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Quantifying energy losses in hot water reheat systems

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1 ABSTRACT

We developed a new method to estimate useful versus wasted hot water reheat energy using data obtained from typically installed instrumentation that applies to all pressure independent VAV terminal units with discharge air temperature sensors. We evaluated the method using a year of 1-minute interval data for a 11,000 m² building with 98 terminal reheat units, and found a 14% upper bound for the uncertainty associated with this method. We found that just 21% of gas energy is converted to useful reheat energy in this building. The distribution losses alone were 44% of the heat output from the boiler. The results raise questions regarding the tradeoffs between hot water heating systems, which have significant distribution losses, and electric heating systems, which effectively have zero distribution losses. In this building, and likely many others, an electric reheat system supplied by a small photovoltaic panel system would have a lower operating energy cost and a lower initial cost than the hot water reheat system. Further investigations using this method will be relevant to designers and standards developers in deciding between electric and hot-water reheat, particularly for modern designs using dual-maximum controls and low minimum airflow setpoints.

Keywords: Hot water reheat; boiler; distribution system efficiency; virtual meter; energy efficiency; variable air volume.

2 INTRODUCTION

Where a central Heating Ventilation and Air Conditioning (HVAC) system supplies multiple zones with the same temperature air, heating coils are needed at the terminal units (i.e. at the zone or room level) in zones that may require heating when there is a demand for cooling elsewhere in the building. An air handling unit (AHU) single-duct system serving multiple zones with variable air volume (VAV) terminal units is a very common type of HVAC system in commercial buildings. Each terminal unit has a damper to control airflow to meet ventilation air requirements and so that it can increase airflow to provide cooling as needed up to its design maximum flow rate. In all but the warmest climates, perimeter zones require heating coils at the terminal units. However, heating coils are also often needed in interior zones to ensure that these zones are not overcooled when supplying (typically cool) ventilation air. For example, when the supply air leaving the AHU must be quite cool to meet a need for cooling in the building (e.g. a west facing zone, operating at maximum air flow), this supply air temperature may be too low for other (e.g. interior) zones served by that AHU. These terminal unit heating coils are typically known as ‘reheat coils’ as there are some times of the year when the AHU has cooled the supply air, only to then ‘reheat’ it at some terminal units. Either electricity or a hot water distribution system serves as the energy source for these coils. Hot water reheat systems, typically served by a gas-fired boiler, are more widely used in buildings because of lower utility costs than for electric reheat systems. Electric reheat is even prohibited in some codes and standards (such as California Title 24 [1]).

For context, Zhang et al. [2] recently concluded a large study of distribution losses for a conceptually similar type of system: open-loop centralized domestic hot water recirculation systems fed by natural gas boilers. The results showed that the delivered hot water energy in 28 different buildings averaged just 35% of the source gas energy consumption [2], that is, there was a 65% loss in the system, due in equal part to distribution losses and losses at heating equipment. There are numerous studies [3–13] of heat transfer and demand within open-loop domestic hot water systems for a range of different applications in both commercial and residential buildings. The overall system efficiency and distribution system losses vary very widely depending on design and application. Chapter 50 of the ASHRAE Handbook of Applications [14] summarizes this well as “Energy losses from hot-water distribution systems usually amount to *at least* 10 to 20% of total hot-water system energy use in most potable water-heating systems [3], and are often as high as 50%; losses of over 90% have been found in some installations [15]”. This includes both heat lost from the distribution system, as well as water and heat wasted at the fixture as the user waits for water to reach a usable temperature. These findings show that the overall system efficiency is far lower than expected based on idealized analysis of these systems.

In comparison, though there is no water waste in a closed-loop reheat system, many losses are similar to open loop domestic hot water systems. In a closed-loop hot water system used for primary heating, where the majority of the building is operating in heating mode, these losses are of less concern as long as the losses occur within the building envelope. Though they may cause control problems (e.g. overheating in some zones), the losses still contribute to heating the building overall. In contrast, for reheat systems, the demand for heat is typically only from a relatively small number of zones in the building, under conditions that vary widely based on the time of day, supply air temperature, heat load in the zone, minimum airflow rates, heat transfer through the envelope, etc.. Reheat demand often occurs during times of the day and year in which the majority of the zones in the building require cooling, or in large buildings where the interior zones always require cooling independent of outdoor weather conditions. In this case, the losses will also be a significant component of overall hot water system efficiency.

Lastly, recently developed control strategies used to improve the energy efficiency of VAV systems, such as those described and demonstrated in [16–20] and recently formalized in ASHRAE Guideline 36 [21], successfully avoid most unnecessary reheat energy. These new strategies reduce the minimum airflow setpoint at the VAV terminal unit to a more appropriate level. Historically, designers have defined this as a fixed percentage of the design maximum airflow, typically 30%, or often even higher (e.g. 50%) for VAV terminal units with reheat coils that use single-maximum control logic¹. Using dual-maximum control logic [18] or time-averaged ventilation [20] allows the minimum to be set to the correct value — the design ventilation airflow requirement for the zone². Many of the heating system losses described above are constant and do not vary with the need for reheat in the building. Thus, they become proportionally more significant when the overall useful reheat demand decreases.

In this paper we focused on closed-loop systems serving reheat coils at variable air volume (VAV) terminal units, commonly known as ‘VAV boxes’. We performed a thorough literature review of likely sources of publications on this topic and were unable to find prior studies that analyzed this specific case. The energy wasted within these systems occurs due to a number of factors:

- Heat lost through insulated and uninsulated piping and fixtures, both during flow conditions and when non-flowing water reaches steady state with the surrounding environment. Hiller [3] describes these losses in detail and illustrates them using a number of example calculations.
- Heat lost by passing valves unnecessarily supplying hot water to a reheat coil, a problem that is unique to the nature of hot-water heating systems of any kind.
- Electrical motor losses serving circulation pumps.
- Boiler combustion, standby and parasitic losses.

In contrast, electric reheat systems have minimal distribution losses, no passing valves, no boiler losses, and lower initial installation costs, but typically have much higher unit energy costs.

We formalized the primary research questions that we wished to answer as: (1) How do we cost-effectively quantify intentional reheat energy use in buildings with hot water reheat systems; (2) What are the distribution losses in a real building; and (3) Under what conditions do the initial cost and operating energy cost tradeoffs favor electric reheat?

3 CASE STUDY BUILDING

3.1 DESCRIPTION

We performed a case study of a 5 story, 11,000 m² office building in the California Bay Area. The Bay Area is a Koppen Csb climate zone (California climate zone 3, ASHRAE climate zone 3C) characterized by dry, warm summers and mild winters. Constructed in 1999, the building is predominately open plan with some enclosed offices and conference rooms along the perimeter, and a central core of services and conference rooms. The window-to-wall-ratio is approximately 0.6 on the first floor and 0.45 on all other floors in most zones, and almost all of these windows are not externally shaded. Approximately 400 people work in the building performing typical administrative tasks and the building is typically occupied from 6am to 5pm. The HVAC system in the building recently underwent a complete controls retrofit

¹ Here, the minimum airflow setpoint in deadband must be the same as that which is required to provide the heating capacity for the design heating condition.

² This does not reduce the amount of outside ‘fresh’ air entering the building, as that is controlled at the AHU and remains constant when zone minimum airflow setpoints change. Decreasing the minimum airflow setpoints in the building simply reduces the total amount of air circulating in the building’s HVAC system, reducing fan power and wasted reheat energy use, while increasing the fraction of outside air in the supply air leaving the AHU.

which has brought it up to current industry best practice, almost identical to the more recently published ASHRAE Guideline 36 [21]. The building has a single-duct, variable air volume (VAV) system, served by two rooftop air handling units with direct expansion (DX) cooling coils and evaporatively cooled condensers. A gas-fired hot water boiler supplies hot water to the terminal (two-row) reheat coils distributed throughout the building, present at most VAV boxes, and those VAV boxes use industry best practice dual-max³ control sequences [18]. The hot water system has a flow meter and temperature sensors on the supply and return at the boiler, commissioned and calibrated as part of the controls retrofit. The retrofit also included replacing all of the existing reheat coil valves with new, high quality valves. The air handling units also use current industry best-practice supply air temperature and duct static pressure resets based on temperature and pressure requests from the individual zones in the building.

3.2 ZONE AND AHU INFORMATION

The building has a total of 144 VAV terminal units (or VAV boxes) among which there are 98 zones with reheat coils, all served by the same closed-loop hot water system. The first step of the analysis investigated the availability and consistency of the data monitored at each VAV reheat box. We logged data from the Building Automation System (BAS) at 1-minute intervals from 1st September 2016 to 31st August 2017, i.e. 525,600 records for each monitored variable involved in the analysis. We did not apply any aggregation procedure to the dataset before the analysis. Values monitored at each VAV terminal unit that are relevant for this analysis include: airflow rate, discharge temperature, room air temperature, and reheat valve position. Relevant values monitored for the AHUs include: outside air temperature, supply air temperature and fan speed. The AHUs operated during occupied hours and for a short period beforehand governed by an optimal start algorithm, and thus we identified the operating hours using the fan speed signal from the AHUs throughout the paper. Outside of operating hours, the reheat coil valves were closed throughout the vast majority of the building⁴. However, the hot water boiler plant and distribution system operated continuously due to concern that gaskets in grooved end pipe couplings may contract when water temperatures drop, allowing water to leak from the fittings. Though this wasted a significant amount of energy during these hours, this does provide a unique opportunity to bound the uncertainty associated with the intentional reheat energy estimation method proposed in this paper, which we discuss in §5.7.

3.3 BOILER AND HOT WATER SYSTEM INFORMATION

The building has two outdoor Rite 225W gas boilers, which each have a nominal maximum output of 527kW with a rated efficiency of 80% according to the manufacturer's manual [22]. We measured total flow from the boilers with an Onicon F-3500 Series insertion magnetic flow meter, with a stated accuracy of $\pm 1.0\%$ of reading from 0.61 m/s to 6.1m/s (2 to 20 ft/s) and ± 0.0061 m/s (± 0.02 ft/s) below 0.61 m/s (2 ft/s). We measured the supply and return temperatures using immersion thermistor probes with a stated accuracy of ± 0.36 °F (± 0.2 °C). The boilers are the only gas consuming device supplied by the utility gas meter, and those boilers only supply the hot water reheat coils in the building. The hot water distribution system is variable flow with two-way zone valves throughout, with the exception of three three-way valves to reduce boiler cycling and to prevent deadheading the hot water pumps. A thorough review of boiler operation during the analysis period showed the lag boiler and pump cycle on frequently, starting partway through 2016, when one boiler and pump were overridden to 'enabled'. The pumps are dedicated, so any time the pump is running, it moves hot water through the boiler. Even though the second boiler is not firing, this will still cause higher boiler losses than a single boiler operating as designed and commissioned. We discuss the implications of this in §5.4

3.4 ENERGY COSTS

We obtained the actual utility tariffs for this case study building, which uses the PG&E G-RN1 tariff for gas consumption, ranging between \$0.72 and \$1.09 per therm depending on season and level of consumption. We simplified this to an average \$0.0311/kWh (\$0.91/therm). The building operated under the PG&E A-10-Secondary electricity tariff, which ranges between \$0.11/kWh and \$0.22/kWh for energy consumption, and from \$8.31 - \$17.83/kWh (peak monthly) for demand, depending on the time of day and year. The utility estimates the total annual average cost for a 'typical' consumer in this tariff band as \$0.204/kWh, and we simplified the further analysis to use that value. The ratio

3 Single-maximum control sequences have a single maximum air flow rate at the design cooling condition. The minimum airflow rate in the deadband between heating and cooling modes is often set by the airflow required at the design heating condition. In contrast, dual-maximum control sequences allow a VAV box to control the airflow to a much lower minimum in the deadband, saving a significant amount of reheat and fan energy. See [18] for more detail.

4 A very small number of reheat coil valves infrequently open during non-operating hours due to a zone heating setpoint setback issue, which in some zones was 18 °C (65 °F). However, the AHU fans are off during this period, so air flow and losses from these coil are small.

of electricity to gas price (6.6 in this case) has a significant effect on this analysis, and it varies widely by location, utility provider, time of day, season, and even from year to year.

4 METHOD

4.1 OVERVIEW

We aimed to quantify the losses in the hot water distribution system by measuring the hot water energy supplied by the boiler and comparing this to the *intentional* reheat energy use, aggregated for every zone – i.e. at each reheat coil. We calculated the heat supplied from the boiler by performing a heat balance calculation using the water flow rate and the boiler supply and return temperatures, assuming the standard properties for water.

We used a novel approach to calculate the amount of intentional reheat energy used by the reheat system. Similarly to open loop systems, where it is difficult to determine what the water use would have been without any distribution losses, here it is not feasible to measure energy on the waterside at each reheat coil for a representative building due to the number and cost of the sensors that would be required. Thus, we proposed a new method to estimate intentional reheat using a modified airside heat balance calculation. This ‘virtual heat flow meter’ uses data acquired from the BAS - taking advantage of sensors that are typically installed in these systems - to obtain new information that would otherwise be uneconomical to measure [23]. ‘Virtual water flow meters’ (using methods such as those described in [24–26]) present other common examples of virtual meters. However, we were unable to find prior publications that have applied this concept to heat flow meters in which additional — non-process related — heat transfer is known to occur between the two temperature measurement locations, as we have done in this paper.

Some control strategies require a dynamic estimation of hot water reheat energy consumption using the instrumentation that is typically installed in a building management system. One example is a cost-responsive supply air temperature reset strategy [27], which dynamically selects the optimal AHU supply air temperature setpoint to minimize the combined cost of fan, cooling, and reheat energy consumption. The work presented in this document is in part a validation and refinement of the earlier method used to estimate reheat energy consumption by that control strategy.

4.2 METHOD FOR ESTIMATING INTENTIONAL REHEAT ENERGY USE

We estimated the intentional reheat energy use by performing a modified air side heat balance using the reheat valve position, discharge air temperature and airflow measured in each zone, and AHU supply air temperature.

We first estimated the long-term temperature difference (ΔT_e^i) between supply air and discharge air for each VAV box when the reheat valve has been closed for an extended period. This accounts for the effects of sensor error and duct heat transfer between the AHU supply air measurement and the VAV box discharge air measurement. It also accounts for the effect of passing valves⁵, which we consider to be an unintentional use of reheat energy, and a source of waste or loss in the overall system. The equation below performs this calculation. In simple terms, we exponentially smoothed

ΔT_e^i when the reheat valve is closed. We performed this smoothing only when both of the following conditions are met:

- the reheat valve supplying that reheat coil is currently commanded fully closed, and has been commanded fully closed for longer than the last 3τ minutes, where τ is the time constant of the hot water coil. For the building in this paper, we approximated the value of τ as 4 minutes for each VAV box based on a timeseries analysis for a randomly selected number of zones, discussed in more detail in §5.3.1.
- the AHU was operating (i.e. the fan is running and there is measured airflow through the VAV box).

Requiring the valve to be fully closed for a period of 3τ minutes ensures the majority of heat has transferred from the hot water that remains in the coil after the valve has closed – i.e. that the system is approaching steady state – before updating the ΔT_e^i estimate. When either condition above is not met, we kept the ΔT_e^i estimate constant.

⁵ Passing valves allow fluid to pass through them when the valve actuator appears fully closed. This can occur over time due to obstructions preventing the valve from fully closing, damage to the valve seat or plug, or actuator failure.

$$\alpha \cdot \Delta T_{e,t-1}^i + (1-\alpha) \cdot (T_d^i - T_s^i), \prod_{j=1}^{3\tau} \Omega_{r,t-j}^i = 0$$

$$\Delta T_{e,t-1}^i, \prod_{j=1}^{3\tau} \Omega_{r,t-j}^i > 0 \wedge \dot{v} \wedge \dot{v}$$

$$\Delta T_{e,t}^i = \dot{v}$$

Equation 1

Where, for the i^{th} VAV box, $\Delta T_{e,t-1}^i$ is the temperature error term at the previously sampled time period [°C],

α is a parameter that controls the amount of smoothing, T_d^i is the discharge air temperature leaving the reheat coil [°C], T_s^i is the supply air temperature leaving the AHU serving that VAV box [°C], $\Omega_{r,t-j}^i$ is the reheat valve position, and τ is the approximate time constant of the hot water coil (in integer steps of the sampling period). Note here that larger values of α cause the value of e^i to be more representative of the long-term average value, whereas smaller values make e^i more responsive to shorter term changes in the system. We used $\alpha = 0.98$ for this paper, and discuss and estimate the effect the value of α has in the supplementary material. Regarding initialization, we initialized this calculation using the instantaneous temperature difference at that time ($t=0$), i.e., by setting $\Delta T_{e,-1}^i$ equal to $T_d^i - T_s^i$.

We then used this long term temperature difference in a modified airside heat balance calculation to compute the intentional reheat used at the i^{th} VAV box (q_v^i) by applying the following equation:

$$q_v^i = \begin{cases} 0, \prod_{j=1}^{3\tau} \Omega_{r,t-j}^i = 0 \\ v_a^i \cdot \rho_a \cdot c_a \cdot (T_d^i - T_s^i - \Delta T_{e,t}^i), \prod_{j=1}^{3\tau} \Omega_{r,t-j}^i > 0 \end{cases}$$

Equation 2

Where, for the i^{th} VAV box, T_d^i is the discharge air temperature leaving the reheat coil [°C], v_a^i is the air flow rate [m³/s], T_s^i is the supply air temperature leaving the AHU serving that VAV box [°C], ρ_a is the density of dry air [1.205 kg/m³] at 20°C at sea level, and c_a is the specific heat capacity of air [1.005 kJ/kg°C].

Similarly to the temperature error calculation above, we performed the modified heat balance calculation only when the AHU was operating, the reheat valve was commanded at least partially open, or was commanded partially open within the previous 3τ minutes. Outside of the above conditions, the intentional reheat for a particular VAV box is zero as the reheat valve is closed, has been closed for an extended period, and is not intentionally supplying heat to the zone from the hot water distribution system. As above, the reasoning behind including the 3τ minutes after the valve closes in this calculation is to account for the transient heat transfer from the hot water that remains in the coil after the valve closes. This can be a significant component of heat transfer for VAV boxes that change in and out of heating mode frequently. We investigate this effect of including or excluding this transient phenomenon in §5.3.1.

4.3 METHOD FOR CALCULATING ENERGY SUPPLIED BY THE BOILER AND OVERALL SYSTEM EFFICIENCY

We first computed the heat supplied from the boiler (q_b) by applying the following equation:

$$q_b = v_b \cdot \rho_w \cdot c_w \cdot (T_{sw} - T_{rw}) \quad \text{Equation 3}$$

Where T_{sw} and T_{rw} are the supply and return water temperature leaving and entering the boiler respectively [$^{\circ}\text{C}$], v_b is the measured water flow rate through the boiler [m^3/s], ρ_w is the density of water [$1000 \text{ kg}/\text{m}^3$] and c_w is the specific heat capacity of water [$4.182 \text{ kJ}/\text{kg}^{\circ}\text{C}$].

We then calculated the hot water distribution system loss (q_l). This heat loss should include components of: (a) heat lost through the insulated water distribution system, (b) heat lost through uninsulated sections of the distribution system, and (c) unintentional, wasted reheat caused by passing valves at the reheat coils.

$$q_l = q_b - \sum_{i=1}^{98} q_v^i \quad \text{Equation 4}$$

The hot water distribution efficiency (η_d) and boiler system efficiency (η_b) (the input or thermal efficiency), and the overall system efficiency (η_o) are then:

$$\eta_d = \frac{q}{q_b} \quad \text{Equation 5}$$

$$\eta_b = q_b / q_g \quad \text{Equation 6}$$

$$\eta_o = \eta_b \eta_d \quad \text{Equation 7}$$

Where q_g is the energy content of the gas^{6,7} supplied to the boiler system.

Note that these equations assume that everything happens contemporaneously, when there are clearly transportation delays involved. For example, it takes several minutes for the water to circulate from the boiler, through the distribution system, and back to the boiler, so when a reheat coil valve opens, that heating demand will be seen at the boiler several minutes later. Though this will on average (or in aggregate) cancel out, we ignored transportation time effects in this analysis. This yielded a small fraction of samples (less than 0.5%), typically under low load conditions, where the supply water temperature was below the return water temperature, creating a short-term negative heat balance across the boiler.

5 RESULTS

5.1 ANALYSIS OF THE HOT WATER DISTRIBUTION SYSTEM

The analysis considered operating and non-operating hours separately for the boiler and AHU variables. We defined the operating hours as when the AHU fan speed is greater than zero, and defined all other intervals as non-operating hours. Typically, operating hours range from 4:00 am to 5:00 pm during weekdays - the start time varies slightly across the

6 We used the higher heating value for natural gas (i.e. energy content including water condensed into liquid state) to allow for a clear and direct comparison between condensing and non-condensing boilers later in the paper.

7 Additionally, the gas meter is a flow meter that measures in units of 100 cubic feet with an accuracy of $\pm 2-3\%$. We converted these flow measurements to SI units using the assumption that 100 cubic feet is equal to 1 therm or 29.3kWh. However, the volumetric energy content of gas varies depending on composition, which varies by source. The 'therm factor' for gas supplied to this building varies from approximately 1.02 and 1.04 according to utility data, so the actual energy content of gas is typically between 2% and 4% higher than the assumed. We ignored this effect in the paper.

year due to an optimal start algorithm. Table 1 compares summary statistics for boiler and AHU variables, subset by operating and non-operating hours after a data cleaning step using the ‘tsclean’ function available in the R ‘forecast’ package [28] to discard outliers. The cleaning step helped remove extreme outliers, but has limited relevance to mean and quantiles values due to the large quantity of data in the dataset. For reference, we included the same table prior to applying this function in the supplementary material. The subsequent analysis and charts throughout this paper use the cleaned dataset. Figure 1 illustrates the distribution of the primary variables here, the heat supplied by the boiler and the intentional reheat, and the supplementary material contains many similar plots for other variables for further reference.

The hot water system supplied an average of 10 kW (0.9 W/m²) of heat to the building during non-operating hours. This indicates a minimum level of distribution heat losses even when all reheat valves are closed, and have been closed for an extended period of time. This is likely due to a combination of heat loss from the insulated and uninsulated sections of the distribution system, as well as any passing valves serving the reheat coils in individual VAV boxes, though this latter component is likely minimal in this case as all of the valves in the building have recently been replaced with high quality valves. Furthermore, the heat lost due to passing valves will be much lower compared to during operating hours because there is little to no airflow through the ductwork and reheat coil.

	(kW)Heat supplied from boiler	(kW)Intentional reheat	(W)Recirculation system	Boiler return temperature (°C)	Boiler supply temperature (°C)	Boiler flow (l/min)	Outside air temperature (°C)	Average AHU supply temperature (°C)	AHU 1 supply temperature (°C)	AHU 2 supply temperature (°C)
<i>Operating hours</i>										
Min.	-30.9	-21.5	-49.6	52.8	71.3	0.0	0.8	5.7	5.1	3.8
1st Qu.	10.6	1.9	3.1	68.9	80.6	19.3	13.1	12.4	12.1	12.3
Median	19.3	7.0	9.7	72.1	82.2	31.2	16.5	14.7	13.8	14.9
Mean	27.7	15.5	10.5	71.8	81.7	38.9	16.8	14.5	14.2	14.8
3rd Qu.	34.9	18.9	18.0	75.3	83.5	46.8	20.5	16.4	16.5	17.3
Max.	131	88.5	66.0	82.7	89.1	147	35.6	26.2	28.3	27.5
<i>Non operating hours</i>										
Min.	-18.5	-	-18.5	70.3	74.1	0.0	2.1	-	-	-
1st Qu.	4.6	-	4.6	76.3	81.6	14.9	12.0	-	-	-
Median	8.5	-	8.5	77.3	82.8	26.9	14.6	-	-	-
Mean	10.0	-	10.0	77.7	82.8	30.6	15.2	-	-	-
3rd Qu.	13.2	-	13.2	79.6	84.0	41.0	17.5	-	-	-
Max.	46.6	-	46.6	83.5	89.2	128	36.1	-	-	-

Table 1 – Summary statistics for each monitored variable with data cleaning applied using the ‘tsclean’ package in R.

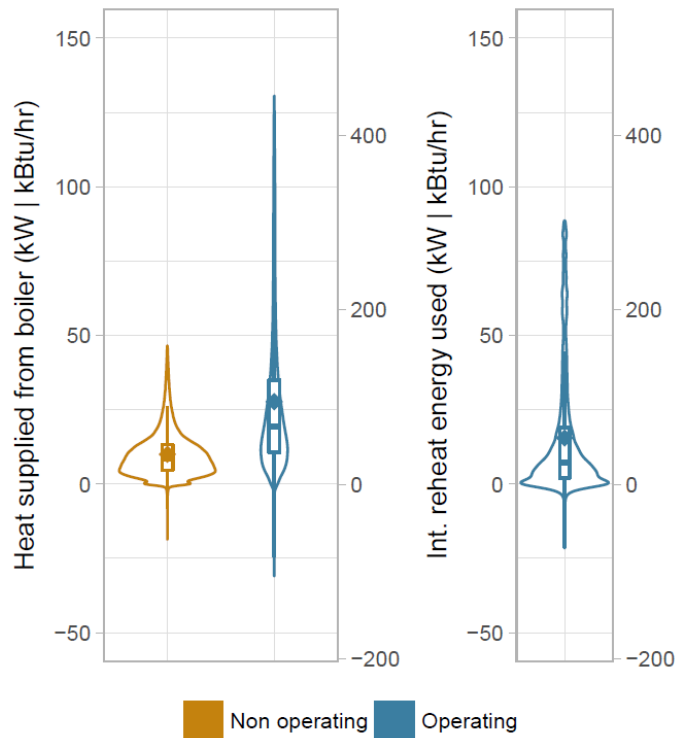


Figure 1 – Violin plot with boxplot overlay showing the distribution of heat supplied from boiler for operating and non-operating hours (left), and intentional reheat energy use (right). The AHUs do not run during non-operating hours, and thus the intentional reheat energy use is zero. Note that the small number of datapoints below zero in these distributions are due to transport times and other transient effects, instrument error, and errors in the estimation technique (e.g. no cooling occurs). The violin plot lines indicate a kernel density of the distribution (i.e. a smoothed line fit through the top of each bar of a histogram). The diamond shape indicates the mean of the distribution and the solid horizontal line indicates the median. We did not display outlier dots on the boxplot (due to the size of the dataset), however we used these outliers in generating the density lines.

5.2 CORRELATION ANALYSIS

Table 2 and Table 3 report the Pearson and Spearman correlation coefficients for each variable pair for operating and non-operating hours respectively. Each of the correlations were as expected for this system, and we highlight some key points during operating hours below:

- The heat supplied by the boiler strongly correlates with the intentional reheat energy used, and negatively correlates with the outside air temperature (Pearson coefficient equal to 0.84 and -0.52, respectively).
- The recirculation system heat loss correlates with the boiler supply water temperature (Pearson $r = 0.41$) – heat loss from the distribution network should increase as the temperature differential between the water and surrounding environment increases.
- The heat supplied by the boiler weakly correlates with the recirculation system heat loss (Pearson $r = 0.21$) – more reheat energy used means more active reheat coils in the building, which increases the heat losses from fixture branches off the main trunk recirculation line and skin losses due to higher fluid velocities.
- Average AHU supply temperature strongly negatively correlates with outside air temperature (Pearson $r = -0.56$) due to the supply air temperature setpoint resets in operation in the building. Additionally, there is a strong negative correlation between intentional reheat energy used and outside air temperature (Pearson $r = -0.61$). Lower supply and outside air temperatures push more VAV boxes into reheat mode, and requires more reheat energy from those already in reheat mode.
- The heat supplied by the boiler and intentional reheat energy used by the reheat coils strongly negatively correlate with boiler return water temperature (Pearson $r = -0.62$ and -0.53 respectively).

Figure 2 shows how intentional reheat increases significantly with decreasing outside air temperature below 15 °C (60 °F); the supplementary material contains additional scatter plots for other variables. Figure 3 shows how intentional reheat varies during the day, with a significant peak in the morning during the winter season, as expected for a building

and system of this type⁸. Figure 3 also shows that the distribution losses are almost constant regardless of time of day (excluding the winter morning peak), which is as expected and provides a further indication that the overall method for calculating distribution losses is reasonable.

	Heat supplied from boiler	Intentional reheat energy used	Recirculation system heat loss	Boiler return temperature	Boiler supply temperature	Boiler flow	Outside air temperature	Average AHU supply temperature
Heat supplied from boiler		0.84	0.21	-0.62	-0.30	0.88	-0.52	0.05
Intentional reheat energy used	0.70		-0.24	-0.53	-0.43	0.83	-0.61	0.09
Recirculation system heat loss	0.46	-0.12		-0.15	0.41	0.00	0.16	-0.06
Boiler return temperature	-0.60	-0.49	-0.20		0.45	-0.39	0.06	0.26
Boiler supply temperature	0.00	-0.09	0.28	0.34		-0.45	0.07	0.16
Boiler flow	0.84	0.64	0.31	-0.24	-0.09		-0.57	0.12
Outside air temperature	-0.36	-0.45	0.14	0.00	-0.16	-0.44		-0.56
Average AHU supply temperature	-0.02	0.00	-0.08	0.30	0.27	0.09	-0.60	

Table 2 - Pearson correlations (upper right triangle) and Spearman correlations (lower left triangle) for each variable pair during operating hours.

	Heat supplied from boiler	Recirculation system heat loss	Boiler return temperature	Boiler supply temperature	Boiler flow	Outside air temperature
Heat supplied from boiler		1.00	0.16	0.53	0.73	-0.37
Recirculation system heat loss	1.00		0.16	0.53	0.73	-0.37
Boiler return temperature	0.18	0.18		0.57	0.56	-0.61
Boiler supply temperature	0.51	0.51	0.58		0.37	-0.64
Boiler flow	0.75	0.75	0.56	0.36		-0.47
Outside air temperature	-0.34	-0.34	-0.64	-0.63	-0.47	

Table 3 - Pearson correlations (upper right triangle) and Spearman correlations (lower right triangle) for each couple of variables for non-operating hours. This table excludes many of the variables found in Table 2 as the AHU fans do not run during non-operating hours.

⁸ In this case (and many others) the term ‘reheat’ may be misleading, as it sometimes has the negative connotation of wasted energy. During morning warmup mode, the AHU outside air dampers are closed and the AHU operates to provide heat to the zones that require it. Here the heat just happens to be provided at the terminal units, instead of at a coil in the AHU itself.

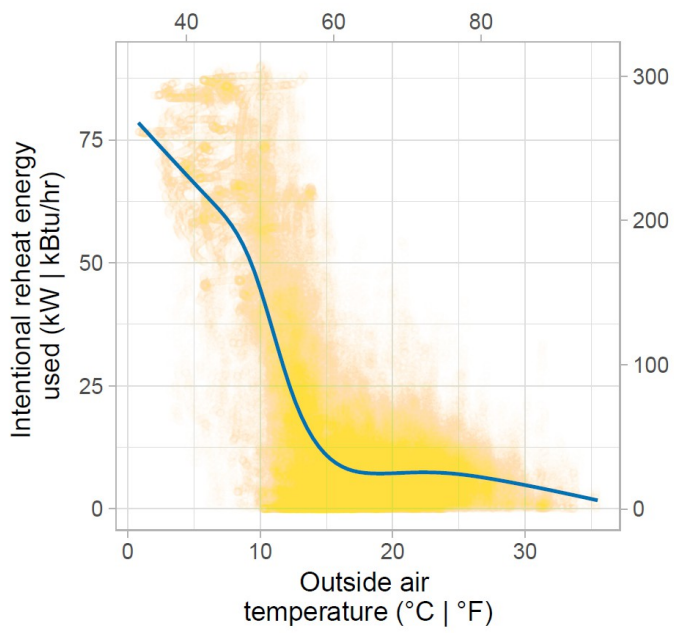


Figure 2: Scatter plot of outside air temperature and intentional reheat energy use during operating hours. The blue line represents a loess fit to the dataset.

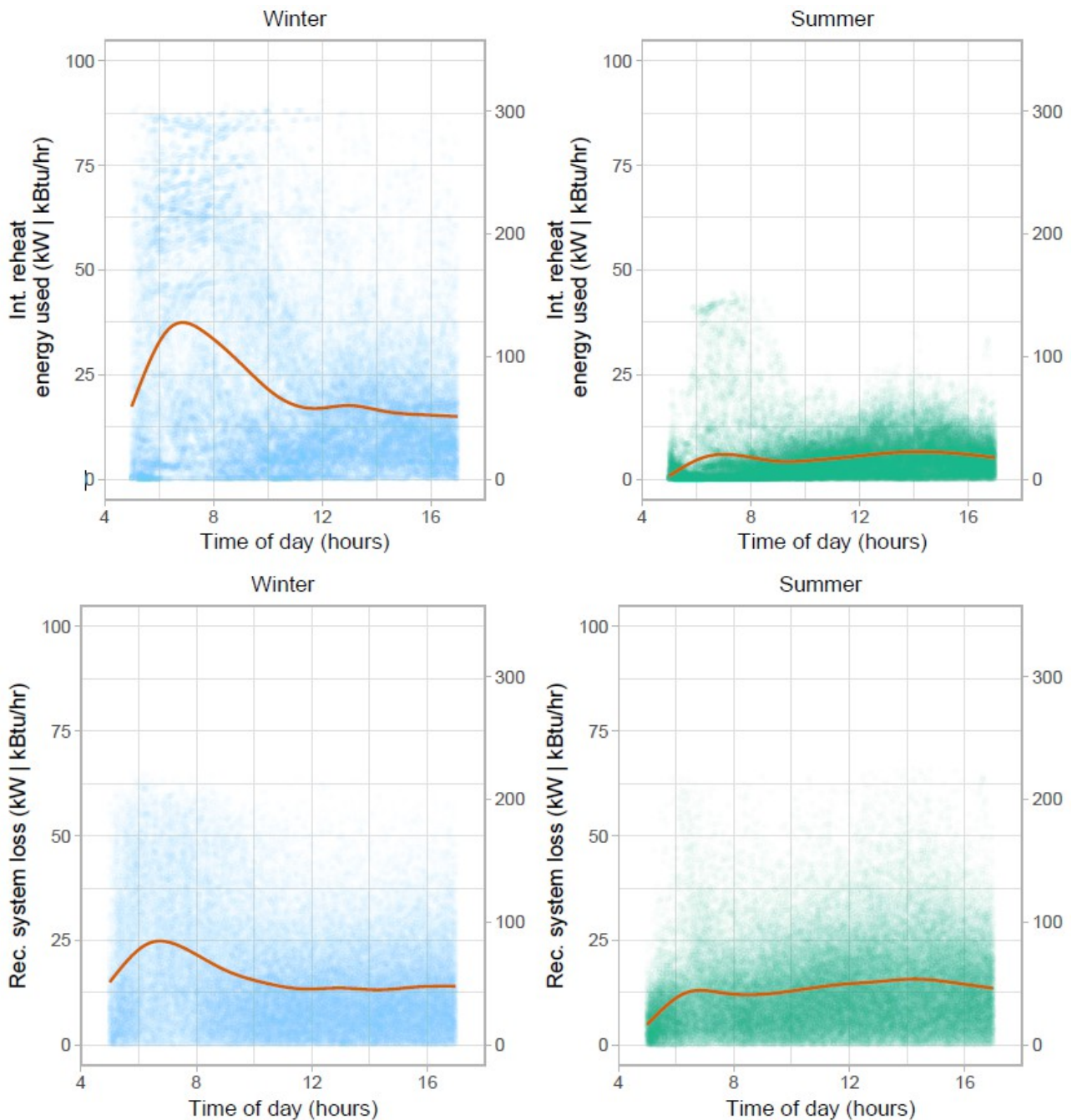


Figure 3 - Scatter plots showing intentional reheat by time of day, subset by season, for both intentional reheat (top) and recirculation system (i.e. distribution) losses (bottom), during operating hours only. We defined months between May and October as summer and months between November to April as winter..

5.3 ANALYSIS OF THE VAV ZONES

For each VAV box, we computed a set of features related to reheat usage, activity, and design characteristics. We calculated all features based on operating hours only, as the reheat valves were typically closed outside operating hours.

- Associated AHU
- Cumulative heat energy per year
- Percentage of heat energy used – computed as a percentage of the total heat energy used across all the VAV boxes, i.e. recirculation system loss excluded.
- Time spent in heating mode (daily average for the year)
- Time spent in heating mode (hourly sum for the year)
- Number of mode changes (i.e., switches from heating mode inactive to active) (daily average for the year)
- Design maximum reheat capacity from original mechanical schedules

- VAV box maximum airflow set point when in heating mode, from BAS data
- Temperature error ΔT_e^i (average for the year)
- Location within the building (either by exterior exposure, or an internal zone)

Table 4 shows the above features for the 10 VAV boxes (of 98) with the highest reheat energy consumption, which are responsible for 40.8% of total reheat energy, demonstrating that no one zone, or small subset of zones, is responsible for the majority of reheat energy consumption. The supplementary material expands this table to all zones, and also includes the details of a k-means clustering analysis. Overall, we noted that the slight majority of VAV boxes (45 of 98) used a negligible amount of intentional reheat. This is consistent with the fact that these spent very little time operating in heating mode, on average, less than 59 hours in the year. Overall, neither the location of the VAV box in the building (i.e., if zone is internal or exposed on one or more façade), nor the AHU supplying the VAV box have a relevant impact on heat usage. The only spatial pattern that we observed is that 9 out of 18 VAV boxes in the top 4 reheat consuming clusters are on the first floor, which is likely due to larger window to wall ratios and design occupancies. Higher design occupancy means higher required minimum airflow rates for ventilation purposes, increasing reheat energy use.

VAV ID	AHU	Annual intent. reheat energy [kWh]	% of bldg. total intent. reheat energy [%]	Avg time per day in reheat mode [hr]	Total annual time in reheat mode [hr]	Avg count of daily mode changes	Design max reheat cap. [kW]	VAV max airflow set point in heating [m ³ /s]	ΔT [°C]	Locale
2-1-19	2	4930	8.1	3.17	1150	2.0	5	0.2	3.9	W
1-2-5	1	3100	5.1	2.47	900	5.5	3.8	0.4	1.8	S
1-3-6	1	3020	5.0	2.38	867	5.0	4.7	0.4	1.6	S
1-5-4	1	2310	3.8	2.43	815	4.0	7	0.3	1.9	W
2-1-11	2	2310	3.8	2.87	641	2.0	13	0.1	2.1	E
2-2-18	2	2210	3.6	1.85	563	3.0	5	0.2	2.4	N
2-1-18	2	1970	3.2	1.70	480	1.0	3.5	0.2	3.6	W
2-2-15	2	1820	3.0	1.03	327	2.0	5	0.5	1.8	N
2-1-20	2	1800	3.0	3.00	582	1.0	3.5	0.1	2.3	SW
2-1-12	2	1750	2.9	2.10	638	3.0	2.6	0.1	2.0	E

Table 4 – Top 10 highest reheat energy consuming VAV boxes.

5.3.1 Transient effects: heat lost from the hot water remaining in the coil after the valve closes

In Table 4, we included a metric to count the number of switches into heating mode that occur on an average day for each zone. We did this because frequent opening and closing of the reheat coil valve would yield additional inaccuracies if the method did not account for the transient heat lost from the hot water that remains in the coil after the valve is closed. However, the table shows that this is a minor effect for most zones. Figure 4 displays a density plot for the average count of daily mode changes at each VAV box. The median value of mode changes per day was 2 across all VAV boxes in the building. Only five zones in the building had an average of above five mode changes per day (an average of 144 minutes between mode changes).

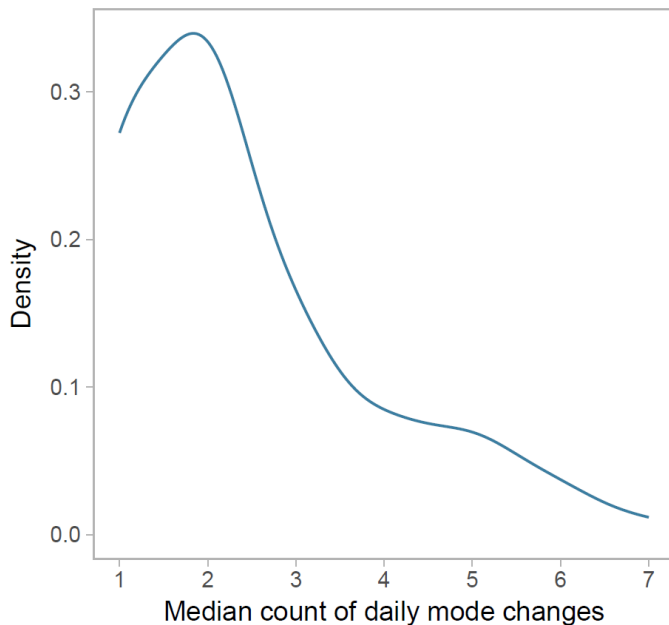


Figure 4 - Density plot for median count of daily mode changes at each VAV box.

In this building, the time constant⁹ of the reheat coil was typically between 3-6 minutes depending on the zone selected and the conditions. The supplementary material contains a figure that illustrates this phenomenon for a representative zone. After 3 time constants have passed, 95% of the temperature change will have occurred and the majority of the heat will have been lost from the (static) water in the coil. Thus, we can estimate the heat emitted by the hot water remaining in the coil after the valve closes using the time constant, which we simplify by assuming that all zones have approximately the same value (4 minutes). We illustrate this transient heat loss effect with an example using the average data for zone 2-1-19 (the highest energy consumer in Table 4). The annual average power consumption is just under 4.3 kW and the average daily time spent in heating mode (i.e. with the valve open) is 3.2 hours. Approximating to a first order system, the reheat energy emitted by the coil after the valve closed is equivalent to 99% of to the original heat transfer rate for one time constant, which we round to 100%. Given the time constant of 4 minutes, when the valve closes, the cumulative heat lost as it reaches steady state with the surrounding environment is 0.3 kWh. Thus, the error incurred by ignoring the transient effect is approximately 2.2% in this case. This error varies widely by zone; more mode changes per day, longer time constants and less time spent in heating mode per instance of valve opening, will all increase the percentage error due to ignoring this transient effect, and vice versa.

We calculated the magnitude of this effect at the whole building level. Excluding these transient effects yields an intentional reheat energy estimate that is 5.4% lower than obtained by using the method described at the beginning of this document, which includes this transient heat loss component.

5.4 ANALYSIS OF THE BOILER

5.4.1 Boiler input efficiency

Both boilers have a nominal capacity of 527 kW (total of 1054 kW). They are currently staged, however, it is not clear if the original design intent was to size the boilers to be redundant, or to size them for staged operation. The combined capacity is at the high end of a local industry rule-of-thumb for boiler size per building floor area, which was 20-30 Btu/h/ft² (63-94 W/m²) at the time of construction. That rule-of-thumb has decreased over time and is now 10-20 Btu/h/ft² (31-63 W/m²). The total design reheat coil capacity in the building (from the original mechanical schedules) is 396 kW, indicating that the boiler plant is oversized in either case.

The actual heating loads occurring in the building are of course far lower than the installed capacity. Table 1 shows that the median heat supplied by the boiler was just 19.3 and 8.5 kW (1.8 W/m² and 0.8 W/m² of building floor area) during operating and non-operating hours respectively. Compared to the nominal installed capacity for one boiler, this is a median part load ratio of 0.037 and 0.016 respectively. Given the pump override issue noted in the building description,

⁹ That is, the time for the discharge air to reach 63% of the change from initial to final (close to steady-state) temperature

causing the lag pump to cycle and water to flow through the second boiler, we also report the part load ratios based on the total capacity for both boilers: 0.019 and .008 respectively. Thus, the reheat system operated at ultra-low part load most of the time. This is due to differences between oversized design loads and those experienced in actual operation, as well as the implementation of new control sequences (and potentially other changes to the building of which we are not aware) that significantly reduce unnecessary reheat energy use since the building was designed. The 99th and 99.9th percentiles of heat supplied by the boiler plant were 214 kW and 283 kW respectively. This corresponds to part load ratios of 0.4 and 0.54 respectively for one boiler (0.2 and 0.27 for both boilers), even though the outside air conditions (minimum outside air temperature of 0.8 °C) did reach below the heating design day condition for this location (the 99.6% heating design day condition is 3.8 °C). This confirms that the boiler plant is larger than necessary for this building, though this is certainly not a rare practice.

Note also that even if the boiler was sized much smaller than usual, and only one boiler ran, it would still typically operate at very low part load. For context here, sizing the boiler to the actual installed capacity of the reheat coils would yield a median part load of 0.049 on the boiler during operating hours. Sizing it to 283 kW - the 99.9th percentile of the distribution of reheat actually required by the building during a year when the outside air temperature was 3 °C lower the design heating day temperature - would still yield a median part load of just 0.068 on the boiler. Both of these are still very low part loads where the majority of boilers operate inefficiently (< 0.1 part load).

Figure 5 shows the boiler part load against the input efficiency achieved by the boiler plant in the building, along with a density plot showing the part load distribution. Here, we calculated part load based using the capacity of a single boiler.

The median boiler system efficiency, $\tilde{\eta}_b$, is very low - just 23.6% for the entire annual dataset due to the frequent, ultra-low part load operation, and the issue with the second pump. Subset by operating and non-operating hours, the median efficiency is 33.3% and 19.3% respectively. Note that because of long-tail part-load distribution, the median efficiency is lower than the mean efficiency, at 26.5%, 35.8%, and 20.9% respectively for the three values above. Similarly, when we aggregated the gas consumption and hot water output data before calculating the average efficiency (instead of calculating efficiency at each measurement instance, and presenting the mean of that distribution), these aggregate mean efficiency values are 28.9%, 37.2%, and 21.1% respectively.

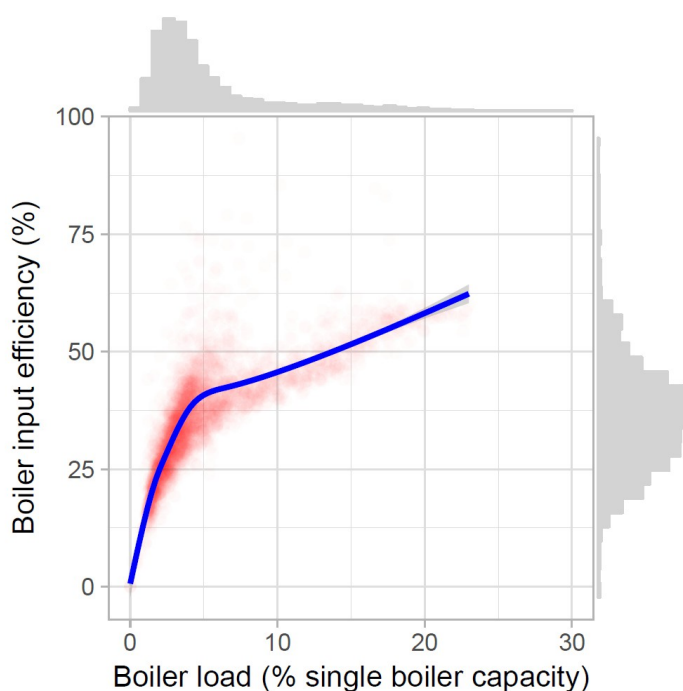


Figure 5: Boiler part load distribution and efficiency normalized to the capacity of a single boiler (though for many hours, hot water flows through the second, equally sized boiler). Note the truncated x-axis scale, which stops at 30% of the capacity of a single boiler.

Many factors affect boiler efficiency. Firstly, there are the losses which appear in the rated capacity of the boiler, quoted by the manufacturer (in this case 20%). Though there is not yet a recognized industry standard for boiler efficiency that assesses part load operation, there is a provisional standard under development (ASHRAE Standard 155P). From preliminary testing using that proposed standard, boiler losses seem slightly higher than the ratings even at nominal

conditions (e.g. [29,30]). Secondly, boiler losses increase over time in real systems due to issues such as scaling. Scale in the tubes can cause a significant efficiency loss (4% at 1/64" of scale up to 27% at 3/16" of scale according to the manufacturer, with more general estimates proved by the US Department of Energy [31]). Thirdly, boiler efficiency decreases at low load due to relatively fixed jacket and stack loss components. This is particularly pronounced in atmospheric boilers without stack dampers (such as the boilers in this building) as they have a fixed loss component (approximately 1-2% of nominal capacity). When the boiler is not firing it operates like a water-to-air heat exchanger, rejecting heat from the hot water flowing through the tubes, which then leaves the jacket through the open stack. Lastly, boiler short cycling occurs when the part load is lower than the burner turndown capability. When cycling, the losses further increase due to the pre- and post- fire purges used to flush out accumulated combustible gas mixture at each cycle [32]. Thus, boiler input efficiency decreases precipitously when operating at low part loads (e.g. < 0.1 part load) and efficiency reaches zero as the load on the boiler approaches zero.

Modern condensing gas boilers have significantly higher nominal ratings (>90% input efficiency) than atmospheric non-condensing boilers. They also typically have stack dampers which reduces heat loss when cycling, as the damper closes when the boiler is not purging or firing, and they typically have higher turn-down ratios. However, efficiency still decreases significantly with decreasing part-load, with efficiency dropping off quickly at part loads below 0.1 [29]. Some modern boilers with very high turndown ratios (e.g. 20:1) can maintain a reasonably high efficiency down to part loads of approximately 0.05.

Boilers rarely (if ever) operate close to design capacity when serving only a hot water reheat system. Boiler efficiency decays rapidly at very low part loads, which unfortunately is likely to be the most prevalent operating condition for reheat systems throughout the year. This issue could be mitigated by designed staged boiler systems, operating as intended, which would improve efficiency and add redundancy, but with added first cost and complexity. Furthermore, these staged boilers would need to be sized much smaller than usual to significantly benefit from efficiency gains, or be unevenly sized, which adds further controls complexity, cost, and potential for operational issues.

5.5 ANALYSIS OF THE OVERALL SYSTEM

Overall, the total heat supplied by the boiler was 90.8 MWh and 54.1 MWh for operating and non-operating hours respectively during the analyzed year. The boiler gas consumption was 244 MWh (8,330 therms) and 257 MWh (8,750 therms) respectively (corresponding to aggregate mean boiler efficiencies of 37.2% and 21.1%). During operating hours, the intentional reheat energy consumed at the VAV boxes was 50.9 MWh, indicating a 44% distribution system heat loss when the system is operating. The gas and intentional reheat energy use during operating hours were just 22.2 kWh/m² per annum and 4.6 kWh/m² per annum respectively, or 6.8 W/m² and 1.4 W/m² respectively averaged over the operating hours.

It is useful to assess the costs associated with reheat energy use in their entirety, which we show in Figure 6. This figure uses the energy cost data for the case study building, including these assumptions:

- Hot water pumping power consumption. Based on the design characteristics of the pumps of 11.4 L/s (180 gpm), 225 kPa (75 ft. water) and 3.4 kW (4.65 BHP), we conservatively estimated this to be 9 MWh/annum of electricity for operating hours.
- Boiler auxiliary (or 'parasitic') electricity consumption assumed at 100 W. For context, for the three different boilers tested in [29] this varied from 5 W - 357 W in idling mode, and from 75 W - 759 W at high fire rates. These losses are typically quite low for a modern boiler when idling, but for the older, non-condensing model, the lowest auxiliary electricity consumption even when idling was 357 W.
- Electricity transmission losses of 0.5%. The meter (and thus utility tariff rates) are after the transformer in this building, so this value solely accounts for minor transmission losses in the building.

There are also a number of factors which we ignored to simplify the analysis as they are either very minor, or they vary so much by location that it is not very generalizable to include in the analysis, or we did not have the data to do so in this case. Thus, we ignored:

- Temporally variable gas and electricity energy pricing, or electricity demand pricing.
- Hot water distribution losses that would occur during warmup and cooldown periods before and after operating hours if the boiler was switched off during those periods. These may be significant, but we do not have data from this case study to determine a reasonable value as the boiler operated continuously.

Figure 6 shows that the actual intentional reheat energy use is just 17% of the total cost of energy spent on providing hot water reheat throughout the building.

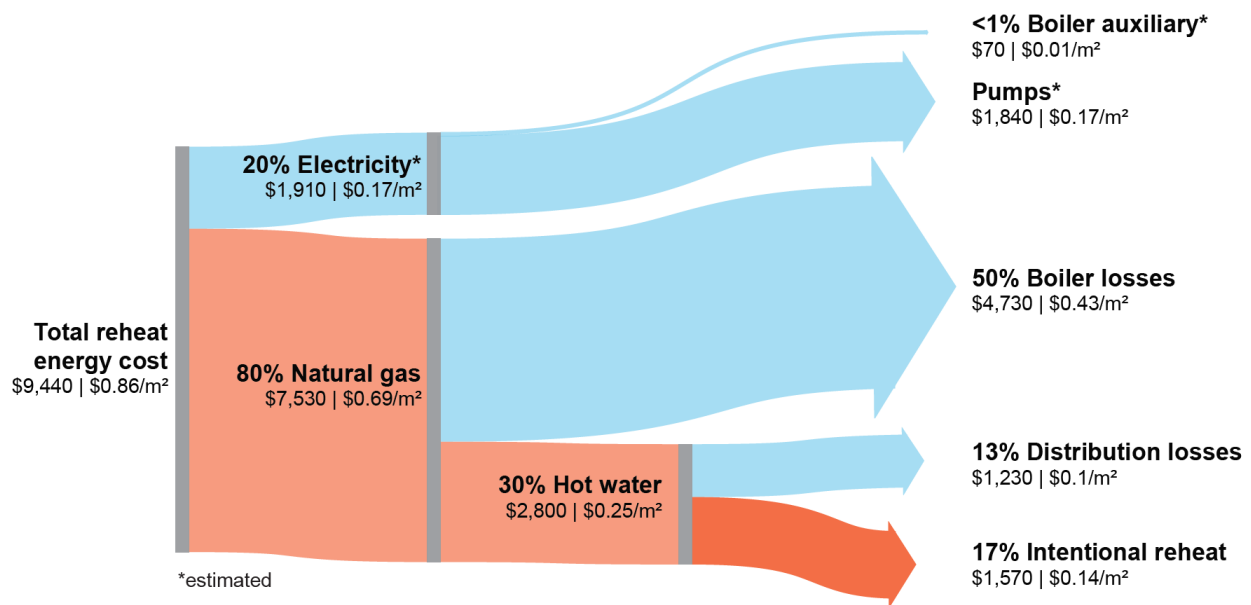


Figure 6 – Sankey chart of annual energy cost flows in the hot water reheat system (for operating hours only).

5.6 COMPARISONS TO OTHER BUILDINGS

We re-analyzed the reheat data from a previous study by the authors [27] in a different¹⁰ building using an earlier version of the reheat estimation method¹¹. In that building, extrapolating from the 6-month study period (September to February), the intentional reheat result was 9.7 kWh/m² per annum, or an average of 1.1 W/m² during operating hours.

For further context, we also re-analyzed the gas consumption data from another study [17], in 5 office buildings in the same climate zone in which we know that the only gas consuming devices were boilers supplying hot water reheat, and the HVAC system was operating with relatively low minimum airflow setpoints. The average gas consumption¹² over a one year period for the 5 buildings was 27.2, 49.7, 49.9, 55.1, and 68.6 kWh/m² per annum, from lowest to highest respectively. Lastly, the average gas consumption for heating in office buildings in this climate zone was 54 kWh/m² per annum based on the stratified random sample of 2,800 commercial facilities in the California Commercial End-use survey (CEUS) data [33]. However, this includes all building heating consumption - both primary heating (e.g., at the AHU) and reheat at reheat coils.

Overall, the values in this current study are in the same region, but on the low-end, of the data in these prior studies. This is in line with the controls retrofit which included new reheat valves and low minimum airflow setpoints. Lastly, we found one paper [34] that reported operating a hot water reheat system for a period of an hour with all reheat valves closed in a similar¹³ office building and measured the hot water supplied. The authors described this as a “reheat system standby loss test” and it is effectively equivalent to the night-time distribution loss as described in this paper. Though the data is clearly limited in time, and showed high variability over the one hour measurement period, it was 3 W/m² – approximately 3 times the distribution loss in this paper’s case study building (0.9 W/m²) – providing another indication that these losses are significant in practice.

¹⁰ The 10-year old building is in the same climate zone, is 13,000 m², 7 floors, and comprises mostly office space. However, the HVAC system operates 24 hours per day and the zone minimum air flow rates are relatively higher than the building analysed in this paper.

¹¹ The method did not account for transient losses from the coil after the valve had closed. We also do not have waterside measurements to validate the intentional reheat method, or for calculating the distribution losses.

¹² We do not have the zone-level data required to calculate intentional reheat in these buildings.

¹³ The paper provides little additional relevant information as its primary focus is not on the reheat system. However, the building is a similar sized (10,000 m²) office building, built in the same year (1999), in the same climate zone (located in San Jose). However, no further relevant information is provided as the paper does not focus specifically on analyzing the reheat system.

5.7 ACCURACY OF THE REHEAT ESTIMATION METHOD

5.7.1 Uncertainty analysis of the waterside measurements at the boiler

We calculated the uncertainty in the waterside heat flow measurement using the methods defined in [35]. This uncertainty is a function of the uncertainty in the flow measurement, the uncertainty of both the supply and return water temperature measurements, the measured mass flow rate, and the difference between the measured supply and return water temperatures. The highest uncertainty occurs when there is a small difference between supply and return water temperature, and at low flow rates. The mean and median propagated uncertainty across the whole dataset was 0.71 kW and 0.52 kW respectively, the mean and median percentage errors (relative to the heat flow measured at that time) were 5.6 % and 4.8% respectively. At the extremes of the uncertainty distribution, the 10th and 90th percentile uncertainty was 0.31 kW and 1.67 kW, or in relative terms were 2.2% and 8.7%.

5.7.2 Approximately bounding the uncertainty of the method

The fact that the hot water recirculation system runs in this building even when the AHUs are not operating provides a unique opportunity to approximately bound the overall uncertainty of the intentional reheat estimation method itself. This is because the energy consumption during non-operating hours gives a valid *lower* bound for the distribution losses. The distribution losses when the AHUs operate will be higher due to additional losses from the hot water in branch piping supplying water to each open reheat coil valve, as well as increased airflow (and increased heat transfer) through coils that have closed, but passing valves. Conversely, there is no clear phenomenon that would cause losses to significantly increase when the AHUs do not operate: the water temperatures are the same (or lower) in all components of the hot water system as they are when the AHUs operate; and the sensors that measure supply and return water temperatures are inside the building, therefore differences in losses from exterior piping (e.g. due to colder night time temperatures when the AHUs do not operate) do not affect the measured heat supplied from the boiler.

Working from the data in Table 1, the mean heat supplied by the boiler was 27.7 kW during operating hours and 10 kW during non-operating hours (representing the lower bound for losses). The mean value for intentional reheat during operating hours using the proposed method was 15.5 kW. As Figure 7 illustrates, we attributed the 2.2 kW remainder to an unknown combination of (a) error in the method, (b) additional distribution losses when the AHUs are operating and reheat valves are open, and (c) uncertainty in the measurements used to calculate the waterside heat balance at the boiler. Note that the positive sign of the remainder is consistent with additional heat loss that is likely to occur during operating hours as discussed above. Ignoring (b) and (c) - i.e., assigning all of that remainder to error in the estimation method - yields an upper bound for the 'true' value for the mean reheat energy consumed of 17.7 kW, versus the estimated value of 15.5 kW. Thus, the method underestimates intentional reheat energy consumption by at most 14% in this building, and likely less than that given the additional distribution losses that occur only during operating hours, such as (b) above.

Figure 7: Approximately bounding the upper limit of the error associated with the reheat estimation method.

6.1 ASSESSING HOT WATER REHEAT DISTRIBUTION LOSSES

Table 1 shows that the distribution losses during operating hours (where some reheat coil valves are open) and non-operating hours (where almost no reheat coil valves are open, and the AHUs do not run) were quite similar. The distribution losses were slightly higher during operating hours, which we expect as there is additional heat transfer from the piping connecting the main trunk piping to the reheat coil valves when those valves are open than when they have been closed for an extended period and the water in the pipe has reached relatively steady state conditions in the return air plenum. Overall the method of performing a modified air side heat balance using the AHU supply and zone discharge air temperature adjusted by ΔT_e^i appears valid based on this dataset, and the uncertainty/accuracy analysis (§5.7).

Distribution system heat loss may be beneficial or undesirable depending on where and when it occurs, even though it is not an intentional part of the control strategy. For example, it may be beneficial in a winter month if it occurs in an exterior zone, but the opposite applies for a summer month for the same zone. Heat lost from the distribution piping in the return air plenum will follow the return air path to the AHU, yielding a slightly higher return air temperature. Higher return temperatures allow the AHU to operate at lower outside air temperatures before requiring heating at the AHU (if a heating coil is present) or reheat coils at terminal units, but will require more mechanical cooling when the outside air temperature is greater than the return air temperature. In other words, this distribution heat loss only affects energy consumption when either the primary heating or cooling coil are active *and* that AHU is not operating on economizer (i.e. operating at minimum outside air). Neither of these conditions are common for this climate (where AHUs operate on economizer a significant fraction of the year, and where typically AHUs do not have heating coils) and thus, we ignore this effect for this case study building.

For this building, reheat use is higher when it is cold outside, driven by heat loss through the building envelope. In other buildings reheat usage may be higher when it is warm outside due to the supply air temperature reset strategy, which often needs to supply colder air in summer to satisfy a small subset of zones with high cooling loads. Zones with low cooling loads, particularly those with high minimum airflow setpoints, then require substantial reheat [19]. Whether or not this effect counteracts the heat lost through the envelope is a function of climate, the actual zone loads compared to the original design, the size of the zone minimum airflows, the amount of perimeter vs interior zones in the building, the building envelope, the type of supply air temperature reset strategy in operation, and many other factors.

When constructed, the building was governed by the 1998 version of the California Building Energy Code. The applicable¹⁴ pipe insulation requirements in the most recent (2016) code have not changed significantly since then. Both require 1.5" of insulation for pipe less than 1.5" in diameter, and the newer code increases this to 2" of insulation for larger diameter pipes (diameters from 1.5" to 8"). In this building most pipes are 1.5" diameter or smaller with larger pipes only in the risers and a short horizontal segment on each floor. The 1998 code included a 0.5" insulation requirement for 'runouts' (defined as lengths of 12' or less connected to terminal unit), which has just recently increased to 1.5" of insulation. Heat loss from runouts will clearly occur when the terminal unit is in, or was recently in, heating mode (or the valve is passing) as there will be hot water in the pipe which will gradually lose heat to its surroundings. For context, Hiller [3] estimated the time needed for the temperature difference between the water in the pipe and surrounding air to drop to half its initial value at approximately 20 minutes for an uninsulated ½" pipe to approximately 60 minutes for a ¾" pipe with ¾" foam insulation. Note also that even when the valve has been closed for an extended period, there is still some small amount of heat lost from runouts and branches due to convection-driven flow and conduction losses.

In standard HVAC construction, insulation is not usually installed on the last section of hot water piping and fittings at each coil (valves, strainers, test plugs, etc), nor on the coil frame, header and u-bends, as Figure 8 shows. The 2013 version of ASHRAE Standard 90.1 [36] defined the term piping to include all elements that are in series with the fluid flow to address this. The 2019 version of the California Building Energy Code introduced a similar requirement. Nevertheless, it is still standard practice to leave these sections of piping, fittings, coil frame and u-bends uninsulated, which may contribute to the significant distribution losses observed.

¹⁴ i.e. in the fluid temperature range for this system (141 - 200 °F)



Figure 8 – Typical hot water pipe installation at VAV reheat coil with uninsulated piping and fittings.

6.1.1 Effect of passing valves

Heat lost due to a passing valve will be rejected first into the zone and then out through the return air. Focusing only on the steady state condition for simplicity, this heat loss can be beneficial (when the temperature in that zone is in the deadband, but would otherwise require heating to meet its heating setpoint), detrimental (when that zone is in cooling mode¹⁵, or is in cooling mode but would otherwise be in the deadband), or could have no effect (when that zone is in the deadband and would remain in the deadband regardless of the additional heat input).

There does not seem to be a valid approach to estimate this effect due to the lack of information about the spatial distribution of these heat losses, and whether the losses are from the piping (i.e through insulated and uninsulated sections) or from passing valves. However, we can infer some information from the ΔT_e^i values calculated for each zone, which is a median of 2.3 °C. The median supply fan speed is just 33% in this building during the operating period, and this amount of duct heat gain is consistent with a VAV system that operates at low duct air speed. An analysis of the locations of the zones with higher ΔT_e^i values indicated that those zones are at the end of long runs of ductwork, and thus that duct heat gains are higher for these zones. Figure 9 displays the density plot for average ΔT_e^i computed at each VAV box. Ignoring sensor error and considering the expected amount of duct heat gain at low duct air speed, it seems unlikely that there are a lot of badly passing valves in this building. The fact that the reheat valves were replaced throughout the building just a few years ago corroborates this assumption.

¹⁵ This can have a disproportionately large effect in zones that are ‘critical’ for the supply air temperature reset strategy. In that case, the passing valve requires the AHU to supply colder air than otherwise necessary, increasing energy use for the entire system. Critical zones are those in which the zone heat gain requires the design maximum air flow rate at the current supply air temperature leaving the air handling unit in order to maintain the zone cooling temperature setpoint. When there is higher heat gain in a critical zone due to a fault (e.g. a passing reheat coil), this wastes energy at the zone, but also often wastes energy at the whole building level, as the AHU must supply cooler air than would otherwise be necessary to all of the other zones served by that AHU. See [16,27] for more detailed discussion.

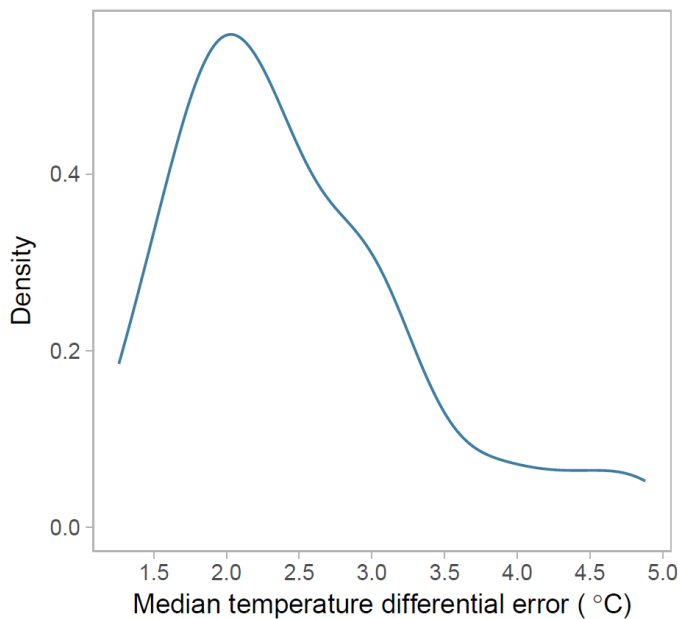


Figure 9 - Density plot for median temperature differential error at each VAV box.

6.2 COMPARISON TO AN ELECTRIC REHEAT SYSTEM

6.2.1 HVAC installation costs for hot water and electric reheat

Hot water reheat is far more common in the United States than electric reheat even though it has higher initial costs [37]. Designers typically justify the higher cost of hot water reheat on the basis of operating energy cost. The few areas of the US where electric reheat is prevalent are those that have relatively low electricity prices compared to gas, typically due to a local hydroelectric power generation resource.

Typical aggregate HVAC cost estimates per building floor area (for design, labor, equipment, and materials) for VAV systems range from \$380-490/m² in Northern California in 2014, according to a detailed worked example from the American Society of Professional Estimators [38]. However, this region has a relatively high cost of labor compared to other regions. Other resources [39,40] estimated the HVAC installation costs (excluding design) in 2010 and 2012 at the national level at \$200/m² and \$226/m² for VAV systems. Approximately 20-40% of these HVAC costs were due to hot water distribution piping, insulation, and coil hookup alone, indicating that the install cost of hot water reheat coils is a significant component of overall HVAC costs.

Though it is widely known that electric reheat systems have lower overall installation costs than hot water reheat systems, we found very few resources that quantify this difference directly. A technical design guide for VAV systems from 2007 [37] estimated the cost premium for hot water reheat over the cost of electric reheat in an overhead VAV system at \$27/m². The only other resource we found that made a direct comparison (in 2015) was from a utility, Madison Gas and Electric [41]. They estimated the cost difference for 20 VAV boxes (each with a reheat coil rated at 3.1 kW) at \$8,000. Scaling this up to the number of VAV reheat boxes in the case study building – ignoring larger reheat coils (median: 5.6 kW) in the case study building - the premium for hot water would be \$40K more than electric reheat. However, neither of these references include details about what is or is not included in that comparison – for example, labor, hot water distribution piping, etc.

We then evaluated existing publications for estimates of either hot water reheat or electric reheat costs even if they did not directly compare them within the same publication. We only found detailed cost estimates for hot water reheat systems. Leach et. al. [39] estimated the cost at a national level, of the hot water distribution piping, boiler, pumps and water balancing for a VAV system at \$32/m², again indicating that the hot water distribution initial costs are significant.

For the much more detailed and recent example in [38] in the same region as the field study building, the VAV box unit purchase cost was \$500, just 2.5% of the total cost per zone, whereas hot water piping, coil hookup, and insulation was 51%.

Looking at the whole building level, the initial cost of the gas boiler plant is also significant (approximately \$40K per boiler in this building, or \$7/m²) and would have been unnecessary with electric reheat. Lastly, we ignore the cost

associated with installing a natural gas service on site, though that would also no longer be necessary with electric reheat in this building.

These costs estimates are for illustrative purposes only as initial costs vary widely based on a large number of factors, and there are no detailed direct cost comparisons publicly available between hot water and electric reheat. An electric reheat system will clearly increase electrical panel and distribution costs, as well as wiring and labor costs as it requires line voltage power at the VAV box instead of 24V control wiring for the controller and actuator in a hot water reheat system. However, a hot water reheat system requires an entirely separate distribution system and trade to install throughout the building, while an electric reheat system expands on already required electrical infrastructure. A detailed cost comparison between hot water and electric reheat systems should estimate the ‘typical’ savings due to the hot water coil, coil hookup (valves, reducers, etc.), piping, hot water distribution piping and labor, pumps, boilers, as well as the associated labor, supervision and overhead. Similarly, such an analysis should also account for the additional costs of a high quality electric coil¹⁶, increased wiring and electricity feeder costs, as well as the associated labor, supervision and overhead. Though such a detailed analysis of costs would be valuable to the industry, it is outside the scope of this paper.

Based on the above and our engineering experience, hot water reheat is considerably more expensive to install than an electric reheat system. We use the above references to bound the range of initial cost difference. For the low side of the cost difference we take the lowest estimate from the above references: \$11/m² based on the Madison Gas and Electric publication [41] including the cost of the boiler plant. For the high side of the estimate, we take the highest estimate from the above references: \$61/m² including the hot water piping labor in the more detailed study in the same region as the case study building [38]. We round to one significant digit to indicate the high level of uncertainty, yielding a cost difference range of \$10-60/m². To be clear, we believe that both sides of this range are unreasonably low and high respectively, and that the likely ‘typical’ cost difference is approximately \$30/m².

6.3 COMPARISON OF ALTERNATIVES DESIGNS

We used the intentional reheat data from this building to compare both operating and first costs for several different system types:

1. An electric reheat system: We assumed a high quality electric reheat system that uses a modulating Silicon Controlled Rectifier and an electronic airflow sensor to allow control down to low airflow.
2. As in the previous case, but with a rooftop photovoltaic (PV) system that offsets 10% of the annual energy cost of the electric reheat system.
3. As in Case 1 with a rooftop photovoltaic (PV) system that offsets 50% of the annual energy
4. As in Case 1 with a rooftop photovoltaic (PV) system that offsets 100% of the annual energy. Note that this integrated system has a net zero energy cost on an annual basis.
5. The actual case study building: Here we assumed the boiler plant performance during operating hours only (37.1%).
6. A good boiler plant scenario: representative of two staged, modern, condensing gas boilers where each is sized to half the mid-range of the local rule of thumb (10-20 Btu/h/ft² or 31-63 W/m²). We assumed it is operating as designed, and has regular, high-quality maintenance to prevent scaling and other issues that affect boiler performance over time.
7. An exceptionally well designed and operated boiler plant: as in case 6, but using a boiler with a 20:1 turndown ratio, and with each boiler sized to half of total installed reheat coil capacity in the building.
8. An air source hot water heat pump system with an relatively high average coefficient of performance (COP: 4).

For case 1, we assumed the same electricity costs as made previously in the paper and assumed the intentional reheat loads remain the same as the estimate for the case-study building.

For cases 2, 3 and 4, we sized the PV system to generate sufficient annual revenue to offset 10, 50, and 100 percent of the operating cost of the electric reheat system respectively. We made several simplifications and assumptions to perform this analysis:

- We used the National Renewable Energy Laboratory’s solar PV potential estimation tool [42] with all default inputs for the building’s location to determine an estimate of 1531 kWh/annum generated per installed kW of solar PV.

¹⁶ Conventional electric reheat coils require high minimums. In this case, we assume the electric reheat system can control down to the same airflow as the hot water reheat cases – i.e., using a modulating Silicon Controlled Rectifier and an electronic airflow sensor capable of measuring down to low air flow rates.

- We used relatively recent installed system cost data [43] for Q1 2016 to estimate the installation cost for commercial roof PV systems of a reasonable size (>10kW) at \$2.13/ W (DC).
- We used the same fixed electricity price as above, ignoring time-of-use pricing and related effects¹⁷.

For case 6 and case 7, we assumed the boiler has the measured part load curves from Test Unit 2 and Test Unit 3 respectively, from this laboratory study, for 140 °F return water [29]. Note that this is above the condensing temperature, however operating at a lower return water temperature would require a complete redesign of the reheat coils, and the estimate for losses would no longer be accurate as the temperature differential between the building and the water would decrease significantly. For the load distribution in this building during operating hours throughout the year, these part load curves yielded an aggregate boiler efficiency of 72% and 86% boiler efficiency for Case 6 and 7 respectively. For cases 6, 7 & 8, we assumed that hot water pumping power and distribution losses did not change from case 5, the actual case study building.

Table 5 compares each of the above design alternatives by estimating the total operating cost, the initial install cost, and the payback compared to an electric reheat baseline (case 1). It clearly indicates that the operating energy savings provided by hot water reheat do not have a reasonable payback period in this building, with Net Present Values (NPV) that remain negative even after 20 years. This applies to all of the evaluated hot water reheat cases, including the combination of the most efficient hot water reheat system with the most conservative estimate of the initial cost premium for hot water reheat system, with a significant margin for error. All of the electric reheat cases that include PV are cheaper to install than any of the hot water system by a large margin. Furthermore, the electric reheat systems with PV typically also have a lower operating energy cost than the hot water reheat systems. The cases with a substantial amount of PV (cases 3 & 4, 50% and 100% PV respectively) each cost less to install and cost less to operate than every hot water system case by a large margin.

	Electric reheat				Hot water reheat				
	Base	10% PV	50% PV	100% PV	Case study building ^A	Air source heat pump	Good boiler plant	Excel. boiler plant	
All cost values are in \$/m²									
Annual operating cost	Electricity	-0.95	-0.85	-0.47	0.00	-0.17	-0.55	-0.17	-0.17
	Natural gas					-0.69		-0.36	-0.30
	Total annual operating costs	-0.95	-0.85	-0.47	0.00	-0.86	-0.55	-0.53	-0.47
	Savings to baseline		0.09	0.47	0.95	0.09	0.39	0.42	0.47
	Percent savings to baseline		10%	50%	100%	-9%	42%	44%	50%
Install cost	Reheat install cost compared to baseline					-10, -60	-10, -60	-10, -60	-10, -60
	PV install cost		-0.61	-3.3	-6.4				
	Total compared to baseline		-0.61	-3.3	-6.4	-10, -60	-10, -60	-10, -60	-10, -60
Comparison	Simple payback (years)		6.8	6.8	6.8	110, 670	25, 150	24, 140	21, 130
	Lower total installation cost than any hot water reheat option?	Yes	Yes	Yes	Yes				
	Lower operating cost than the poorest performing hot water option?	No	Yes	Yes	Yes		Yes	Yes	Yes
	Lower operating cost than the best performing hot water option?	No	No	Yes	Yes	No	No	No	
	5-Year Net Present Value (7% rate)		-0.22	-1.3	-2.6	-9.6, -60	-8.4, -58	-8.3, -58	-8.1, -58
	10-Year Net Present Value (7% rate)		0.06	0.09	0.23	-9.3, -59	-7.2, -57	-7.1, -57	-6.7, -57
	20-Year Net Present Value (7% rate)		0.4	1.8	3.6	-9.1, -59	-5.8, -56	-5.6, -56	-5.0, -55

^A For operating hours only. Including night boiler operation, the actual building annual operating cost is far higher (\$1.87/m²).

17 This simplification is likely conservative for several reasons: the PV system will reduce demand charges as it generates power concurrently with peak demand hours in typical commercial buildings, and electricity use rates are often relatively low at times when reheat systems operate.

Table 5 - Operating, install, and overall cost comparisons for the 8 design alternatives. Numbers rounded to 2 significant digits. Install cost estimates for the install cost premium of hot water reheat over electric reheat use a low and high range of 10 and 60 \$/m² respectively

For example, the electric reheat option that is most expensive to install (case 4, 100% PV, or net zero energy cost on an annual basis) is between half and 1/10th the cost premium of any hot water reheat system (using the low and high range of the hot water reheat initial cost premium that §6.2.1 describes). For further context, case 7 (50% PV) still outperforms any hot water reheat system in terms of operating costs and is between a quarter and 1/40th the cost premium of any hot water reheat system.

Perhaps even more importantly given the effect of greenhouse gas emissions on the global climate, the electric reheat cases with substantial PV (cases 3 & 4), all have lower site and source energy use than any hot water reheat option by a large margin. Case 4 allows the HVAC system to be all-electric, requiring no fossil fuel combustion on site. This indicates that an electric reheat system served by a PV system can be a far better choice than a hot water reheat system from an initial cost, operating energy cost, and (site and source) energy use perspective.

6.3.1 Other considerations in the comparison of design alternatives

Photovoltaic panels require roof space which may not be available. To provide context here, the PV system sized to 10% of the total annual reheat load (case 2) is a 3.5 kW PV system which requires approximately 27 m² of roof area, or 1.4% of the total roof area of this 5-floor building. The boiler plant that is currently on the roof – which only serves the reheat system - covers more than this (3% of the roof area).

Lastly, electric reheat has potential advantages compared to hot water reheat that we do not evaluate in this paper. For example, maintenance costs are typically lower and leaks are not possible. Passing valves and the associated energy waste both at the zone and at the AHU, are not possible. Additionally, electric reheat coils have a negligible airside pressure drop, whereas hot water reheat coils typically add 38 - 75 Pa (0.15-0.3 in.w.c) that the AHU fan must overcome, increasing initial costs and operating costs.

6.4 LIMITATIONS

6.4.1 Limitations of the paper

We investigated one building in this paper and the findings must be taken in that context. This building has modern controls at the zone level (dual-maximum control logic and low minimum airflow rates) which reduce the amount of unnecessary reheat compared to older buildings. However, the building code (California Energy Code) requires these technologies for all new buildings and retrofits. Furthermore, the analysis compared the measured reheat gas consumption in this building to a large dataset of measured total heating gas consumption in other office buildings in the same region. This analysis showed that the building consumed less gas than the average of that dataset, but not unreasonably so given that those buildings are on average older, with far less modern controls, and have at least some additional gas heating consumption that is not due to the reheat system. Also, the magnitudes of the differences in first and operating costs shown in Table 5 are large enough that even doubling the reheat consumption would not change the overall cost related findings.

There are building level control strategies that can potentially reduce both boiler losses and hot water distribution losses. These strategies were not in place in this building and could potentially reduce reheat use, boiler losses, and distribution losses. For example, disabling the boiler unless a zone (or zones) requires significant reheat, or decreasing the hot water supply temperature when there is little demand for reheat. However, how much of a reduction would depend highly on implementation. For example, at least one reheat valve was open in the building for 95.5% of the AHU operating hours, so a strategy of disabling the boiler in the absence of any reheat demand would have little effect.

Lastly, this case study was in a mild winter climate, and the need for intentional reheat could be even lower in a hot climate, further favoring electric reheat. However, that is not certain as it depends highly on the type of AHU supply air temperature control strategy operating in the building, as well as the layout of zones in the building and the AHUs that serve those zones. Conversely, colder climates are likely to move towards favoring a hot water reheat system. However, the design of the HVAC system and the building would have a large impact on those results. For example, buildings with relatively little exterior-exposed floor area, or buildings with independent AHUs serving different exposures of the building, could handle most of the heating requirements at the AHU heating coil instead of the zone-level reheat coil. Again, this depends highly on the effectiveness of the supply air temperature reset strategy. Thus, it is clear that the building, HVAC and controls design will impact this comparison, and likely more so in more extreme climates than the climate in which this case study building is located.

6.4.2 Limitations of the reheat estimation method

This method relies on data from sensors that are typically required and installed as part of a modern VAV system. This has the benefit that analysts can apply the same method to other buildings without requiring additional hardware, but it is important to bear in mind that those sensors are relatively poor quality and uncalibrated. Though §5.7 indicates that the accuracy is reasonable in this building, this may not be the case in other buildings due to any combination of reasons such as different manufacturers, age, installation, etc. Furthermore, the characteristics of the HVAC system itself may affect the accuracy of the method. For example, though it is not that common, in HVAC systems where the supply air temperature sensor is upstream of the fan the method would also need to account for fan heat gain. This varies with both airflow and pressure, and that variation would add an additional source of error to the ΔT_e^i calculations. Similarly, all of the reheat coil valves were recently replaced in this building so there are likely few, if any passing valves. This is unlikely to be the case in other buildings with older (or poorer quality) valves. Passing valves are a source of wasted energy consumption and comfort issues in buildings, but they would also add a new source of error in the method that would be difficult to quantify independently from the distribution losses.

Lastly, we typically assume the ‘best case’ scenario for the comparison of design alternatives for this building. Real buildings never operate this way, and it is difficult to predict how those alternatives would operate in practice, given confounding effects like manual overrides, faults, etc.

7 CONCLUSIONS

This paper provides a method for estimating intentional reheat energy consumption which uses only the data available from sensors typically installed with a modern VAV system. We demonstrated and validated that method on a large office building over a one year period. For the first time to date, we assessed the overall efficiency of the reheat system in a real building, quantifying intentional reheat energy use, hot water distribution losses and boiler efficiency. In the case study building, the overall hot water system efficiency – from gas consumed by the boiler system to the sum of the intentional reheat use at all VAV terminal units - was just 21% during operating hours. The 79% losses were due to a combination of boiler and hot water distribution system losses.

The data from this study indicates that intentional reheat energy use is low in this building compared to the total installed reheat coil capacity – the median reheat use is just 2%, and the maximum in the year is 23% of installed capacity. This is mostly due to modern controls such as dual-max logic and low minimums, both of which are required by the current applicable energy code in this location. The total hot water supplied by the boiler during operating hours over the year was 91 MWh (8.3 kWh/m² of floor area per annum, or 2.5 W/m² averaged over operating hours). Of that, 40 MWh (3.6 kWh/m², 1.1 W/m²) was lost due to hot water distribution losses. That was 44% of the total heat supplied by the boiler. The remaining intentional reheat energy use was 51 MWh (4.6 kWh/m², 1.4 W/m²). Also, the median of the 1-minute interval data for hot water supplied by the boiler during operating hours was 19 kW (1.8 W/m²), which is just 4.8% of the total installed reheat capacity. These very low median percentage values caused the boiler system to typically operate at ultra-low part load, where boilers are extremely inefficient. This building has an oversized boiler plant and staging issues which exacerbates this issue. However, these median part load ratios are so low that most boiler plant designs would still operate inefficiently the majority of the time.

This paper analyzes a single building in detail and that building was operating with current best practice controls that significantly reduce reheat energy consumption (dual-maximum control logic and low minimum airflow setpoints). However, a high-level comparison of gas consumption data to a large dataset of other buildings demonstrates that this building (22 kWh/m² per annum gas consumption) is comparable to office buildings in this climate zone (mean of 54 kWh/m² per annum gas consumption). While we do not have the detailed data to calculate boiler losses, distribution losses or intentional reheat for buildings in this dataset, the gas data indicates that this building is not a distant outlier in terms of gas consumption. This, and comparisons to other datasets in the paper, imply that the findings of this paper are likely more broadly applicable than the results of a single case study building evaluation suggest.

Finally, this paper indicates that end-to-end efficiency is extremely relevant to reheat system selection. In comparisons of hot water to electric reheat to date, analysts often focus on the utility cost of energy and the nominal boiler efficiency, nominal part load efficiency, while assuming relatively high reheat energy use and ignoring hot water distribution loss and realistic boiler standby and cycling losses. This study has shown each of these to be a significant consideration for system selection. For this case study building, an electric reheat system served by a photovoltaic panel system would have had a far lower initial cost, a lower operating energy cost, and a lower (site and source) energy cost. This applied even with the most conservative initial cost assumptions and across all hot water cases, even with an exceptionally

efficient redesign of the boiler system. Though further case studies are needed to confirm if this finding is generalizable, the implications are potentially very significant for the building industry and policy makers. It indicates that the current – almost universal – practice of serving a reheat system using a gas-fired boiler and hot water distribution system may not be the best choice in many cases.

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