Title

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ABSTRACT

Variable air volume systems with hydronic reheat at terminal units are a common Heating Ventilation Air Conditioning (HVAC) system type in medium and large commercial buildings. This study measured HHW heat loss in detail in a 66,000 ft^2 (6,200 m^2) office and lab building, built in 2000, in Davis, California. We used methods adapted from Raftery et al. (Raftery, Geronazzo, et al. 2018) to calculate the HHW distribution losses from BAS measured data, and then measured unintentional heat loss at the whole building level including losses from distribution and passing HHW valves. We further measured HHW distribution losses in greater detail on a single HHW distribution branch removing loss contributions from other potential issues, such as passing HHW valves.

For the whole building, using newly installed, calibrated water flow meter and matched pair calibrated RTD HHW supply and return temperature sensors, typical HHW setpoints, with all air handlers turned off, the steady-state unintentional heat loss was 4.4 W/m^2 (1.4 $Btu/h.ft^2$) when all VAV terminal unit HHW valves were commanded shut, and 3.2 W/m^2 (1.0 $Btu/h.ft^2$) when one HHW valve was commanded open.

Focusing on one HHW branch, during normal building operation over a two-month period in the heating season, we used BAS readings for air flow rate, supply air temperature, and discharge air temperature and measured a distribution heat loss of 2.86 W/m^2 (0.91 $Btu/h.ft^2$) and 40% HHW distribution efficiency. Using separately installed, calibrated temperature sensors yielded a similar result (2.43 W/m^2 (0.77 $Btu/h.ft^2$), 49%), and further correcting air flow rates with passive flow hood single point calibration of BAS reported flow rates also yielded a similar result (2.76 W/m^2 (0.87 $Btu/h.ft^2$), 42%). The close agreement between the results using BAS and calibrated sensors suggest that existing buildings can be screened for heat loss reduction interventions using only BAS data.

The magnitude of the measured HHW losses are small compared to design day loads, but they occur for a large number of hours so reducing these losses can save substantial energy. Further, during the cooling season the losses both waste heat and increase cooling loads. Paths forward include adopting aggressive heating hot water supply temperature resets, reducing unnecessary reheat operation, improving HHW pipe insulation practices, and/or changing design strategies to seasonal switchover or electrically driven distributed systems such as electric resistance or terminal unit heat pump equipment.

INTRODUCTION

Many large and medium commercial buildings use HVAC system architectures based on Variable Air Volume (VAV) air handler unit (AHU) air supply with hydronic reheat coils in terminal units to control space temperatures. Heating Hot Water (HHW) is distributed from the central water heating equipment, typically boilers or heat exchangers, to the terminal units throughout the building and then back to the central mechanical room in insulated pipes. Unintentional heat lost from the distribution pipes leads to increased energy consumption through both a reduction in intentional heating service provided for a given amount of energy consumed, and also through unintentional heating that is either unnecessary, or at worst, potentially

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requires additional cooling.

The magnitude of the HHW unintentional heat loss has not been well studied. Raftery et al published a method for estimating HHW unintentional heat loses based on calculations from building automation system (BAS) sensor readings with results in one recently retrofitted office building in San Mateo California with high quality controls commissioning showing that, over the course of a year, distribution losses accounted for 44% of the heat output from the boiler adding up to 1.1 W/m^2 (0.35 Btu/h.ft^2), whereas the intentional reheat energy use added up to 1.4 W/m^2 (0.44 Btu/h.ft^2) (P. A. Raftery 2018). UC Davis joined the team from that earlier publication to extend the analysis to measurements from a total of 7 commercial buildings with results ranging from 0.8 to 2.0 W/m^2 (0.25 to 0.63 Btu/h.ft^2) (Raftery, et al. 2023). Recent work (Wendler, Raftery and Cheng 2023) has also performed full scale laboratory testing of VAV reheat terminal units, highlighting losses from the valve train and piping serving the coil, as well as substantial reductions in heating capacity due to damper position while maintaining the same airflow.

Few building energy models include unintentional heat loss from HHW piping, and very few simulation engines even provide the capability of modeling it. For context, typical average measured office plug loads are 3 W/m^2 (1 Btu/h.ft^2) and are considered large enough to warrant consideration.

METHODS

Building Selection

Buildings were reviewed for physical and data access, completeness of mechanical plans, use of VAV reheat HVAC, existence of BAS with terminal unit air flow and discharge air temperature sensors, and reasonable BAS data outputs leading to selection of Ghausi Hall on the UC Davis campus in Davis California, US climate zone 3B (California Climate Zone 12). Ghausi Hall was built in 2000 and is a mixed office and laboratory building with three stories and 6175 m^2 (66,462 ft^2). There are five air handler units (AHU) serving single duct distribution systems with terminal units serving thermal zones throughout the building. The terminal units use an actuated damper to regulate airflow to the zone in response to zone temperature requirements and use flow stations to measure air flow rates for feedback control. For the detailed heat loss measurements, one HHW branch on the top floor was selected that feeds 9 VAV terminal units that serve 14 private offices, 3 computer labs, the copier area and foyer of a small open office area, and a section of hallways, with a total floor area of 355 m^2 (3,855 ft^2). The top floor branch serves perimeter zones with private office rooms that have exterior walls facing West, North, and East, as well as three core rooms that are lightly used computer labs, and the adjoining hallways. The HHW piping, along with the VAV boxes and all ductwork, are in the return air plenum, above the drop ceiling. Heat lost from the piping into the return plenum returns to the air handler. The HHW distribution pipes are insulated, but the HHW valves and 6 inch to 4 foot sections of pipe upstream and downstream are uninsulated.

Identification of Intentional Reheat and Calculation of Unintentional Heat Loss

We estimated the intentional reheat energy use and unintentional heat loss by performing both a water side heat balance and an air side heat balance. We operated the whole building with all AHUs off so there would be no intentional air flow and all HHW valves commanded shut so that the only HHW heat loss is unintentional heat loss. This whole building unintentional heat loss includes losses from the pipes and from passing HHW valves[1]. We performed a water side heat balance, equation 1, using the HHW temperatures and flow rates at the whole building HHW supply and return risers where $q_{building,tot}$ is the water heat loss rate.

[1] Passing valves allow fluid to pass through them when the valve actuator appears fully closed, also called valve leakage.

$$q_{building,tot} = \dot{m}_w \cdot c_w \cdot (T_{sw} - T_{rw}) \tag{1}$$

Where:

- $\dot{m}_w$ is the mass flow rate of the HHW
- c_w is the heat capacity of water
- T_{rw} is the HHW riser return temperature
- T_{sw} is the HHW riser supply temperature

For the selected top floor branch of the HHW loop serving 9 VAV terminal units with normal building operation, subtracting the intentional reheat air side heat gain energy from the total water side heat loss energy gives the unintentional heat loss energy. We estimated the intentional reheat energy using a modified air side heat balance, from Raftery et al 2018, calculated with the reheat valve position, discharge air temperature, airflow measured in each zone, and AHU supply air temperature (Raftery, Geronazzo, et al. 2018). We eliminated bias error between the AHU supply and zone discharge temperatures by first estimating the long-term temperature difference (T_e^i) for each terminal unit when the reheat valve has been closed for an extended period and subtracting this bias error when calculating air side capacities. This corrects for the effects of sensor bias error, duct heat transfer occurring between the AHU supply and the Terminal unit discharge air temperature sensors, and eliminates the effect of passing HHW valves.

$$T_{e,t}^i = \begin{cases} \alpha \cdot T_{e,t-1}^i + (1-\alpha) \cdot \left(T_d^i - T_s\right), & \prod_{j-1}^{3\cdot\tau} \Omega_{r,t-j}^i = 0 \\ T_{e,t-1,}^i & \prod_{j-1}^{3\cdot\tau} \Omega_{r,t-j}^i > 0 \end{cases} \tag{2}$$

Where, for the *i*-th terminal unit:

- $T_{e,t-1}^i$ 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 is the supply air temperature leaving the AHU serving that terminal unit [°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).

We performed this smoothing bias error estimation only when both of the following conditions are met:

1. the reheat valve supplying that reheat coil is currently and has been commanded fully closed for longer than the last 3 τ 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 min for each terminal unit based on a time series analysis.
2. the AHU was operating (i.e., the fan is running and there is 50 cfm or greater measured airflow through the terminal unit).

Requiring the valve to be fully closed for a period of 3 τ 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 T_e^i estimate. When either condition above is not met, we kept the T_e^i estimate constant to avoid an impact on the bias error estimate during a reheat call.

We used α = 0.98 for this paper. Note here that larger values of α cause the value of T_e^i to be more representative of the long-term average value, whereas smaller values make T_e^i more responsive to shorter term changes in the system. Regarding initialization, we initialized this calculation using the instantaneous temperature difference at that time (t = 0), i.e. by setting $T_{e,-1}^i$ equal to $T_d^i - T_s$.

We then used this long-term temperature difference in the airside heat balance calculation to compute the intentional reheat used at the *i*-th terminal unit (q_{TU}^i) and summed for all 9 terminal units on the branch by applying the following equations:

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

$$q_{TU,tot} = \sum_i q_{TU}^i \quad (4)$$

Where, for the i-th terminal unit:

- T_d^i is the discharge air temperature leaving the terminal unit reheat coil [°C]
- v_a^i is the air flow rate [m^3/s]
- T_s is the supply air temperature leaving the AHU serving that VAV terminal unit [°C]
- ρ_a is the density of dry air [1.205 kg/m^3] 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 3 τ min. Outside of the above conditions, the intentional reheat for a particular terminal unit 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 3 τ 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 terminal units that change in and out of heating mode frequently.

Examining the time series data shows that the discharge air temperature does not rise until the HHW valves have reached values of approximately 12% open. Building on the methods from Raftery et al, we defined the HHW valves to be open above a 12% value and to be closed below the 12% value as a HHW valve mechanical backlash correction. We also used a minimum flow rate of 50 CFM (1.42 m^3/h) was used to determine times when transient behavior in the bias temperature error calculation should be eliminated.

For the selected top floor branch, this intentional heating capacity with normal building operation was calculated using a combination of installed calibrated sensors and BAS measurements. The Intentional heating capacities were then totaled for each Terminal unit and totaled again over all the Terminal units in the studied branch. To determine the distribution heat loss, we subtracted this total intentional air heating capacity from the water total heat loss in the branch. The total heat loss rate ($q_{b,tot}$) was calculated via (eq5) using a matched pair of RTDs on the water side of the top floor branch.

$$q_{b,tot} = \dot{m}_w \cdot c_w \cdot (T_{sw} - T_{rw}) \quad (5)$$

Where:

- $\dot{m}_w$ is the mass flow rate of HHW in the branch
- c_w is the heat capacity of water [4.190 kJ/kg°C].
- $T_{rw}$is the HHW branch return temperature leaving the branch
- T_{sw} is the HHW branch supply temperature entering the branch

Instrumentation

For the whole building measurements, the following sensors and equipment were used. The building automation system (BAS) is a Siemens Apogee (PXC100-PEA) controller with and RTD temperature sensors (AI Pt 1000). The BAS system exports data to an OSIsoft PI System and SkySpark database repository system. The existing whole building HHW flow meter was replaced with a new Onicon F-3500-11-C3-1211 insertion electromagnetic flow meter accurate to plus or minus 1.0% of rate from 0.61 – 6.1 m/s (2.0 - 20.0 ft/sec) velocity and plus or minus 6.6 mm/s (0.02 ft/sec) below 0.61 m/s (2 ft/sec), calibrated against a NIST traceable primary standard accurate to within 0.1%. The existing whole building HHW supply and return temperature sensors were replaced with MAMAC Systems immersion 3-wire 1000-ohm platinum RTDs (TE-703-C-3-A-2: ± 0.1% @ 0°C) with custom matched pair calibration to reduce any potential bias between the two sensors over a temperature range from 49 to 93°C (120 to 200°F) to more accurately estimate the heat loss from the hot water loop.

The top floor HHW branch selected for this study serves 9 VAV terminal units. The 9 VAV terminal units were instrumented with entering air and leaving air temperature using calibrated RTDs with the measuring resistance element distributed across the length of the probe to capture an average temperature from the bottom of the duct to approximately half the duct height (Omega PR-10L: ±.35ºC @ 100°C). For the branch HHW supply and return temperatures and for each terminal unit the hydronic reheat coil entering water and leaving water temperatures were measured using calibrated pipe surface RTDs, with extreme care for good adhesion and insulation (Omega SA1-RTD: ±.35C @ 100°C). Before installation, all the RTDs were clustered together and highly insulated while monitoring their readings over time, matched pairs of RTDs showing minimum offset and similar trend in temperature readings were selected to measure the change in temperature between the entering and leaving flows to minimize offset bias error. The branch HHW supply and return flow rates were measured using Ultrasonic flow meters (Flexim F601 Ultrasonic Liquid Flow Meter with ultrasonic shear wave transducers: ±1.6 %). HHW flow rates were measured at the branch supply, and branch bypass so that supply, bypass, and total return flows can be calculated. To position the ultrasonic flow meter transducers far enough from flow disturbances given constraints of the existing piping layout required measurement downstream of the first terminal unit and an extra measurement of the HHW flow for that terminal unit, with the reading subsequently added to the supply flow and subtracted from the bypass flow. All data acquisition used DT85 series 3 DataTaker data loggers.

For the top floor branch, the data collected by the existing BAS sensors was also analyzed including the supply air temperature (SAT) leaving each of the rooftop air handlers, the discharge air temperature (DAT) for each terminal unit (SEIMENS 535-741-4: ±0.5°C), and the air flow rate for each terminal unit. The terminal units use AeroCross Multi-point Center Averaging flow stations to measure air flow rates claimed to have "+/-5% accuracy, no matter what the inlet conditions may be" (Titus 2002).

RESULTS

Whole Building – No Air flow

Whole building unintentional heat loss was measured while unoccupied, with all fans commanded off so that there was no intentional air flow. To avoid any effects from unusual space temperatures or heating recovery, the building HVAC use the normal occupied settings for the preceding two days even though the building was unoccupied. The HHWS and HHWR temperatures were measured using a custom calibrated matched pair of RTDs to minimize any bias between the two sensors and the temperature difference was multiplied by the mass flow rate of water and the water specific heat. This whole building HHW unintentional heat loss was measured with all HHW valves closed so that there was no intentional hot water flow through any of the terminal unit coils, fan coils, or air handlers during the measurements and for a period more than one hour before. Building controls were set and data for the heat loss calculation was selected after sensor readings had reached a steady value. For the no air flow and closed HHW valves measurement the result of 4.43 W/m^2 (1.4 Btu/hr.ft^2) of unintentional heat loss from the HHW system with average conditions of: Outdoor Air Temperature of 12.9°C (55.3°F), HHW flow rate 0.94 l/s (14.9 gpm), end of line differential pressure 83 kPa (12 psi), and HHWS temperature of 82.2°C (179.9°F), and HHWR temperature of 75.4°C (167.5°F) , as shown in Table 1. The measurement was repeated with one HHW valve commanded open, 0.025 l/s (0.39 gpm) flow rate, in one terminal unit on the top floor with a results of 3.21 W/m^2 (1 Btu/hr.ft^2) of unintentional heat loss from the HHW system with average conditions of: Outdoor Air Temperature of 13.5°C (56.3°F), HHW flow rate 1.07 l/s (17.0 gpm), end of line differential pressure 90kPa (13 psi), HHWS temperature of 82.2°C (180°F), and HHWR temperature of 78°C (172°F), as Table 1 shows.

Table 1. Whole Building HHW Unintentional Heat Loss with Fans Off

Average over test period	HHW Valves Closed	One Terminal Unit HHW Valve open (0.39 gpm flow)
Waste Heat, std dev	4.43 W/m^2, 0.47 (1.4 Btu/hr.ft^2)	3.21 W/m^2, 0.46 (1.0 Btu/hr.ft^2)
Outdoor Air Temperature	12.9°C (55.3°F)	13.5°C (56.3°F),
HHW Flow Rate	0.94 l/s (14.9 gpm)	1.07 l/s (17.0 gpm)
End of Line Differential Pressure	83 kPa (12 psi)	90kPa (13 psi)
HHWS	82.2°C (179.9°F)	82.2°C (180°F)
HHWR	75.4°C (167.5°F)	78°C (172°F)

Top Floor Branch – Normal Operation

The distribution heat loss of the top floor branch reheat HHW supply and return system was calculated from BAS outputs and also measured using calibrated sensors over a two month winter period during normal building operation. HHW flow rates and temperatures and flow rates were measured using ultrasonic HHW flow meters and a calibrated matched pair of HHW temperature sensors measuring the branch inlet and outlet water temperatures to determine the total reduction in thermal energy carried by the HHW. During this period the average HHWS temperature entering the branch was 73°C (164°F) and HHWR temperature leaving the branch was 34°C (94°F).

Supply air temperatures and discharge air temperatures were measured with BAS or calibrated sensors with any long-term bias between the entering and leaving air temperature readings subtracted from the measured temperature difference to determine the air heat gain corrected for sensor bias and in the BAS sensor case where SAT is at the AHU also corrected for duct temperature gain and remove impact of passing HHW valves. Intentional air heat gain was defined as the cumulative increase in thermal energy from the terminal unit air entering temperature to the leaving air temperature minus the long-term temperature bias, for times when the BAS control command for the HHW valve rose to more than 12% open and lasting until the valve command was reduced to less than 12% plus a 12 minute time constant to account for the intentional air heating that happed after the HHW valve closed.

The air flow rates for each terminal unit were measured at a 50% air flow rate setpoint using a passive flow hood on each supply diffuser with three replicate measurements that were each individually compared to the BAS air flow reading output at the time of each measurement. The average passive flow hood measured air flow was 15.7% higher than BAS outputs with the range across terminal units from 4% to 34%, see Figure 1.

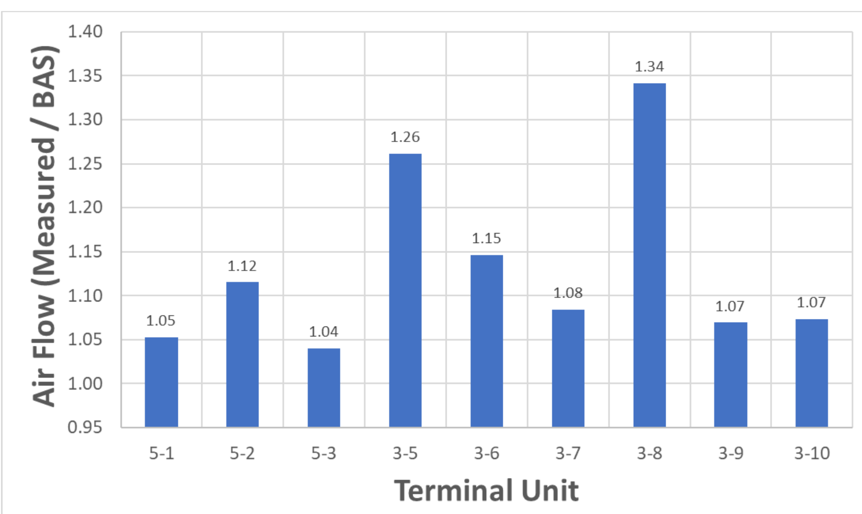

Figure 1 Passive flow hood measured air flow rate divided by the simultaneous BAS air flow rate output for each terminal unit on the top floor branch.

HHW distribution heat loss was calculated by subtracting the intentional air heat gain from the total HHW thermal energy reduction using BAS outputs and also using the calibrated sensor measured values, Figure 2. Using the BAS output for air (temperatures, flow rates), along with the readings from the installed calibrated sensors for HHW flow rates and HHW entering supply and leaving return temperatures gives a HHW distribution heat loss result of 2.86 W/m^2 (0.91 Btu/h.ft^2), and intentional air heating of 1.92 W/m^2 (0.61 Btu/h.ft^2) for an HHW branch delivery efficiency of 40.2%. Using the corrected BAS output for air flow rates, along with the readings from the installed equipment for terminal unit entering and leaving air temperatures, HHW flow rates, and HHW top floor branch entering and leaving temperatures gives a HHW distribution heat loss result of 2.43 W/m^2 (0.77 Btu/h.ft^2), and intentional air heating of 2.36 W/m^2 (0.75 Btu/h.ft^2) for an HHW branch delivery efficiency of 49.3%. These results show a 17.7% larger HHW distribution heat loss when using BAS sensor data compared to when using calibrated instruments. The majority of this difference in results is due to the air flow rate correction. There is significant uncertainty in applying the air flow rate correction based on a single damper position to a range of actual damper position operation. Using the calibrated sensor readings with the BAS reported air flow rate gives a HHW distribution heat loss result of 2.76 W/m^2 (0.87 Btu/h.ft^2), and intentional air heating of 2.03 W/m^2 (0.64 Btu/h.ft^2) for an HHW branch delivery efficiency of 42.3%.

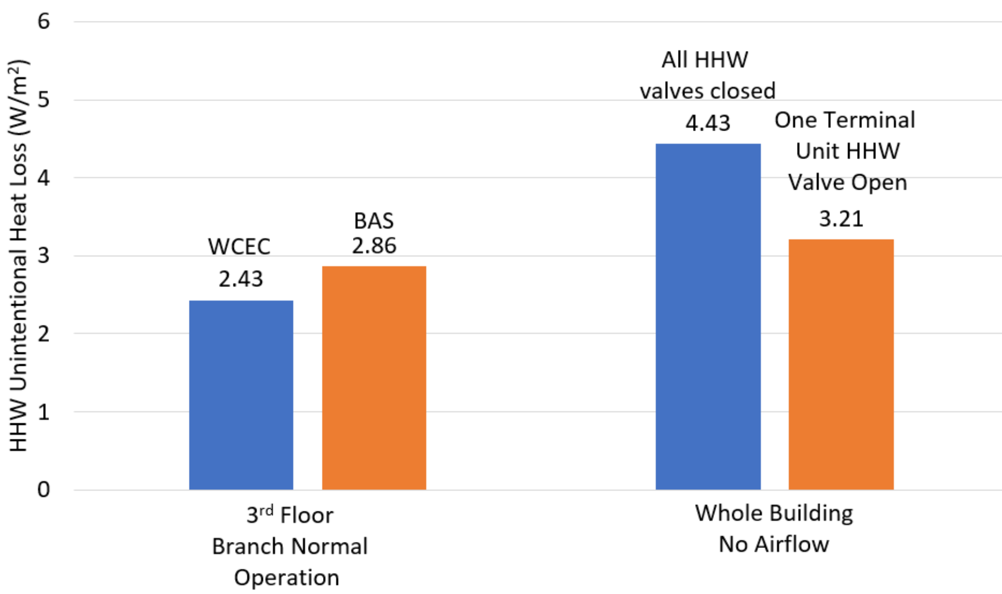

Figure 2 HHW unintentional heat loss results

DISCUSSION

The BAS and calibrated instrument HHW distribution heat loss results are relatively close with the main differences caused by the air flow rate correction. Future work can use calibrated instruments to map the air flow rate through each terminal unit across multiple damper positions to better assess BAS air flow accuracy impacts on HHW distribution heat loss results in normal operation.

The HHW unintentional heat loss of 2.43 to 4.43 W/m^2 (0.77 to 1.4 Btu/h.ft^2) results from this study are larger than the 0.8 to 2.0 W/m^2 (0.25 to 0.63 Btu/h.ft^2) found by Raftery et al (P. A. Raftery 2018), but similar order of magnitude. The magnitude of the measured unintentional hot water heat loss suggest that significant energy savings are possible by reducing unintentional losses. Possible paths forward include adopting aggressive heating hot water supply temperature reset schedules, reducing unnecessary reheat operation, improving insulation practices, and or changing reheat strategies to electric resistance.

The HHW valve backlash correction using 12% threshold shows 8.5% higher HHW distribution heat loss compared to using a threshold of 2%. Future work using BAS data to screen buildings for interventions can assess a range of HHW valve backlash correction thresholds to identify a reasonable value where the results are less sensitive to changes in the threshold.

CONCLUSION

For the whole building, using new calibrated water flow meter and matched pair calibrated RTD HHW supply and return temperature sensors, typical HHW control setpoints, with all fans turned off, the steady-state unintentional heat loss was 4.4 W/m^2 (1.4 $Btu/h.ft^2$) when all VAV terminal unit HHW valves were commanded shut, and 3.21 W/m^2 (1.0 $Btu/h.ft^2$) when one HHW valve was commanded open. These unintentional heat loss results are larger than the 0.8 to 2.0 W/m^2 (0.25 to 0.63 $Btu/h.ft^2$) found by Raftery et al. (Raftery, et al. 2023), but similar order of magnitude.

Focusing on one HHW branch, during normal building operation over a two-month period in the heating season, we used BAS readings for air flow rate, supply air temperature, and discharge air temperature and measured a distribution heat loss of 2.86 W/m^2 (0.91 $Btu/h.ft^2$) and 40.2% HHW distribution efficiency). Using separately installed, calibrated temperature sensors yielded a similar result (2.43 W/m^2 (0.77 $Btu/h.ft^2$), 49.3%), and further correcting air flow rates with passive flow hood single point calibration of BAS reported flow rates also yielded a similar result (2.76 W/m^2 (0.87 $Btu/h.ft^2$), 42.3%). The HHW distribution efficiency results from this study show less than half of the HHW heat going into intentionally heated air, similar to the 56% HHW distribution efficiency found by Raftery et al. for a different office building using BAS measurements (Raftery, Geronazzo, et al. 2018).

The relatively close agreement between the heat loss results using BAS and calibrated sensors suggest that existing buildings can be screened for heat loss reduction interventions using only BAS data. The magnitude of the measured HHW unintentional heat loss and distribution losses are relatively small but occur for a large number of hours, so reducing or avoiding them can save significant amounts of energy. During the cooling season HHW losses both waste heat and increase cooling loads. Paths forward include adopting aggressive heating hot water supply temperature resets, reducing unnecessary reheat operation, improving HHW pipe insulation practices, and/or changing design strategies to seasonal switchover and or electrically driven distributed systems such as electric resistance or terminal unit heat pump equipment.

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