Title

Hot Water Heating: Design and Retrofit Guide

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Publication Date
2024-05-21

Hot Water Heating

Design and Retrofit Guide

Version 1.0 | March 18, 2024

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Table of Contents

Table of Figures

Preface

This guide was developed with funding from the California Energy Commission's Public Interest Energy Research (PIER) program as part of the project titled Getting Out of Hot Water: Reducing Gas Consumption in Existing Large Commercial Buildings (PIR-19-013) which was led by the University of California Berkeley, Center for the Built Environment.

1

Introduction

1 Introduction

Most natural gas consumption in commercial buildings in the US is for space heating. Unitary systems are common in smaller buildings, whereas space heating is typically accomplished using a hydronic heating hot water (HHW) system for medium and large commercial buildings, often with gas-fired boilers. These HHW systems are generally the predominant end use for natural gas consumption and represent a large fraction of total greenhouse gas emissions in commercial buildings.

Reducing natural gas consumption in commercial buildings is important not only for minimizing operating costs but as a decarbonization strategy for both new construction and existing buildings. Recent research has highlighted a number of findings indicating that boilers and HHW systems often do not operate as designed or intended. For example, Figure 1 shows a Sankey chart representing the fate of heating energy costs in a commercial office building where only 17% of the cost was converted to intentional space heating (Raftery, Geronazzo, Cheng, & Paliaga, 2018). Significant losses were measured at the boilers and in the hot water distribution, far exceeding what is conventionally assumed for HHW systems.

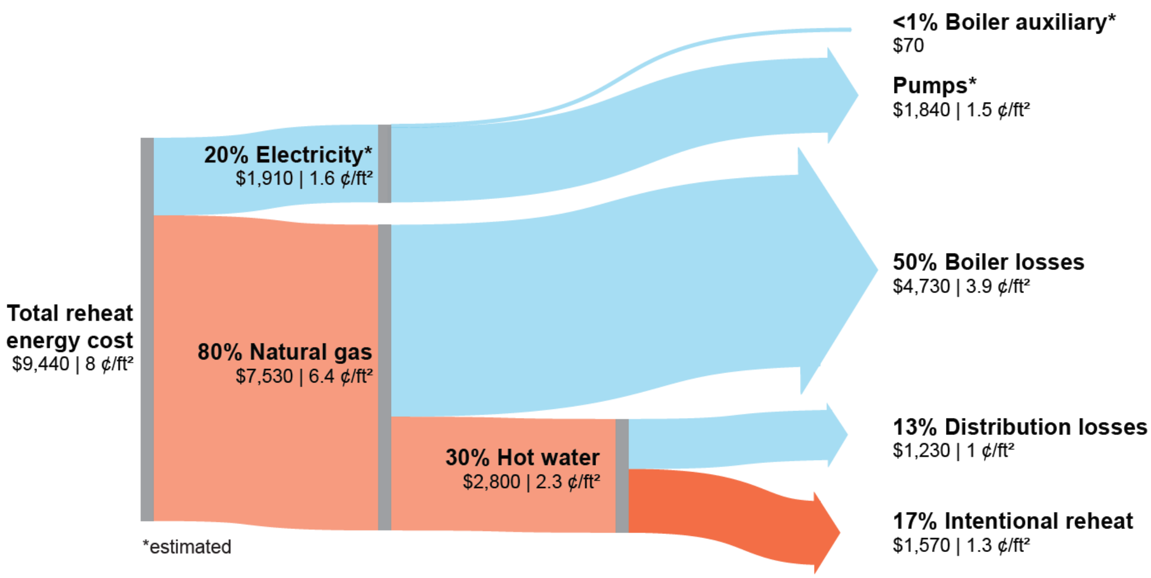


Figure 1 – Fate of Heating Energy in a Commercial Office Building

These research projects and other recent efforts have suggested key areas for attention for improving energy efficiency in HHW systems, often with opportunities for both significant and cost-effective savings. Figure 2 shows the before and after natural gas consumption for a commercial office building that underwent an HVAC control system retrofit, resulting in a reduction in natural gas use of over 50 percent with a simple payback of 7 years (Taylor Engineers, 2018). These strategies for reducing natural gas consumption in HHW systems can be considered for general application but are also important first steps for projects considering converting to all-electric systems. Though some of the strategies for reducing heating energy consumption apply to the HHW equipment directly, many of the most effective strategies are upstream in the control of the airside equipment. Improving airside system control can often significantly reduce the annual total and peak heating energy loads that need to be met by the HHW systems.

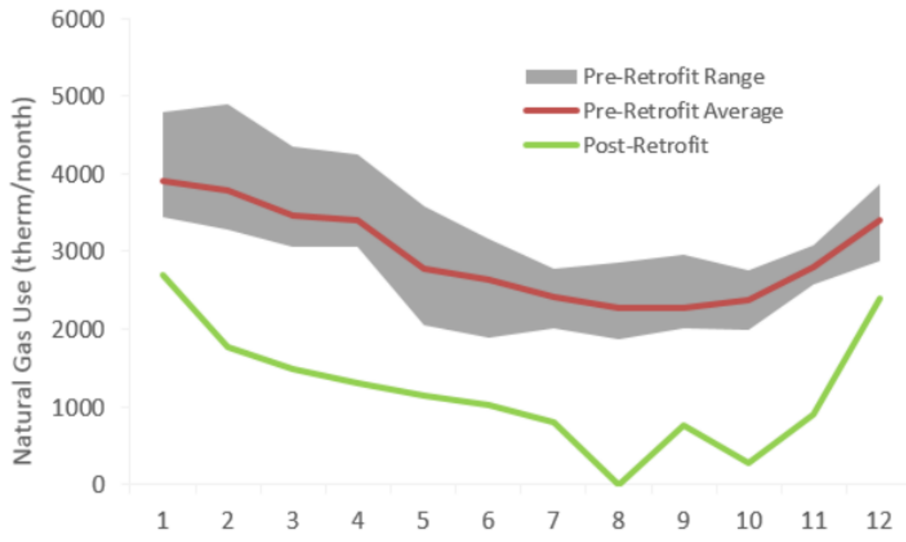


Figure 2 – Natural Gas Savings from Commercial Building Control Retrofit

This guide describes key design issues, then provides information on strategies to reduce hot water loads and improve heating system efficiency. This guide is intended for a broad audience, including designers, energy analysts, installers, commissioning providers, and building operators. Building owners and property managers may also benefit from general information presented and retrofit opportunities.

2

Key Design Issues

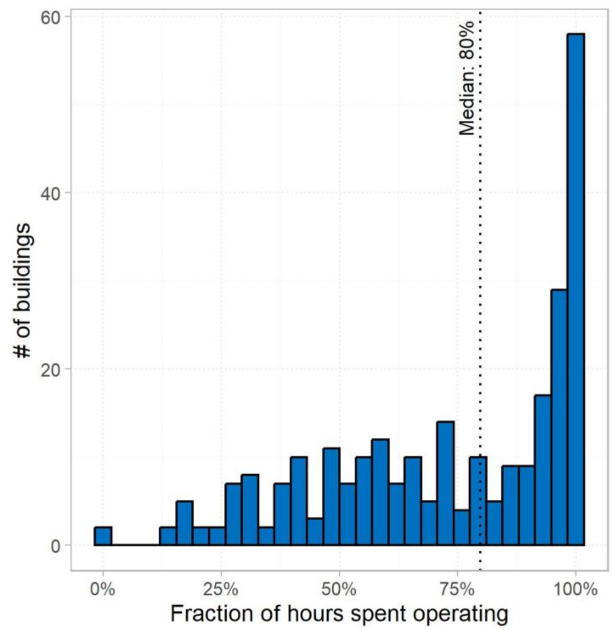
- Boiler Run Time
- Boiler Sizing
- Condensing Boilers
- Distribution Losses

2 Key Design Issues

2.1 Boiler Run Time

Many hot water plants are operated continuously 24/7 or for longer hours than required to meet building loads. A large scale data collection study of hot water systems in commercial buildings found that the median system operated for 80% of the total time period evaluated (Raftery, Singla, Cheng, & Paliaga, in press) and that 40% of the systems operated more than 90% of the time (Raftery, Singla, Cheng, & Paliaga, in press). Though some of the buildings in the data set are continuously occupied (i.e., labs and healthcare), most are buildings with offices and typical occupancy hours and ventilation requirements. The data show that many hot water systems operate continuously without switching off at night, on weekends, or during the summer. Based on designer interviews (Lamon, Raftery, & Schiavon, 2022) and other feedback, common reasons include:

1. Lack of awareness that the hot water plants were running unnecessarily
2. Poorly configured or implemented controls
3. Concerns of leaks from pipe fittings if the systems were allowed to cool
4. Manual overrides that were inadvertently left indefinitely in place



(Raftery, Singla, Cheng, & Paliaga, in press)

Figure 3 – Hot Water Plant Operating Hours

Solutions to address challenges with #2 are discussed in Section 4.1.

Real or perceived concerns that grooved end pipe couplings (e.g., Victaulic) will leak is a commonly cited reason for why hot water systems are intentionally run continuously. Grooved end couplings are a common way of joining pipes where grooves formed near the ends of the pipes are held together with a coupling, with gaskets to form a seal.

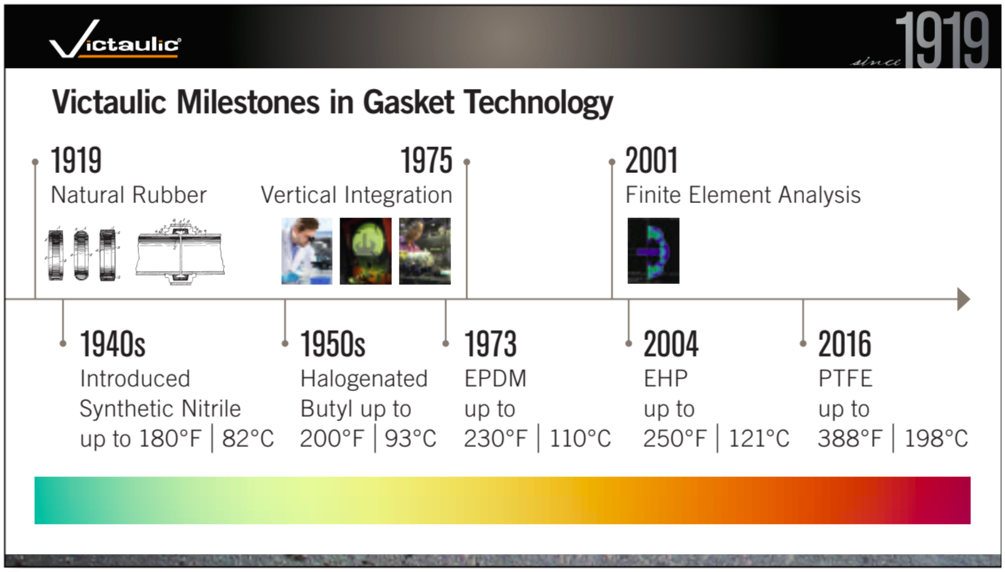
Based on correspondence with Victaulic (Lafferty, 2023), the gasket materials used in their couplings have evolved over time from natural rubber to various synthetics to improve performance and longevity (Figure 4). Gaskets are vulnerable to degradation if exposed to temperatures above certain upper limits for each type of material, potentially compromising sealing quality when the pipes are allowed to cool down. Victaulic used EPDM gaskets starting in the early 1970s, which were rated to a maximum temperature of 230°F. The subsequent development of EHP

40%

OF BOILER PLANTS RUN CONTINUOUSLY

(more than 90% of the time, even in summer)

gaskets, rated up to 250ºF, in 2004 was in response to reported issues with leaks due to cooled loops. A possible concern was older boilers that did not have good temperature control or safeties to prevent these high temperature excursions. Degradation can be positively identified if gaskets removed and sent to a lab for testing. Installation issues such as improper tightening, damaged gaskets, or bad seating are not likely to be culprits because these would begin leaking upon initial pressurization.



(Victaulic)

Figure 4 – Victaulic Gasket Technology Development

Nevertheless, many hot water systems are operated continuously in buildings constructed since 2004 based on this concern. A possible scenario is that building operators were trained in older buildings with systems where this was a real issue, and have taken the strategy of running the plants continuously to newer buildings where the issue does not apply, or have passed down to subsequent generations of operators the conventional wisdom that all Victaulic couplings are prone to this issue.

2.2 Boiler Sizing

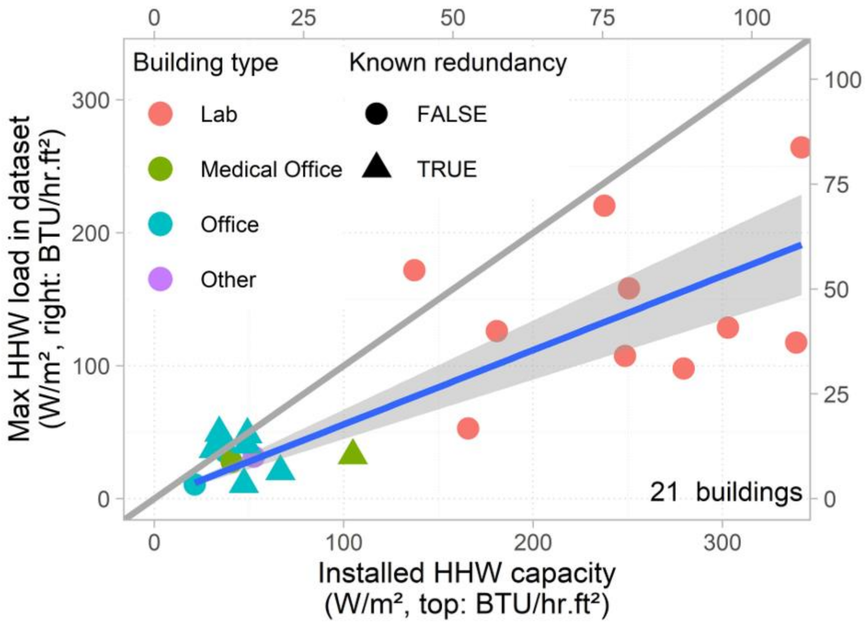
Hot water plants are frequently oversized due to a number of common designer practices and actual hot water load profiles are often skewed heavily toward low part loads. Oversizing and operation at low part loads are important factors that impact system efficiency and longevity, and may become more critical as more hot water systems are replaced with all-electric plants.

Oversizing and Load Profiles

A large scale data collection study of hot water systems in commercial buildings found that design heating plant capacities were on average twice as large as the maximum observed peak load (Raftery, Singla, Cheng, & Paliaga, in press). For buildings with at least a year of data including weather conditions close to the design temperature, Figure 5 shows the maximum observed peak load plotted against the total design capacity. The blue diagonal line represents the average oversizing factor, oversized by more than a factor of 2, with many buildings

The average boiler plant is oversized by a factor of two.

close to a factor of 3 oversized. Many lab buildings have greatly oversized plant capacities, which may reflect oversizing for future capacity and flexibility, but most office buildings are also heavily oversized. Where extra capacity for redundancy is known, it is excluded from the total capacity reported.



(Raftery, Singla, Cheng, & Paliaga, in press)

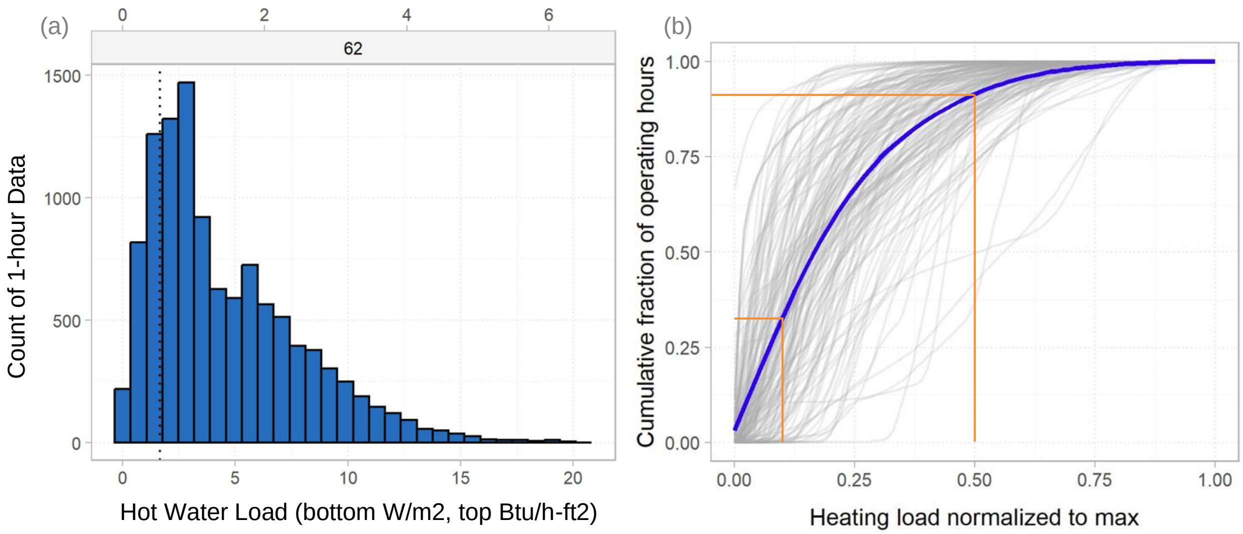
Figure 5 – Hot Water Plant Oversizing

Oversizing of hot water plants may be due to a range of factors. Conventional load calculation practices are generally very conservative, evaluating envelope loads at the design ambient temperature in a steady state condition, rather than with a more-typical diurnal temperature profile. Typical load calculations assume occupied heating setpoints, even though zero internal gains or solar gains are assumed and the time of day when the coldest ambient temperatures are observed are generally prior to occupancy when space temperature setpoints are setback. Many designers also assume minimum ventilation and include a safety factor or a recovery factor for morning warmup. The ASHRAE Handbook of Fundamentals suggests that oversizing factors of 20 to 25% for warmup and safety are common (ASHRAE, 2021). Because boilers are only available in discrete sizes, selecting the next-size-up equipment may further increase actual plant capacity.

For boiler plants, the conventional wisdom is that there are few downsides to oversizing. It is safer to oversize because larger equipment are only incrementally more expensive and incrementally larger in physical size, whereas buildings that don't recover in time from night setback run the risk of complaints that plants are undersized, presenting a professional liability concern.

The same study found that almost all buildings have load distributions significantly skewed toward low part loads, that the majority of the systems spend most of the time operating at relatively low loads. Figure 6(a) shows the hot water load distribution for a typical building within the data set, where the vast majority of operating hours are at very low part loads. Figure 6(b) shows the cumulative fraction of operating hours for each plant, normalized to each plant's maximum load, with the blue line representing the median distribution. The median building spends 94% of the time operating at loads at or below 50% of the maximum, and 30% of the time at loads below 10% of the maximum.

Sections 3.3 and 4.5 include discussion of some considerations around boiler sizing and selections.



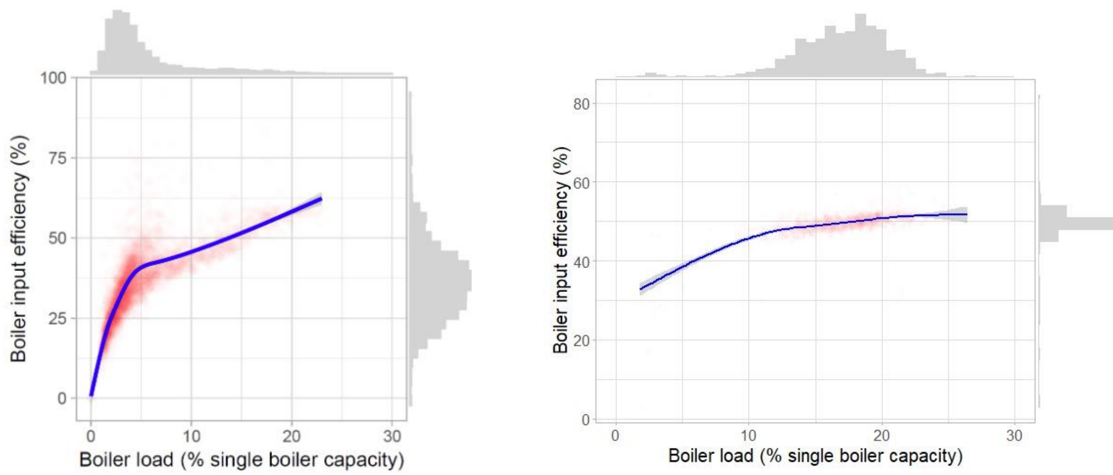
(Raftery, Singla, Cheng, & Paliaga, in press)

Figure 6 – Hot Water Plant Load Distribution

Impacts of Oversizing

Though hot water plant design efforts traditionally focus on design conditions to ensure adequate capacity, attention must be paid to typical operating conditions as well to ensure efficient operation. The natural tendency for building heating load distributions to be skewed to low part loads combined with the industry tendency for oversizing means that many boilers operate at loads that are below their minimum turndown ratio for a large portion of the year. Consider a typical plant with two equally-sized boilers serving a building with the average load profile shown Figure 6(b), assuming that the plant is oversized by a factor of 2 greater than the actual peak load according to Figure 5. A boiler with 5:1 turndown will spend nearly 60% of the time cycling below its minimum firing rate. A boiler with 10:1 turndown will spend 30% of the time cycling below its minimum firing rate.

Boiler efficiency is generally only rated at full load. Though condensing boiler efficiency improves slightly at low loads because of improved heat exchange, efficiency for boilers in general falls dramatically when short cycling at loads below the minimum turndown limit. Each time a boiler cycles on and off, there is incomplete combustion. Pre-purge and post-purge cycles clear unburned fuel, along with some of the heat from the boiler. The boiler jacket losses are relatively constant but represent a larger portion of the total gas input at low part loads, compared to at full load. Non-condensing boilers generally have much less turndown capability and are at higher risk of low operating efficiencies due to short cycling, compared to condensing boilers. Figure 7 shows low measured boiler efficiencies at very low part loads for non-condensing boilers in operating buildings. The boiler plant at left consisted of two greatly oversized natural draft boilers with limited turndown capability. Over a full year of operation, the vast majority of operation was at loads below 10 percent of the single boiler capacity, with annual average efficiency around 33 percent (Raftery, Geronazzo, Cheng, & Paliaga, 2018). The boiler plant at right consisted of a single forced draft boiler with 2:1 turndown capability. With operation largely between 10 to 25 percent of capacity, the boiler efficiency averaged about 50 percent (Raftery, et al., 2024). The very low part load efficiencies displayed in Figure 7 appear to be consistent for boilers with limited turndown capability (Raftery, Singla, Cheng, & Paliaga, in press). With boilers typically oversized and buildings tending to operate at low part loads below minimum turndown limits, extensive boiler short cycling frequently leads to annualized efficiencies far below the rated values. Modern condensing boilers tend to have variable firing rates and lower turndown capabilities (e.g., 10:1 or 20:1). The improved turndown reduces the risk of excessive cycling, and generally these boilers display actual efficiencies that are much closer but still often lower than the rated efficiencies (Raftery, Singla, Cheng, & Paliaga, in press). Attention to preventing short-cycling at low load conditions may be more important for overall plant efficiency than selecting high efficiency boilers (Peterson, 2018).



(Raftery, Geronazzo, Cheng, & Paliaga, 2018) (Raftery, et al., 2024)

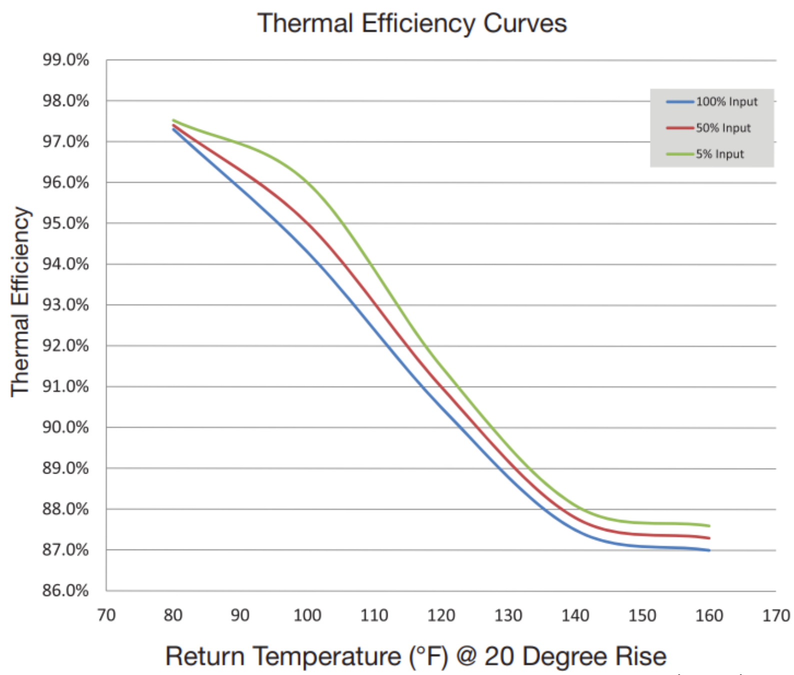
Figure 7 – Boiler Part Load Efficiencies

For all-electric hot water plants, equipment oversizing has a much larger impact on first cost and physical equipment size. Air-to-water heat pumps (AWHP) are significantly more expensive, roughly 5

times more expensive than boilers, so the incremental cost for larger capacities has a much larger first cost impact, not including the impact to electrical infrastructure costs. AWHPs are also physically much larger and heavier than gas-fired boilers. Oversizing thus exacerbates the challenge of finding sufficient space for AWHP plants, particularly for large commercial and retrofit projects, which in some cases require structural upgrades to support the heavier equipment.

2.3 Condensing Boilers

Condensing boilers achieve higher efficiencies by allowing water vapor in the flue gases to condense to preheat the entering water (either within the same heat exchanger, or using an additional heat exchanger), recovering the latent heat of vaporization which would otherwise be lost in conventional boilers. To achieve condensing conditions, the boiler entering water temperature must generally be around 130°F or lower, with efficiency increasing as the entering water temperature decreases further. Figure 8 shows typical condensing boiler efficiency curves as a function of entering water temperature. Condensing boilers are effectively required by ASHRAE Standard 90.1 and California's Title 24 building energy standards for boilers with capacities greater than 1 million Btu/h (ANSI/ASHRAE/IES, 2022) (California Energy Commission, 2022). Both standards further require that the hot water distribution system be designed <u>and</u> operated to achieve hot water return temperatures of 120°F or lower, through coil and heat exchanger selection and by minimizing bypass flow.



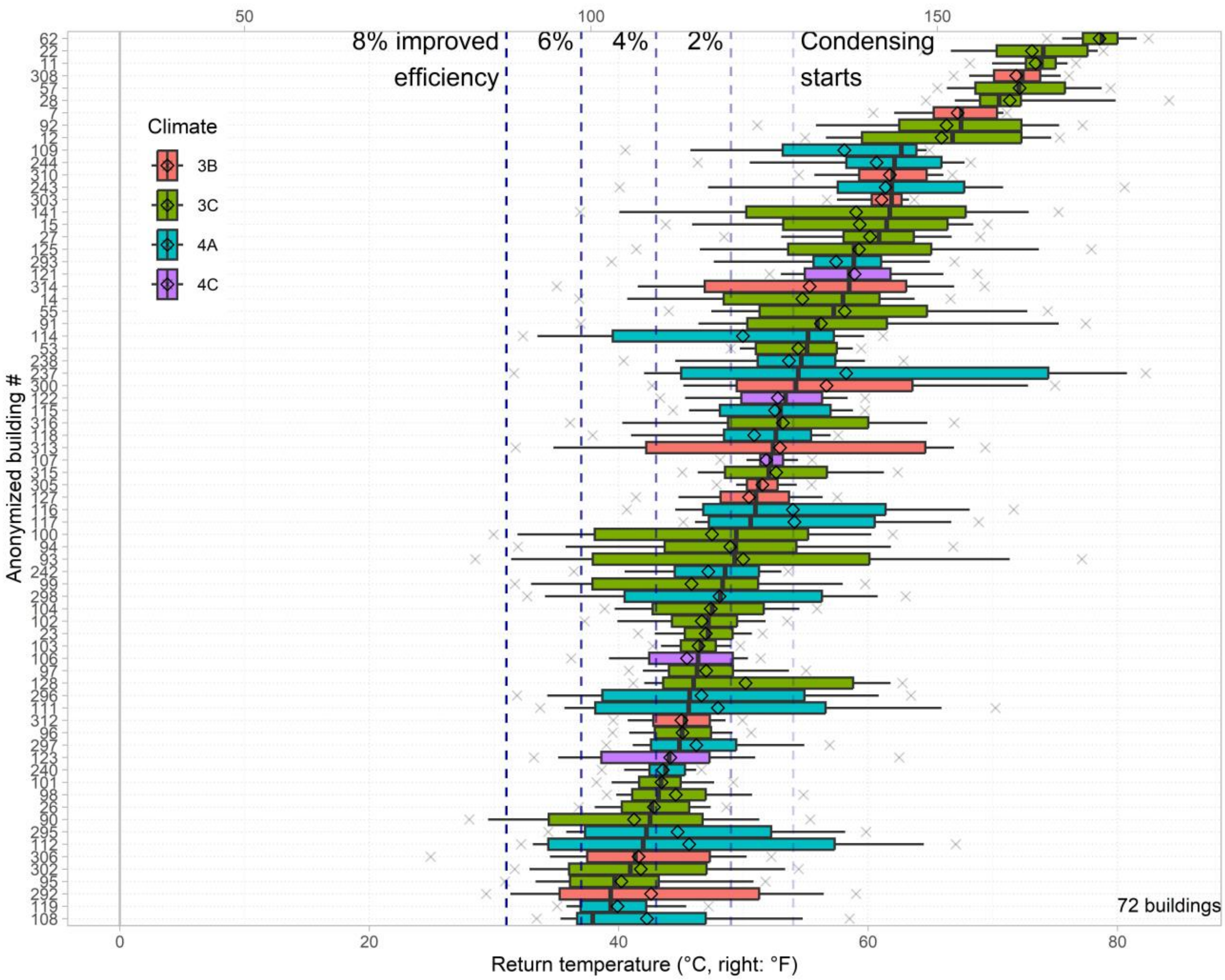
(Aerco)

Figure 8 – Condensing Boiler Efficiency

A large scale data collection study of hot water systems in commercial buildings evaluated how often the condensing boilers achieved condensing conditions by analyzing return water temperatures (Raftery, Singla, Cheng, & Paliaga, in press). Figure 9 shows return water temperature distributions for over 70 buildings with condensing boilers through box and whisker plots. Each box represents the interquartile (25th to 75th percentiles), with black lines and diamond-shaped points indicating median and mean, whiskers indicating 10th/90th, and X-shaped points indicating 5th/95th percentiles. The blue vertical dashed lines roughly represent nominal efficiency increases due to condensing operation. Roughly a quarter of the buildings rarely if ever achieve condensing conditions, based on return temperatures that exceed 130°F the vast majority of the time, and roughly half of the buildings spend half of the time above condensing conditions. Very few buildings have consistently low return temperatures that achieve half of the condensing potential for a substantial portion of the operating hours.

Roughly half of buildings with condensing boilers rarely, if ever, achieve condensing conditions.

Though it is relatively straightforward to design hot water systems to condense at design conditions, achieving condensing conditions in actual operation presents a number of additional challenges. Operators may increase the supply temperature setpoint to improve capacity on cold days, or due to familiarity with higher temperatures in systems in other buildings. In the large data set evaluated, supply temperature setpoints were often found to be relatively constant among the same data set, suggesting that setpoint resets were not working effectively even when known to be implemented. Though hot water coils may be selected for a certain delta-T at design conditions, the delta-T is invariably much lower at off-design conditions. At part load conditions, bypass flow for minimum boiler flows or to keep loops engaged may also become limiting, further raising return water temperatures.



(Raftery, Singla, Cheng, & Paliaga, in press)

Figure 9 – Condensing Boiler Return Water Temperatures

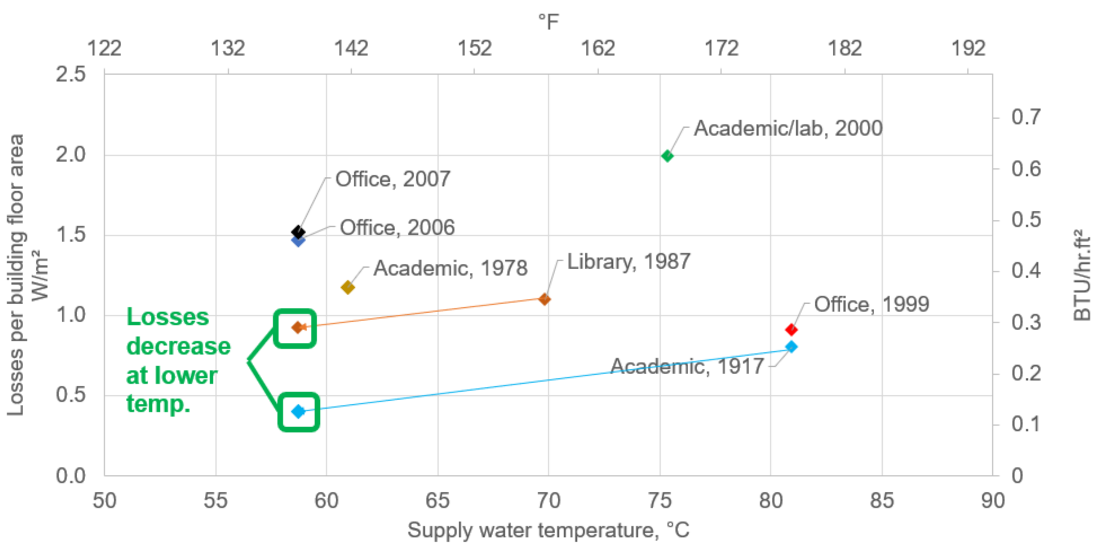
Considerations for maintaining low entering water temperatures and improving condensing boiler performance are discussed in Sections 4.1, 4.2, and 4.3. Many of these strategies are also applicable to improving performance with all-electric hot water plants.

2.4 Distribution Losses

Energy loss through hot water pipe distribution is a factor contributing to heating system efficiency that has largely been overlooked in heating hot water systems. ASHRAE Standard 90.1 even requires that "piping losses shall not be modeled" in the performance approach requirements (ANSI/ASHRAE/IES, 2022). Separate studies of two different office buildings found that over 40% of the heat produced by the hot water plant over the course of a full year was lost through the pipe distribution (Raftery, Geronazzo, Cheng, & Paliaga, 2018) (Raftery, et al., 2024). A recent study measured losses of a median of 0.38 Btu/h-ft^2 (normalized by building gross floor area) from the HHW recirculation piping in 7 large commercial buildings (Raftery, Vernon, Singla, & Nakajima, 2023). For comparison, this is roughly equivalent to one third of average office plug loads. Figure 10 shows the range of losses across the buildings tested and illustrates how losses may vary as a function of hot water supply temperature. Losses can be significant over time because of long piping runs and because most hot water valve assemblies are not insulated (Peterson, 2018). While these losses may have limited detrimental effect in very cold outdoor conditions as they (mostly) occur within the building envelope, medium and large commercial buildings typically have some demand for hot water year round so these losses also occur during the cooling season, placing additional burden on the cooling system to reject the heat which has been added to the building.

In one building, over 40% of the heat produced by the hot water plant was lost through pipe distribution.

Several studies have investigated these losses in other contexts in similar systems, such as (Zhang, 2013) which found that an average of 33% of input natural gas energy was lost annually from DHW recirculation piping in 28 multi-family residential buildings and (Hiller, 2006) measuring these losses in laboratory conditions, further highlighting that these losses are not negligible.



(Raftery, Vernon, Singla, & Nakajima, 2023)

Figure 10 – Hot Water Pipe Distribution Losses

Solutions to help minimize pipe distribution losses are discussed in Sections 4.1 and 4.4.

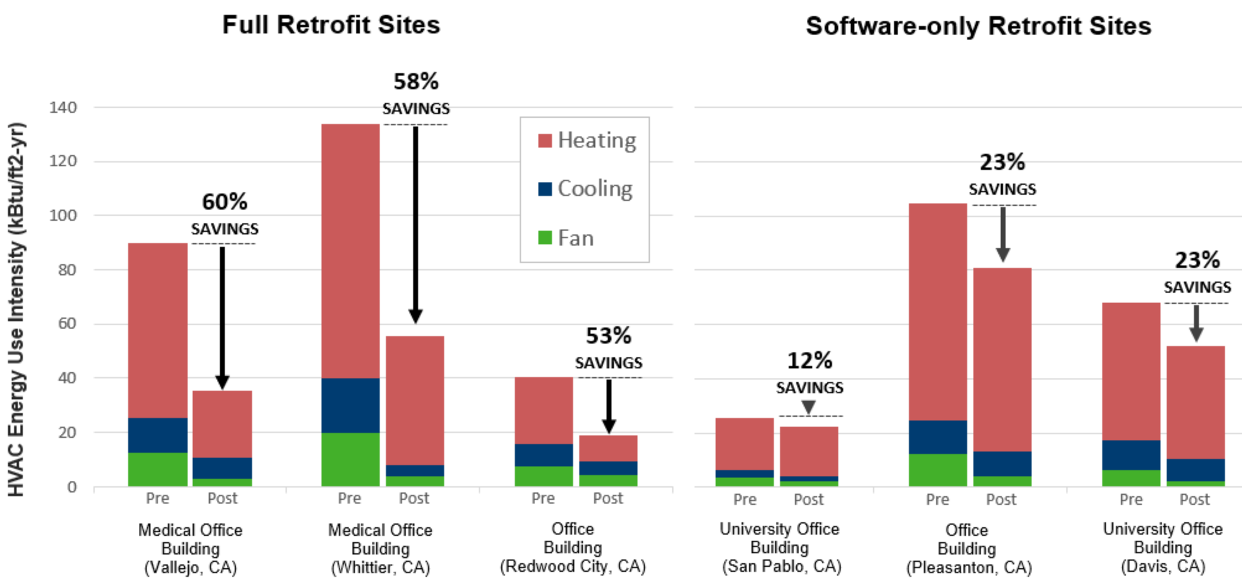
3

Reducing Hot Water Loads

- Reduce VAV Minimum Airflows
- Supply Air Temperature Reset
- Improve Morning Warmup Performance
- Demand Controlled Ventilation
- Discharge Air Temperature Control

3 Reducing Hot Water Loads

The first step in reducing natural gas consumption in building HVAC systems is to reduce the hot water loads. Recent research has identified significant and cost-effective opportunities to reduce hot water loads by addressing the downstream HVAC equipment, primarily with changes to how the equipment are controlled. A study funded by the California Energy Commission retrofitted the HVAC controls in several existing buildings using control sequences from ASHRAE Guideline 36 (Cheng, Singla, & Paliaga, Final Project Report. Demonstrating Scalable Operational Efficiency Through Optimized Controls Sequences and Plug-and-Play Solutions, 2022) (ASHRAE, 2021). At sites that underwent "full retrofits" of the HVAC control hardware and software, HVAC energy use was reduced by 50 to 60 percent compared to the baseline. Note that some of the savings achieved in these projects were likely due to addressing deferred maintenance issues such as non-performing economizer dampers and leaking control valves. At sites that had modern digital controls throughout, HVAC energy use was reduced by 10 to 20 percent based on simply revising the controller programming to follow Guideline 36 sequences. Figure 11 shows the energy savings for both retrofit types, with component energy end-uses broken out. Note that heating energy was reduced by 50 to 60 percent for the full retrofit sites, and up to 20 percent for the software-only sites. These demonstrations illustrate the opportunity for significant decarbonization of the existing building stock using cost-effective retrofit strategies (less than 10 year simple payback), and without replacing major equipment.



(Cheng, Singla, & Paliaga, 2022)

Figure 11 – Guideline 36 Retrofit Energy Savings

Some of the most effective control strategies that are applicable to reducing hot water loads are described in further detail in this section.

3.1 Reduce Variable Air Volume Minimum Airflows

The zone minimum is the airflow provided when a VAV zone is in deadband mode. Conventional practice has been, and in some cases still is, to set this rate equal to a fixed percentage of the zone design flow, between 20 to 50 percent of the design cooling maximum flow. This setpoint must generally be non-zero to provide minimum ventilation to occupied spaces, but it is often set higher than minimum ventilation due to concern over VAV box controllable minimums, concerns of negative thermal comfort impact at low flows (e.g., dumping), and lack of awareness of the importance of this setpoint. ASHRAE Standard 90.1 and California's Title 24 limit VAV minimums and require them to be set equal to minimum ventilation (ANSI/ASHRAE/IES, 2022) (California Energy Commission, 2022). However, the vast majority of existing buildings have unnecessarily high VAV minimums due to their age.

Dual Maximum VAV Logic

Unnecessarily high VAV minimums are often coupled with the conventional VAV control method referred to as "single-max" logic whereby the minimum flow setpoint applies in deadband and heating modes, and must be set high enough to provide the required heating capacity at design conditions (Figure 12). Better energy and comfort performance, in addition to code compliance, can be achieved using "dual-max" logic, which varies the heating airflow between the required ventilation minimum (typically lower than the 20-50% of cooling max flow used by conventional controls) and the heating maximum airflow (typically about 50% of cooling maximum flow).

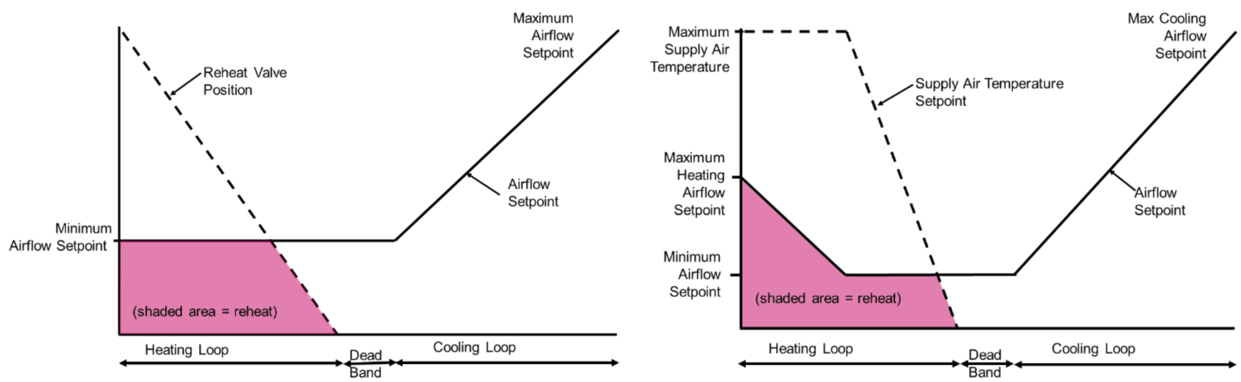
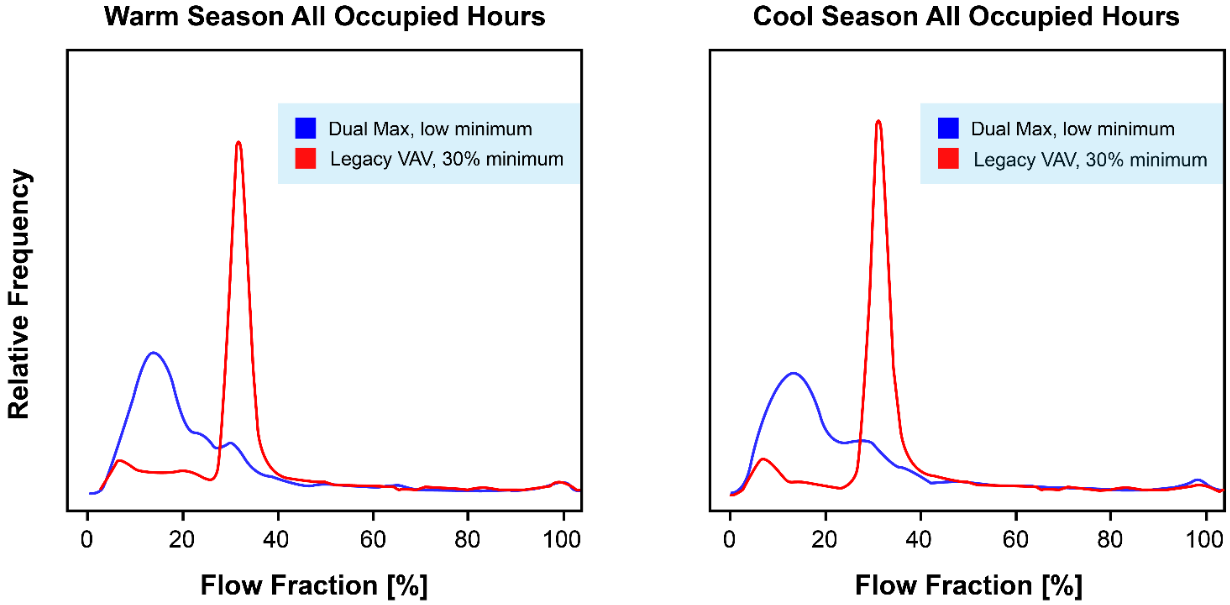


Figure 12 – Conventional Single-Max (left) vs. Dual-Max (right) VAV Control

Reducing zone minimum airflows can save 10 to 30% of HVAC energy use.

The benefits of lowering minimums to the required point for ventilation are well supported by research. ASHRAE research project RP-1515 compared the energy and thermal comfort performance between conventional 30% zone minimums and dual maximum VAV logic with low minimums (Arens, et al., 2015; Paliaga, Zhang, Hoyt, & Arens, 2019). Figure 13 below shows that typical office building cooling loads require zone airflows that are far below 30% of the cooling maximum. In the baseline case (red), in both warm and cool seasons, the frequency plots show that the zones spend the vast majority of their time at the minimum limit of 30%. When the zone minimum is reduced, the zones operate for the majority of the time at lower airflows and still satisfy zone heating and cooling needs. The lower minimums not only reduce fan power, but also result in reductions in system cooling and reheat energy. Overall HVAC energy savings from reducing the zone minimum airflows ranged from 10 to 30% at the study buildings in RP-1515 (Arens, et al., 2015).



(Arens, et al., 2015)

Figure 13 – Measured Zone Airflow Fractions for Dual Maximum vs Legacy VAV Logic

To achieve these benefits, the specifying engineer must set zone minimums accordingly. Ideally, they will calculate the minimum airflow at each zone based on actual area and expected occupancy, rather

> If upon reviewing the zone schedule, the controls installer (or commissioning provider) finds that the zone airflow minimums are consistently more than 20% of the design maximum, they should issue an RFI and ask the designer to consider lower minimums. This is particularly true if each zone's minimum is the same percentage of design airflow, which suggests that the designer did not evaluate minimum airflow requirements individually for each zone.

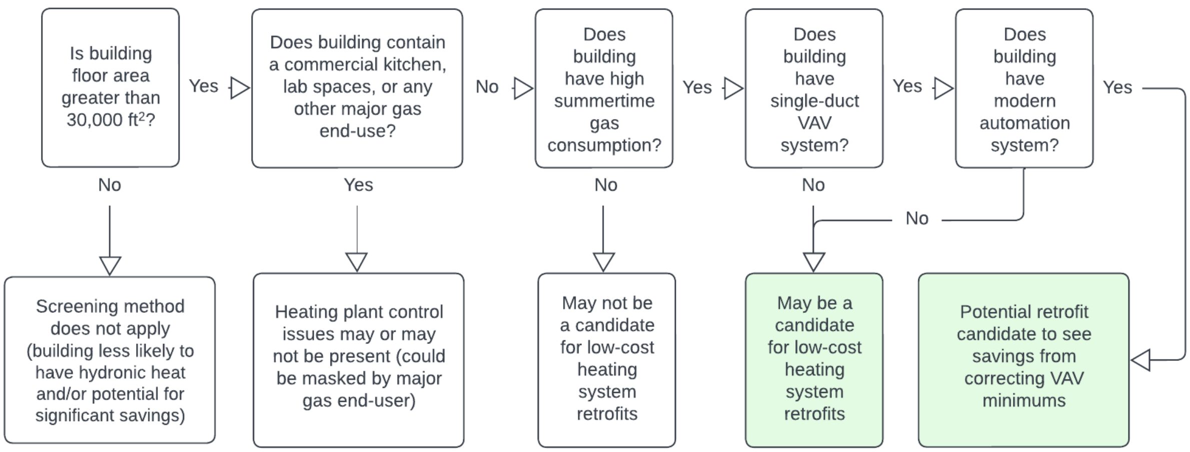
than as a fixed percentage of the design cooling airflow as has been typical historical practice. **Current versions of ASHRAE Standard 90.1 and Title 24 require the minimum airflow in deadband to be no larger than the ventilation requirement** (with some exceptions).

Zone Ventilation Calculations

A free spreadsheet tool is available to streamline the process of determining zone ventilation airflow requirements in accordance with either ASHRAE Standard 62.1 (simplified approach) or California Title 24 (https://tayloreng.egnyte.com/dl/tjSgwpRxaN/Zone_Ventilation_Calculation_Tool.xlsx_). Though the tool may be used for new construction as well, it is targeted for application to existing buildings. The tool takes input information gathered from drawing takeoffs such as VAV schedule data, zone areas, and the types of spaces served to generate a revised VAV schedule that provides airflows and minimum outdoor rates for each zone. The tool also allows for consideration of demand-controlled ventilation (DCV) and occupied standby controls in its selection of minimum airflow values. The zone airflow setpoints selected by the tool provide the user with necessary inputs to follow control sequences in ASHRAE Guideline 36.

Screening Tool

Researchers at the UC Berkeley Center for the Built Environment developed a screening tool designed to identify priority buildings for heating system retrofit or controls improvements (Kemp & Raftery, 2023). The tool is intended to be used with either large or small portfolios, taking basic building information such as building type, size, and gas consumption to create filters to narrow down the buildings to a shortlist of those most likely to present opportunities for retrofits or controls improvements. The filters first identify candidate building types (those without large non-HVAC gas uses such as kitchens or labs), then identify buildings most likely to have heating system operation issues (those with high gas consumption in the summertime, when heating should be minimal). These filtered buildings are further narrowed down to those most likely to have high VAV reheat minimum airflows (those with high summertime gas use which also have single-duct VAV systems). A flow chart for this screening method is shown in Figure 14.



(Kemp & Raftery, 2023)

Figure 14 – Screening Method Flowchart

To test the screening tool, the researchers screened a small portfolio of 22 buildings at California State Polytechnic University, Humboldt. Using the filtering methods described above, and with more detailed knowledge of the buildings provided by the operators, they were able to apply the screening tool to identify 2 buildings as having high summertime gas use and high VAV reheat minimum airflows, with controls systems modern enough for a low-cost controls upgrade (findings were confirmed by building operators and by reviewing the mechanical drawings).

The researchers also screened a large portfolio of 3,318 buildings in Washington, DC. From this large portfolio, the screening tool identified 6 buildings as viable candidates for correcting VAV minimum flows. However, as communication with building operators was not practical for such a large portfolio, the screening of these buildings relied on less granular methods and more unconfirmed assumptions than the one conducted on the smaller portfolio at Cal Poly Humboldt. This shows that while the screening tool can still effectively identify priority buildings in a large portfolio, more accurate and actionable results can be obtained when screening a smaller portfolio with more detailed information available (Kemp & Raftery, 2023).

Limitations

Some factors may limit the extent to which zone minimums can be readily reduced in certain systems without major upgrades or retrofits.

- Pneumatic and configurable (non-programmable) zone controllers may only have the capability of having a single minimum airflow setpoint, which must be set high enough to meet design heating requirements for reheat zones. Some configurable zone controllers can be set to largely achieve dual maximum VAV logic, or a reasonably close approximation[1].
- Electric reheat terminals have a minimum airflow requirement as a safety for enabling the electric resistance coil, where this minimum is generally higher than the corresponding ventilation requirement. This only affects the minimum flow in heating mode (not in deadband), but can be limited by using modulating (silicon controlled rectifier, or SCR), instead of staged capacity control, and electronic airflow switches[2], instead of differential pressure or paddle type switches.
- Packaged air-conditioning units with direct expansion and/or furnace heat may also have relatively high minimum airflow requirements at the system level as safeties. Though these apply at the system level, these thresholds may limit how low the minimums can effectively be set at the corresponding zones and still effectively ensure adequate flow for the system to enable cooling or heating.

3.2 Supply Air Temperature Reset

A significant source of heating energy in buildings occurs through overcooling in VAV systems, which subsequently leads to additional heating energy consumption. This tends to occur, in part, when the supply air temperature (SAT) setpoint is set too low in cooling relative to the actual demand of the zones. Effective demand-based SAT reset strategies can minimize unnecessary heating and cooling energy consumption.

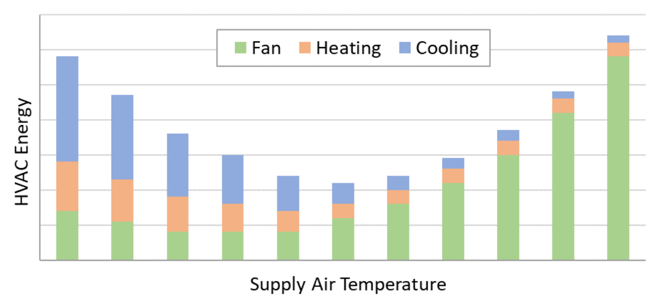


Figure 15 – HVAC Energy and Supply Air Temperature

The optimal SAT setpoint for minimizing HVAC energy consumption while meeting cooling demand is a balance between the different energy end-uses (Figure 15). The SAT setpoint can be reset upward to minimize cooling and reheat energy, but this may come at the expense of increased fan energy. Operating with a higher SAT would require increased airflow for a given cooling load, and fan energy, compared to a lower SAT. Because of the fan laws, at some point the increased airflow requirement causes the resulting increase in fan power to exceed the mechanical cooling savings.

[1] For example, some Siemens TEC controllers have separate variables for CLG FLOW MIN, HTG FLOW MIN, and HTG FLOW MAX, and allow airflow in heating to increase with increasing heating loop outputs.
[2] For example, Thermo-V from Thermo-lec

Trim & Respond Reset

SAT reset sequences typically operate using trim & respond (T&R) logic, which resets the SAT setpoint at the AHU based on the cooling demand of the associated VAV zones. T&R is one of the principal energy savings strategies used in G36 (Taylor S. T., 2015). T&R resets the setpoint by slowly but continuously *trimming* the setpoint upward during periods where there is no demand for additional cooling (i.e., raising the cooling setpoint). This continues until the T&R loop receives a call for cooling from downstream equipment, at which point it responds by resetting the setpoint in the opposite direction (e.g., lowering the cooling setpoint) to satisfy the zone demand. G36 further imposes an upper limit on the SAT setpoint that varies as a function of outdoor air temperature to address the balance between minimizing both cooling and fan energy. When the temperature outside is warmer, there is less potential for airside economizer operation and perimeter cooling loads are likely to be higher, so the G36 SAT setpoint reset strategy drives the setpoint down. Figure 16 illustrates the interacting components of the G36 SAT reset strategies for a few different cases.

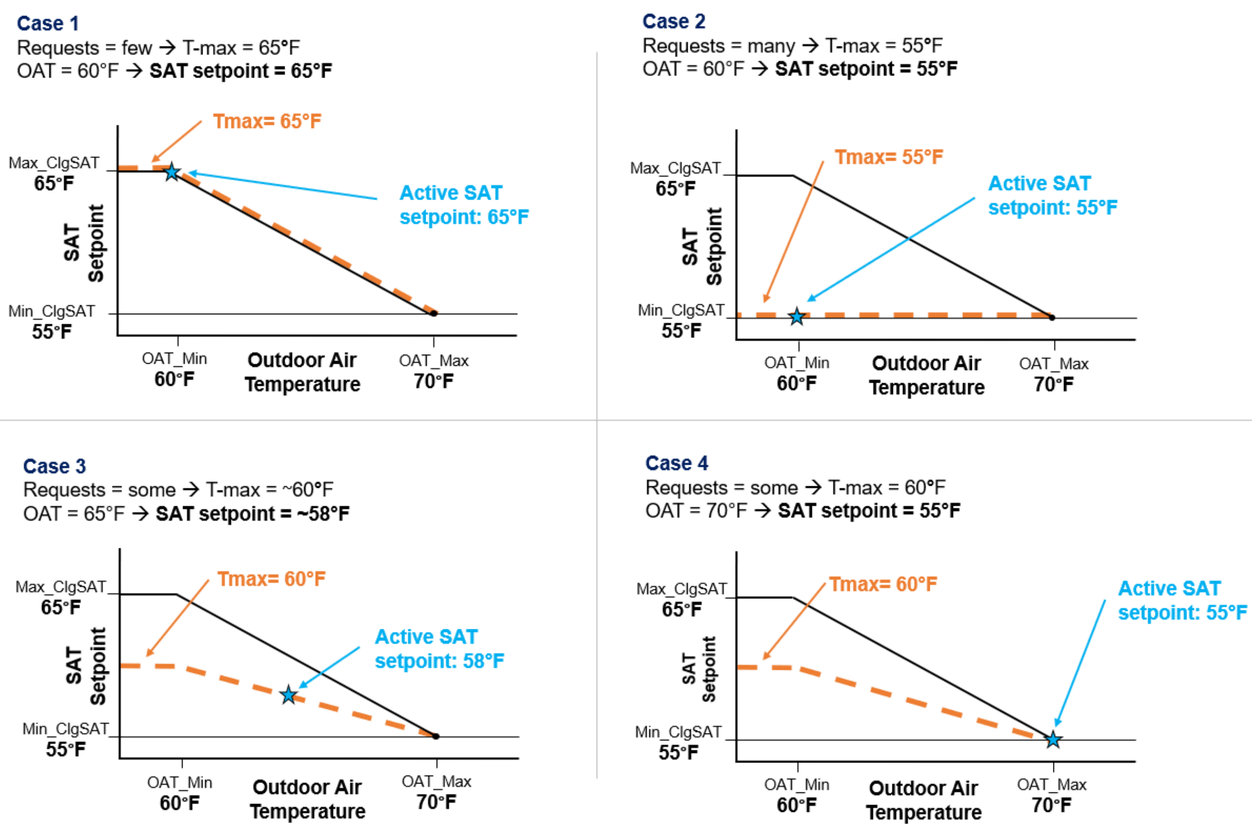


Figure 16 – Examples of SAT Reset by Demand and OAT

Rogue Zones

Rogue zones are individual zones that too-frequently generate requests to a T&R loop. Left unresolved, rogue zones can prevent a T&R loop from resetting effectively, eliminating any energy savings from the reset across the entire system. Rogue zones can occur due to programming error, design error (e.g., undersized VAV box), conditions in the zone (e.g., someone put a coffee pot in front of the thermostat, generating false cooling demand), or operator overrides (e.g., a damper or valve output manually overwritten to 100%).

It is critical to identify and remediate rogue zones. At any point in time, different zones may generate requests, which is normal. A rogue zone is one that generates requests continuously or much more frequently than other zones. Guideline 36 provides logic to easily identify rogue zones by evaluating the cumulative “%-Request-Hours” and comparing this across similar zones. This is made trivial by simple inspection of an effective zone summary graphic (see Figure 17 for an illustration) but is otherwise very difficult to do effectively. The “%-Request-Hours” value should be evaluated separately for each type of Request—static pressure, heating, and cooling—as a zone may exhibit rogue behavior relative to one parameter but not others. Any request % value that greatly exceeds those for other similar equipment should be investigated.

												Static Pressure Reset		
	Operating Mode	Zone Temp ($^{\circ}$F)	Htg SP ($^{\circ}$F)	Clg SP ($^{\circ}$F)	Airflow (CFM)	Airflow SP (CFM)	Damper Pos (% Open)	HW Valve Pos (% Open)	DA Temp ($^{\circ}$F)	DA Temp SP ($^{\circ}$F)	CO2 Level (PPM)	Requests	Cumulative Req Hrs (%-req-hrs)	Importance Multiplier
VR-19-1	Deadband	73.3	71	74	342	330	29	0	56.9	55	n/a	0	10	1
VR-19-2	Cooling	73.9	71	74	278	280	31	0	57.2	55	n/a	0	2	1
VR-19-3	Cooling	73.1	70	73	275	279	24	0	57.2	55	n/a	0	1	1
VR-19-4	Deadband	72.9	71	74	245	235	16	0	57.2	55	565	0	6	1
VR-19-5	Deadband	72.3	71	74	184	180	25	0	58.7	55	n/a	0	3	1
VR-19-6	Cooling	70.8	68	71	1147	1144	45	0	57.2	55	n/a	0	6	1
VR-19-7	Deadband	73	71	74	102	100	32	0	59.1	55	n/a	0	1	1
VR-19-8	Deadband	72.9	71	74	252	250	41	0	57.5	55	n/a	0	6	1
VR-19-9	Deadband	73.1	72	75	508	500	15	0	58.4	55	n/a	0	1	1
VR-19-10	Deadband	72.4	71	74	308	300	31	0	57.2	55	n/a	0	9	1
VR-19-11	Deadband	73	71	74	290	270	32	0	58.1	55	n/a	0	1	1
VR-19-12	Deadband	74.3	72	75	368	360	11	0	57.1	55	n/a	0	3	1
VR-19-13	Deadband	73.7	71	74	512	500	32	0	58.6	55	505	0	14	1
VR-19-14	Cooling	73.8	71	74	673	675	54	0	56	55	n/a	0	$\boxed{86}$	1

Rogue zone

Figure 17 – Rogue Zone Identification

For rogue zones that cannot be easily remedied, consider locking them out of the T&R loop or increasing the number of ignores, so the reset can correctly respond to variable demand in other zones. Locking out a zone is most easily done by setting the rogue zone’s Importance Multiplier factor to zero. Either approach involves a tradeoff between maintaining airflow and temperature control vs. minimizing HVAC energy use. Where there are practical limitations to resolving mechanical design issues, consider the use of personal comfort stations. Desk fans are inexpensive, and low energy and can improve occupant comfort in warm conditions by locally increasing air movement. Chair heaters and electric space heaters may also be appropriate in cold conditions if they allow the overall HVAC system to operate more efficiently.

This process of screening for rogue zones is critical to good system performance because most of the energy efficiency benefits may be lost if rogue zones are allowed to drive the resets.

The Advanced BAS Best Practices Guide provides more detailed guidance on T&R logic, including guidance on setting and tuning T&R parameters, and common causes of rogue zones (Cheng, Singla, & Paliaga, 2022).

3.3 Improve Morning Warmup Performance

For many buildings, a large portion of the annual heating energy consumption occurs during morning warmup periods. Morning warmup is the period prior to scheduled occupancy when the HVAC system is recovering space temperatures from overnight or weekend setbacks during the heating season. There are multiple factors that can impact heating system performance and energy consumption during morning warmup.

Optimal Start

Optimal start is a common strategy to minimize HVAC system run time, waiting as long as possible before starting the HVAC systems in the morning, and then recovering space temperatures as fast as possible prior to the beginning of the scheduled occupancy period. During mild weather the recovery period may be very short, compared to longer recovery periods during colder weather conditions. Typical strategies use learning algorithms and real time readings of outdoor air temperature, zone temperatures, and zone temperature setpoints to determine the start time. Most BAS manufacturers have internalized optimal strategies built into their systems, some proprietary and some with published algorithms, and optimal start is required by building energy codes (ANSI/ASHRAE/IES, 2022) (International Code Council, 2021) (California Energy Commission, 2022).

Unfortunately, verbal feedback from a range of stakeholders suggests that optimal start is rarely implemented effectively and is often quickly replaced by conservatively early scheduled start times. Possible reasons include:

- The manufacturer's optimal start logic simply does not work to correctly determine the required recovery duration. If space temperatures are not consistently recovered to comfortable conditions, leading to occupant complaints, building operators may be compelled to disable optimal start and/or simply schedule the system start times much earlier to ensure comfort conditions are maintained.
- The optimal start tuning was not completed properly or does not adapt to changes in the building over time. The tuning or learning process is often skipped during system start up or simply cannot be completed properly because of seasonality (e.g., setting up a system during the summer would not allow for tuning of optimal start for winter warmup conditions), leaving the default settings in place. If the system fails to recover to comfortable temperatures on time, the logic may be disabled or overridden as above.
- Lack of awareness by operators of optimal start capabilities. Some operators simply are not aware that the BAS includes optimal start logic, and schedule the HVAC systems to start early in order to consistently recover on time, as if there were no optimal start feature. The end result of this lack of awareness is the unnecessary extension of system run times. For example, to consistently recover in time for expected occupancy at 8 am, an operator might schedule the system to start every day at 5 am (3 hours early). If the optimal start is also enabled, it may start up the HVAC systems as early as 2 am in order to try to recover spaces to comfortable temperatures by 5 am (instead of the intended 8 am).

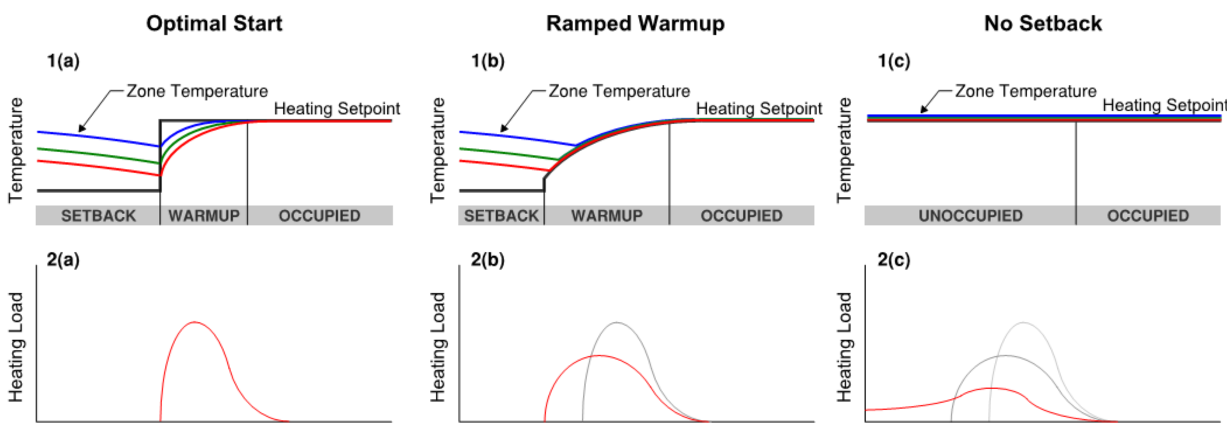
The potential negative impacts of non-functional optimal start logic and fixed, conservatively early start times are increased energy HVAC consumption.

Dedicated Warmup Mode

Guideline 36 defines a dedicated warmup mode of operation that differs from other operating modes. In particular, a critical factor is that warmup mode occurs prior to the period of expected occupancy. Because buildings are expected to be unoccupied during morning warmup, ventilation is not required and should not be provided, in order to avoid the energy use with tempering of the ventilation air and to ensure that available system heating capacity is available for recovering space temperatures. When morning warmup is simply accomplished by scheduling the system start time early by a fixed duration every day (e.g., starting at 5 am every day to be recovered by 8 am), the end result is that the HVAC system is unnecessarily providing minimum ventilation during the unoccupied warmup period, increasing heating energy use and prolonging the duration of the recovery period.

Ramped Morning Warmup

The conventional wisdom of deploying optimal start to recovering as fast as possible to minimize energy use may not actually be appropriate for modern HVAC systems and may have some negative consequences for heating system performance. Much of the original research around optimal start is decades old, prior to the widespread use of modulating capacity control in HVAC equipment, like variable speed drives for fans and modulating boilers. Fans and heating systems with modulating capacity control generally operate more efficiently at part load conditions than at full load. Recovering as fast as possible also results in creating as large of a peak heating load as possible, with negative consequences of unnecessarily staging on additional equipment, preventing hot water supply temperature reset, furthering a perception for the need for more capacity. For all-electric heating plants, larger peak heating loads may also impact utility peak demand charges in some climates and may eventually impact grid capacity as more buildings shift to become all-electric.

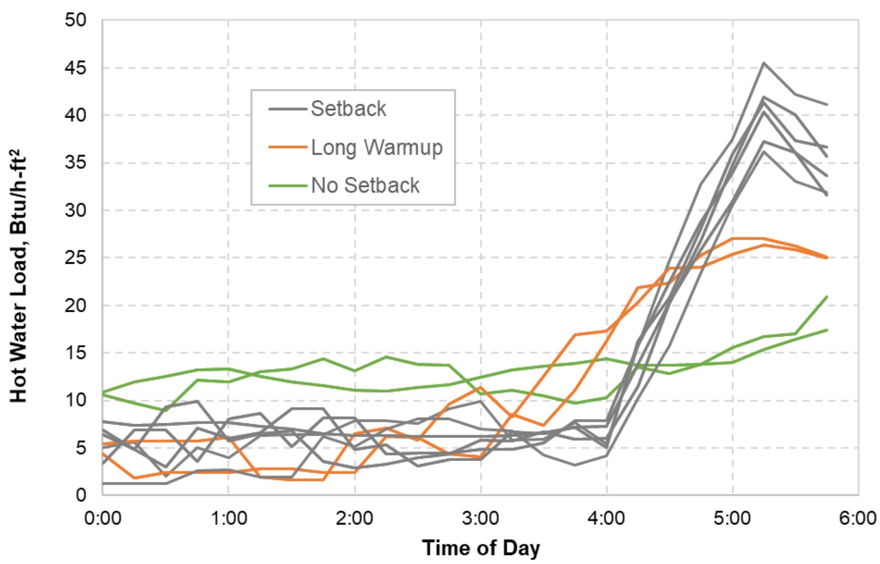


(Cheng, Raftery, & Wendler, 2024)

Figure 18 – Warmup Strategies and Associated Heating Loads

Applying a ramped recovery approach that extends the morning warmup period over a longer duration may help reduce heating peak loads and improve HVAC system efficiency. Figure 18 schematically illustrates three alternative warmup concepts and the associated heating loads. The optimal start strategy deploys a step change in heating setpoint at the onset of the warmup mode (1a) and other control strategies to recover as fast as possible, resulting in as large of a peak heating load as possible (2a). The ramped warmup instead increases the zone heating setpoints according to a decaying exponential rise (1b), extending the duration of warmup and reducing the peak heating load (2b). In the extreme case with no night setback (1c), a constant heating setpoint is maintained, eliminating the morning recovery load (2c) (Cheng, Raftery, & Wendler, 2024). Note that an earlier start to the warmup period but with a conventional step change in setpoints would simply shift the (2a) peak to be earlier without significantly impacting the magnitude of the peak. The peak in (2b) is flattened by limiting the heating demand in each individual zone and varying the time of onset of heating mode based on when each zone temperature intersects with the rising heating setpoint.

The ability to significantly limit peak heating load in morning warmup was field demonstrated with zone setpoint strategies similar to, but not exactly the same as, shown in Figure 18. Figure 19 shows measured heating hot water loads for different morning warmup strategies (Cheng, Raftery, & Wendler, 2024). Each line represents the load for a particular day with similar average outdoor air temperatures (i.e., similar recovery heating loads). The baseline “setback” approach used a decaying exponential rise but had a limited and untuned warmup duration that compressed the recovery period into a short period, resulting in consistently high peaks loads. The “long warmup” approach followed the same decaying exponential rise strategy but with a longer duration and tuning to effectively spread out the recovery period and significantly reduce the peak heating load. In the extreme case, the “no setback” approach maintained a constant heating load overnight, eliminating the morning warmup recovery load altogether. The “no setback” approach significantly increased heating energy use with higher envelope loads maintained overnight, compared to the other strategies, but represents a possible control intervention to meet comfort conditions where capacity is limited (e.g., a building could operate the “no setback” approach only on the most extreme days, while operating with “long warmup” the vast majority of the year, and have a much smaller than typical HHW plant) or there are reasons to significantly curtail peak heating load.



(Cheng, Raftery, & Wendler, 2024)

Figure 19 – Field Demonstration of Heating Loads for Various Warmup Strategies

The “long warmup” approach did not measurably reduce heating energy compared to the baseline “setback” approach in (Cheng, Raftery, & Wendler, in press) but only very limited data were available. The building’s condensing boiler plant was also controlled in a way that prevented taking advantage of improved efficiency from better condensing conditions. Though more study is needed to evaluate this approach, it suggests that conventional optimal start and warmup strategies should be re-evaluated and that ramped warmup approaches may be an important consideration for all-electric projects, particularly for retrofits. If peak loads can be reliably reduced, all-electric heating equipment can be smaller, reducing first costs and space requirements. Furthermore, the reduction in zone heating loads may allow for existing heating coils and pipe distribution to be reused in all-electric retrofits. Air-to-water heat pumps can generally only produce water temperatures of 120 to 130°F, compared to typical boilers that are designed to generate 140-160°F (condensing) or 180°F (non-condensing). The resulting reduction in coil and distribution capacity due to lower temperature supply water may otherwise warrant costly and disruptive replacements, but this study suggests that peak heating loads may potentially be reduced through simple control strategies to avoid the need for prohibitive heating system replacements.

3.4 Demand Controlled Ventilation

Demand-controlled ventilation (DCV) and occupied-standby controls are energy efficiency measures that allow zones to reduce their ventilation airflow rates from design levels during periods of partial occupancy based on real-time sensing. Both strategies reduce zone level airflow rates as well as system-level outdoor airflow rates during partial occupancy. This in turn contributes to heating energy savings by reducing the amount of outdoor air that needs to be conditioned and zone airflow that needs to be reheated.

DCV is required by Standard 90.1 and Title 24 for spaces with high design occupant densities. Carbon dioxide (CO2) is a bioeffluent that is produced through respiration and is a common indicator used for occupant sensing for DCV. ASHRAE Standard 62.1 and California Title 24 set zone ventilation requirements with area- and occupant-based components that are compiled in different ways to determine the system-level outdoor airflow rates. Though the control strategies for each ventilation standard are slightly different, both approaches reduce the zone ventilation requirement to the area-based component during periods of low occupancy. As the zone CO2 concentration increases, zone airflow and ventilation rates increase to respond to the increase in occupancy. The system-level outdoor air requirements are also reduced during periods of partial occupancy and respond accordingly as CO2 concentrations increase. Guideline 36 provides detailed control sequences implementing DCV in accordance with Standard 62.1 and Title 24 (ASHRAE, 2021). Note that in addition to indicating which zones are to be equipped with CO2 sensors, designers must also indicate zone level outdoor air ventilation requirements.

Occupied-standby is a control strategy required by Standard 90.1 and Title 24 that reduces the area-based ventilation component to zero when spaces are sensed to be vacant. In addition, occupied-standby applies temperature setbacks to the space heating and cooling setpoints, reduces zone airflow to zero when in deadband, and reduces the system-level outdoor air requirement by the corresponding amount. A simulation study found that occupied-standby mode saved 20 to 40% of HVAC energy use across different U.S. climate zones for a prototype medium office building that was modified to include detailed thermal zones and stochastic occupancy profiles (Pang, et al., 2020).

3.5 Discharge Air Temperature Control

Maximum Discharge Air Temperature Limit

One facet of the dual maximum VAV logic in ASHRAE Guideline 36 is that the hot water reheat valve is controlled to maintain a resetting discharge air temperature (DAT) setpoint, rather than controlling the valve directly based on the heating loop as per conventional practice (see Figure 12). Excessively high DAT temperatures can lead to stratification and hinder mixing of heating supply air with the room air, which would impede effective heating of the spaces and ventilation effectiveness. Controlling to a maximum DAT setpoint of 90 to 95°F is required by Standard 90.1 and Title 24.

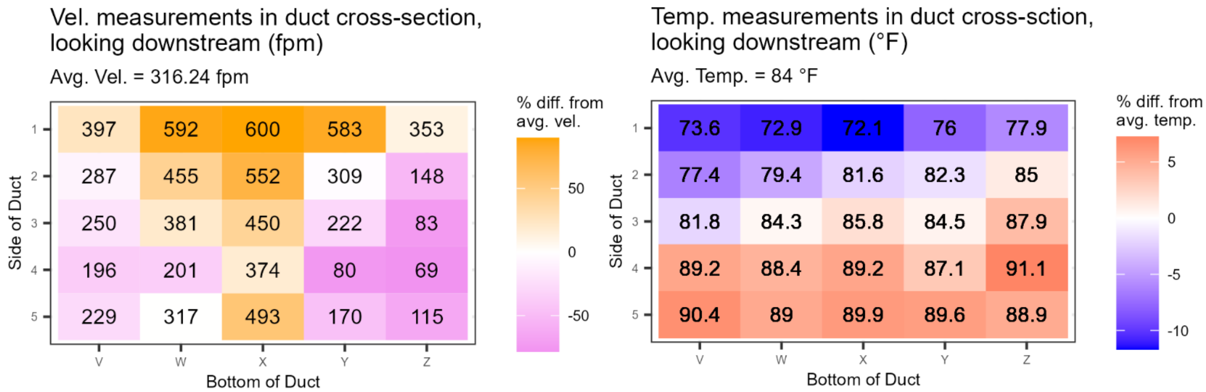
Controlling 2-way reheat valves to DAT setpoints has additional benefit of allowing the hydronic loop to be self-balancing, each valve will only take as much flow as is needed to maintain DAT at setpoint (Taylor S. T., 2002) (Taylor S. T., 2017). Leaving the system unbalanced enables better reset of the hot water supply temperature setpoint by allowing for higher-than-design water flows when needed at certain zones during part load conditions. Further, compared to letting the coil be "wild" at times (valve commanded full open by zone heating loop, flow dictated by hydronic loop conditions), maintaining a DAT setpoint will limit the water flow and improve delta T, reducing pipe losses and pumping energy, and improving condensing efficiency where applicable. This effect can be readily demonstrated with coil selection software. For a standard 8-inch VAV box with 2-row coil, 140°F supply water temperature, and 285 cfm of 65°F entering air:

- Controlling the DAT to 90°F requires 0.44 gpm with a waterside ΔT of 36.4°F and leaving water temperature of 103.6°F.
- A wild coil producing 105°F requires 2.13 gpm with a waterside ΔT of 11.9°F and leaving water temperature of 128.1°F (barely condensing).

Discharge Air Temperature Stratification

A 2022 study evaluated temperature stratification in VAV reheat boxes under varying conditions (Wendler, Raftery, & Cheng, 2023). Temperature stratification is a challenge that can impact system performance and efficiency. If temperature readings are low compared to the actual average temperature, the high DAT supply air may not mix effectively with the room air, with negative impacts to zone heating capacity. If temperature readings are high compared to the actual average temperature, the actual heating capacity may be insufficient to meet loads. Operator interventions to overcome capacity issues in either case may further impact system performance and efficiency (e.g., increasing airflow setpoints, overriding setpoints, etc.).

Across various combinations of VAV box configurations and operating conditions, 5x5-point velocity traverses were taken at the coil outlet and 5x5-point temperature measurements were taken further downstream in the cross-section of the duct. The results show that at more-closed damper positions, air velocity tends to increase toward the top of the duct. The areas of higher velocity coincide with lower discharge temperature and vice-versa, revealing that both airflow and temperature become more stratified at more-closed damper positions: faster, colder air occurring at the top of the duct and slower, warmer air occurring at the bottom of the duct. The pattern of velocity stratification toward the top of the duct is related to the damper position and rotation, with typical VAV dampers rotating open from the top, directing air upward at partially open positions. Figure 20 shows the results for a representative test (8" box, 3-row oversize coil, 120°F HWST, 37% open damper).



(Wendler, Raftery, & Cheng, 2023)

Figure 20 – Typical Velocity (left) and Temperature (right) Stratification in VAV Reheat Boxes

These results have important implications for reheat boxes using single-point DAT sensors. With this degree of temperature stratification in the outlet plenum, the accuracy of discharge air temperature readings depends heavily upon the mounting location of single-point temperature sensors. Where used, single-point sensors should be mounted such that the tip is as close to the centerline of the duct and as far from the coil as possible. This may require use of multiple probe lengths for different size coils, where common practice is to instead provide a single probe length for all VAV boxes. An alternative solution is to use rigid averaging temperature sensors. Rigid averaging sensors generally have 4 thermistors equally spaced across the length of a rigid probe. Though the sensors are more expensive than single-point probes, the labor costs are similar and the improved control with averaging probes may reduce the risk of call backs and heating complaints.

The representativeness of single-point DAT readings also depends on the damper position, with less representative readings taken when there is more stratification due to partially closed dampers. More-closed damper positions are common when boxes are in heating due to the lower airflow setpoint limits. This effect is even more justification for the use of static pressure setpoint reset sequences to help keep damper positions as open as possible and to reduce the risk of airflow and temperature stratification.

4

Improving
Plant Efficiency

- Controls
- Hot Water Loop Configuration
- Coil Selection
- Pipe Insulation
- Boiler Sizing and Selection
- Commissioning

4 Improving Plant and Distribution Efficiency

In this chapter, we describe a range of design considerations and strategies for improving hot water plant and distribution efficiency.

4.1 Controls

Hot Water Supply Temperature Control

Hot water supply temperature is a factor that strongly affects both equipment efficiency and distribution efficiency.

- For condensing boilers, achieving condensing conditions and maximizing condensing efficiency requires low return water temperatures, which in turn requires low supply water temperatures. Effectively resetting hot water supply temperatures during part load conditions can be an effective way to maximize condensing potential.
- For air-to-water heat pumps (AWHP), the equipment efficiency is directly tied to supply water temperatures, with the coefficient of performance (COP) decreasing at higher temperatures, and most heat pumps effectively limited to supply temperatures of about 120 to 130°F. Effective supply temperature resets may be critical for maximizing the annualized COP for AWHPs, even if the COP at design conditions is relatively low.
- Distribution losses in heating hot water piping increase with higher water temperatures, as described in Section 0. Limiting maximum supply water temperatures in design and using setpoint resets to minimize temperatures in operation can help limit the impact of these losses for heating energy, as well as the corresponding cooling energy to handle those additional loads in some conditions.

ASHRAE Guideline 36 provides control sequences for resetting the hot water supply temperature reset using T&R logic, with requests generated by hot water valve position (ASHRAE, 2021). Important factors for achieving effective setpoint reset include:

- Tuning the T&R parameters for the project design conditions. Most implementations of Guideline 36 simply use the default parameters, rather than adjusting and tuning them to the project-specific needs. An addendum to the 2021 version of Guideline 36 changed the default number of ignores from a fixed value of 2 to a value that is instead determined as a percentage of the total number of associated zones in the reset logic. The selected number of ignores is a tradeoff between maximizing energy efficiency (more ignores) and meeting demand (fewer ignores) but generally should scale as a function of the size of the system. The Advanced BAS Best Practices Guide provides a succinct summary of recommendations of how to adjust T&R parameters for project conditions (Cheng, Eubanks, & Singla, 2022).
- Monitoring for rogue zones. See Section 3.2.

For many buildings, most of the heating energy is consumed during the morning warmup process to recover space temperatures to the occupied heating setpoints. During the first hour or two of hot water plant operation, limiting the maximum hot water supply temperature setpoint may help ensure that condensing is achieved for condensing boiler plants. Because warmup occurs prior to occupancy, any capacity shortfall from the lower supply temperatures can be overcome by extending the duration of the warmup period to meet the recovery load. With the Guideline 36 T&R approach, this limiting approach can be achieved by setting the initial setpoint SP0 to a value lower than design, and increasing the delay timer Td from the default of 10 minutes to a value of 1 hour or longer.

Boiler Plant Staging

Many boiler plants are enabled manually, based on a schedule, or when the outdoor air temperature is below a threshold limit. With digital controls, boiler plant operating hours can be minimized by instead enabling the plant based on demand from hot water coils and valves. ASHRAE Guideline 36 includes control sequences that enable the boiler plant when “plant requests” exceed an adjustable minimum threshold (ASHRAE, 2021). Attention should be paid to only enabling the plant when there is enough demand to minimize excessive short cycling (Peterson, 2018). For example, wait to enable the plant until several reheat zones are demanding heat, rather than based on any single zone. Short cycling can significantly reduce system efficiency and lead to premature equipment wear.

Many boilers come with optional factory controls that manage equipment staging. These factory controls stage on lag equipment when the supply temperature setpoint falls below the setpoint for a certain amount of time. Though convenient and simple, this strategy does not take into account whether the additional capacity of the lag boiler is actually needed; the factory controls generally do not know system flow and cannot evaluate the system load when staging equipment up or down. Guideline 36 provides control sequences that stage on additional boilers when it is more energy efficient to do so, in addition to when required due to loss of temperature control. The Guideline 36 logic also provides flexibility to provide tuning to help prevent unnecessary short cycling. For example, a lag boiler might always be enabled in the first few minutes of plant operation each day while the cold water in the loop is initially being warmed up and the supply temperature setpoint cannot be maintained. The loss of setpoint control in this transient condition is not necessarily a sign that the actual building heating load is high enough to warrant additional capacity. Though the additional capacity from the lag boiler may help provide faster recovery, there are energy penalties if the lag boiler only runs briefly before cycling off. These penalties include purge cycle less and the extra heat consumed to warm the mass of the boiler and the water contained within it, which is lost to ambient when the boiler cycles off. The Guideline 36 staging logic can be tuned with a low initial hot water supply temperature setpoint, plus a longer delay and larger error threshold for the failsafe stage up conditions to help prevent the lag boiler from being enabled during this transient condition at startup.

Boiler plant staging thresholds should also be tuned to provide stable operation and maximize staging efficiency. Condensing boilers are generally more efficient at part loads above the minimum turndown limit than at full load, so staging on additional lag boilers earlier may improve system efficiency. However, staging on lag equipment may be detrimental for efficiency if the need is brief and the lag boiler cycles back off after a short period. A staging cycle includes the negative energy effects of heating up the mass and water within a cold boiler, as well as pre and post purge cycles that exhaust unburned fuel and cool down the boiler mass.

4.2 Hot Water Loop Configuration

There are a number of factors with the design of the hot water distribution that affect system efficiency and performance.

Variable Flow

Variable flow distribution is a critical requirement for hot water plants with condensing boilers, and an important factor for efficiency for any hot water system to minimize distribution losses. Most condensing boilers have a minimum flow requirement, but achieving the lowest possible entering water temperature requires minimizing bypass flow as much as possible to avoid blending hot supply water with the return water. ASHRAE Standard 90.1 and California Title 24 require condensing boiler plants for certain system sizes, and require that systems be designed and operated to maintain return water temperatures below 120°F, such as by selecting appropriate coils and minimizing bypass flow.

Achieving low entering water temperatures for condensing requires minimizing bypass flow as much as possible.

Bypass flow commonly occurs in many locations in hot water distribution systems.

Primary-secondary hot water distribution. Primary-secondary hot water distribution is incompatible with condensing boilers. This is a strategy that maintains constant flow across the boilers in the primary loop, with a secondary distribution loop that is either variable or constant flow to the hot water coils. With a constant primary, variable secondary distribution, the primary supply water will often be blended with the secondary return water, limiting or preventing condensation at the boilers.

Variable flow, primary-only hot water loops. Variable flow, primary-only hot water loops often include a bypass with control valve to maintain boiler minimum flow requirements (Figure 21). For buildings with a high degree of load turndown, the boiler minimum flow can become a limiting factor for a large portion of the time. Attention should be paid to selecting boilers with low or no minimum flow requirements as well as controlling systems to minimize bypass flow as much as possible. The boiler minimum flow often is not indicated on equipment schedules and often incorrectly configured in control systems. Figure 22 shows the hot water return temperatures at the building return and the boiler inlet (with the rise in boiler inlet temperature due to bypass flow) for one plant as a function of system flow, where the bypass is controlled to a minimum flow setpoint of 100 gpm. For system flows above that threshold, the building hot water returns directly to the boiler and there is no bypass. However, the system operates with the bypass open nearly all of the time to maintain the minimum flow requirement, diluting the boiler inlet water with bypassed supply flow. The building return water temperature is predominantly below the condensing threshold (as indicated by histogram at right in Figure 22), but the high degree of bypass causes the boiler inlet temperature to instead be well above the condensing threshold, resulting in a median temperature rise of 13°F. Though there are a range of reasons why condensing boilers may not be operated under condensing conditions, the boiler minimum flow requirement is the sole reason in this case.

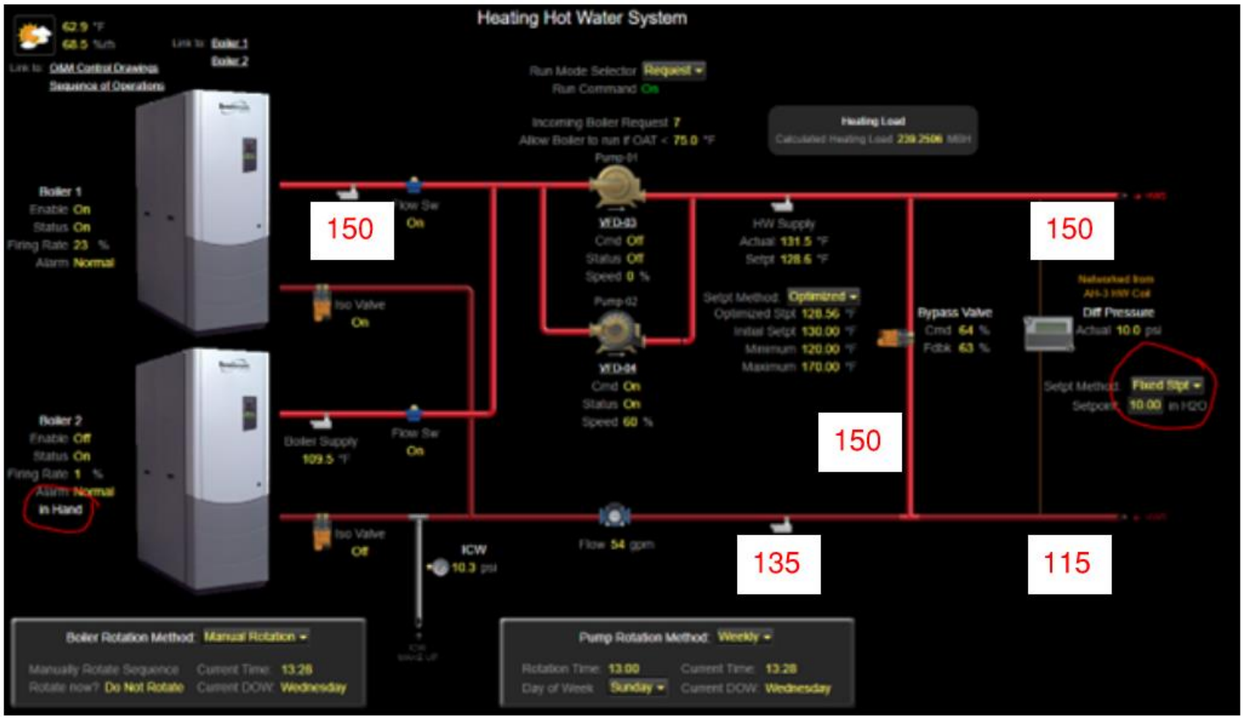


Figure 21 – A Primary-only, Variable Flow Hot Water Loop with Bypass

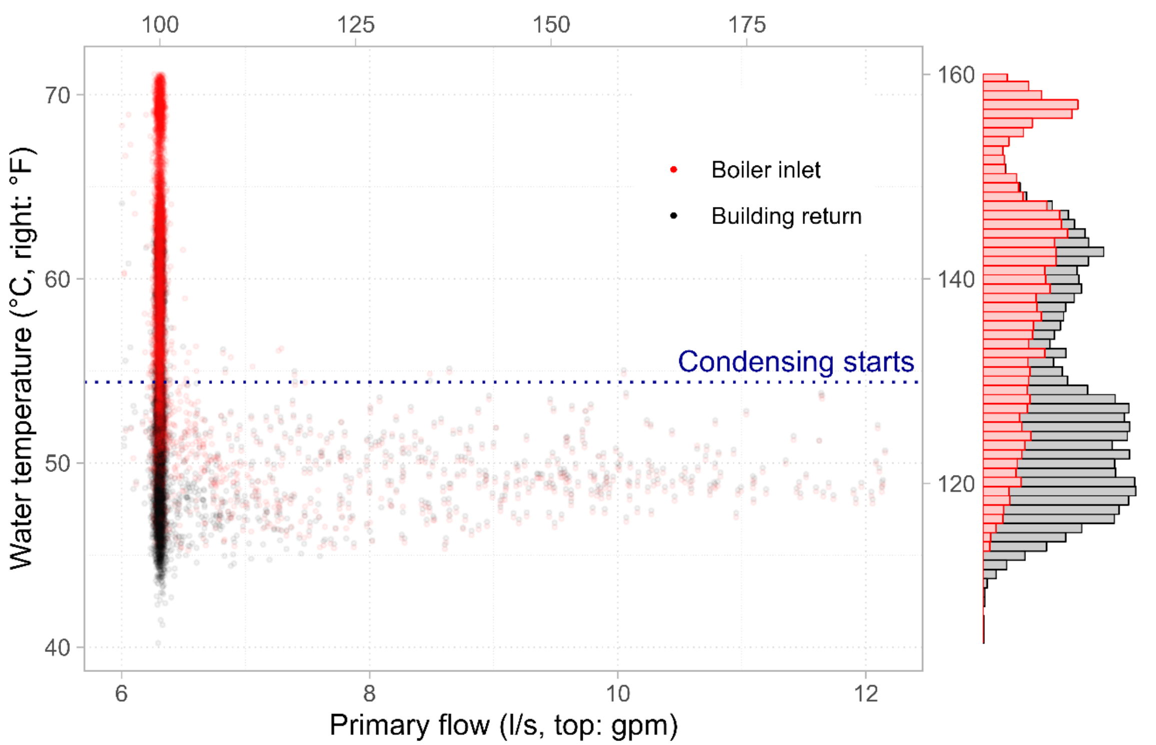


Figure 22 – Impact of Boiler Minimum Flow on Condensing Boiler Inlet Water Temperature

3-way valves. Hot water distribution systems were commonly designed as constant flow with 3-way valves at the coils in the past. Today, variable flow distribution is more common in commercial buildings with most hot water coils served by 2-way valves, and occasional 3-way valves. Potential reasons for the use of 3-way valves and counter arguments are noted below:

- Avoid deadheading the pumps. With digital controls at each of the valves, this risk can be prevented instead by coordinating the pump operation with coil demand, disabling the pumps if all associated valves are shut.
- Help keep loops warm. During off peak conditions, hot water valves may periodically open and close in different locations. With only 2-way valves and purely variable flow, the supply water in some branches of the piping may cool down to ambient at times when there is no flow and as heat is lost to the surroundings. Maintaining an end-of-the-line 3-way valve or bypass valve at the end of each branch helps keep the loop warm and instantly ready for any demand for heat. In a situation with only 2-way valves, it would not take long for a cold loop to recover once there is active flow and under these conditions the time constant of most spaces would not require hot water to instantly be available. Given the continuous distribution losses associated with keeping the loop warm, this justification for end-of-the-line bypass is not warranted.
- Engage the thermal mass of the water in the piping. 3-way valves are commonly used to help engage the mass of the water in the piping to improve system stability. Close coupled systems often suffer from excessive boiler short cycling during low load conditions. An alternative approach to increase circulating water system volume is to use a buffer tank located in the return or minimum flow bypass leg at the plant. This approach provides increased system volume at low load conditions without the penalty of increased pipe distribution losses that occurs when relying on the pipe volume and 3-way valves, assuming that the buffer tank is insulated (Peterson, 2018).
- Maintain the boiler minimum flow requirement. Since the boilers may also have minimum flow requirements as well, many designers opt to use end-of-the-line 3-way valves or bypasses to meet this need. A disadvantage with this approach, compared to a bypass at the plant, is that the bypasses are uncontrolled or not directly controlled to maintain the minimum flow at setpoint. When control valves elsewhere in the system are open and meeting the minimum flow requirement, these additional bypasses will continue to flow. The additional blending of supply water into the return stream may negatively impact the efficiency of condensing boiler systems. Using end-of-the-line bypasses to meet minimum flow requirements also increases pipe distribution losses.

Some newer models of condensing boilers, such as the Lochinvar Crest, have a minimum flow requirement that varies as a function of the boiler firing rate. Rather than setting a fixed minimum flow setpoint, such boiler would allow for a variable setpoint to minimize bypass at part load conditions.

Balancing

Common practice is for designers to require that water flows to hot water coils be balanced to meet design flow requirements. Balancing is nominally done to ensure that each coil is able to receive its share of flow, and so that coils closest to the pumps do not take disproportionately more flow than ones that are further away. One challenge with this strategy is that the calculated flows are not necessarily accurate. Designers must make a number of assumptions when performing load calculations that may or may not match actual operating conditions. Balancing may also not be needed with variable flow systems using 2-way valves. If 2-way valves are controlled to maintain discharge air temperature at setpoint, they will only take as much flow as is required and the system effectively becomes self-balancing (Taylor S. T., 2002) (Taylor S. T., 2017).

Further, balancing water flows may be detrimental to a system's ability to reset the hot water supply temperature setpoint during part load conditions. For a given heating load, more flow will be required when supply water temperatures are lower, with some coils occasionally requiring more than design flow at times, even during part load conditions. A balanced system may prevent these coils from receiving increased flow, thereby limiting how much the supply water temperature can be reset.

4.3 Coil Selection

Proper coil selection to increase the waterside temperature difference (ΔT) of reheat coils has multiple interactive benefits, including improving compatibility with lower design hot water temperatures (120-140°F) for condensing boiler or all-electric plants, reduced distribution losses due to reduced flows and lower return water temperatures, and improved efficiency for condensing boiler plants. Improving ΔT can also reduce flow rates, pump size, and pipe sizes. Designers can increase waterside ΔT by increasing the heat transfer surface area of the coils such as with oversized coils, higher coil row counts, and high-capacity coils (with 12 fins/inch as opposed to the typical 10 fins/inch). Oversized coils, see (Taylor S. T., VAV Box Duct Design, 2015), provide improved ΔT at a lower airside pressure drop, whereas the two other strategies increase airside pressure drop. The performance of these coil options can be evaluated with data from standard VAV box selection software. For a 2-row coil with an 8-inch inlet and given heating load, Figure 23 shows the waterside ΔT as a function of hot water supply temperature for different coil (standard 10 fins/inch vs. high capacity 12 fins/inch) and box casing options (standard vs. oversize).

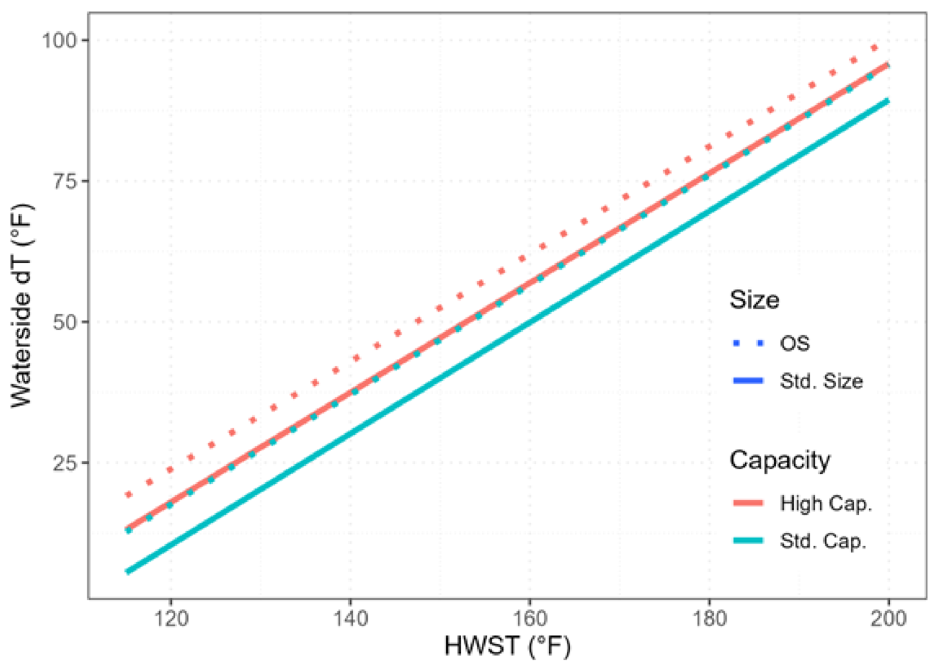
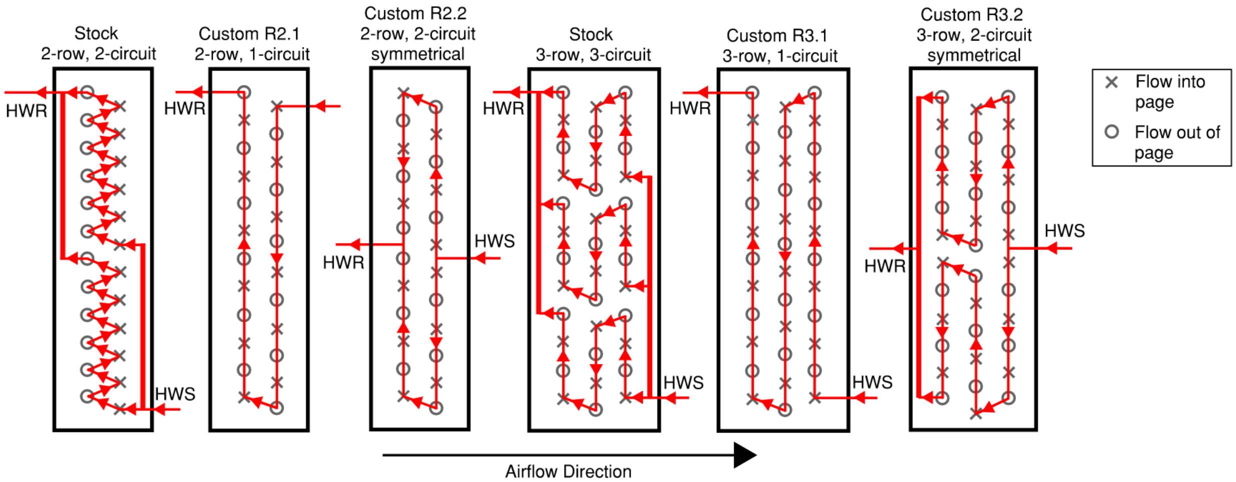


Figure 23 – Performance of Various 2-row VAV Reheat Coils

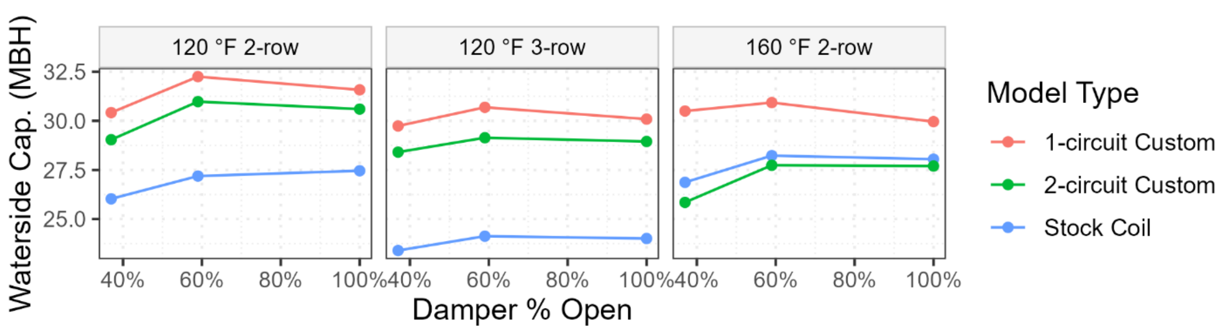
Another way that waterside ΔT can be increased in reheat coils is by improving the coil circuiting. A lab study evaluated the performance of standard (or "stock") 2-row and 3-row reheat coils against custom circuited alternatives (Wendler, Raftery, & Cheng, 2023). The custom designs focused on changing the coil circuiting to allow for reduced circuit count, greater symmetry in heat distribution, and to eliminate paths of water flow that run in parallel flow (as opposed to counterflow) with the airstream. Figure 24 depicts the custom and stock coil circuiting configurations that were evaluated.



(Wendler, Raftery, & Cheng, 2023)

Figure 24 – Custom Coil Circuiting Designs

Figure 25 shows the results of the custom coil tests, comparing the waterside capacity of the custom coils to that of the typical stock designs, with results broken out by HWST and coil row count. Changing the coil circuiting substantially improved low-HWST capacity over the stock designs, especially for the single-circuit custom designs which had capacity increases of nearly 20% in some tests. This increase in capacity was driven by a proportional increase in waterside ΔT for each of the custom coil designs. Another way to consider these results is that the single circuit coils can achieve the same heating capacity with less water flow and better waterside ΔT, using a coil that uses the same amount of material (actually slightly less copper because header is eliminated), and no negative impact to the airside of the coil. The single circuit coils increased fluid pressure drop, which would increase pumping energy, but this is a minor disadvantage whose impact is mitigated by the fact that the extra pump energy effectively becomes electric heat added to the loop.



(Wendler, Raftery, & Cheng, 2023)

Figure 25 – Custom Coil Circuit Performance

Based on the lab results that were published in 2023, at least one VAV box manufacturer has developed the single circuit coil as an available option at no additional cost compared to the standard, and completed more extensive performance testing confirming the improved waterside performance compared to conventional coils. The single circuit coil option is a simple but key innovation that will improve the performance and cost effectiveness of condensing and all-electric hot water plants.

4.4 Pipe Insulation

California's Title 24 began requiring in the 2019 version that all piping, valves, fittings, coil housings, and coil tube bends associated with reheat systems be insulated. Despite this code requirement, it is common to find these components uninsulated in new buildings with VAV reheat systems because this is a change from longstanding conventional insulation practices (Figure 26). While many may consider losses from these components to be negligible, laboratory testing has revealed that these distribution losses could represent nearly 10% of the full-load coil capacity, though the losses vary as a function of hot water supply temperature. This potentially represents a substantial amount of heat lost to the plenum or space. While these losses may be useful in some locations and conditions (i.e., adding heat to space that is cold), they may be detrimental and even increase cooling loads in other cases. Insulating the valve trains at each coil per code requirements could reduce losses by over 40% (Wendler, Raftery, & Cheng, 2023). Greater enforcement of these insulation requirements is therefore essential to avoiding excessive energy loss and lowering natural gas consumption in buildings with gas boilers.

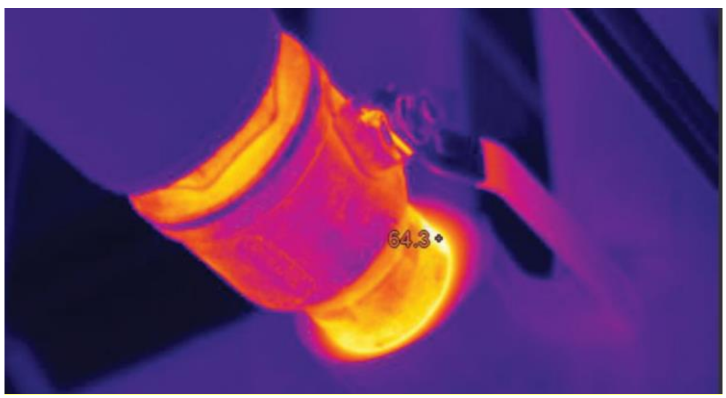


Figure 26 – Typical Uninsulated Components at a VAV Reheat Box

4.5 Boiler Sizing and Selection

Boiler plants are frequently oversized for a range of reasons as discussed in Section 2.2. Nevertheless, there are a number of detrimental consequences with oversizing. System efficiency may be sacrificed with high minimum flow requirements that lead to extra bypass flow and frequent operation below the boiler minimum turndown limit will result in short cycling. Short cycling in turn may lead to premature equipment wear, further degrading efficiency. Concrete data showing the negative impacts of oversizing may encourage designers to evaluate sizing practices more carefully, particularly where first cost and equipment size may be more critical factors for all-electric plants.

Retrofit Sizing

In addition to re-evaluating load calculation assumptions for new construction, retrofit projects offer opportunities for right-sizing based on actual measured heating loads. For retrofits, rather than simply matching existing equipment capacities, designers should consider installing metering, if needed, and evaluating observed peak heating loads where possible.

- Note that the design heating condition may not necessarily occur during a given winter season so evaluation of measured peak heating loads should include review of outdoor air temperatures as well.

- Consider applying time-averaging to the trended data. Trend data points are typically recorded as the instantaneous value at a particular moment in time, rather than as an averaged value over the trend interval. Instantaneous values may reflect transient spikes that are not representative of actual peak heating demand, such as due to transport delays. In addition, there may be an artificial peak at system startup when a boiler generates hot supply water but the return water temperature is still representative of overnight equilibrium temperature. This load represents the boiler output but not necessarily the load from the building. Consider averaging recorded values over 60 minutes to evaluate the representative building peak.
- Pay attention to opportunities for reducing peak loads prior to establishing equipment sizes. Often there will be simple control opportunities that can effectively reduce peak heating loads, such as ensuring that there is a dedicated warmup mode that operates without ventilation, and/or applying a longer, slower morning warmup strategy (Section 3.3) either every day, or on days approaching design heating conditions.

Redundancy

For buildings with redundancy requirements, redundancy is often evaluated as a percentage of design capacity. Because building heating load profiles are so often skewed toward low part loads (see Section 2.2), a more effective strategy may be to consider percent of time instead. For example, for a typical load profile, a plant with two boilers each sized for 50 percent of design capacity could meet the heating loads for 87 percent of time with one boiler failed. Rather than provide an additional equally-sized boiler for N+1 redundancy, consider whether the two-boiler plant provides acceptable coverage, or consider three smaller boilers that may provide additional redundancy and turndown capabilities.

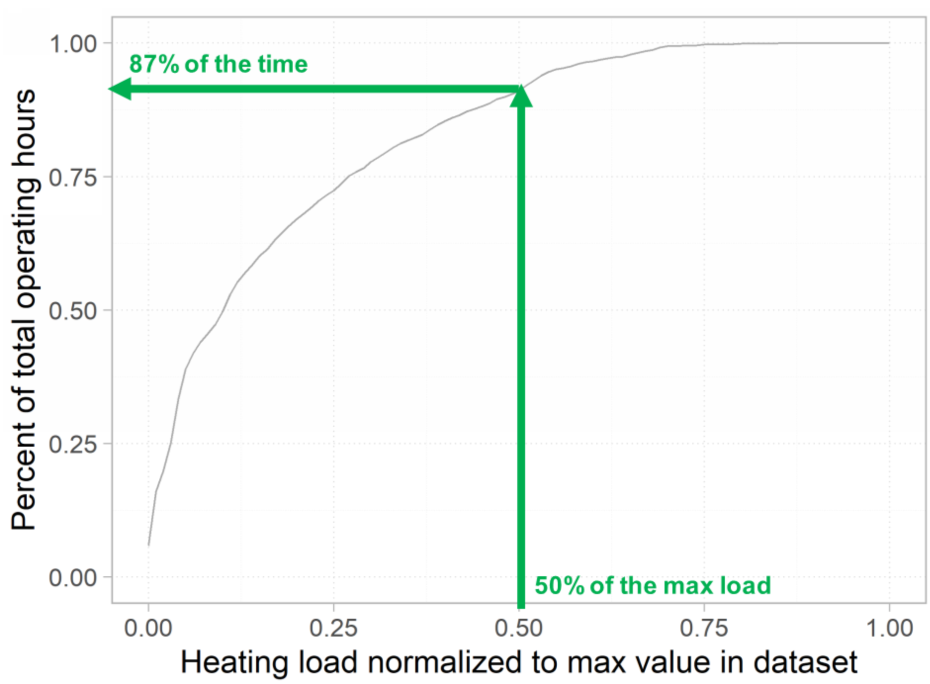


Figure 27 – Boiler Plant Redundancy

Boiler Selection

In addition to capacity and rated efficiency at full load conditions, boiler selection should consider other factors that impact operational performance.

Given that boilers for typical buildings spend the vast majority of the time at very low part loads, plant turndown capability and minimum system load should be considered to minimize the amount of time that boilers are short cycling. Plant turndown can be achieved by selecting boilers with individual turndown capability (e.g., many manufacturers offer models with 10:1 and 20:1 turndown capability) or by selecting multiple boilers (e.g., two equally sized boilers with 5:1 turndown capability result in a plant with an overall 10:1 turndown capability). Peterson suggested that systems with zonal heating can commonly produce systems loads that are 5% of the design peak (Peterson, 2018). This is reinforced by measured load data (see Figure 6), indicating that a combined plant turndown of 20:1 or more is recommended (e.g., two boilers each with 10:1 turndown capability and each sized for half of the design load).

The boiler minimum flow requirement may also be a factor that directly impacts system operating efficiency for variable flow systems. Modern boilers tend to be low-mass and have smaller water volumes to reduce cost and improve rated efficiency but have minimum flow requirements to ensure stable temperature control and protect the boilers from overheating. Some boilers have minimum flow requirements that vary as a function of a firing rate. Other high mass boilers are available with no minimum flow requirements. In addition to noting these minimum flow requirements when selecting equipment in design, the plant controls should be coordinated to match these requirements to ensure that bypass flow is minimized as much as possible in operation. Selecting multiple, smaller boilers may also allow a reduction in the minimum flow requirement for the system.

Carefully consider boiler turndown capability and minimum flow requirements.

Designers should specify minimum turndown and minimum flow requirements for their projects and carefully review submittals to ensure that these requirements are met.

4.6 Commissioning

As with other aspects of HVAC and control system installation, commissioning is a critical process for ensuring that HHW systems are designed, installed and initially operated to meet design intent, owner requirements, and code requirements. The number of issues that may impact heating system performance are too numerous to describe exhaustively but this section highlights a few items that are often overlooked:

- **Boiler tuning.** Boilers are configured and tuned by factory-trained technicians to provide stable operation and more. Effective boiler start up should not be assumed to just be a given. Startup documentation should be reviewed and stable boiler operation should be reviewed in trends. Often there are challenges with boiler tuning for stable operation to prevent flameouts, and desired turndown capability may not necessarily be configured properly. Often multiple rounds of review and repeat visits by boiler startup technicians are required with detailed review and consistent oversight to provide a system that is stable and reliable. For buildings installed in summer conditions, an additional review should be completed during the first winter to ensure that any boiler issues can be resolved during the warranty period.

- **Optimal start tuning.** As noted in Section 3.3, optimal start algorithms requiring tuning and/or learning through a training period to work effectively. Many learning algorithms work through a trial and error approach, so training the optimal start to work properly in warmup mode may require learning during winter conditions and cannot be done effectively otherwise. Confirm that algorithms are tuned rather than set to default settings (e.g., are the settings identical across dissimilar zones?), require in-season tuning if appropriate, and confirm that the operators understand that HVAC systems should be scheduled to start with expected occupancy, rather than with a fixed warmup period if optimal start is deployed.
- **Trim and respond reset tuning.** Confirm that setpoints are resetting effectively during post-occupancy trend reviews. Proper reset tuning cannot simply be tested under artificial functional testing conditions. As noted in Section 2.3, many operating buildings with hot water supply temperature reset were observed to largely operate with setpoints fixed at the maximum. Ensure that the number of ignores is set to appropriately balance the need for temperature control and energy efficiency. Identify and address rogue zones, and ensure that BAS graphics appropriately display T&R parameters, allow for adjustments within the graphics, and calculate %-request-hours (if required). See Sections 3.2 and 4.1 for additional discussion on T&R resets and rogue zones.
- **Boiler entering water temperatures.** For condensing boilers, evaluate entering water temperatures under a range of realistic operating conditions through trend review. Building energy codes require that boilers (within a certain size range) be designed and operated to maintain return temperatures of 120^{o}F or lower. If actual temperatures are higher than this, confirm that the supply temperature control is appropriate and confirm that bypass flow is appropriately minimized.
- **Review BAS graphics.** Often BAS graphics are the limiting factor that hinder the operators' ability to monitor and adjust system operation effectively. The BAS graphics are the human-machine interface for operators to understand how the HVAC systems are operating. All key inputs and outputs should be shown on graphics, clearly and intuitively (e.g., measured values displayed next to setpoints, in relevant locations on system schematics), and setpoints and parameters should be adjustable within the graphics (not requiring users to make adjustments in programming). There should be hyperlinks between related systems and logical system trees to provide ease of navigation. Key monitoring points should be configured for long term trending and, where possible, basic trend graphs should be set up and saved to allow for quick review of key system operation.

The Advanced BAS Best Practices Guide provides additional recommendations on commissioning that may improve heating system performance (Cheng, Eubanks, & Singla, 2022).

5

Educational Resources

- Trainings and Seminars Available Online
- Live Trainings
- Reference Documents

5 Educational Resources

5.1 Trainings and Seminars Available Online

BEST Center 2020 Annual Institute: "ASHRAE Guideline 36 – High Performance Sequences of Operation for HVAC Systems"

Steve Taylor (Taylor Engineers) presented a seminar for the Building Efficiency for a Sustainable Tomorrow (BEST) Center at Laney College in 2020. The training and lab demonstration were recorded and posted online here:

https://www.youtube.com/watch?v=g2bvUCDKGEU

https://www.youtube.com/watch?v=fIxaJWouAVw

AMCA insite Webinar: "VAV Systems Part 1: VAV Design Tips"

Steve Taylor (Taylor Engineers) presented a webinar for the Air Movement and Control Association (AMCA) insite webinar series in 2021. The webinar is available online here:

https://amca.wistia.com/medias/mrwuppyiw4

ASHRAE Hawaii Chapter Meeting: "Guideline 36: Best in Class HVAC Control Sequences"

Steve Taylor (Taylor Engineers) presented at the January 2021 ASHRAE Hawaii Chapter meeting. The recorded meeting is available online here:

https://public.3.basecamp.com/p/cpu2bisnAAnLxC5mTnidc1AY

5.2 Live Trainings

ASHRAE Instructor-Led training

Since 2018, ASHRAE has offered live seminars titled "Guideline 36: Best in Class HVAC Control Sequences". These have typically been three-hour classes, offered two times per year alongside the Winter and Summer ASHRAE meetings.

Find out information about upcoming classes through the ASHRAE Learning Institute and through the ASHRAE Conference course listings:

https://www.ashrae.org/professional-development/all-instructor-led-training/instructor-led-training-seminar-and-short-courses

https://www.ashrae.org/conferences

https://www.ashrae.org/professional-development/all-instructor-led-training/instructor-led-training-seminar-and-short-courses/guideline-36-best-in-class-hvac-control-sequences

PG&E Energy Centers

The PG&E Energy Centers have periodically offered live classes on G36 and advanced HVAC controls, ranging in length from three to eight hours. Classes are free of charge and are generally available in an online webinar format, either synchronous or on-demand from previously recorded offerings.

Find out information about upcoming classes here: https://pge.docebosaas.com/learn

5.3 Reference Documents

ASHRAE Guideline 36-2021 High Performance Sequences of Operation for HVAC Systems

https://www.techstreet.com/standards/guideline-36-2021-high-performance-sequences-of-operation-for-hvac-systems?product_id=2229690

Advanced Building Automation System Best Practices Guide. Version 1.0. June 2022.

The Advanced BAS Best Practices Guide is intended to be a resource for a wide range of stakeholders and presents overviews on the importance of BAS, the background and savings potential of using ASHRAE Guideline 36, explanations of key control sequences, and guidance on other aspects of control system design and operation.

https://tayloreng.egnyte.com/dl/phXTDfFQb8/2022-06-13_BAS_Best_Practices_Guide_v1.0.pdf_

Fundamentals of HVAC Control Systems, by Ross Montgomery and Robert McDowall. 2011. ASHRAE Learning Institute.

This book provides a thorough introduction and a practical guide to the principles and characteristics of HVAC controls. It describes how to use, select, specify, and design control systems.

https://www.techstreet.com/ashrae/standards/fundamentals-of-hvac-control-systems-i-p?gateway_code=ashrae&product_id=1771686

Advanced Variable Air Volume System Design Guide. Energy Design Resources. Pacific Gas and Electric Company. Second Edition. March 2007.

The Advanced Variable Air Volume System Design Guide is written for HVAC designers and focuses on built-up VAV systems in multi-story commercial office buildings.

https://tayloreng.egnyte.com/dl/jzNGKhmF1C/EDR_VAV_Guide.pdf_

See also the list of references in Chapter 6.

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