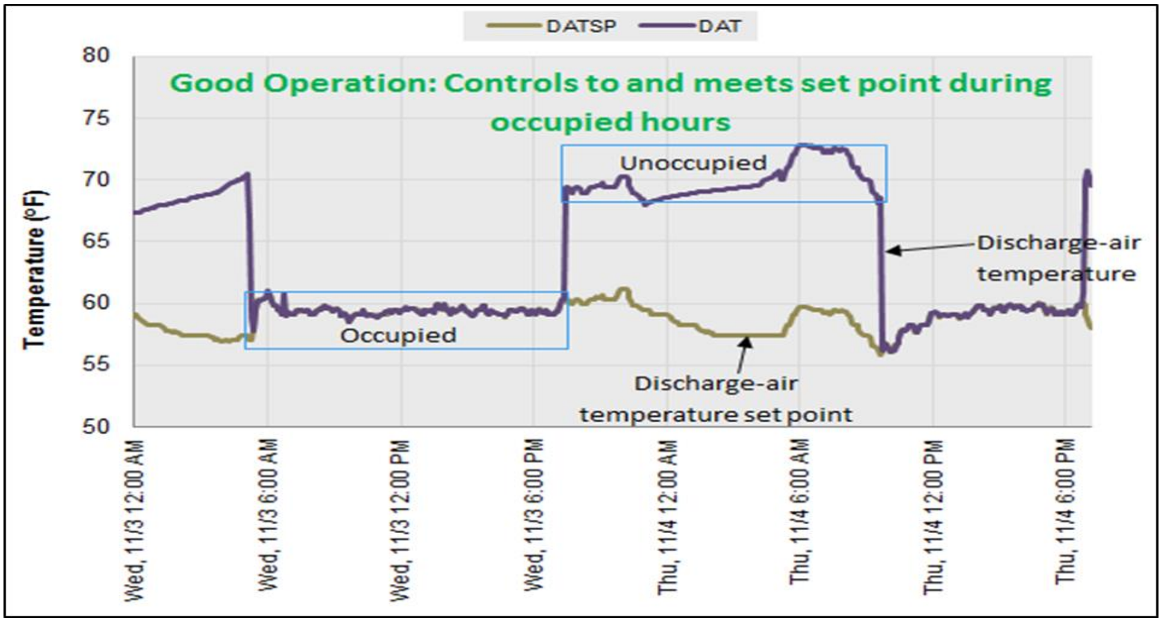

Figure 6.7. DAT Reset based on Zone Feedback

6.2.4 Is the Discharge-Air Temperature Setpoint Too High, Too Low, or Unstable?

To determine whether the DAT setpoint is too high, too low, or unstable, review the zone VAV box reheat valve positions versus time. Ideally, zone VAV box reheat valves should modulate between 10% and 30% open. If most of the zone VAV box reheat valves are closed during cooling, the DAT setpoint is most likely too high. If most of the zone VAV reheat valves are wide open during cooling, the DAT setpoint is most likely too low. Figure 6.8 shows the corresponding zone VAV box reheat valve positions versus time for the AHU that serves the zones.

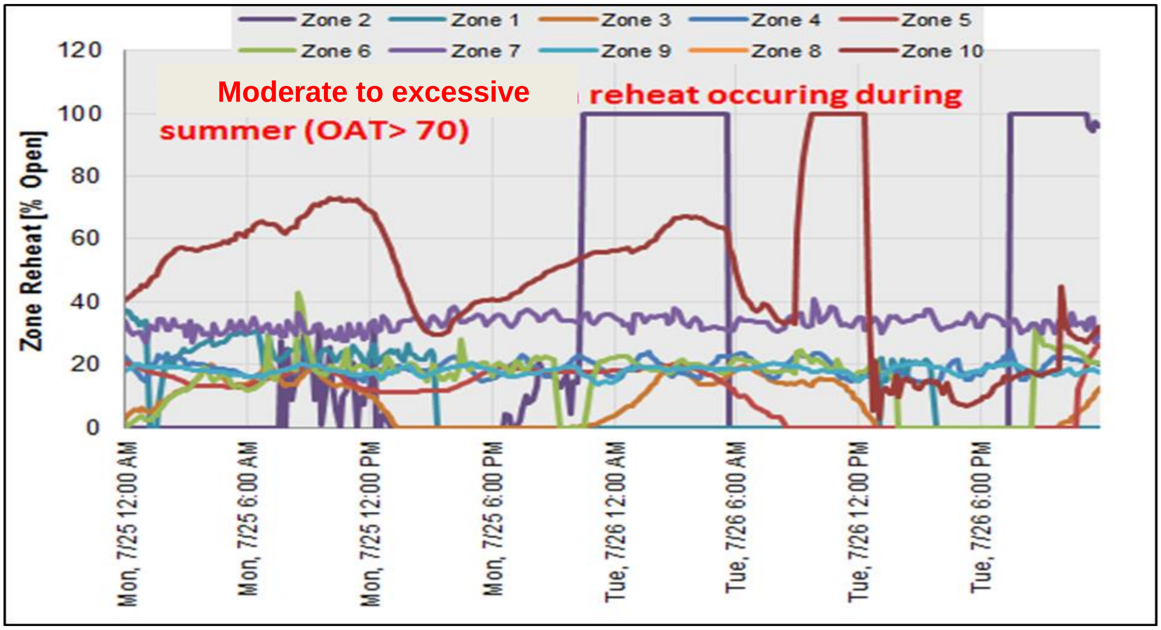

Figure 6.8. Zone VAV Box Reheat Valve Position Trend

This AHU serves 10 zones, and all of the zone VAV box reheat valves are less than 40% open during the trend sample period (except for two of the zones, which appear to be over-cooled, and therefore require more reheat). This is an indicator that the DAT setpoint is probably not optimal. The trend data is from the last week of July when OATs are warmer than 70°F.

Figure 6.9 shows an example of bad operations for an AHU that has too low of a DAT setpoint. In Figure 6.9 all of the zone VAV box reheat valve positions are more than 75% open, which indicates that the DAT is too low. If the majority of the boxes have reheat valve positions more than 40% open, the DAT setpoint should be increased because the DAT setpoint is too low.

A DAT reset control sequence should do this automatically, if it is set up and working correctly.

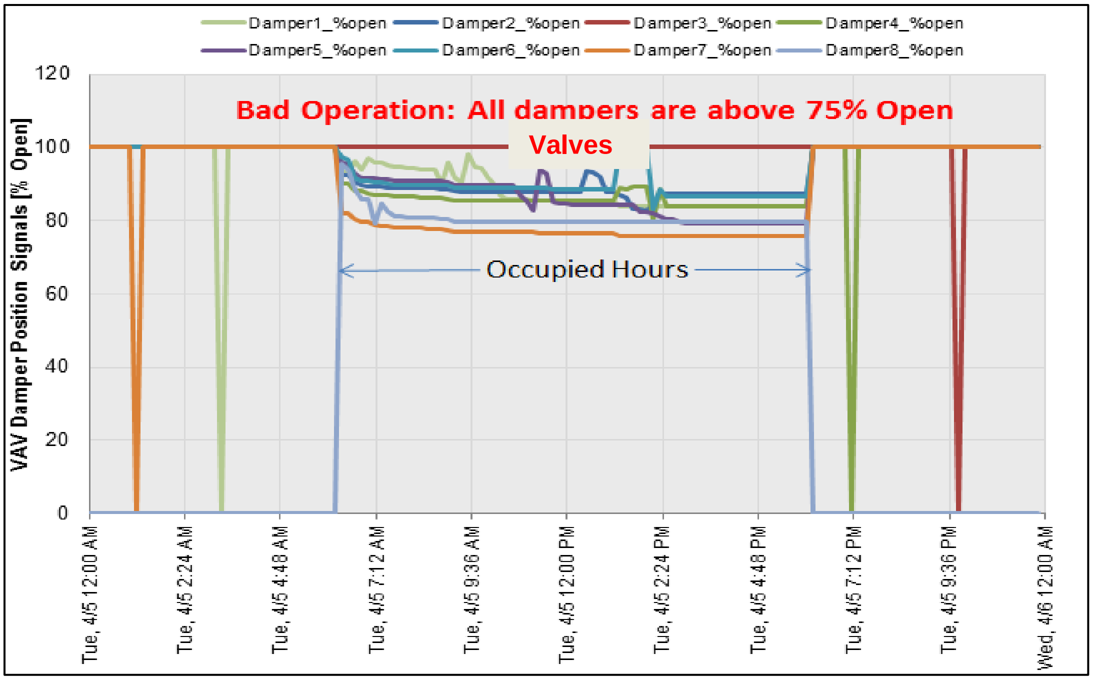

Figure 6.9. Zone VAV Box Reheat Valve Positions vs. Time (DAT Setpoint is Too Low)

6.2.5 Identifying and Correcting Problems that May Interfere with the Functional Discharge-Air Temperature Reset

This section of the guide covers some common zone VAV box and AHU DAT issues, things to look for to help identify these issues (through use of the BAS interface and trend data), and a list of possible problems that would cause the issue.

1. Identify zone VAV box reheat valves that are not modulating with changing indoor and outdoor conditions, and zone VAV box reheat valves that are not being controlled or not responding to control signals.
   - Zone VAV box reheat valve commands that are always reading 100% open while the AHU DAT values are 60°F or less, and the zone temperatures are less than the zone heating setpoint may indicate failed component(s), such as:
     – Zone VAV box reheat valve has failed closed (actuator motor failure, loose wire, power failure, or mechanical failure of reheat valve – motor linkage).
     – Balancing valve or manual isolation valve for the reheat coil is closed.
     – Controller/electronics have failed.
     – If the VAV box includes a DAT sensor (leaving reheat coil temperature) and the value is within 2-5°F of the AHU DAT value while the reheat valve command is 100% open, this may confirm that the valve has failed (or other upstream valves are closed).
   - Zone VAV box reheat valve commands that are always reading 0% open with zone temperature readings that are higher than the zone setpoint while the AHU DAT is less than 60°F may indicate failed component(s).
     – Zone VAV box reheat valve has failed open (actuator motor failure or mechanical failure of reheat valve – motor linkage).

   - If the VAV box includes a supply-air temperature sensor (leaving reheat coil temperature) and the value is more than 5-10°F greater than the AHU DAT value while the reheat valve command is 0% open, this may confirm that the valve has failed open.

3. Look for failed or improperly located DAT sensors.
   - DAT sensors that are always reading significantly lower or higher than the DAT setpoint, may indicate failed component(s), such as:
     - DAT sensor is wired incorrectly or has a loose wire.
     - The sensor has failed.
     - The sensor location is not correct (located before the cooling coil or further downstream to not accurately detect the leaving coil temperature).

4. Identify AHU heating coil control valves that are not modulating or responding to control signals.
   - AHU heating coil control valve commands that are always reading 100% open with very low AHU DAT readings may indicate failed component(s), such as:
     - AHU heating coil control valve has failed closed (actuator motor failure, loose wire, power failure, or mechanical failure of the heating coil control valve – motor linkage).
     - Balancing valve or manual isolation valve for the heating coil is closed.
     - Controller/electronics have failed.
     - AHU cooling coil control valve has failed open or is overridden open and heating coil control valve is responding to the over-cooled AHU DAT.
   - AHU heating coil control valve commands that are always reading 0% open with very high AHU DAT readings may indicate failed component(s), such as:
     - AHU heating coil control valve has failed open (actuator motor failure, loose wire, power failure, or mechanical failure of the heating coil control valve – motor linkage).

5. Identify AHU cooling coil control valves that are not modulating or responding to control signals
   - AHU cooling coil control valve commands that are always reading 100% open with very high AHU DAT readings may indicate failed component(s), such as:
     - AHU cooling coil control valve has failed closed (actuator motor failure, loose wire, power failure, or mechanical failure of the cooling coil control valve – motor linkage).
     - Balancing valve or manual isolation valve for the cooling coil is closed.
     - Controller/electronics have failed.
     - AHU heating coil control valve has failed open or is overridden open and cooling coil control valve is responding to the over-heated AHU DAT.
   - AHU cooling coil control valve commands that are always reading 0% open with very low AHU DAT readings may indicate failed component(s), such as:
     - AHU cooling coil control valve has failed open (actuator motor failure, loose wire, power failure, or mechanical failure of the cooling coil control valve – motor linkage).

7.0 Boiler Lockout

The purpose of this boiler lockout implementation guide is to show how the lockout of boilers can be efficiently controlled and what the indicators are for both good and bad operations. This guide also offers some common implementation strategies.

7.1 Why Consider Implementing a Boiler Lockout?

Boiler lockouts are simple to implement, track, and administer. The goal for any building with properly implemented boiler lockout strategies is to see energy consumption reductions during outdoor temperature conditions when the building should not need heating, or the requirement for heating is very low. Boiler lockouts achieve energy consumption reductions by disabling boilers and associated hot water pumps, whenever they are not needed and refraining from starting up boiler systems when loads associated with vacancy periods (nights and weekends) are minimal.

A boiler hot water system is typically designed to provide hot water to help maintain minimum temperature setpoints in a building. Historically, minimum temperature requirements have been associated with cold weather (seasonal) operations. However, in climate zone locations that have mechanical cooling systems configured for dehumidification (over-cool the discharge air below the dew point temperature to cause moisture condensing in the primary heating, ventilation, and air-conditioning [HVAC] system), the requirement for zone reheat may exist (because discharge-air temperatures may be as low as 50°F). When this occurs, reheating at the zones being served is often a comfort requirement, unless the zones experience significant cooling loads (solar and other internal heat gains). In these situations, boilers and their associated equipment (pumps, etc.) may be required to operate at much higher outdoor-air temperature (OAT) values, to ensure the minimum temperature requirements during occupancy of the building are being met.

When a boiler is enabled, the associated pumps are enabled to operate, with the assumption that the building's end-use loads are active and have a valid heating requirement. End-use loads that may have a valid heating requirement can include AHUs, rooftop units (RTUs), variable-air-volume (VAVs) boxes, fan-coil units (FCUs), perimeter baseboard heating systems and similar HVAC systems.

A boiler cycle consists of a firing interval, a post-purge, an idle period, a pre-purge, and a return to firing. Boiler efficiency is the useful heat provided by the boiler divided by the energy input (useful heat, plus losses) over the cycle duration. Because boilers are often sized for design load conditions (cold weather), it is not uncommon to find boilers that are short-cycling. Short-cycling is often a result of a boiler's inability to "turn down" (reduce its firing rate or rate of natural gas combustion). When a boiler has very low load, the boiler (once it begins firing) will quickly overshoot the internal setpoint for the boiler (temperature if hot water, pressure if steam). Short-cycling losses occur when an oversized boiler quickly satisfies process or space-heating demands, and then shuts down until heat is again required. Boiler efficiency decreases when short-cycling occurs. This decrease in efficiency occurs, in part, because fixed losses are magnified under lightly loaded conditions. For example, if the radiation loss from the boiler enclosure is 1% of the total heat input at full load, at half load the losses increase to 2%, while at one-quarter load the loss is 4%. In addition to radiation losses, pre-and post-purge losses occur. In the pre-purge, the combustion fan operates to force air through the boiler to flush out any combustible gas mixture that may have accumulated. The post-purge performs a similar function. During purging, heat is removed from the boiler(s) as the purged air is exhausted from the building and takes heat energy from the boiler with the exhausted air. Implementation of a boiler lockout when heating loads are low or nonexistent can help alleviate boiler short-cycling.

All boilers that are equipped with reliable controls are good candidates for implementing boiler lockouts, under the following assumptions:

- The BAS and boiler controllers are reliable (communication between boilers and the supervisory controllers is more than 95% reliable and any integration issues for specialty systems such as boilers are totally resolved).
- The boilers are designed for on/off cycling, as well as thermal contraction and expansion within vendor-recommended guidelines. Some boilers, due to age or technology, are not recommended for cycling on/off and once started for the heating season, are left in an operational state (by design). In certain situations, the hot water pump that serves the boiler loop can be locked off, but the boiler can be left in an operational state (hot standby) so that the boiler does not "see" temperature swings.
- The installed piping, if Victaulic design, is designed with the proper fittings and seals that can tolerate temperature fluctuations as loop temperatures reset and as water temperatures cool down over nights, weekends and even extended periods of lockout (warm weather, summer shutdown periods, etc.).

7.2 What Systems Should Be Considered for Boiler Lockout Implementation?

Some of the typical HVAC systems that may be found in various buildings that should be considered for boiler lockout include the following:

- condensing hot water boilers
- non-condensing hot water boilers (where vendors approve significant temperature swings)
- non-condensing hot water boilers (where the boilers can remain in a hot-standby condition, while the circulating pumps are off).

Note: Ensure that the installed piping, if Victaulic design, is designed with the proper fittings and seals that can tolerate temperature fluctuations as loop temperatures reset and as water temperatures cool down over nights, weekends and even extended periods of lockout (warm weather, summer shutdown periods, etc.).

Note: Most steam boilers are not designed to cycle off and remain off for long periods of time, due to condensate issues that might occur when the boiler re-starts (water-hammer as well as thermal expansion/contraction issues). The O&M team should be aware of steam-water hammer issues related to poor steam trap operations and manage steam systems that startup in the fall (after an extended shutdown period) to avoid steam contacting standing or pooled water (large quantities) that could create water hammer response in the steam piping. All automatic steam management operations should be thoroughly tested to ensure all components are working properly to ensure mitigation of steam-water hammer.

7.3 Methodology for Boiler Lockout

This section of the implementation guide goes through some common strategies for implementing boiler lockout, including

- boiler lockout (definition and how it works)
- night setback response (definition and how it works)
- scheduled HVAC systems – morning warm-up.

Note that this list is not intended to cover every implementation technique, but to present a few techniques in detail to provide some context for what a good boiler lockout implementation might look like. All final boiler lockout parameters should be thoroughly discussed with the owner/operator of the BAS and the building(s) where the boiler lockout and related strategies (night setback and extreme weather operations) may be applied. The implementation strategy may vary by building type, system type, or other variables that this guide does not directly highlight.

The goal for any building with properly implemented boiler lockout strategies is to see energy consumption reductions during outdoor temperature conditions when the building will not need heating, or the need for heating is very low. This is typically based on the OAT. A boiler lockout will disable the boiler and corresponding hot water pumps when the OAT is greater than the configured lockout setpoint. Lockout setpoint values often vary from 50°F to as high as 70°F. As lockout setpoint values increase, boilers may operate under low loads, which contributes to boiler short-cycling.

The simplest boiler lockout strategy is a constant OAT lockout setpoint. The boiler and corresponding hot water pumps are locked out (disabled) anytime the OAT is greater than the lockout setpoint. This strategy can be very effective but care must be taken when choosing an appropriate lockout setpoint. Typically, for dry climates the heating loads will be small to nonexistent when the OAT is greater than 60°F. In climate zones where dehumidification of the outdoor air is required, boilers and their associated equipment (pumps, etc.) may be required to operate at much higher OAT values to ensure minimum temperature requirements during occupancy of the building are being met. As lockout setpoint values increase, boilers may operate under low loads, which contributes to boiler short-cycling.

The lockout setpoint value can also be based on a scheduling feature that automatically adjusts the lockout setpoint, (based upon the time of day and/or day of week) to a higher or lower OAT. For example, the lockout setpoint could be set to a higher value during occupied periods and then setback to a lower value during unoccupied periods.

7.3.1 Implementing Boiler Lockout – Why Is this Important?

When boiler lockouts are put in place, the immediate challenge for most configurations is how the building responds during the vacancy period when most of the HVAC systems are turned off. At a minimum, the HVAC controls for the zones should be configured to automatically re-activate the required minimum HVAC systems to allow the building to maintain the zone temperatures at their required setback values.

Typical zone temperature setback values for heating are between 55°F and 65°F. This means that the HVAC systems will remain off, until the zone temperatures (lowest or average values) drop to the heating setback value. If the BAS does not have setback values configured or does not have the control code that would automatically activate the required HVAC systems to maintain the building at the setback values, than it is likely that the building will over-cool during cool/cold weather.

It is imperative that setback values are properly configured. It is also imperative that the automated responses that should be occurring during night and weekend periods are actually occurring. This includes automatic restart of the boiler(s) and associated hot water circulating pumps.

7.3.1.1 Winter Night Setback Operation Verification Steps

1. Verify that auxiliary systems (boilers, pumps) are active when AHU and zone HVAC systems activate during night setback actions, as well as during morning warm-up periods.

2. Verify that the heating hot water being delivered during the night setback or morning warm-up periods is sufficient to satisfy the heating loads (AHU discharge temperature setpoints, FCUs, VAV zone reheats, etc.).
3. If the building has a perimeter hot water heating system that serves perimeter radiation (fin tube baseboard) or perimeter zone terminal boxes, ensure that the hot water heating system activates early enough to maintain minimum perimeter space temperatures in the building, prior to occupancy. This is especially true for optimal start/scheduling measures that rely on heating systems to recover the building.

7.3.1.2 Shoulder Season/Summer Verification Steps

1. Verify that the heating hot water being delivered for zone reheat is sufficient to satisfy the heating loads (VAV zone reheats, etc.). Typically, this would be when OATs are above 55–60°F and the loop should not be any higher than 120°F (for condensing boilers) and 150°F (for non-condensing boilers). For warm weather reheat requirements, the hot water heating system should be disabled when occupancy ends, as most AHUs turn off during the unoccupied period. If AHUs turn back on to maintain the building's zone setback temperature setpoints (typically 80-85°F), there should not be any need for reheat, as the cooler AHU discharge air will help drive the zones back to the desired setback temperature, minus the dead band differential (typically 2-5°F below the setback setpoint).

7.3.1.3 BAS Data Needed to Verify Boiler Lockout

Analyzing and detecting boiler lockout problems and opportunities can be achieved by using trend capabilities through the BAS. The following parameters should be monitored using the trending capabilities of the BAS:

- heating hot water supply temperature
- heating hot water pump status
- boiler status
- zone temperatures.

The recommended frequency of trend data collection is between 15- and 30-minute intervals for a minimum of 1–2 weeks. When analyzing the heating hot water or boiler systems, the trend data to look for include the following:

- Do the zone temperature trend data indicate that zone temperatures are dropping at night and during weekends (during cool-cold weather vacancy periods) to the configured heating setback values (55–65°F)? When setback actions are occurring, the AHU, RTU, FCU, and boiler/pumping systems should also indicate temporary operations to maintain the setback values.
- Do the trend data for the boilers, pumps, and zone heating systems indicate that they are disabled and not operational when the OAT is above the lockout setpoint?
-
- Do the trend data for the zone temperatures and the AHU discharge-air temperatures indicate that temperatures are rising during the morning warm-up period, to properly warm the building prior to occupancy (in an optimal manner)?

If the trend data for these systems indicate that boilers are not locking out when expected to, or that the boilers are not enabling when expected to, a thorough review of the control sequences for the boilers and pumps should be performed and any issues resolved. Failure to do so may result in equipment overrides, which will result in excess energy consumption as well as excess equipment wear and tear that may lead to shortened equipment life.

8.0 DOAS Discharge-Air Temperature

The purpose of this dedicated outdoor-air system (DOAS) discharge-air temperature (DAT) implementation guide is to help highlight how the DOAS DAT can be efficiently controlled and what the possible indicators are for both good and bad operations. This guide assists in identifying DOASs that are good candidates for implementing a DAT reset, and offer common implementation strategies and guidelines. DOASs can be significantly more complicated than typical air-handling units (AHUs).

8.1 Why Consider Implementing a DOAS DAT Reset?

DOASs are designed to provide large quantities of outdoor air to meet whole building ventilation and building pressurization requirements in many buildings. Typical DOAS applications rely upon building exhaust air to transfer energy via one or more methods into (or from) the ventilation air being introduced from outside the building. This is referred to as energy-recovery ventilation (ERV) and may involve the transfer of sensible heat only via a heat wheel, heat pipe, or run-around coil. It may also involve the transfer of sensible and latent energy (via an enthalpy wheel or a fixed plate). The ERV is the energy-recovery process of exchanging the energy contained in building air (that is exhausted) and using that energy to treat the incoming outdoor ventilation air in heating, ventilation and air-conditioning (HVAC) systems (for both heating and cooling of the outdoor air). The benefit of using energy recovery is the ability to meet the ASHRAE ventilation and energy standards, while improving indoor-air quality (IAQ) and reducing total HVAC equipment capacity.

Most DOASs supplement the free energy transfer from the ERVs with additional heating and/or cooling coils. When heating and cooling functions are not properly sequenced with the heat-recovery portion of the DOAS or with the temperature requirements of the downstream systems served by the DOAS, excess heating energy may be delivered by the DOAS. Excess heating energy will need to be removed at the DOAS (cooling coil) or by the downstream AHUs or the various downstream zone terminal equipment (variable-air-volume [VAV] boxes, fan-coil units [FCUs], chilled beams, etc.).

The heating and cooling energy required to temper the outdoor air can account for a significant percentage of a building's total energy consumption, but the design and operations of the DOAS subsystems can significantly reduce those energy requirements. A common strategy with some DOAS configurations is to deliver "neutral air" from the DOAS to the downstream loads. Neutral air is generally considered conditioned air that is close to the desired space conditions (between 70–75°F and 40–60% relative humidity [RH]). Because the DOAS effectively de-couples the ventilation requirements of the other HVAC systems from outdoor conditions, the other HVAC systems can be down-sized to deal with comfort heating and cooling loads. The DOASs (when designed properly) also help deal with building air pressurization issues (infiltration) and building IAQ issues such as carbon dioxide (CO_2), odor, and moisture mitigation.

The likelihood of energy improvement opportunities related to proper sequencing and DOAS DAT control is high. However, the need to coordinate the control of an array of subsystem components (heat recovery, heating, cooling, frost mitigation, dehumidification, or humidification) can result in complicated control sequences that may not be optimized.

Enthalpy wheels are designed to help offset the required cooling energy expended for dehumidification in humid climate zones, or to help the building retain acceptable moisture levels in dry climate zones without the need for (or to reduce the amount of) supplemental humidification. Most modern heat wheels and enthalpy wheels are designed to rotate via a variable-frequency drive (VFD) that is controlled to maintain a DOAS DAT setpoint, with supplemental heating and cooling coming into play when the wheel reaches 100% speed (except during certain heating season operational limitations).

Heat pipes (which can be designed for multiple configurations) are intended for applications where two air streams are side by side in a horizontal plane, or one air stream is above the other with minimal distance between them. It is the heat exchanger of choice in applications where cross contamination of the discharge air by the exhaust air is not permissible such as labs, hospitals, and other health care facilities. With no moving parts and a long service life, this product is the most reliable heat exchanger in its class. Heat pipes can be designed for either predominantly heating or predominantly cooling seasons or equal heating and cooling seasons. Primary designs may include the ability to “tilt” the coils for liquid Freon to flow from the cooler air stream to the warmer air stream where it will vaporize (phase change) and flow back up to the cooler coil as a gas, which allows heat transfer to continue as long as air is flowing.

When the OAT drops low enough to cause the exhaust-air stream temperatures to drop below freezing (as heat transfer is occurring), the risk of developing frost on the exhaust coil occurs. Frost that develops on coils can result in reduced airflow rates, which can exacerbate this condition. Leaving exhaust coil temperature sensors are required for proper control sequences that address frost mitigation. To mitigate frost buildup, the control sequence will either slow the heat wheel (via VFD speed reduction), stop or divert the flow in the hydronic loop to the exhaust coil or flow to the exhaust heat pipe, or use face-and-bypass dampers (on either the supply side or the exhaust side) to bypass air around the heat-recovery coil so it does not frost up in the exhaust air stream, when temperatures are close to freezing in the exhaust air stream. In all cases (heat wheel speed reduction, modulating valve, or modulating face-and-bypass damper operation), the result is a reduction or cessation of heat recovery. If a heating coil is located upstream of the heat-recovery coil (see Figure 8.1), it can be configured to activate once the heat-recovery system is either maximized or when recovery actions must be reduced or cease, due to frost conditions. Use of a face-and-bypass damper also reduces the DOAS fan pressure losses that occur when air was flowing across a coil (but is now being bypassed around the coil). The control sequences should optimize the fan power pressure drop across heat recovery coils or heat wheels by using bypass dampers to re-direct airflow around the heat recovery coils or heat wheels when not in use.

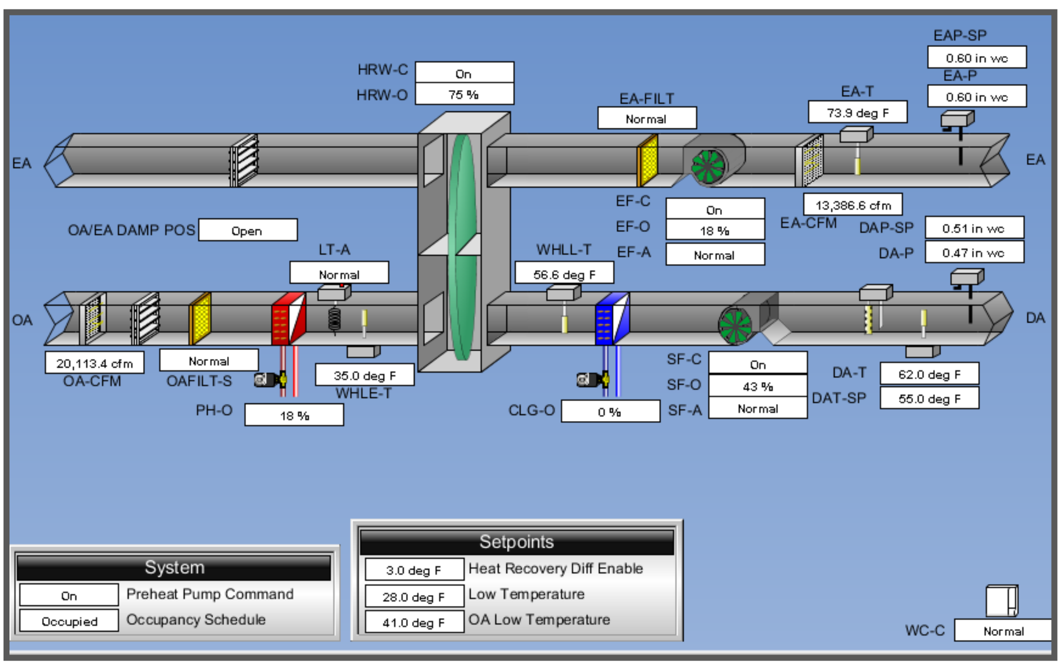

Figure 8.1. Example DOAS Heat-Recovery System for 100% Outdoor-Air Ventilation

As already stated, DOAS designs can be found in many different configurations. Figure 8.1 provides *one example* of a DOAS system that is designed to use an enthalpy wheel for sensible heat recovery and also some latent heat (moisture) removal. The DOAS configuration shown in Figure 8.1 does not provide any active humidification capability (but it could), and it shows the active heating coil upstream of the wheel (typical). This design allows for active heating to occur with or without the aid of the enthalpy wheel. If the enthalpy wheel is active, the heating coil load is significantly reduced. In Figure 8.1, the heat wheel is increasing the leaving hot water coil temperature by almost 22°F (35.0°F up to 56.6°F) in an attempt to meet the DAT setpoint (55°F). However, the final DAT is 62.0°F, which is 5.4°F higher than the setpoint. The added energy most likely is from the supply fan motor and highlights improvement opportunities for the heating coil and heat wheel controls. Because the configuration shown does not include an exhaust-air temperature sensor downstream of the heat wheel, the controls cannot determine the leaving exhaust-air temperature that would indicate values that are below freezing.

It appears that the controls use a fixed low-temperature setpoint (28°F), which most likely refers to either the OAT or to the leaving hot water heating coil temperature. Once this condition is true (too cold outside or too cold on the leaving-air temperature for the hot water coil), the heat wheel would either slow down (reduce its speed) or it would be disabled to ensure that the heat-recovery wheel in the exhaust-air stream does not ice up (potentially restricting airflow). Reduction or cessation of heat recovery will result in more energy being required from the mechanical heating coil (to make up for the heat recovery reduction). The heating coil location shown in Figure 8.1 allows for heat-recovery actions during colder weather as opposed to a heating coil location that is downstream of the heat-recovery system. If the heat-recovery wheel is the first energy-transfer device on the supply side, it may be disabled for more hours in the heating season due to the colder temperatures that could result in the heat-recovery wheel in the exhaust air-stream icing up.

A design improvement that provides for a leaving heat wheel (exhaust-air stream) temperature sensor would allow a more intelligent control sequence that would disable the heat-recovery wheel or slow it down, when the leaving heat wheel (exhaust-air stream) temperature drops below 35°F–40°F. Most engineers that design heat-recovery wheel control sequences, configure the heat-recovery wheel leaving-air temperature setpoint (exhaust-air stream) for a 35–40°F setpoint value to account for temperature sensor accuracy anomalies and to ensure that the exhaust-air components for heat-recovery systems do not frost over as the moisture from the building impinges on surfaces that are below freezing.

DOASs come in many different configurations relative to how they interface with building supply and exhaust systems, and they are typically designed to provide outdoor air (ventilation) to a building, while also exhausting most of the building air. This typically requires outdoor and exhaust ducts to intersect at the same location to enable energy transfer between the two air streams (outdoor air and exhaust air). The intersection of outdoor air and exhaust air streams is not a requirement for run-around loops that are based on hydronic (pumped) loops, which move energy via pumps, and control valves, which modulate to vary the heat transfer rate (or reduce the rate for frost protection requirements). The intersection of air streams for heat pipes (while recommended) is not a hard requirement (some separation is allowed). Heat pipes rely on some elevation difference for the working fluid (refrigerant) to migrate up or down the pipes as phase changes occur.

The following methods are discussed and will be the basis for different DAT control sequences:

- The DOAS provides all ventilation air to downstream AHUs via the AHU's economizer (outdoor-air damper connection).
    - Typically, these systems are designed such that the DOAS is providing no more than 15%–25% of the total AHU design airflow.

- This design allows for much lower DOAS DATs (cool-cold weather) and much higher DOAS DATs (warm weather), because the AHU will always be mixing DOAS air with the AHU return air.
- The DOAS primary role is ventilation and moisture removal (when needed). If the DOAS includes a heat wheel that is capable of moisture removal, it can be used to help reduce the dehumidification loads on the DOAS and downstream mechanical cooling coils.
- Downstream AHUs most likely will include mechanical cooling coils for final tempering of the air, along with additional moisture removal.
- The DOAS should be optimized via scheduling to not start up in the morning until just prior to the beginning of occupancy (to help remove stale air and ventilate the building).
- Temperature control sequences:
  - During cold weather (OAT <50°F), the DOAS setpoint may be as low as 45–50°F (or as determined by the design engineer). Lower setpoints reduce the DOAS mechanical heating and allow for heat recovery to be the primary energy input (transfer energy from the exhaust-air stream to the discharge-air stream), while supplementing any additional energy from the DOAS hot water heating coil. Lower setpoints take advantage of larger volumes of downstream air at the AHU that mix AHU return air with smaller volumes of discharge air from the DOAS to obtain a neutral temperature that may require additional cooling or heating (minimum quantity) to achieve the desired AHU DAT setpoint.
  - During warm weather (OAT >75–80°F), the DOAS setpoint may be as high as 65–70°F (or as determined by the design engineer). Higher setpoints reduce the DOAS mechanical cooling and allow for heat recovery to be the primary energy input (transfer energy from the exhaust-air stream to the discharge-air stream), while supplementing any additional energy from the DOAS chilled-water cooling coil. Higher setpoints take advantage of larger volumes of downstream air at the AHU that mix AHU return air with smaller volumes of discharge air from the DOAS to obtain a neutral temperature that may require additional cooling (minimum quantity) to achieve the desired AHU DAT setpoint.
  - During warm and humid weather, the DOAS DAT setpoint may need to be overridden to obtain a lower dew point delivery temperature, to ensure moisture removal is occurring. Use of latent energy heat wheel designs should be maximized for this possibility. The DOAS DAT setpoint may be automatically calculated, based upon the outdoor air humidity ratio, outdoor dew point, or the indoor space humidity (if properly and accurately sensed). Calibrated and accurate outdoor humidity sensors are also a requirement.

- The DOAS provides all ventilation air into the building via dedicated VAV boxes with dedicated discharge ductwork that feeds other HVAC systems. This configuration can be found with FCUs, passive or active chilled beams, where the VAV box discharge ductwork is connected to the return-air side of the FCU, is separately ducted into the same space as the active chilled beams, or is directly ducted to the suction side of and active chilled beam system, which employ blower coil units (BCUs) to ensure adequate ventilation. This design may include VAV boxes that provide the dedicated return/relief/exhaust-air requirements and are designed to help balance the building to ensure building pressures are not negative or positive pressure (excessive).
  - The DOAS primary role is ventilation and moisture removal (when needed). If the DOAS includes a heat wheel that is capable of moisture removal, it can be used to help reduce the dehumidification loads on the mechanical cooling coil.
  - Downstream FCUs most likely will include mechanical cooling coils for final tempering of the air, along with moisture removal and they may include heating coils (a two-pipe system where

one coil is configured for heating or cooling, or a four-pipe system where dedicated heating and cooling coils are provided).

- Caution should be exercised with active chilled beams that rely upon BCUs to provide air from the DOAS. If the DOAS DATs are too low, this could result in condensation at the chilled beams. Chilled beam water temperatures will generally be no lower than 55–58°F (to avoid dew point/condensation issues), but the DOAS DAT setpoint should also be configured to minimize condensation issues. Chilled beams, while configured for radiative cooling, are generally not designed to remove moisture and if condensation occurs, this may result in unwanted water dripping to surfaces below the chilled beam panels (office desks/furniture/etc.).
- FCUs that are part of a variable refrigerant flow (VRF)-designed system, are configured for discharge temperatures that are much lower than typical AHU systems (35–45°F). The FCU determines the cooling requirement, in response to zone temperature deviations from the zone cooling setpoint(s). When the DOAS provides over-cooled air to the return-air side of the FCU, an intelligent DOAS DAT reset that considers FCU cooling and/or heating requirements should be employed where the DAT setpoint is reset based upon the FCU that has the greatest cooling demand.
- If VAV boxes are the primary DOAS air delivery method, over-cooling may occur if the VAV box minimum flow rates are excessive.
- Temperature control sequences:
  - During cold weather (OAT <50°F), the DOAS setpoint may be as low as 45–50°F (or as determined by the design engineer). Lower setpoints reduce the DOAS mechanical heating and allow for heat recovery to be the primary energy input (transfer energy from the exhaust-air stream to the discharge-air stream), while supplementing any additional energy from the DOAS hot water heating coil. Lower setpoints take advantage of larger volumes of downstream air at the zone that mix zone air with smaller volumes of discharge air from the DOAS that may require additional reheating to achieve the desired zone temperature setpoint. For zones that require cooling, the lower DOAS discharge-air temperature may be enough to minimize additional cooling from the FCU or the chilled beam systems.

    Caution: If the zone system (FCU/chilled beam) does not come with any ability to reheat (at the ceiling or along the wall – baseboard heating, etc.), then occupant complaints may arise if the DAT is too low.

    Caution: Colder air with chilled beam zonal designs may result in unwanted condensation at the chilled beam systems, with potential for dripping condensation (on occupants) as a result. If the potential for this exists, DOAS DAT setpoints less than 58–60°F are not encouraged.
  - During warm weather (OAT >75–80°F), the DOAS setpoint may be as high as 65–70°F (or as determined by the design engineer). Higher setpoints reduce the DOAS mechanical cooling and allow for heat recovery to be the primary energy input (transfer energy from the exhaust-air stream to the discharge-air stream), while supplementing any additional energy from the DOAS chilled-water cooling coil. Higher setpoints take advantage of zonal systems (FCUs, chilled beams, etc.) that mix zonal air with smaller volumes of air from the DOAS to obtain a neutral temperature that may require additional cooling to achieve the desired zone temperature setpoint.
  - During humid weather, the DOAS DAT setpoint may need to be reset lower to obtain a lower dew point delivery temperature, to ensure moisture removal is occurring. Use of latent energy

heat wheel designs should be maximized for this possibility. The DOAS DAT setpoint may be automatically calculated, based upon the outdoor air humidity ratio, outdoor dew point, or the indoor space humidity (if properly and accurately sensed). In this situation, if the DOAS serves a chilled beam system, then the ability to reheat the over-cooled air must occur or the potential for unwanted condensation of moisture on the chilled beams is. Calibrated and accurate outdoor humidity sensors are also a requirement. With chilled beam systems, system design and controls must be in place to raise the delivery temperatures of both water and air above the dew point temperature to ensure condensation is not occurring.

- The DOAS provides all ventilation air into the building, either directly via discharge ducts and diffusers or via VAV boxes that have two ducted connections (one from the DOAS and one from the AHU). The VAV box will serve dedicated locations (offices, hallways, and other spaces). This design may include VAV boxes that provide the dedicated return/relief/exhaust-air requirements and are designed to help balance the building's pressure, to ensure there are not any issues with negative or positive pressure (excessive) issues.
  - The primary role of the DOAS is ventilation and moisture removal (when needed). If the DOAS includes a heat wheel that is capable of moisture removal, this functional capability should be validated because there may not be any downstream systems that provide additional moisture removal.
  - With active chilled beams (ACB), the designed control sequences have to be cautious about delivering DOAS DAT values that could result in condensation at chilled beams.
  - Delivery of DOAS air to FCUs that are part of a VRF design are designed for temperatures that are much lower than typical AHU systems (35–45°F).
    - The FCU determines the zone cooling requirements.
    - In the case of a DOAS that delivers over-cooled air to the FCU, there is no method for relieving this excess cooling (except for the FCU to switch over to heating).
  - Temperature control sequences:
    - During cold weather (OAT <50°F), the DOAS setpoint may be as low as 45–50°F (or as determined by the design engineer). Lower setpoints reduce the DOAS mechanical heating and allow for heat recovery to be the primary energy input (transfer energy from the exhaust-air stream to the discharge-air stream), while supplementing any additional energy from the DOAS hot water heating coil. Lower setpoints take advantage of larger volumes of downstream air at the zone VAV box that mix zone air with smaller volumes of discharge air from the DOAS that may require additional reheating to achieve the desired zone temperature setpoint. For zones that require cooling, the lower DOAS DAT may be enough to minimize additional cooling from the VAV boxes.
    - During warm weather (OAT >75–80°F), the DOAS setpoint may be as high as 65–70°F (or as determined by the design engineer). Higher setpoints reduce the DOAS mechanical cooling and allow for heat recovery to be the primary energy input (transfer energy from the exhaust-air stream to the discharge-air stream), while supplementing any additional energy from the DOAS chilled water cooling coil. Higher setpoints take advantage of zonal systems (VAV boxes) that mix zonal air with cooler air from the primary AHU and warmer air from the DOAS to obtain a neutral temperature to achieve the desired zone temperature setpoint.
    - During humid weather, the DOAS DAT setpoint may need to be overridden to obtain a lower dew point delivery temperature to ensure moisture removal is occurring. Use of latent energy heat wheel designs should be maximized for this possibility. The DOAS DAT setpoint may be automatically calculated, based upon the outdoor air humidity ratio, outdoor dew point, or

the indoor space humidity (if properly and accurately sensed). Calibrated and accurate outdoor humidity sensors are also a requirement.

Figure 8.2 shows a DOAS DAT reset based on the OAT. Two lines represent two different reset scenarios and are shown for illustrative purposes only. The **blue line** is a fixed OAT reset (50–70°F DAT) as the OAT rises from 40–70°F. The **black line** is similar, but also reduces in value for moisture control, once the OAT exceeds 60°F. Moisture control can be based on the outdoor dew point, outdoor air humidity ratio, or space humidity conditions and is meant to show that the DOAS can be used to reduce moisture that is introduced into the building. Dehumidification actions may require active reheat capability in the zones, so this must also be taken into consideration before applying DOAS DAT resets that may over-cool without reheat capabilities (avoid occupant comfort complaints).

Heat pipe designs offer the capability to over-cool the outdoor-air stream for dehumidification purposes, while using the incoming outdoor air to transfer heat to the downstream side of the mechanical cooling coil (which provides a free reheat capability and also helps to cool the warm-hot incoming outdoor air). Dual heat wheels also provide similar capabilities. Typically, the first heat wheel would be designed for latent energy transfer (temperature and vapor) between the outdoor air and the exhaust air while the second heat wheel would be designed for sensible energy (temperature) transfer only. The second heat wheel provides some reheat capability while further cooling the exhaust air, which helps to further cool the outdoor air.

Generally, the DOAS DAT should be controlled such that the supply temperature it provides reduces the mechanical heating and cooling requirements at the DOAS, while assisting the building ventilation and moisture reduction requirements. In humid climate zones or where humidity control is critical, the DOAS DAT setpoint may be driven to a much lower setpoint, as a first stage of dehumidification, based on outdoor humidity ratios or outdoor dew point values that assume increasing moisture levels (inside the building) via the DOAS. Other moisture-management systems may rely on space humidity sensors to determine zone humidity conditions which automatically reset the DOAS DAT setpoint to drive the mechanical cooling coil into dehumidification (when or as needed).

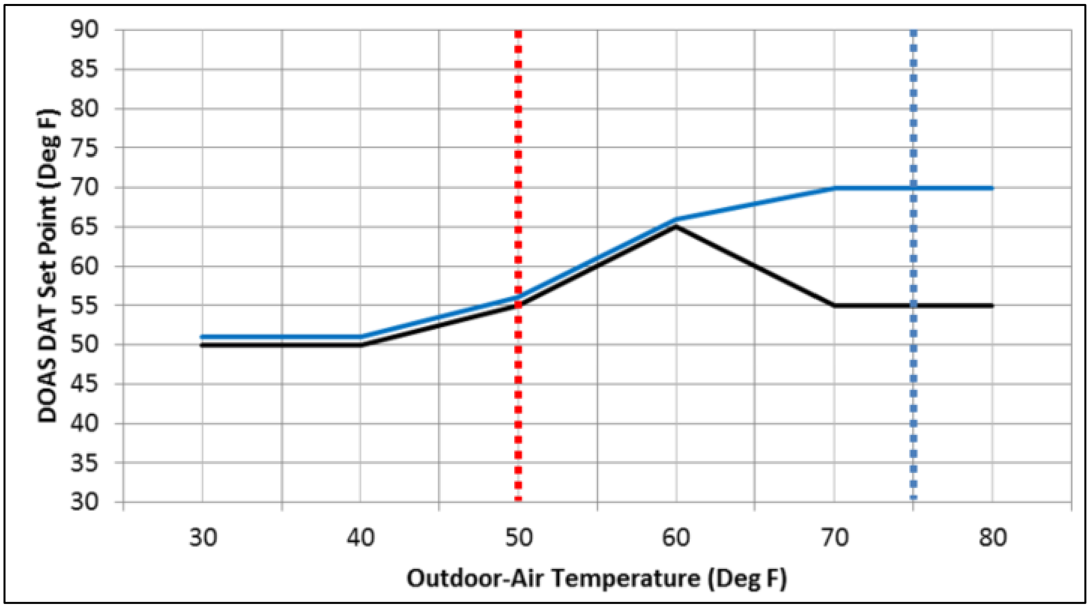

Figure 8.2. Typical DOAS DAT Reset and Heat-Recovery Lockout Setpoints for Heating and Cooling

For heat recovery to occur with most heat-recovery systems (heat wheel, heat pipe, run-around loop, etc.), there must be at least a 3–5°F temperature difference between the exhaust air and the discharge-air streams. If the building exhaust-air temperature is around 75°F, the ability to pre-cool the warmer outdoor air will not occur until there is at least 3–5°F difference (the OAT has to be greater than 78–80°F). If the heat-recovery system is activated when the ΔT is less than 3°F, it may actually result in heat being added to the incoming outdoor air (increasing the mechanical cooling load), due to sensor inaccuracies in either the exhaust-air stream or the outdoor-air stream (or both). Heat recovery for pre-heating is generally not needed until the OAT is less than 45–50°F.

When the OAT is greater than 45–50°F and less than 75°F, the DOAS heat-recovery system should be off (no heat recovery occurring), while the cooling coil is enabled to provide mechanical cooling (based upon the DOAS DAT setpoint). Because most DOAS fans and motors (depending upon the fan and motor size) will contribute 2–6°F of additional heat, it is assumed that with no active heat recovery during outdoor-air conditions that are as low as 45°F, the resultant DOAS DAT could be as low as 47°F and as high as 51°F (with the DOAS heat-recovery system disabled). If the OAT is as low as 50°F, the resultant DOAS DAT could be as low as 52°F and as high as 56°F (with the DOAS heat-recovery system disabled).

A typical heat-recovery control sequence for the DOAS would lock out the heat-recovery system between 50°F (heating lockout) and 75°F (cooling lockout)—or as configured—to ensure that energy is not inadvertently recovered from the exhaust air stream, only to be removed by the cooling coil in the DOAS or vice versa (energy is not inadvertently added to the discharge air-stream by the heating coil, only to be transferred by the heat-recovery system into the exhaust-air stream).

Once the OAT drops below the heating lockout or rises above the cooling lockout, the heat-recovery system should be activated and controlled (modulated via coil hydronic flow, wheel speed, or heat pipe airflow via a face-and-bypass damper or refrigerant flow via a control valve, depending upon the heat-recovery design) to recover as much energy as possible, without exceeding the DOAS DAT setpoint and possibly activing mechanical cooling. The **blue** and **red** vertical dashed lines shown in Figure 8.2 typically indicate when the heat-recovery system would be activated for heating (**red line**) or for cooling (**blue line**) based on typical OAT lockout setpoint values. When humidity control is required, the cooling lockout (**blue line** in Figure 8.2) would be closer to 60°F.

When the DOAS DAT setpoints are held constant (too high or too low), this may result in increased energy consumption in the building (at the DOAS, at the AHU, or at the zone HVAC systems).

When the DOAS DAT is configured to only deliver air to downstream zone devices and the DOAS DAT is too cold (less than 55°F) during the cooling season (OAT greater than 75–80°F), this may result in increased reheat energy at the zone devices and requires mechanical cooling at the DOAS that is greater than what would be required if the DAT setpoint were allowed to remain warmer than typical DAT setpoints during warmer outdoor conditions. This may be especially true if the building is designed with zone cooling capabilities (FCUs or chilled beams, etc.).

DOAS DATs that are less than 55°F can be acceptable, especially when air is delivered to downstream AHUs that have the ability to mix cooler DOAS air with warm building return air. This may help reduce zone cooling at the downstream AHUs, while de-humidifying air at the DOAS. If humidity issues linger in the zones, then AHU cooling may still be required.

When the DOAS DAT is too warm during the heating season (OAT conditions <50–55°F) and the DOAS DATs are greater than 65°F, this may result in increased heating energy at the DOAS and may require mechanical cooling at the downstream AHUs that is greater than what would be required if the DOAS DAT setpoint were allowed to remain lower than typical DAT setpoints during cooler outdoor conditions.

Optimal conditions for automatically changing the DAT setpoint can be detected through the BAS in near real time. Failure to identify and implement DAT reset may lead to increased heating and cooling energy consumption. The DAT reset automatically changes the DAT setpoint in response to the OAT. The automatic reset for the DOAS DAT should result in energy savings (versus a fixed DAT setpoint) because the DOAS is designed to maintain minimum outdoor air ventilation rates at temperatures that are significantly less energy-intensive compared to other outdoor air ventilation system designs that are not provided with the ability to temper the air using energy-recovery methods (heat wheel, heat-recovery coil, or heat pipe).

8.2 What DOASs Should Be Considered for Discharge-Air Temperature Reset Implementation?

All DOASs should be evaluated for correctly implemented DAT reset as well as subsystem sequencing (heat recovery, heating coil, cooling coil, etc.). All DOASs that are equipped with reliable controls are good candidates for implementing DOAS DAT reset, under the following assumptions:

- The BAS and field controllers are reliable.
- The DOAS and its mechanical components are reliable and *fully integrated* and communicating reliably to the BAS network without any apparent problems.
- Sensors are reliable, calibrated, and functioning.

8.2.1 Methodology for Implementing a DOAS Discharge Temperature Reset

This section of the implementation guide provides common strategies for implementing a DOAS DAT reset. DOAS DAT reset is dependent upon the specific DOAS and downstream loads being served. This section provides some context for what an economical DOAS DAT reset implementation might look like. All final reset parameters should be thoroughly discussed with the owner/operator of the BAS and the building(s) where the reset strategy will be applied. The implementation strategy may vary by building type, system type, or other variables that this guide does not directly highlight.

8.2.1.1 BAS Data Needed to Verify DOAS Discharge-Air Temperature Reset

Analyzing and detecting DOAS DAT control problems and opportunities can be achieved by using trend capabilities derived from the BAS. The following parameters should be monitored using the trend capabilities of the BAS:

- DOAS discharge (duct) DAT
- DOAS discharge (duct) DAT setpoint
- OAT.

Data should be collected at recommended intervals of between 5 and 30 minutes for a minimum of 1–2 weeks. When analyzing the DOAS DAT reset, the data trends to look for include the following:

- Is there an existing reset for the DOAS DAT?
- Is the DOAS DAT setpoint too high or too low?

8.2.1.2 Identifying and Correcting Problems that May Interfere with Functional Discharge-Air Temperature Reset

This section of the guide covers some common DOAS subsystems (sequencing) and DOAS DAT issues, things to look for to help identify these issues (through use of the BAS interface and trend data), and a list of possible problems that would cause the issue.

1. Look for failed or improperly located DAT and OAT sensors.
   - With DOAS designs, there should be leaving sensors after the heating coil, heat-recovery section, and the cooling coil, as well as the exhaust coil, to enable proper control and diagnostics.
   - DAT sensors that are always reading significantly lower or higher than the DAT setpoint, may indicate failed component(s), such as:
     - DAT sensor is wired incorrectly or has a loose wire.
     - The sensor has failed.
     - The sensor location is not correct (located before the cooling coil or further downstream to not accurately detect the leaving coil temperature.
2. Identify DOAS heating coil control valves that are not modulating or responding to control signals.
   - DOAS heating coil control valve commands that are always reading 100% open with very low DOAS heating coil leaving-air temperature readings may indicate failed component(s), such as:
     - DOAS heating coil control valve has failed closed (actuator motor failure, loose wire, power failure, or mechanical failure of the heating coil control valve – motor linkage).
     - Balancing valve or manual isolation valve for the heating coil is closed.
     - Controller/electronics have failed.
   - DOAS heating coil control valve commands that are always reading 0% open with very high DOAS heating coil leaving-air temperature readings may indicate failed component(s), such as:
     - DOAS heating coil control valve has failed open (actuator motor failure, loose wire, power failure, or mechanical failure of the heating coil control valve – motor linkage).
3. Identify DOAS cooling coil control valves that are not modulating or responding to control signals.
   - DOAS cooling coil control valve commands that are always reading 100% open with warm DOAS DAT readings may indicate failed component(s), such as:
     - DOAS cooling coil control valve has failed closed (actuator motor failure, loose wire, power failure, or mechanical failure of the cooling coil control valve – motor linkage).
     - Balancing valve or manual isolation valve for the cooling coil is closed.
     - Controller/electronics have failed.
   - DOAS cooling coil control valve commands that are always reading 0% open with very low DOAS DAT readings may indicate failed component(s), such as:
     - DOAS cooling coil control valve has failed open (actuator motor failure, loose wire, power failure, or mechanical failure of the cooling coil control valve – motor linkage).
4. Identify DOAS heat-recovery systems that are not modulating or responding to control signals.

- DOAS heat-recovery system commands that are always reading 100% capacity with DOAS heat-recovery system leaving-air temperature readings that are very close to the OATs during cool-cold weather may indicate failed component(s), such as those listed for the systems below.
  - DOAS heat-recovery system includes a coil and run-around pump design:
    - DOAS heat recovery coil control valve has failed closed (actuator motor failure, loose wire, power failure, or mechanical failure of the cooling coil control valve – motor linkage).
    - Balancing valve or manual isolation valve for the heat-recovery coil is closed.
    - Heat-recovery coil is dirty or too cold (minimizing heat transfer or accumulating frost on the exhaust coil from exhaust air moisture).
    - Failed chilled water coil control valve (failed open)
  - DOAS heat-recovery system includes a heat wheel:
    - DOAS heat wheel VFD or motor has failed off (motor or VFD failure, power failure, or mechanical failure of the heat wheel).
    - DOAS heat wheel is belt-driven by the VFD/motor and the belt is slipping or is broken
    - Heat wheel surface is dirty or too cold (minimizing heat transfer or accumulating frost from the exhaust air moisture).
    - Wheel bypass is stuck open.
    - Failed chilled water coil control valve (failed open)
- DOAS heat-recovery system commands that are always reading 100% capacity with DOAS heat-recovery system leaving-air temperature readings that are very close to the OATs during warm-hot weather may indicate failed component(s), such as those listed for the systems below.
  - DOAS heat-recovery system includes a coil and run-around pump design:
    - DOAS heat-recovery coil control valve has failed closed (actuator motor failure, loose wire, power failure, or mechanical failure of the cooling coil control valve – motor linkage).
    - Balancing valve or manual isolation valve for the heat-recovery coil is closed.
    - Heat-recovery coil is dirty (minimizing heat transfer).
    - Failed hot water coil control valve (failed open)
  - DOAS heat-recovery system includes a heat wheel:
    - DOAS heat wheel VFD or motor has failed off (motor or VFD failure, power failure, or mechanical failure of the heat wheel)
    - DOAS heat wheel is belt-driven by the VFD/motor and the belt is slipping or is broken
    - Heat wheel surface is dirty (minimizing heat transfer).
    - Failed hot water coil control valve (failed open)

9.0 Air-Side Economizer Control

This air-handling unit (AHU) air-side economizer control implementation guide is intended to show, through examples of good and bad operations, how the air-side economizer can be efficiently controlled and what the indicators are for good and bad operations. This guide is also intended to assist in identifying AHUs that are good candidates for improved energy-efficiency via air-side economizer control, and offer some common implementation strategies and guidelines.

9.1 Why Consider Implementing Air-Side Economizer Control Improvements?

AHU heating and cooling energy can account for a significant percentage of a building's total energy consumption. If the AHU serves a large area, is sized for more than 10,000 cfm, or operates 10 or more hours each day, the likelihood of energy-saving improvement opportunities related to the air-side economizer control function of the AHU is very high. AHU supply fans are often equipped with heating and/or cooling coils for tempering of the outdoor air as well as the mixed air (which is a mixture of return air and outdoor air). Heating and cooling coils are designed and provided to temper the air streams. Newer AHUs may come with heat-recovery coils or heat-recovery wheels designed to temper the outdoor air used for ventilation. In dry climates, AHUs may be configured to have humidification capabilities. All of these subsystem components (outdoor-air tempering, heating, cooling, dehumidification, and humidification) challenge most building operation teams because the control complexity increases accordingly. Optimizing the control of the AHU air-side economizer translates into energy and cost savings without sacrificing comfort in the spaces served by the AHU(s).

This air-side economizer control guide shows, through use of examples of good and bad operation, how air-side economizers should be used and controlled.

Incorrect control of an air-side economizer may go unnoticed because mechanical heating or cooling will compensate to maintain the discharge-air temperature (DAT) at the desired DAT setpoint. This may include periods of time when too much outdoor air is being introduced to the AHU (when the economizer control is attempting to maintain a minimum outdoor-air setpoint) or when not enough outdoor air is being introduced to the AHU (when the economizer control is attempting to bring in the maximum amount of outdoor air). Failure to correct/mitigate this situation, in all likelihood will lead to increased energy consumption.

An air-side economizer is a duct/damper arrangement in an AHU and has automatic controls that allow an AHU to use outdoor air to reduce or eliminate the need for mechanical cooling. When cooling is needed and outdoor-air conditions are favorable for cooling (the outdoor-air temperature [OAT] is less than the return-air temperature [RAT] or the OAT has less total energy than the RAT), unconditioned outdoor air can be used to meet all of the cooling energy needs or supplement mechanical cooling. In a properly configured economizer control sequence, the outdoor, return, and exhaust dampers sequence together to mix and balance the airflow streams to meet the AHU DAT setpoint.

Air-side economizers (Figure 9.1) are designed using ducts, dampers, sensors, and controls to deliver the most economical mixture of outdoor air and return air (via mixing dampers) to optimize the total energy consumption required by the AHU to either heat or cool the mixed-air stream before the air is delivered to the downstream building spaces.

Air-side economizer definition
"A duct-and-damper arrangement as well as an automatic control system that allow a cooling system to supply outdoor air to reduce or eliminate the need for mechanical cooling during mild or cold weather." (*ASHRAE Standard 90.1-2004*).

When air-side economizer dampers are improperly controlled and introduce more outdoor air than is required during hot, humid, or cold weather, this will often result in increased energy consumption in the building (at the AHU or downstream reheat coils) as more energy from the chilled-water plant or hot water plant will be required to temper the excess outdoor air. When the outdoor-air conditions are optimal for the air-side economizer to introduce 100% outdoor air (or a mixture of outdoor air and return air that exceeds the minimum ventilation requirements), but the economizer is only delivering the minimum required outdoor air (for ventilation requirements), mechanical cooling energy can be expended at a higher rate than if the air-side economizer system were working optimally.

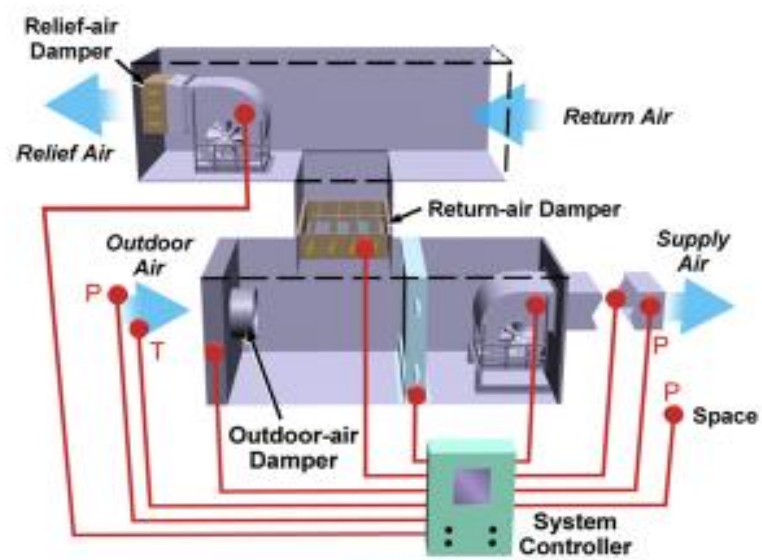

Figure 9.1. Typical Air-Side Economizer Damper and Control Arrangement

The AHU may have a relief fan or return fan as shown in Figure 9.1 or it may not have a relief or return fan. A return fan would be upstream of both the return-air damper and exhaust/relief air damper. The control sequence for the return/relief fan (or no fan) configurations should be evaluated.

Normally, automatic controls will operate the economizer cycle. The relief damper and relief fan can be problematic because they can provide another path for outdoor air to enter the mixing plenum if the relief dampers fail to tightly close.

Figure 9.2 shows a typical control sequence statement from the 1990s. Current economizer control sequences are more complicated/complex, because they often include requirements for carbon dioxide (CO_2) control for demand control ventilation (DCV), minimum ventilation air control (often measured with airflow stations), and may include additional requirements for duct or building pressurization control.

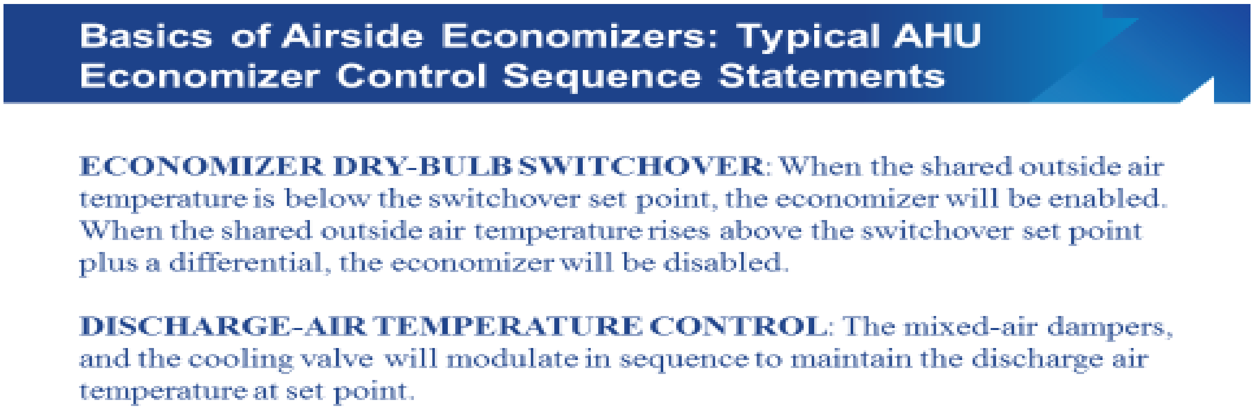

Figure 9.2. Typical Air-Side Economizer Control Sequence – 1990s

As long as the outdoor-air stream has a lower energy content (either the measured or the calculated enthalpy or measured dry-bulb temperature) than the return-air stream, the outdoor-air stream should be used even if mechanical cooling is required. In humid climates, when enthalpy-based economizers are not used, the economizer should be active when the OAT is 5°F to 10°F below the RAT.

There are times when economizing should not be used. This includes during building warm-up periods and cool-down periods when the outdoor conditions are not favorable for economizing, or during unoccupied periods when the supply fan is operating unless introduction of outdoor-air is advantageous to the unoccupied (night setback) or cool-down periods.

Other ventilation requirements may be required for air-side economizer control integration, adding complexity to traditional air-side economizer controls. This can include DCV controls that use CO_2 sensors, to ensure that air quality (based on return-air sensing, zone sensing or other engineered parameters) is maintained and it may include building pressurization controls that are integrated to the air-side economizer controls where additional outdoor air is introduced to help maintain the building pressure requirements.

Optimal conditions for controlling the air-side economizers can be detected through the building automation system (BAS) in near real time. Failure to identify poorly controlled air-side economizers and implement improvements in all likelihood will lead to increased heating and cooling energy consumption.

9.2 What AHUs Should Be Considered for Air-Side Economizer Improvements?

Any AHU that is designed with a standard air-side economizer design (outdoor-air, return-air and relief/exhaust-air dampers that are sized for approximately similar area and have properly installed damper actuators/linkages and include good damper blade and damper seal construction) should be considered for air-side economizer improvements. The air-side economizer should have controls that are integrated into the existing digital controller that controls the AHU. The actuator(s) for the economizer dampers should be electrically actuated, but other actuators (pneumatic) will allow for optimization, as long as the instrument air system is properly maintained and the pneumatic actuators are in good condition and properly calibrated. Electric and pneumatic actuators should be verified for proper calibration (i.e., they correctly stroke the dampers in sequence with each other).

Damper assemblies are mechanical structures that include dampers, damper blade seals, damper linkages and the required minimum number of actuators that are properly sized to provide the required power (torque) to stroke the dampers as they rotate from fully open to fully closed and back again. Besides the frictional losses incurred from blades, seals, and linkages, the actuators also have to overcome the air pressure developed by both the supply and return fans while they stroke from fully closed to fully open positions.

All AHUs that are equipped with reliable dampers, damper actuators, and controls are good candidates for implementing air-side economizer control improvements, under the following assumptions:

- The BAS and field controllers are reliable (communication between AHU field controllers and the supervisory controllers is more than 95% reliable).
- AHU controllers and connected economizer actuators are digital controls-based (pneumatic signals and pneumatic actuators are not used at the economizer dampers; if they are, they must be in well-maintained condition).
- AHU economizer dampers are in well-maintained condition (damper blade seals are in good condition, damper linkages are configured correctly, and dampers are not broken).
- Temperature sensors used for air-side economizer control are calibrated and properly located.

Note: Temperature sensors for air-side economizer control should (in most situations) include averaging sensors that are either 8 feet or up to 20 feet in length and installed to sense the mixing plenum's cross-sectional area. Where mixing plenums are too large, the control design should consider installing at least 2 sensors to measure different sections of the mixing plenum and average the values in the control code. This also allows the use of the lower temperature value for cold-weather freeze mitigation response.

Most AHUs and control systems allow for embedding the calculated outdoor-air fraction (OAF) of the air-side economizer. The OAF values can be used to determine air-side economizer performance by comparing the OAF value to the actual damper command. In most cases, there will be significant difference between the OAF and the outdoor-air damper (OAD) command, especially at low OAD values. Therefore, using OAD values can be misleading because it does not reflect the true outdoor-air intake.

The calculated OAF should only be used to investigate the true fraction of outdoor-air entering the AHU when the OAT is significantly (± 5ºF) different than the RAT. As these two temperatures get closer to each other, the OAF calculation may not be accurate. Reviewing the plot of outdoor-, return-, mixed-, and discharge-air temperatures versus time can help determine the right time to compare the calculated OAF to the OAD position.

If the calculated OAF deviates from the actual OAD position by more than 30%, it may be an opportunity to investigate the economizer system for improvements.

If the calculated OAF is not automatically calculated in the BAS, the calculated OAF should be added and tracked for all AHUs where the existing temperature sensors already exist (outdoor-air, return-air and mixed-air temperature sensors).

9.3 Methodology for Implementing Air-Side Economizer Control Improvements

This section of the implementation guide highlights some common strategies for implementing air-side economizer control improvements. This list is not intended to cover every implementation technique. It is intended to present a few techniques in detail to provide some context for what the different air-side economizer control improvements might look like, compared to less desirable control sequences. All final control sequence and control parameters should be thoroughly discussed with the owner/operator of the BAS and the building(s) in which the control improvements will be applied. The implementation strategy may vary by building type, system type, or other variables that this guide does not directly highlight.

According to American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Guideline 90.1, the following economizer configurations apply (see Figure 9.3):

- Approach 1: Fixed Outdoor-Air Dry-Bulb Temperature High Limit Setpoint
  - Fixed dry-bulb temperature high limit setpoint for climate zones 1B, 2B, 3B, 3C, 4B, 4C, 5B, 5C, 6B, 7, and 8: OAT >75°F (24°C)
  - Fixed dry-bulb temperature high limit setpoint for climate zones 5A and 6A: OAT >70°F (21°C)
  - Fixed dry-bulb temperature high limit setpoint for climate zones 1A, 2A, 3A, and 4A: OAT >65°F (18°C)
- Approach 2: Differential Dry-Bulb Temperature
  - Differential dry-bulb temperature for climate zones 1B, 2B, 3B, 3C, 4B, 4C, 5A, 5B, 5C, 6A, 6B, 7and 8: OAT > RAT – climate zones 1A, 2A, 3A and 4A exempted
- Approach 3: Fixed Enthalpy High Limit Setpoint with Fixed Dry-Bulb Temperature High Limit Setpoint
  - Fixed enthalpy with fixed dry-bulb temperature for all climate zones: enthalpy outdoor-air (hOA) >28 Btu/lb. (47 kJ/kg) or OAT >75°F (24°C)
- Approach 4: Differential Enthalpy with Fixed Dry-Bulb Temperature High Limit Setpoint
  - Differential enthalpy with fixed dry-bulb temperature for all climate zones: hOA >enthalpy return-air (hRA) or OAT >75°F (24°C)

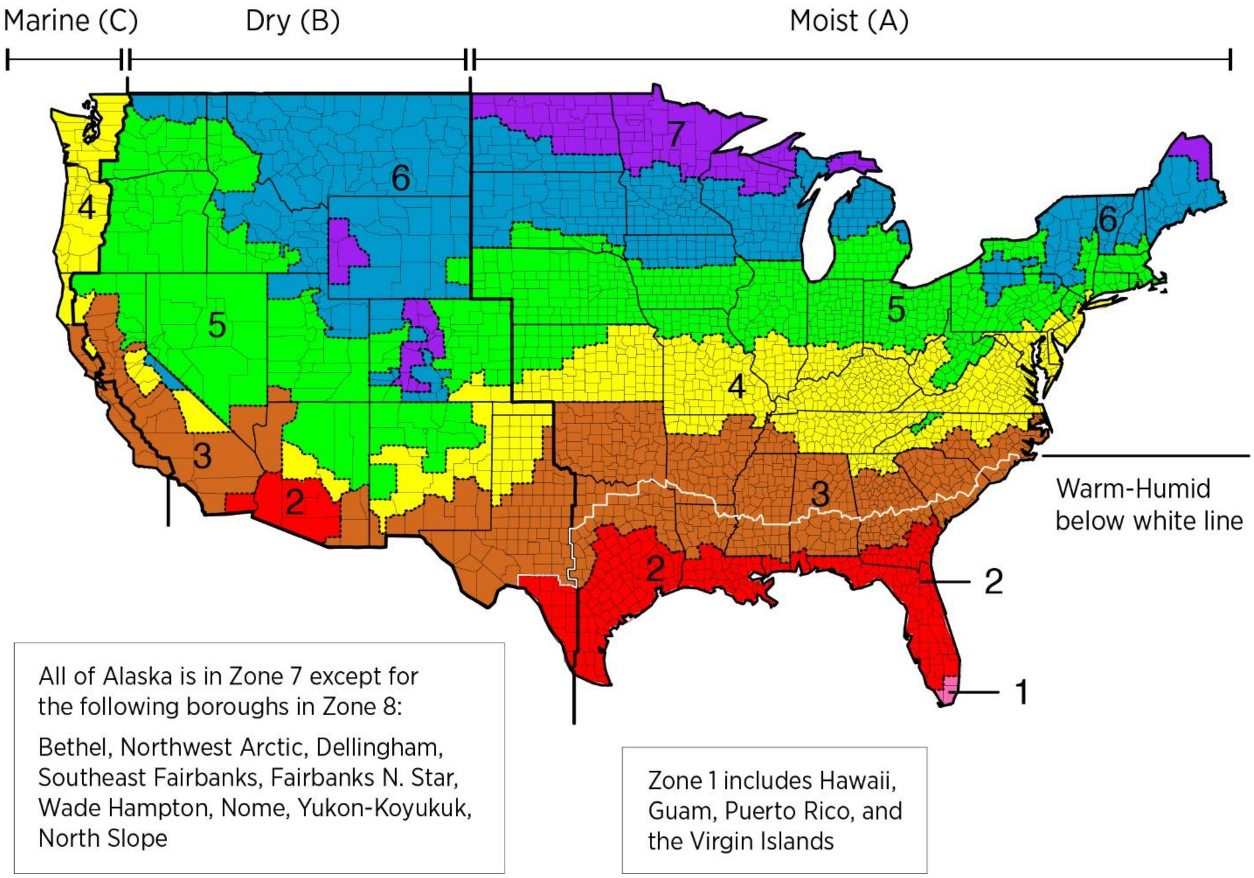

Figure 9.3. ASHRAE Climate Zones

Within these four different approaches, the control sequence should also include a reasonable differential (e.g., 2°F) value that would require the measured variable(s) to rise above (or fall below) the limits noted prior to enabling or disabling the economizer function.

Accuracy, calibration of sensors, energy savings, and long-term operations and maintenance (O&M) costs are all critical when deciding which approach (1–4) to use for economizer control.

- Approach 1 (Fixed Outdoor-Air Dry-Bulb Temperature High Limit Setpoint) only requires the use of an OAT sensor. However, it is critical that the OAT sensor be properly located for reliable and accurate temperature sensing, and the high limit value should be properly configured for the geographical location (see Figure 9.3).
- Approach 2 (Differential Dry-Bulb Temperature) requires the use of an OAT sensor and a RAT sensor (that are continuously compared to each other). As already stated, the OAT sensor accuracy is imperative. The RAT sensor may experience false readings if the return-air plenum is open (not ducted) and passes through an attic or similar space where warmer or colder than interior conditions may exist. This can falsely influence the RAT sensor.
- Approach 3 (Fixed Enthalpy High Limit Setpoint with Fixed Dry-Bulb Temperature High Limit Setpoint) requires the use of two sensors—an OAT sensor and an outdoor-air humidity (OAH) sensor—to enable the accurate calculation of outdoor-air enthalpy. Humidity sensors require periodic (at least once-a-year) calibration to ensure their accuracy but are often neglected by O&M staff. And, the OAT (dry bulb) sensor may be poorly located (as already noted). Both conditions have the potential to result in enthalpy calculation errors. Comparisons of enthalpy, dry-bulb, and humidity

conditions with local weather station data should be considered for accuracy comparisons, if no other options exist.

- Approach 4 (Differential Enthalpy with Fixed Dry-Bulb Temperature High Limit Setpoint) requires the use of four sensors—an OAT sensor, an OAH sensor (to enable the accurate calculation of outdoor-air enthalpy), a RAT sensor, and a return-air humidity sensor (to enable the accurate calculation of return-air enthalpy). As already noted, location and calibration validation of sensors for accuracy is rarely executed for one to two sensors, and this is even less likely for the four sensors required for this approach. Therefore, this approach is deemed to be the least favorable economizer control strategy and also the most costly to implement (due to the number of sensors and the ongoing O&M costs associated with it).

9.4 Suggested Actions

This section of the guide covers some common AHU air-side economizer control issues, things to look for to help identify these issues (through use of the BAS interface and trend data), and a list of possible problems that would cause the issue.

9.4.1 BAS Data Needed to Verify Air-Side Economizer Control Improvement Opportunities

Analyzing and detecting AHU air-side economizer control problems and opportunities can be achieved by using the BAS trend capabilities. The following parameters should be monitored using the trend capabilities of the BAS:

- OAT – either the temperature sensor used globally or locally (choose the most accurate sensor), RAT and mixed-air temperature (MAT)
- OAH (and calculated outdoor-air enthalpy)
- Return-air humidity (and calculated return-air enthalpy)
- OAD signal
- OAF
- In most BASs, the OAF is probably not computed and trended. If the OAF is not recorded in the BAS, it can be computed externally using the outdoor-, return-, and mixed-air temperatures: OAF = abs[(MAT – RAT)/(OAT – RAT)] × 100
- Cooling-coil-valve (CCV) signal.

Data should be collected at recommended intervals of between 5 and 30 minutes for a minimum of 1–2 weeks. When analyzing the discharge-air temperature reset, the trend data to look for should include the following:

- How close is the OAF compared to the OAD signal?
  - They should be within 25% of each other (consistently)
- Is the minimum OAD position reasonable (between 10% and 20%)?
  - Minimum OAD positions that are more than 20% may indicate other control sequences (demand control ventilation [DCV], which may use CO_2 feedback for control).

- Some control sequences use airflow stations and/or building static pressure sensors to determine minimum outdoor airflow rates as well as proper building pressurization when supply and return fans are variable frequency drive (VFD)-driven. Ensure that flow setpoints are reasonable and that building pressurization is slightly positive. Validate the building pressurization readings (when provided) by checking main floor entry points (exterior doors, lobby entry points, etc.).

- Is the OAD open when outdoor conditions are not favorable (OAT > RAT)?
  - If the OAT versus RAT economizer control is being used (Approach 2: Differential Dry-Bulb Temperature), the OAD should be at the minimum position (unless other control schemes are being used – such as DCV). If DCV is being used, ensure that the DCV setpoint and sensors are configured and calibrated appropriately. CO_2 setpoints should be >1,000 ppm and CO_2 sensor values should be <1,000 ppm.
- Is the OAD closed or at the minimum position when outdoor conditions are favorable for economizing and the AHU is in cooling mode?
  - Some economizer control schemes are automatically disabled when mechanical cooling is active. This is no longer considered an optimal control strategy by ASHRAE or other accepted industry best practices. Fix or update the economizer control code (if this is observed).
- Does the cooling coil operate when the OAT is lower than the DAT setpoint?
  - This may indicate a leaking hot water coil in the AHU as the cooling coil opens in response to the added heat from the hot water coil's leak to maintain the DAT setpoint.
  - This may indicate economizer controls that are not working correctly (dampers are not opening up fully to bring in the cooler OAT).
  - This may indicate improperly configured cooling coil lockout setpoints or no lockout setpoints (due to leaking hot water coils).
- When the cooling coil is actively cooling, if the conditions are favorable for economizing, is the OAD fully open?
  - Some economizer control schemes are automatically disabled when mechanical cooling is active. This is no longer considered an optimal control strategy by ASHRAE or other accepted industry best practices. Fix the economizer control code to open the OAD 100% before the cooling coil is allowed to operate, and maintain the OAD at 100% open while the cooling coil operates to meet the DAT setpoint.
- Do OADs restrict their command values (or fully close) for freeze protection (when the MAT drops below the mixed-air low limit setpoint)?
  - What is the mixed-air low limit temperature setpoint value? It should not be any lower than 45°F.
  - It should not be any higher than 55°F (or economizer performance will be negatively impacted)
- Is the MAT value between the OAT and the RAT values?
  - This indicates temperature sensors are reading reasonable values because it is physically impossible for the MAT sensor to be lower (or higher) than the two air streams.

    Note: If the AHU is served outdoor-air from a dedicated outdoor-air system (DOAS), and the DOAS is able to heat or cool, then the MAT sensor could read lower or higher due to the DOAS effect.
- When conditions are not favorable for economizing, is the MAT closer to the RAT or to the OAT?

– If the MAT is closer to the OAT, this often indicates that the outdoor dampers are mostly open when they should be closer to the minimum OAD position.

- When conditions are favorable for economizing, is the MAT closer to the OAT or to the RAT?
  – If the MAT is closer to the RAT, this often indicates that the OAD is mostly closed when it should be closer to the fully open position to take advantage of economizer conditions.

9.5 Is the Minimum Outdoor-Air Damper Position Reasonable (between 10% and 20%)?

To meet ventilation requirements, the AHU must provide a certain amount of fresh air when the building is occupied. The ventilation requirements are determined at the design stage based on zone occupancy and other parameters. The ventilation requirements are then translated to a minimum damper position to meet ventilation requirements. Typically, the minimum damper position is between 10% and 20%. It is easy to check the minimum damper position signal by reviewing the plot of OAT and OAD position signal vs. time. Figure 9.4 below shows these trends for a 2-day period in which the OAT varies between 65°F and 80°F. While this is listed as an example of too high of a minimum OAD position, airflow requirements through the ductwork must be evaluated to ensure the OAD position for ventilation is appropriate. In addition, adding the OAF to the graph will provide additional insights.

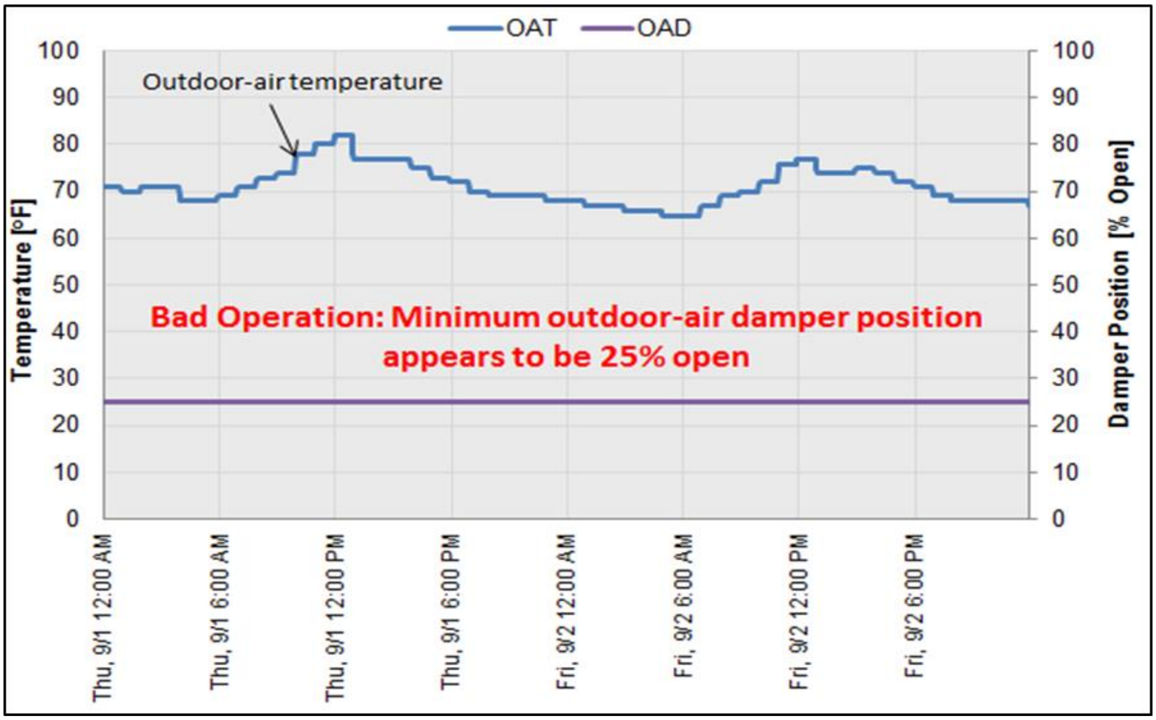

Figure 9.4. Minimum Outdoor-Air Damper Signal Is 25% Open

10.0 Chilled Water Temperature Reset

The purpose of this chilled-water temperature reset implementation guide is to show, through examples of good and bad operations, how chilled-water temperature resets can be efficiently controlled and what the indicators are for both good and bad operations. This guide offers some common implementation strategies.

10.1 Why Consider Implementing a Chilled-Water Temperature Reset?

Chilled-water temperature resets are easy to implement, track, and administer. The goal of chilled-water temperature reset is to optimize the chilled-water temperature that the cooling system is trying to maintain. Whether the cooling system is an older chilled-water system configured with primary and secondary pumping loops or a newer all- variable frequency drive (VFD) chilled-water cooling system with VFD-driven pumps, VFD-driven tower fans, and VFD-driven chiller compressors, the principles are the same: generate chilled water that is just cool enough to satisfy the majority of the cooling loads seen by the various heating, ventilating and air-conditioning (HVAC) systems.

The initial chilled-water supply temperature setpoint is theoretically calculated to satisfy the peak cooling loads. Typical systems will only see their peak (or design) cooling loads for a small fraction of the year; they spend the majority of time operating at part load (or less than design) conditions. During part load operation, design chilled-water supply temperatures are not needed to meet the loads of the chilled-water end uses. Therefore, chilled-water supply temperature setpoints can be increased during part load operation. Because the chilled-water temperature will affect the chilled-water flow needed, an effective chilled-water temperature reset sequence should optimize the combined operation and relative energy consumption of the chillers, chilled-water pumps, and cooling tower fan loads.

When water-cooled chillers are designed to take advantage of ambient wet-bulb temperature conditions that enable lower condenser water temperatures, even greater energy efficiency can occur. Condenser water temperature reset is discussed in another guide (Section 5.0), but is mentioned here because all chiller compressors perform work based on the “lift” that the compressor sees. Lift is defined as the difference between condenser refrigerant pressure and evaporator refrigerant pressure. Using defined pressure-temperature relationships, lift can also be measured with the leaving chilled (evaporator) water temperature and the leaving condenser water temperature. The greater the difference, the greater the lift, which translates into greater work required by the compressor.

The chilled-water temperature setpoint can have greater impacts when the setpoint is warmer than historically configured, so when setpoints are cooler than required, energy efficiency may be negatively affected. **Note**: If chilled water temperatures are too warm, this may negatively impact AHU fan energy consumption.

Because chillers are often sized for design load conditions (hot weather), it is not uncommon to find chillers that are short-cycling during moderate weather. Short-cycling is often a result of the chiller’s inability to “turn down.” When a chiller is operating during very low loads, the chiller will quickly overshoot the internal setpoint and cycle off.

Optimized sequencing of multiple chillers, pumps, and tower fans is the desired outcome for most chilled-water plants. However, this guide focuses on the optimum value to which the chilled-water temperature setpoint should be reset. An optimized chilled-water temperature setpoint can help minimize chiller short-cycling and it can help optimize the chiller’s energy consumption.

If the chiller is configured with VFD-driven compressors, the ability to operate the chiller while minimizing short-cycling is greatly enhanced. When chiller loads are low, this may be noted by low ΔT values (the difference between the chilled-water supply and chilled-water return temperatures determines the ΔT value).

In many cases, the cooling loads served by a chiller are for comfort-cooling only. This means that once the outdoor air temperature (OAT) is less than 55–60°F, the chilled-water system should be turned off (automatically) in the control sequences and may include the use of an OAT cooling lockout function to perform the chiller system shutdown or chiller system re-activation. If cooling is needed in the building when OAT values are less than 55°F, the expectation would be that air-side economizers would provide this cooling. If process loads are served from the same chilled-water system, or if air-side economizers do not exist (or were under-sized), then the chilled-water system may be required to operate continuously.

The other advantage of properly configured chilled-water temperature setpoints is avoiding a constant setting that results in chilled water that is excessively low in temperature. Most building HVAC systems will have a few chilled-water cooling control valves that may leak. This may occur at an air-handling unit (AHU), rooftop unit (RTU) or fan-coil unit (FCU). Another ancillary benefit is that resetting the chilled-water temperature setpoint above the design conditions will reduce the heat transferred to the air-stream per unit volume of chilled-water. This can reduce unwanted cooling (and potential reheating) if a cooling control valve leaks, and can only occur if the chilled-water temperature is intelligently reset to match the setpoint to the true load(s).

A chilled-water system is typically designed to provide chilled water to help maintain maximum temperature (and possibly humidity) levels in a building. Historically, maximum temperature requirements have been associated with hot OAT operations. However, when warm-weather mechanical cooling is operated to de-humidify a building via AHUs that over-cool the discharge air (to ensure condensation and removal of moisture is occurring), hot water for zone reheat capability may be needed. The temperature of the chilled water can be based upon the OAT, or it can be based on a more intelligent zone feedback algorithm that calculates actual cooling requirements to determine the optimal chilled-water temperature setpoint.

The goal for any building that has properly implemented chilled-water temperature reset strategies includes energy efficiency, which is often measured as kilowatts per ton (kW/ton). Historically, air-cooled chillers have often generated low kW/ton values during cool OAT conditions (OAT <65–70°F), because air-cooled chillers do not rely upon condenser water and cooling tower systems (additional pump and fan energy). As the OAT increases, the air-cooled chillers are forced to activate more condenser fans and the compressors have to work harder because the ability to remove heat for an air-cooled chiller becomes more difficult as it gets warmer outside (compared to a water-cooled chiller plant that takes advantage of evaporative cooling). Water-cooled chillers rely upon condenser water pumping systems and cooling towers to aid in heat rejection, and this adds energy to the overall cooling process. Water-cooled chillers, with VFD-driven cooling tower fans, VFD-driven compressors, and VFD-driven pumps (chilled water) are often able to achieve lower total kW/ton values than air-cooled chillers, during warm to hot weather, when these systems are properly controlled and sequenced. However, water-cooled chillers may require greater levels of maintenance due to water-treatment issues, scaling, and additional pumps used for the condenser water/cooling tower systems.

All chillers that are equipped with reliable controls and proper maintenance are good candidates for implementing chilled-water temperature resets, under the following assumptions:

- If the chiller is constant speed, it is assumed that it is configured in a primary-secondary distribution loop with properly configured secondary loop pump controls.

- The building automation system (BAS) and chiller controllers are reliable (communication between chillers and the supervisory controllers is more than 95% reliable and any integration issues are totally resolved).

10.2 What Systems Should Be Considered for Chilled-Water Temperature Reset Implementation?

Some of the typical HVAC systems that may be found in various buildings that should be considered for chilled-water temperature reset include the following:

- water-cooled chillers
- air-cooled chillers.

10.2.1 Methodology for Chilled-Water Temperature Reset

This section of the implementation guide goes through some common strategies for implementing chilled-water temperature reset.

- OAT reset
- average cooling coil valve position reset.

Note that this list is not intended to cover every implementation technique, but to present a few techniques in detail to provide some context for what a good chilled-water temperature reset looks like. All final chilled-water temperature reset parameters should be thoroughly discussed with the owner/operator of the BAS and the building(s). The implementation strategy may vary by building type, system type, or other variables that this guide does not directly highlight.

10.2.1.1 Chilled-Water Temperature Reset based on the Outdoor-Air Temperature

The most common chilled-water temperature reset strategy is the use of OAT to reset the setpoint. This strategy dates back to the days of pneumatic controls and is still the primary reset technique used today. With the advent of the BAS, more intelligent control sequences are available such as a reset based on chilled water return temperature, but are often not used due to the robustness of the OAT reset, and the added complexity of further techniques without additional benefit. Figure 10.1 shows two examples of a temperature reset using OAT.

The **red line** in Figure 10.1 shows the chilled-water temperature reset that is between 52°F and 44°F, as the OAT rises from 70–90°F. The **black line** in Figure 10.1 shows the chilled-water temperature reset that is between 48°F and 40°F, as the OAT rises from 70–90°F.

Depending upon the building age, design, envelope (windows, doors, wall and ceiling insulation) and HVAC operations, the chilled-water temperature that is required for cooling the building may need to be as low as 40–42°F, during hot weather conditions (also known as “design-day” conditions). What are the design-day conditions for your building? This should be on the mechanical design prints or similar documentation, which should include the OAT value for the design day and the minimum chilled-water temperature required.

Many times, O&M staff will lower the minimum chilled-water temperature (low limit) to a value that is lower than the original design value. Sometimes this occurs because the building mission or operation has changed. This may include the introduction of more outdoor air into the building or it may include extreme weather events that result in O&M staff lowering the chilled-water temperature low limit from the original design parameters (or the OAT temperature at which the low limit would be reached). In both cases, this can often result in lower-than-needed chilled-water temperatures, which may result in greater energy consumption to generate the lower chilled-water temperatures.

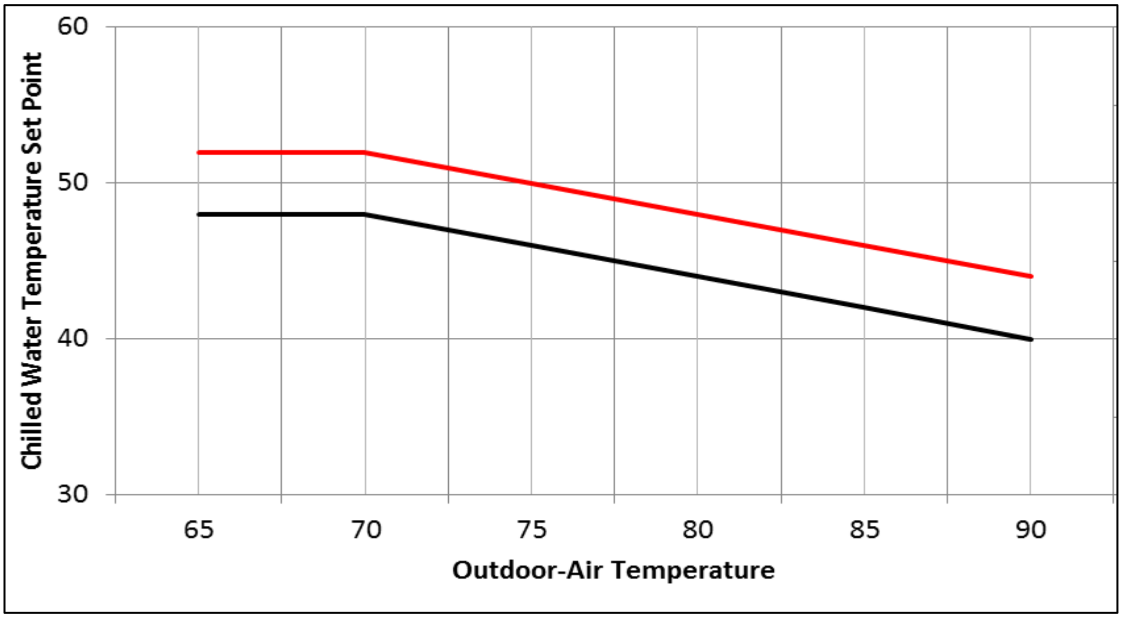

Figure 10.1. Chilled-Water Temperature Reset, Outdoor-Air Temperature Reset Example

Some caveats to this type of reset may include the following:

- Outdoor-air humidity that drives indoor humidity conditions. This may require chilled-water temperatures that are no greater than 48°F to ensure that the surface temperatures of the cooling coil are below the dew point of the air, so that condensation of vapor in the air-stream can occur (dehumidification).
- The cooling coil surface and design of the coil (area, number of rows and fins) all contribute to the coil's ability to remove moisture. There is no "perfect" coil and other factors can affect moisture removal. These factors can include air velocity across the coil (if too high, moisture removal can be impacted), coil cleanliness (dirt is an insulator), or plugged drain pans (moisture will not drain out of the fan system and is re-entrained into the air).
- Dehumidification targets (maximum humidity or dew point temperature) should be carefully selected to ensure excess energy consumption is minimized.
- Temperature (dry bulb/dew point) and humidity sensors should be calibrated periodically to ensure their accuracy.

10.2.1.2 Chilled-Water Temperature Reset – Average Cooling Valve Position Reset

One chilled-water temperature reset strategy is to use average cooling coil valve position feedback to reset the setpoint. With the advent of the BAS, intelligent control sequences that rely upon primary (AHU/RTU) HVAC cooling demand conditions and secondary (zone FCU) HVAC cooling demand conditions that reflect an average cooling demand value can be used to more intelligently reset the chilled-water temperature setpoint. Figure 10.2 shows two examples of a chilled-water temperature reset, using the average cooling coil valve position feedback.

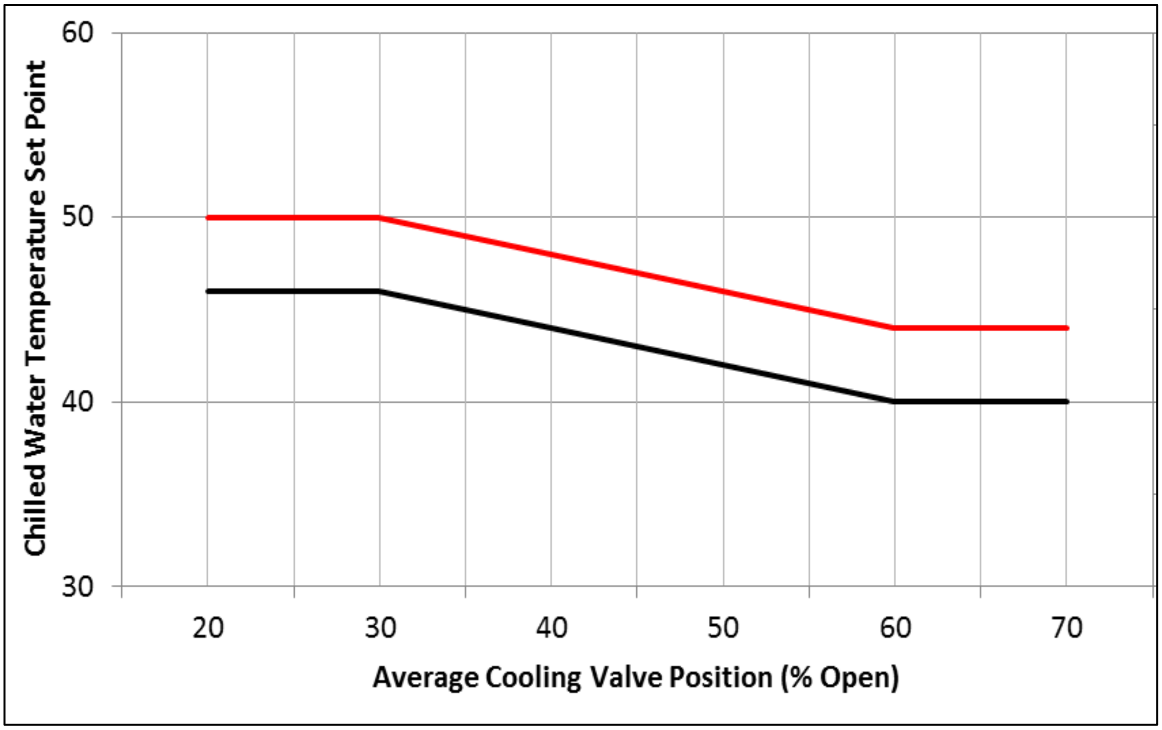

Figure 10.2. Chilled-Water Temperature Reset, Average Cooling Valve Position Reset Example

The **red line** in Figure 10.2 shows the chilled-water temperature reset that is between 50°F and 44°F, as the average cooling valve position increases from 30% open to 60% open. The **black line** in Figure 10.2 shows the chilled-water temperature reset that is between 46°F and 40°F, as the average cooling valve position increases from 30% open to 60% open.

Similar to the first example in Figure 10.1, Figure 10.2 also shows what a chilled-water temperature reset would look like, but the feedback value represents the actual loads that require chilled water, not a value based on the OAT. This method accounts for any number of design and operational changes that may have occurred or are actively occurring, including the introduction of more outdoor air, the improvement of a building's envelope (newer windows, better insulation, reduced internal heat gain from more efficient lights and plug loads, etc.), and over-heating that may be occurring due to DCV strategies (introducing more warm/humid air in the summer) or dehumidification strategies that also result in over-cooling of AHU and/or zone temperatures.

The Figure 10.2 results are from the automatic calculation of cooling coil valves and their demand for flow. If the average coil valve position is low, this would automatically calculate that a higher chilled-water temperature is sufficient, while a higher value would automatically calculate that a lower chilled-water temperature is required. The chilled-water temperature reset algorithm would automatically adjust the chilled-water plant or individual chillers to the calculated setpoint, based on the average of all the specified cooling coil control valve positions. The intent is to meet all or most of the cooling loads while operating the chiller in a region where the compressor lift is reduced.

10.2.1.3 BAS Data Needed to Verify the Chilled-Water Temperature Reset

Analyzing and detecting chilled water temperature setpoint problems and opportunities can be achieved by using BAS trend capabilities. The following parameters should be monitored using the trend capabilities of the BAS:

- chilled-water supply temperature
- chilled-water return temperature
- cooling coil valve position
- zone temperatures
- OAT.

Data should be collected at recommended intervals of between 5 and 30 minutes for a minimum of 1–2 weeks. When analyzing the chilled-water temperature reset, the trend data to look for include the following:

- Do the zone temperature trend data indicate that zone temperatures are rising at night and during weekends (during warm-hot weather vacancy periods) to the configured cooling setback values (78–85°F)?
  - When setback actions are occurring, the AHU, RTU, FCU, chilled water plant and other ancillary HVAC systems should also indicate temporary operations to maintain the setback values.
- Is the chilled-water loop temperature cool enough to keep most of the cooling coil control valves less than 90% open?
- The chilled-water plant should be scheduled to not run during nights and weekends, unless the loads being served include 24/7 process loads that have do not have dedicated cooling equipment.
- If all space temperatures are below the night setback setpoint values (<78–85°F) during night and weekend periods, the entire chilled-water plant should be off, or configured to operate at less capacity during vacancy periods in order to maintain the building at setback conditions.
- During chiller operations, is the ΔT (difference between the chilled-water return and chilled-water supply temperatures) for the running chiller(s), at least 7°F and optimally 10°F?
  - Optimal ΔT values of 10°F (or higher) indicate that the compressor is adequately loaded.
  - If the ΔT values are less than 5°F, this may indicate other problems, such as:
    - The primary/secondary loop is not configured correctly (flow balance issues between the primary and secondary loops that could be design-related, poor secondary loop controls or other O&M issues)
    - When the chilled-water plant consist of more than one chiller, there are more chillers running than is required for the building cooling loads.

11.0 Hot Water Temperature Reset

The purpose of this hot water temperature reset implementation guide is to show, through examples of good and bad operations, how hot water temperature resets can be efficiently controlled and what the indicators are for both good and bad operations. This guide also offers some common implementation strategies.

11.1 Why Consider Implementing a Hot Water Temperature Reset?

Hot water temperature resets are simple to implement, track, and administer. The goal of hot water temperature reset is to optimize the hot water temperature that the heating system is trying to maintain. Whether the heating system uses a hot water boiler or a steam-fed hot water heat exchanger, the principles are the same: generate hot water that is hot enough to satisfy the majority of the heating loads seen by the various heating, ventilating and air-conditioning (HVAC) systems.

Condensing hot water boilers are designed to take advantage of condensing for greater energy efficiency, and the hot water temperature setpoint and return water temperature significantly impacts the efficiency of the boiler. Condensing boilers are most efficient when the inlet (return water) temperature for the boiler is less than 130°F, and this efficiency increases as the inlet water temperature decreases. The energy efficiency of condensing boilers is negatively affected when the hot water temperature setpoints are warmer than required because the return water temperature will also be warmer than required. Non-condensing boilers (by design) are generally not recommended to be operated any lower than 140–160°F to avoid condensing of flue gases, which can have negative ramifications for the boiler (refractory and other boiler components).

Because boilers are often sized for design load conditions (cold weather), it is not uncommon to find boilers that are short-cycling during moderate weather. Short-cycling is often a result of a boiler's inability to "turn down" (reduce their firing rate or rate of natural gas combustion). When a boiler has very low load, the boiler (once it begins firing) will quickly overshoot the internal setpoint for the boiler (temperature if hot water, pressure if steam). Short-cycling losses occur when an oversized boiler quickly satisfies process or space-heating demands, and then shuts down until heat is required again.

The goal for any building with properly implemented hot water temperature reset strategies is to see energy consumption reductions during conditions when heating loads are very low. If the hot water boiler is a condensing boiler, the ability to configure the boiler to a lower setpoint will help the boiler operate longer, minimize boiler short-cycling, and operate the condensing boiler in its "sweet" spot for efficiency. Steam-fed hot water heat exchangers do not have issues with short-cycling.

The other advantage of properly configured hot water temperature setpoints is the ability to not generate hot water that is warmer than needed. Most building HVAC systems will have a few hot water heating control valves which may leak. Such leaks may occur at an air-handling unit (AHU), rooftop unit (RTU), fan-coil unit (FCU) or a variable-air volume (VAV) box. If control valves leak, it is better to leak heated water that is not as hot as would normally be expected, and this can only occur if the hot water temperature is intelligently reset to match the setpoint to the true load.

Note: Ensure that the installed piping, if Victaulic design, is designed with the proper fittings and seals that can tolerate temperature fluctuations as loop temperatures reset and as water temperatures cool down over nights, weekends and even extended periods of lockout (warm weather, summer shutdown periods, etc.).

11.2 What Systems Should Be Considered for Hot Water Temperature Reset Implementation?

A heating hot water system is typically designed to provide hot water to help maintain minimum temperatures in a building. Historically, minimum temperature requirements have been associated with cold outdoor-air temperature (OAT) operations. However, when warm-weather mechanical cooling is operated to de-humidify a building via AHUs that over-cool the discharge air (to ensure condensation and removal of moisture is occurring), hot water for zone reheat capability is needed. The temperature setpoint for the hot water can be based upon the OAT, or it can be based on a more intelligent zone feedback algorithm that calculates actual zone reheat requirements to determine the optimal hot water temperature setpoint.

All hot water boilers or steam-fed heat exchangers that are equipped with reliable controls are good candidates for implementing hot water temperature reset strategies, under the following assumptions:

- The hot water boiler is a condensing boiler (able to reset its setpoint to as low as 100°F).
- If the hot water boiler is a non-condensing boiler, it is assumed that it is designed with a three-way valve to regulate the distribution loop temperatures to as low as 100°F, while keeping the boiler in a hot (non-condensing) state (140–160°F), to protect the boiler.
- The BAS and boiler controllers are reliable (communication between hot water boilers or steam heat exchangers and the supervisory controllers is more than 95% reliable and any integration issues for specialty systems such as boilers are totally resolved.

Some of the typical HVAC systems found in various buildings that should be considered for hot water temperature reset includes the following:

- condensing hot water boilers
- non-condensing hot water boilers (equipped with a three-way valve to regulate the distribution loop temperatures while keeping the boiler temperature >160°F)
- steam-fed hot water heat exchangers.

11.2.1 Methodology for Hot Water Temperature Reset

This section of the implementation guide presents some common strategies for implementing hot water temperature reset.

- OAT reset
- average heating coil valve position reset.

Note that this list is not intended to cover every implementation technique, but present a few techniques in detail to provide some context for what a good hot water temperature reset looks like. All final hot water temperature reset parameters should be thoroughly discussed with the owner/operator of the building automation system (BAS) and the building(s). The implementation strategy may vary by building type, system type, or other variables that this guide does not directly highlight.

11.2.1.1 Hot Water Temperature Reset – Outdoor-Air Temperature Reset

The most common hot water temperature reset strategy uses the OAT to reset the setpoint. This strategy dates back to the days of pneumatic controls (and is still used by pneumatic controllers today). With the advent of the BAS, more intelligent control sequences are available, but are often not used. Figure 11.1 shows two examples of a hot water temperature reset using OAT.

The **red line** in Figure 11.1 shows the hot water temperature reset that is between 140°F and 200°F, as the OAT drops from 60°F to 0°F. The **black line** in Figure 11.1 shows the hot water temperature reset that is between 120°F and 160°F, as the OAT drops from 60°F to 20°F. Depending upon the building age, design, envelope (windows, doors, wall and ceiling insulation) and HVAC operations, the hot water temperature that is required for heating the building may need to be as high as 160–200°F, during cold weather conditions (also known as "design-day" conditions). The design-day conditions for your building should be on the mechanical design prints or similar documentation and should include the OAT value for the design day and the maximum hot water temperature required. Often, it is not uncommon for operations and maintenance (O&M) staff to raise the maximum hot water temperature (high limit) to a value that is greater than the original design value. Sometimes this occurs because the building mission or operation has changed. This may include the introduction of more outdoor air into the building or it may include extreme weather events that result in O&M staff raising the hot water temperature high limit from the original design parameters (or the OAT temperature at which the high limit would be reached). In both cases, this can often result in higher-than-needed hot water temperatures and greater energy consumption to generate the higher hot water temperatures. Unless located in Alaska or the very northern tier states (Climate Zone 7, see Figure 9.3), the maximum hot water temperature for most hot water heating systems should never exceed 180°F (especially in southern climate zones).

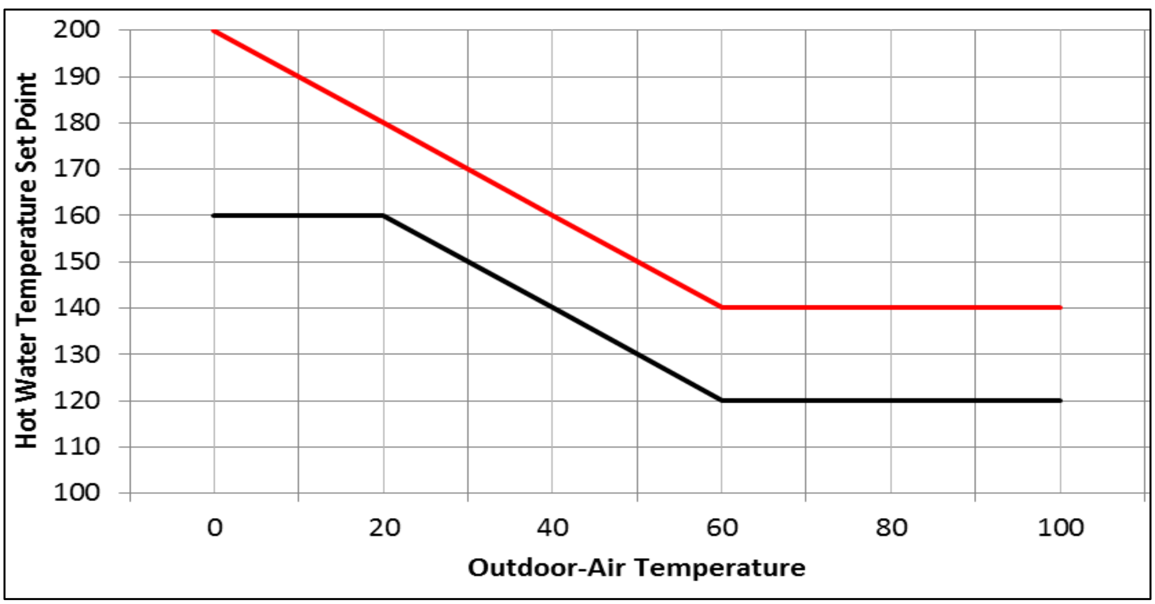

Figure 11.1. Hot Water Temperature Reset, Outdoor-Air Temperature Reset Example

As already noted, if the boilers are non-condensing in design, the higher setpoints are a requirement for the boiler's protection because they are not designed for condensing of combustion gases in the flue, which can return as a liquid to the boiler assembly that may contain corrosive acids that are harmful to the boiler. By keeping the boiler operating in a hot condition (boiler hot water temperatures between 160°F and 180°F), the risk of condensing is greatly reduced. These types of boilers and hot water systems may be designed with a primary/secondary loop that allows the boilers to run hot all the time, and the secondary loop temperature can be reset to much lower temperatures (either by the OAT reset method or the average heating coil valve position method) if the secondary loop design includes a three-way mixing or blending valve that allows the hot primary boiler loop to mix with cooler secondary loop water. This design allows the boiler to remain hot all the time (reduced risk of condensation). If the non-condensing boiler does not have this capability, a temperature reset should not be implemented.

When the OAT reset is applied to a condensing boiler system, if the boiler system operates at temperatures that are greater than 150°F, this will often result in return water temperatures that are 130°F (or higher). Condensing boilers that see return water temperatures greater than 130°F operate at lower efficiency values than condensing boilers that see return water temperatures below 120–130°F. In some cases, this can be as great as 5–10% loss in efficiency (at the boiler).

11.2.1.2 Hot Water Temperature Reset – Average Heating Coil Valve Position Reset

A less common (but more intelligent) hot water temperature reset strategy is the use of average heating coil valve position feedback to reset the setpoint. With the advent of the BAS, intelligent control sequences that rely upon primary (AHU/RTU) HVAC heating demand conditions and secondary (zone VAV/FCU) HVAC heating demand conditions that reflect an average heating demand value can be used to more intelligently reset the hot water temperature setpoint. Figure 11.2 shows two examples of a hot water temperature reset, using the average heating coil valve position feedback.

The **red line** in Figure 11.2 shows the hot water temperature reset that is between 140°F and 180°F, as the average heating valve position increases from 20% open to 60% open. The **black line** in Figure 11.2 shows the hot water temperature reset that is between 100°F and 160°F, as the average heating valve position increases from 20% open to 60% open.

Similar to the first example in Figure 11.1, Figure 11.2 also shows what a hot water temperature reset would look like, but the feedback value represents the actual loads that require heated water, not a value based on the OAT. This method accounts for any number of design and operational changes that may have occurred or are actively occurring, including the introduction of more outdoor air, the improvement to a building's envelope (newer windows, better insulation, reduced internal heat gain from more efficient lights and plug loads, etc.), and over-cooling that may be occurring due to demand control ventilation (DCV) strategies or dehumidification strategies that also result in over-cooling of AHU and/or zone temperatures.

The average heating coil valve (calculated value) is derived from averaging of heating coil valve positions in the building. As the calculated average value drops to a low value, the hot water temperature setpoint is automatically adjusted to a lower value and as the calculated average value rises to a higher value, the hot water temperature setpoint rises to a higher value. The hot water temperature reset automatically updates the boiler or steam-fed heat exchanger with the updated setpoint, based on the current load (average heating valve) conditions. **Note**: Due to the potentially large number of heating valves in a building, the amount of control code required to calculate the average heating coil valve position may be cumbersome. Alternatives to this method could include calculating a sampling of zone reheat valves per AHU and should include zones that serve perimeter heating loads.

Automatic temperature resets below 140–160°F should not be applied to non-condensing boilers, as condensation may damage the boiler. If the hot water system is designed with non-condensing boiler(s), and configured with a secondary loop whose temperature can be reset (via three-way control valve or other means of blending primary hot water with cooler secondary water), then this automatically calculated setpoint can also be applied to the non-condensing boiler's secondary loop (not to the boiler).

This type of design allows the boiler to keep its internal loop hot (160–180°F) while allowing the secondary loop to run cooler.

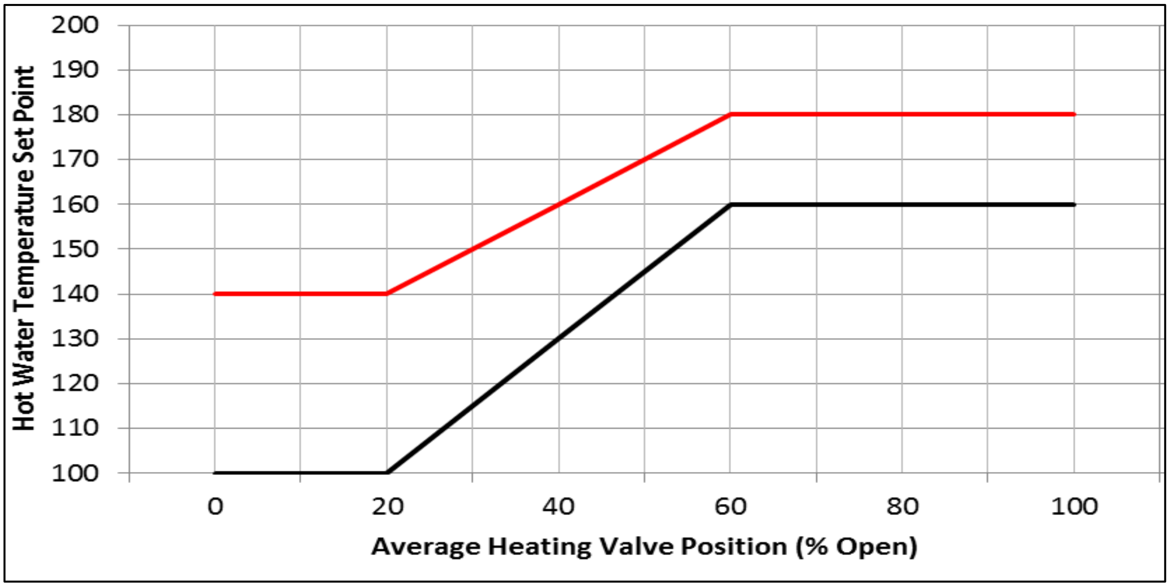

Figure 11.2. Hot Water Temperature Reset, Average Heating Valve Position Reset Example

Some advanced control sequences will start the hot water setpoint as low as 80–100°F (especially if the hot water heating system must remain active year-round for summer reheat loads). Other advanced controls may automatically configure the high limit to be 140°F, but combine the OAT feedback to gradually increase the high limit from 140°F to as high as 160–180°F, as the OAT drops from 20°F down to 0°F (versus holding the high limit at 160–180°F all the time).

11.2.1.3 BAS Data Needed to Verify Hot Water Temperature Reset

Analyzing and detecting hot water temperature reset problems and opportunities can be achieved by using trend capabilities derived from the BAS. The following parameters should be monitored using the trend capabilities of the BAS:

- heating hot water supply temperature
- heating hot water return temperature
- heating coil control valve position
- zone temperatures
- OAT.

Data should be collected at recommended intervals of between 5 and 30 minutes for a minimum of 1–2 weeks. When analyzing the hot water temperature reset, the data trends to look for include the following:

- Do the zone temperature trend data indicate that zone temperatures are dropping at night and during weekends (during cool-cold weather vacancy periods) to the configured heating setback values (55–65°F)? When setback actions are occurring, the AHU, RTU, FCU, hot water plant and other ancillary HVAC systems should also indicate temporary operations to maintain the setback values.

- Is the hot water loop temperature warm enough to keep most of the heating coil control valves less than 90% open? The average heating coil control valve position should be 40% or less.
- If the OAT is >45°F during night and weekend periods, the entire heating hot water plant should be off, or configured to operate at less capacity during vacancy periods.

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