

Performance of a Combination Heating System for Residential Applications

by

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ABSTRACT

Advancements in water heating and control technology have improved the performance of tankless water heaters in recent years. A tankless water heater coupled with a hydronic air-handling unit functions as a novel system known as a combination (“combo”) system that is used for residential space and water heating applications. However, the performance of such a system has not been well studied to date. In this thesis, an experimental method was developed to analyze the performance of the combo system for space heating only operation and simultaneous (space and water heating) operations. To quantify performance of the combo system, the effect of the outlet water temperature from the water heater and the effect of the flow rate of water for domestic usage on first law efficiency and hydronic heat exchanger second law efficiency were studied. With the experimental assembly, it was found that the thermal efficiency of the tankless water heater during space heating and simultaneous operation ranged between 39% and 95%. Two adverse behaviours, short-cycling and flow-cycling, were observed in the combination system operation and were found to have a negative impact on system efficiency. The hydronic heat exchanger exergy analysis results suggested that the combo system has comparable second law efficiencies when compared to other conventional hydronic space heaters. The results of the first and second law analyses suggest that combination systems have equivalent performance similar to mid- and high efficiency furnaces and conventional water heater units, provided that adequate control strategies are in place to optimize combination system operation.

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NOMENCLATURE AND ABBREVIATIONS

Symbols

c_p	specific heat capacity (kJ·kg ⁻¹ ·K ⁻¹)
\dot{E}	rate of energy (kW)
$E\dot{x}$	rate of exergy (kW)
\dot{H}	rate of total enthalpy (kW)
h	specific enthalpy (kJ·kg ⁻¹)
h_f°	specific enthalpy of formation (kJ·kg ⁻¹)
s	specific entropy (kJ·kg ⁻¹ ·K ⁻¹)
\dot{S}	rate of entropy (kW·K ⁻¹)
\dot{W}	rate of electrical work (kW)
\dot{Q}	rate of heat loss (kW)
X	mass fraction
\dot{m}	mass flow rate (kg·s ⁻¹)
\dot{V}	volumetric flow rate (m ³ ·s ⁻¹)
LHV	Lower heating value (kJ·kg ⁻¹)
t	time (s)
T	temperature (°C)

Greek Symbols

η	efficiency
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Subscripts

CV	control volume
in	inlet
out	outlet
gen	generated
ng	natural gas
ca	combustion air
cg	combustion products
a	air (at ambient)
ra	return air
sa	supply air
mw	TWH inlet water line
DHW	domestic hot water line
rw	AHU water outlet line
avg	average
0	dead state (298 K, 1 atm)
i	species
I	first law
II	second law (exergy)
v	water vapour in stack
f	flue gases in stack

1. INTRODUCTION

1.1. RESIDENTIAL ENERGY USAGE IN CANADA

As global warming becomes an increasingly prevalent issue in society, the question of how energy is used within the confines of society falls under scrutiny. In May 2015, the Canadian government set a target under the Paris Agreement to reduce GHG emissions by 30% below 2005 emission levels by 2030 [1]. In Canada, the main sectors that affect GHG emissions are: industrial, transportation and residential [2]. Additionally, the transportation, industrial, and residential sectors are likewise the main sectors with respect to energy usage. In Canada, the residential sector accounted for approximately 16% of the total annual energy used and 14% of the total annual GHG emissions [3]. More specifically, space and water heating accounted for approximately 80% of the annual energy usage for households [3] with natural gas and electricity as the main sources of energy [2]. This is primarily due to Canada's cold climate and the cultural approach to use a high-grade energy sources to meet space and water heating demand. Therefore, reduction of energy usage and associated GHG with residential space and water heating will play a significant role in meeting Canada's climate change goals.

Despite the introduction of regulated medium- and high-efficiency furnaces into the residential sector over a twenty-year period, the total energy usage for space heating over the same time period increased by 4.3% [3]. Similarly, for the same time period, the total energy usage for water heating increased by 8.2% even with the introduction of minimum energy performance standards and higher efficiency water heaters into the residential sector [3]. Overall

energy efficiency gains by the implementation of higher efficiency appliances for space and water heating were not realized when considering total energy usage due to a variety of mitigating factors, including increases in: population, number of households, and larger average household size. Therefore, as Canada's energy demand continues to grow, the need for market-ready water and heating systems that can minimize energy consumption along with other related parameters (emissions, energy costs, energy losses) without compromising occupant comfort or productivity will increase as well [4].

1.2. TANKLESS WATER HEATER COMBINATION SYSTEMS

Advancements in water heating and control technology have improved the performance of tankless water heaters (TWHs) in recent years. In the United States, TWH systems have seen an increase in market share due to utility provider and government incentive programs such as Energy Star and increased marketing from manufacturers [5]. As well, TWHs are high efficiency units for domestic hot water (DHW) production. TWH units have superior energy factor (EF) ratings compared to conventional water heating systems such as storage water heaters (SWH). This result is primarily due to the elimination of standby heat losses as the water in TWH systems is heated on demand as opposed to SWH systems that maintain a large volume of volume at a constant elevated temperature. As such, typical non-condensing TWH units have energy factor ratings ranging from 0.80 to 0.83 [5]. This performance is in contrast to SWHs where typical SWH unit energy factor ratings range between 0.55 and 0.63 [6]. In fact, SWHs with energy factor ratings exceeding 0.80 have yet to be developed [5].

A tankless water heater coupled with a hydronic air handling unit (AHU) form an alternative space and water heating system known as a combination (“combo”) or integrated appliance system. A combo system is unique in that the system is capable of providing space and hot water heating by way of a single, unified system as opposed to the conventional assembly of two independent systems such as a furnace or electric baseboard for space heating and a SWH for domestic hot water (DHW) usage. Improvements in building science have reduced space heating loads and, combo systems have become an alternative for consumers to consider from a cost mitigation and space utilization perspective. Furthermore, from a space heating perspective, combo systems present an advantage in that users are able to exhibit control over the temperature of the air delivered for space heating through modulation of the TWH water outlet setting controller. This control is distinctly different from space heating furnaces, which generally do not allow for modulation of space heating capabilities. The ability to control the DHW temperature set points independently and influence space heating allows the combo system increased flexibility in performance when considering consumer comfort and safety. DHW temperature user set points are a balance between preventing scalding injuries and minimization of bacterial growth (such as legionella bacteria) [7, 8], and in combo systems these set points have direct influence on space heating performance.

The benefits of combo systems with a TWH heating plant include: superior energy factors compared to SWHs [5, 9] for DHW production, high water heating capacity, elimination of water storage losses, low idle period losses, and reduced greenhouse gas (GHG) emissions. Secondary benefits include reduced space requirements due to number of fuel line connections and single intake/exhaust line. This secondary benefit may allow for increased consumer

comfort, as the combo system can be located closer to the end users and alleviate design constraints by reducing air delivery losses and hot water time-to-tap. Generally, tankless water heater combo systems may be more suitable for new household constructions given that proper design of the total system is possible [10].

1.3. COMBO SYSTEM PERFORMANCE FOR RESIDENTIAL APPLICATIONS

A recent U.S. Department of Energy initiative, the Building America program, investigated tankless water heater combination systems for wide scale implementation of combo systems for single- and multifamily residences and concluded that “before committing to wide-scale implementation of such combination space and domestic water heating systems for high performance buildings, whether new or retrofit, design decisions and site conditions affecting performance, maintenance, and occupant acceptability should be well understood.” [10] The report concluded that a key step to combo system implementation included predicting and quantifying expected combo performance for realized consumer comfort gains and potential energy savings [10]. Currently, combination systems have higher initial appliance and installation costs (capital costs) compared to traditional systems [11]. Dieckmann *et al.* estimated that gas-fired tankless water heaters have approximately twice the initial equipment cost of gas-fired SWHs [12]. Additionally, combo systems can be more costly to employ in retrofit applications, where additional construction considerations apply such as ensuring existing gas service has adequate capacity, conducting modification to utility lines, and ensuring compliance with TWH venting requirements [12]. Therefore, due to the increased initial costs, the performance of combo systems should be well quantified such that consumers can be informed on potential cost and energy savings to encourage them to upgrade to higher efficiency

systems [13]. At present, the performance of a combo system is difficult to estimate due to: (1) lack of published studies on tankless water heater integrated combo system performance and (2) limitations in the current North American combo system regulatory documentation and rating standards.

1.4. REVIEW OF RELEVANT LITERATURE

For DHW production, the performance of TWHs under various initial and operating conditions has been well studied. Glanville *et al.* [5] studied short-term performance for small volume, short duration, and intermittent heated water draws. The investigators concluded that for short term operation, performance was adversely affected by several factors including: pressure drop, start-up delay, and fluctuations in outlet water temperatures. Grant *et al.* [6] performed experiments to address questions regarding TWH behaviour including: response to rapid changes in water flow rate, response to preheated inlet water, and impact of water draw characteristics on steady-state efficiency. The study concluded that rapid changes in flow rates can produce water outlet temperature fluctuations of up to 9°C and lasting up to one minute. As well, preheated inlet water testing showed that TWH units used feedback control protocols to dampen temperature fluctuations at low temperature settings. As well, the study found evidence that there was no relationship between DHW flow rates or TWH setting temperature and steady-state efficiency [6]. Healy *et al.* [14] studied the possibility of using a linear model to relate output energy rate to input energy rate under different usage scenarios. The study concluded that a linear model may not be appropriate as model predictions were found to overestimate efficiency up to 4.5%. Dieckmann *et al.* [12] suggested that energy efficiency of a TWH was not impacted by total daily hot water consumption of the residence but was negatively impacted by frequent,

small volume draws. As well, Dieckmann *et al.* [12] found that replacement of conventional SWH units with TWHs could allow for an annual energy savings potential of up to 20% based on the increased water heating efficiency on the system. As well, a study performed by The Canadian Centre for Housing Technology (CCHT) determined that replacing existing natural gas-fired SWH with gas-fired TWHs resulted in an average reduction of natural gas consumption of 230 m³/year [15].

In relation to the use of electric appliances for space heating, electric resistance systems for space heating has been previously studied. Wu *et al.* [16] performed a study to compare energy and exergy performance of several electric and fossil fuel space and water heating systems in Montreal, Canada. Wu *et al.* [16] found that annual energy efficiency for electric baseboard heating systems and electric water heater systems ranged between 64.2% to 73.1% and for hydronic baseboard heating systems and gas-fired water heaters ranged between 79.6% to 81.1%. Furthermore, the results of the study suggested a high priority should be given to improving the exergy performance of natural gas appliances for heating and DHW usage as opposed to electricity from power generating plants as the exergy destruction in power generation and transmission was unavoidable for end-users. A study by the U.S. Department of Energy concluded that electric heat can be more expensive than space heating in a home with fossil fuel appliances due to large efficiency losses associated with electricity generation and transmission of electricity to end-user dwellings [17]. The study estimated that only 30% of the input fuel from fossil fuel (oil, gas and coal) power plants is converted into electricity [17]. Gao *et al.* [18] studied energy consumption of electric and hydronic reheat systems and concluded

that AHU related energy for hydronic reheat systems was lower than in electric reheat systems between 24% to 33%.

Previous studies on combo systems primarily focus on combo systems with boilers or storage water heaters as the heating plant. Butcher [19] previously studied combo systems and demonstrated that a linear input/output relationship can be used to model combo system performance and study how system parameters affect annual performance. Butcher [19] studied several combo systems with boiler and SWH heating plants and found annual performance could be primarily characterized by two combo system parameters: steady-state thermal efficiency at full load and combo system idle losses. As well, the study demonstrated that idle losses in combo systems have a significant impact on both DHW efficiency during the non-heating season and annual combined efficiency. This conclusion by Butcher [19] supported the integration of a TWH as the heating plant in a combo system as opposed to the integration of a SWH unit. Through integration of a TWH in the combo system, the anticipated idle losses of the combo system were noticeably lower than that of a boiler or tank water heater system due to the superior EF ratings of the TWHs. In addition, several field studies have been performed to assess combo systems. A field study performed by The Canadian Centre for Housing Technology (CCHT) determined that a combo system with a condensing SWH was able to have comparable energy performance to a high-efficiency furnace during high space heating demand [20]. However, this study did not evaluate combo systems with a TWH heating source. The U.S. Department of Energy performed experiments involving combination systems with boilers and SWH and TWH systems as the heating plants and concluded that system efficiency can be achieved by minimizing return water temperatures from the hydronic unit [21]. The study concluded that steady-state

space heating efficiency in excess of 85% may be possible for space heating only operation depending on the system configuration [21]. A field study performed in Hennepin County, MN installed and monitored twenty combination units of varying design to examine efficiency and performance. The combination systems studied included systems integrating boilers and SWH and TWH systems as the heating plants. A major conclusion from the study found that the maximum annual efficiency of a combination system operating for space heating with a TWH as the heating plant surpassed the maximum annual observed efficiency for a combo system with a boiler or SWH [11]. However, this study concluded that the results presented were affected by incorrectly installed monitoring devices. As well, in some trial residences in this study, the water heaters and AHUs used were not from the same manufacturer.

1.5. NORTH AMERICAN REGULATIONS AND STANDARDS

At present, combination heating systems are not currently regulated in Canada under the *Energy Efficiency Regulation Act* [22, 23]. This has far reaching implications for combo system usage in industry. TWH units will have a regulated rating (energy factor), but this rating would only pertain to combo system performance as a water heater for DHW provision only. As well, the EF rating would not reflect the water heating performance of the system under simultaneous operation (space and water heating). Similarly, the TWH will perform differently when the combo system is being used for space heating and the EF factor would be an unrelated measure in quantifying space heating performance [21]. Due to the fact that combo systems are not regulated under energy efficiency statutes, combo system products will not have been evaluated for compliance with the relevant energy efficiency standards nor have an energy efficiency report filed with Natural Resources Canada. Furthermore, combo systems will not be identified

with energy efficiency documentation, such as an *EnerGuide* label. This lack of regulation over energy efficiency of combo units serves to hinder the ability of the product to gain traction in the marketplace due to negatively impacting consumer confidence given that conventional space and water heating appliances are typically covered under the energy efficiency statutes. In terms of regulation for combo systems relating to contractor or designer certifications, the Heating, Refrigeration and Air Conditioning Institute of Canada (HRAI) offers a certification program, detailing design selection, installation, and commission of combination systems [24].

In North America, there are two standards relating to the evaluation of the performance of combo systems: ASHRAE Standard 124-2007 [25] and CSA P.9-2011[26]. ASHRAE Standard 124 is an amalgamation of ASHRAE Standard 103 [27] (relating to space heating performance) and ASHRAE Standard 118.2 [28] (relating to residential water heating) to test and rate combo systems. ASHRAE 124 develops a combined annual efficiency (CAE) factor to rate combo systems based on the resulting ratings for the combo system under ASHRAE Standard 103 and ASHRAE Standard 118.2. ASHRAE Standard 124 does not allow for testing the combo system as a complete package, and thus testing the actual combined performance is not possible under ASHRAE Standard 124. Furthermore, the standard mandates testing operating temperatures that may not align with realistic usage patterns and a does not consider manufacturer controls that may improve actual field performance such as advanced controls and smart integration systems [10, 29]. In contrast to ASHRAE Standard 124, CSA P.9-2011 tests the combo system as a complete system, rather than developing a rating based on the performance of the individual components and developing a rating based on the individual component ratings. To assess combo system performance for an entire system, CSA P.9-2011 uses a thermal performance factor

(TPF) to describe the overall actual combo performance. However, as opposed to ASHRAE Standard 124, CSA P.9-2011 does not rate the components of a system based on the individual components testing standards. CSA P.9-2011 does not force set points, allowing manufacturers a large degree of freedom with respect combo system controls [29]. Furthermore, the scope of CSA P.9-2011 currently does not apply to hydronic heat distribution systems, electric, oil, or solar based combos systems and multifamily dwellings with a central heating plant whereas ASHRAE Standard 124 is inclusive. Similar to ASHRAE Standard 124, CSA P.9-2011 has been criticized in several studies as not having accurately predicted actual usage patterns based on the current iteration of the standard [10, 29].

1.6. OBJECTIVES

A study on TWH combo systems will provide valuable insight and guidance to industry and consumers regarding combo TWH-hydronic heater systems for use in residential buildings. Quantification of thermodynamic parameters will establish a baseline for anticipated performance of combo systems from a building science and thermal engineering perspective and will help to address these aforementioned barriers. Therefore, the overall objectives of this study were to:

1. Develop an experimental methodology to study and quantify the impact of variable water temperatures from the TWH to the AHU when the combo system was operating solely as a space heating system on:
 - a. First law efficiency of the system during steady-state operation;

- b. Second law efficiency of the AHU heat exchanger during steady-state operation; and
 - c. Energy lost in flue gas emissions during steady-state operation.
- 2. Develop an experimental methodology to study and quantify the impact of variable water temperatures and DHW water draws for when the combo system was operating for both space and water heating on:
 - a. First law efficiency of the system during steady-state operation and
 - b. Second law efficiency of the AHU heat exchanger during steady-state operation.

Developing and sharing knowledge and data on combo systems will serve to produce better information and allow for government, stakeholders, and the general public to make informed decisions regarding alternative space and water heating systems.

1.7. THESIS ORGANIZATION

This thesis is divided into several chapters. Following the Introduction presented in Chapter 1, Chapter 2 describes background information on combination system equipment and operation. Chapter 3 details the experimental methodology used to test and study the combo system and Chapter 4 describes the theoretical analysis developed to quantify combo system performance. Chapter 5 presents the results and analysis of the experimental tests. Chapter 6 presents the conclusions of this study, and Chapter 7 proposes future work that may be extended from this thesis study.

2. BACKGROUND

The Canadian standard for combo system testing, CSA P.9-11 [26], defines a combo system as “a product or a group of individual components that form an integrated system that is designed to provide space heating and water heating”. The standard broadly defines three categories of combo systems: (1) Type A system: combo system with a fixed capacity for space heating, (2) Type B system: combo system equipped with controls to adjust automatically the space heating capacity based on space heating load, and (3) Type C system: combo system with a thermal storage tank or equivalent component that decouples space heating load from burner control. Across each category, the integral components of the combo system are a heating plant (water heating) and a hydronic distribution system (space heating). In this study, a Type A system was examined with the heating plant and hydronic distribution system of a TWH and an AHU, respectively. Type B and Type C systems are typically more complex variants of the Type A system, and so performance testing of a Type A system will serve as an important benchmark for the expected performance of the complex systems. Figure 2-1 shows an example of a commercial configuration of a Type A combo system for space and water heating.



Figure 2-1: Combination system for space and water heating (Type A) [30].

2.1. OPERATIONAL PRINCIPLE OF TWH UNITS

Figure 2-2 shows the internal configuration of the components of a typical indoor, natural gas fired TWH unit. TWHs are considered as on-demand or “instantaneous” water heaters due to the fact that there is no storage of preheated water. TWHs heat water by passing the water multiple times through an internal heat exchanger within a combustion chamber. TWHs are controlled by a microprocessor unit (PCB) that monitors and controls several functions such as flow rate of gas, water, or combustion air and water temperature.

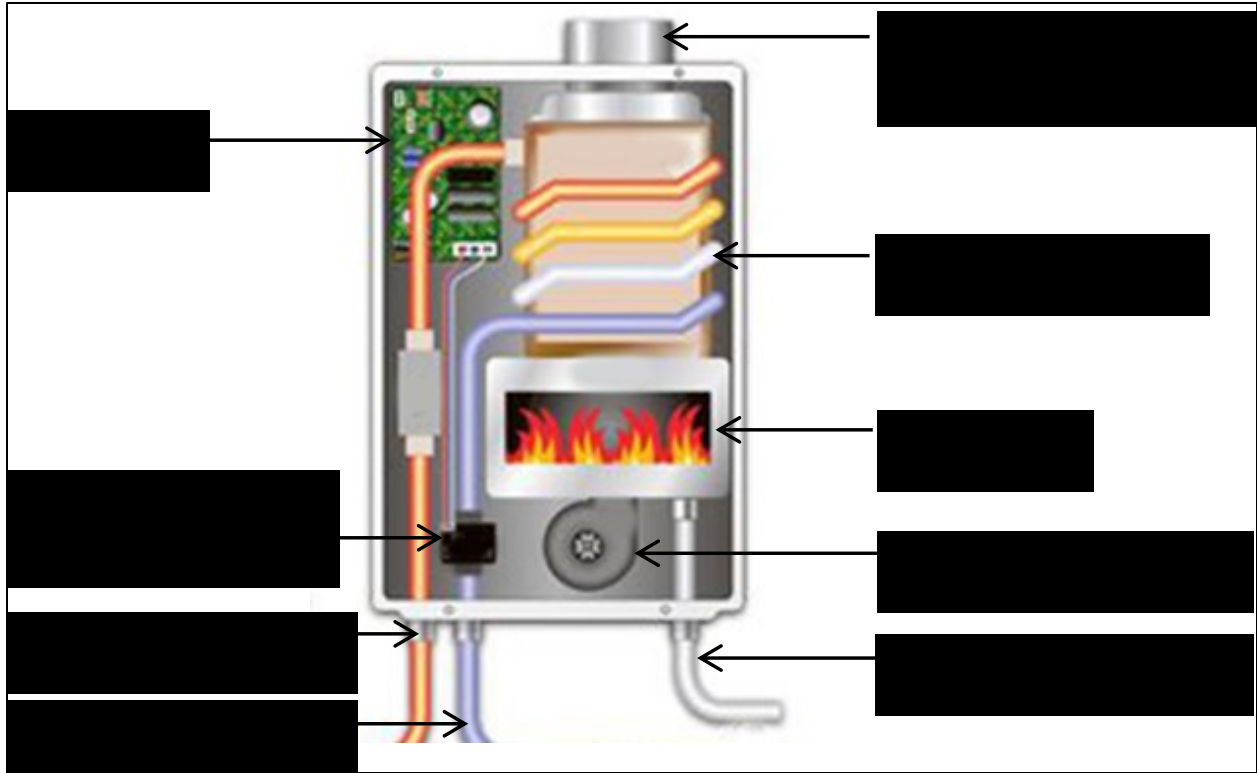


Figure 2-2: Internal components of natural gas fired TWH [31].

The TWH sequence of unit operation can be separated into five stages [29]:

1. *Water Flow Activation*: The TWH unit has a flow sensor that monitors water flow rate. Once the minimum flow rate for activation of the unit is achieved, the ignition sequence initiates.
2. *Combustion Chamber Ignition*: The PCB initiates the combustion chamber fan (see Fig. 2-2) to ensure that there is sufficient fresh air for the combustion process of the natural gas. The PCB modulates to a pre-set low initial firing rate to proof ignition. Once flame rods prove ignition, the spark ignitor ceases to operate.
3. *Normal Operation*: The PCB controller monitors several variables including: flame rods, combustion fan motor, flow rate of water and outlet water temperature during operation.

Natural gas and combustion air inputs to the TWH unit can be modulated by the controller based on the set point water temperature and the water flow. If the unit is unable to heat the water to the set-point temperature, a water flow control valve will modulate the flow rate as needed. The unit will not fire if the minimum firing rate and incoming water temperature would exceed the thermostat temperature to a pre-determined unacceptable value. Some TWH units have a cold water bypass feature in which water that is heated above the set-point temperature is tempered with cold water until the set-point temperature is achieved at the exit of the TWH. The aim of this feature is to reduce outlet temperature fluctuations during changes in hot water demand. This feature was not present in the TWH that was used in this study.

4. Shutdown operation: The PCB controller senses water flow rate below an activation threshold. The gas valve closes and the combustion fan purges the remaining fuel and exhaust gases.
5. Standby operation: The PCB continuously monitors the TWH components and water flow rate.

TWH units have a priority control strategy to ensure that water at the desired set point temperature is provided to users, and that the TWH may regulate water flow to ensure that the water temperature is provided. In industry, when sizing homes for a combination system, the heating load requirements for DHW is the primary performance variable for TWH sizing; thus, for combination systems, the TWH is typically oversized relative to the space heating load requirements [32].

2.2. OPERATIONAL PRINCIPLES OF HYDRONIC AIR HANDLING UNITS

Figure 2-3 shows the internal components of a typical AHU that was tested in this study. The AHU is a hydronic heating system, which used water (or in some cases water-based solutions) to transfer heat to air for space heating. The AHU thermal energy was provided by a tankless water heater and the thermal energy was released into a space for heating by a crossflow, non-mixing water-to-air heat exchanger. The AHU was considered a hydronic system due to the transfer of heat from water to air for space heating use. The piping network in the AHU consisted of a heat exchanger with inlet and an outlet header with staggered finned tubes.

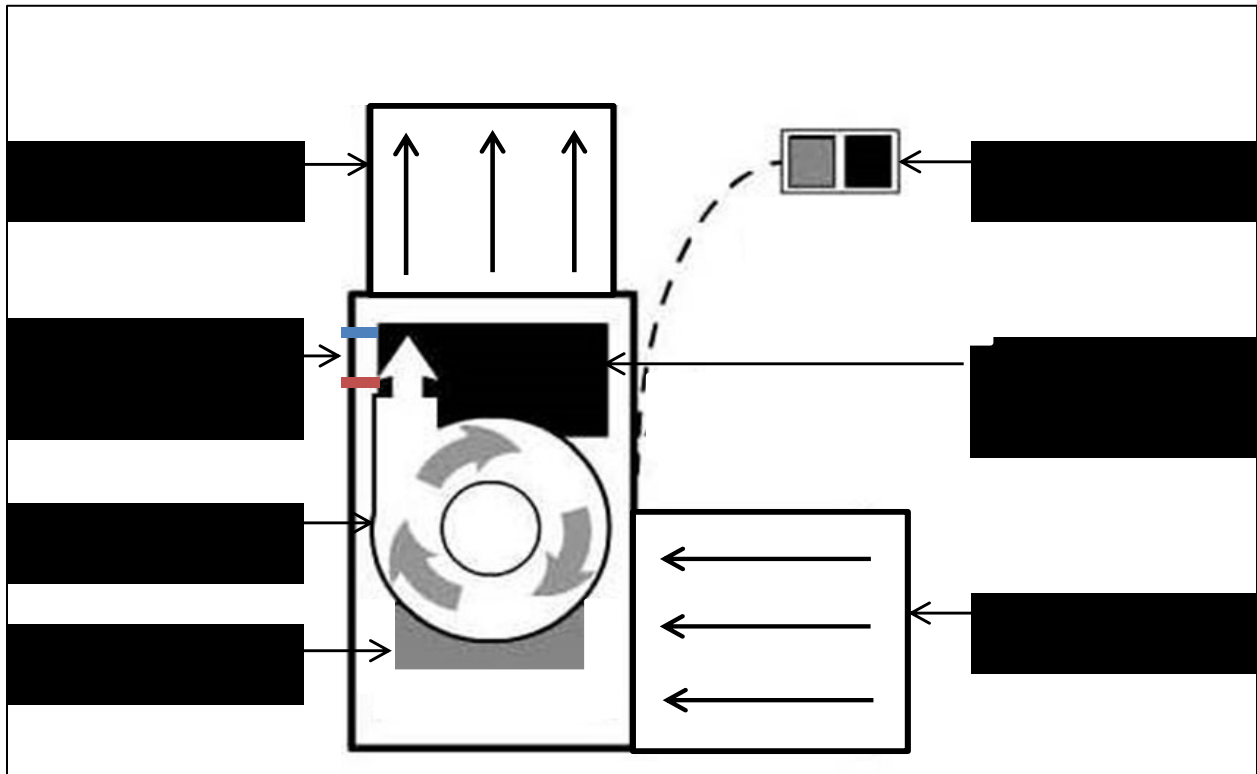


Figure 2-3: Internal components of typical hydronic AHU [33].

A blower passes air over the heat exchanger and the heated supply air is sent to the heating ducts of the residence for space heating. The AHU controller is connected to a thermostat to monitor room temperature and activate the AHU as required. The AHU sequence of operation can be separated into three stages:

1. *Water Flow Activation*: The AHU controller is activated when heating is required. This activation signal would be initiated by a thermostat controller in the residence. An external water flow sensor signals the AHU control panel to energize a circulating pump. Once the pump has been energized and water flow rate was proven, the controller energized the indoor direct drive blower.
2. *Normal Operation*: The controller monitors the thermostat input signal to ensure that the call for heat remains. The external flow sensor continuously monitored the downstream water flow rate from the AHU. If the water supply to the inlet of the AHU was no longer available, the external flow sensor will halt operation of the blower, even if a request for heating was outstanding.
3. *Shutdown operation*: When the thermostat call is removed, the controller first deactivates the circulation pump and the indoor blower is de-energized after expiration of a user-defined time delay.

2.3. OPERATIONAL PRINCIPLE OF A COMBINATION SYSTEM

Figure 2-4 shows a process flow diagram for a typical combo system for space and water heating with a tankless water heater as the heating plant. In practice, combo systems have three

operational modalities: (1) providing DHW only, (2) providing space heating only or (3) providing water and space heating simultaneously.

In the case of DHW only operation for a tankless water heater combo system, the system performance simply models that of a tankless water heater. The sequence of operation would be identical to the sequence presented in Section 2.1. In this mode of operation, combo system performance will have superior performance compared to conventional SWH performance, due to higher energy factor ratings for TWHs than SWHs. The performance of natural gas fired TWH for the provision of DHW under various operating and initial conditions has been thoroughly studied [5, 9, 12, 34] and was not investigated in this study.

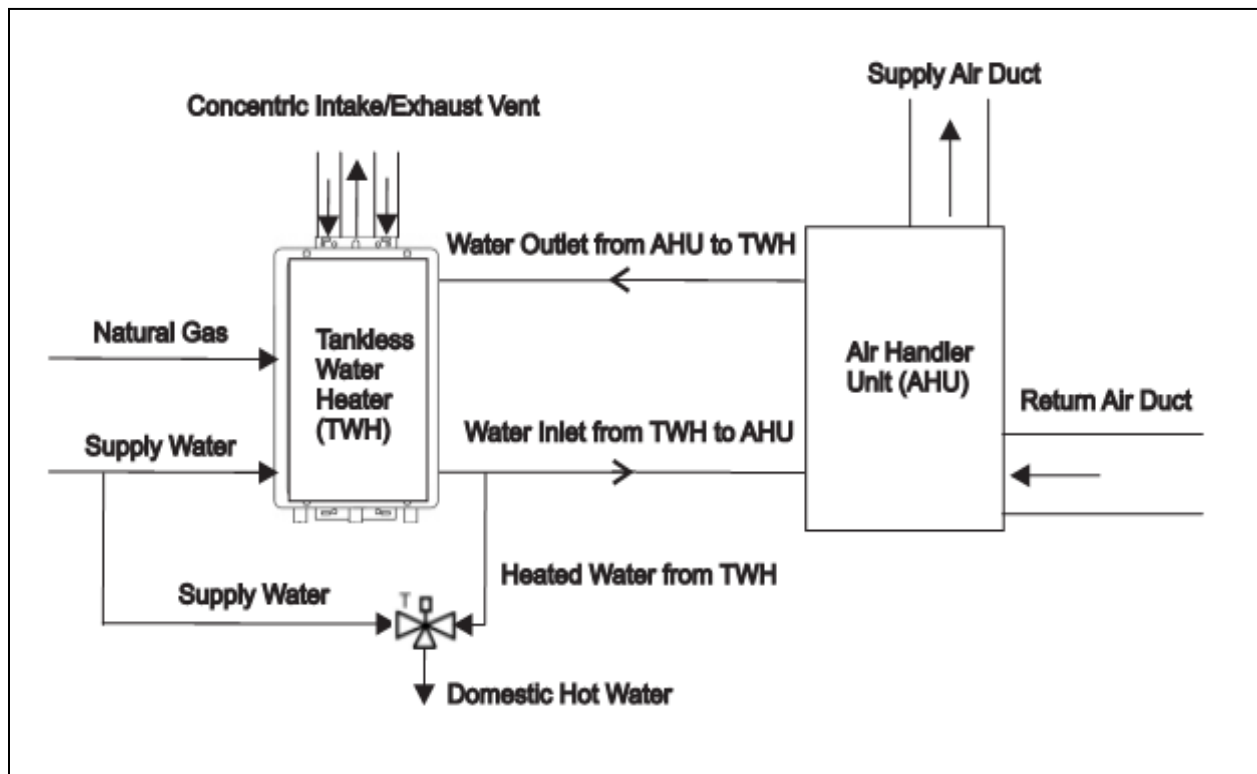


Figure 2-4: Process flow diagram of combo system with TWH heating source.

For space heating only, the TWH functions as a heating plant for the hydronic air heater. In this configuration, the system is considered “closed” since no water is entering or exiting the system. The TWH can either be used in tandem with the AHU unit for space heating in two different configurations: direct or indirect. In a direct combo system, the heated water from the TWH is pumped to the AHU heat exchanger and heat is transferred to the air stream. For an indirect combo system, the heated water is pumped to an external heat exchanger and is kept completely separated from the AHU system. Heat is transferred from the water to a heat transfer fluid (typically treated water) which would then transfer heat via a heat exchanger to the air stream. An indirect system allows for the water used for DHW to be kept separate from the AHU. An indirect system may be used in cases when there is concern of contamination of potable water by the materials used within the AHU [35]. As well, domestic water may contain dissolved minerals or other contaminants may lead to precipitate or fouling of piping and components. Precipitate accumulation and fouling is a particular concern for TWH units [29, 35].

In this study, the combo system configuration for space heating only is a closed direct system. For space heating only operation, the DHW draw line shown in Fig. 4 was closed and no hot water was removed from the system for DHW usage.

The TWH water outlet line feeds the AHU inlet water line. Similarly, the AHU outlet water line feeds the TWH water inlet line and no cold feed water was added to the system. In combo systems for space heating operation, a circulating pump in the AHU activates when there is a request for heating from the residence thermostat. When the minimum activation flow rate of the TWH is surpassed, heated water from the TWH outlet water line is pumped to the AHU inlet

water line. The heated water passes through a header in the AHU where the heated water entered a staggered-tube finned bank heat exchanger. An air blower draws air and passes the air over the heat exchanger and through the duct network. The water exits the heat exchanger and is returned from the AHU outlet water line to the inlet water line of the TWH. This configuration distinguishes itself from the DHW only mode as the TWH inlet water temperature will be higher compared to supply water temperature from the city main water supply.

The heating operation of a natural gas TWH has been previously classified in literature into four main stages [34]. A similar operational cycle for the system temperature trace was defined for the purpose of analysis of the performance of the combo system for space heating that was used in this study.

1. *Standby stage*: The standby stage is classified as a pre-operational state of the combo system. The cycle precludes operation of the TWH independently of the system. This stage represents the initial/ambient system conditions, namely temperature of the water initially in the piping system and makeup air temperature. Energy consumption in this phase is restricted to electricity used to power the units whereas in conventional SWHs, the standby stage includes energy to maintain continuously burning pilot lights. Typically, power consumption in the standby stage is in the range of 1 - 10 W [34]. In this stage, no water is flowing through the TWH or AHU.
2. *Ramp up stage*: The transient part of the cycle is classified as the time at which the TWH initiates heating until the target water temperature no longer fluctuates and is delivered at a stable temperature. Energy consumption in this stage occurs from combusting natural gas in the TWH and from electricity consumed for operation of the controller, blower and

pump. As well, energy is accumulated in the distribution network piping, water contained within the system and the TWH unit itself. The length of this stage is impacted by the combo system control strategy and response, water inlet temperature, and water flow rate. This stage also consists of the ignition of the TWH burner and activation of the AHU circulator and air blower. This stage may differ from manufacturer to manufacturer depending on system control strategies, responses, and temperature setting sensitivity.

3. *Steady-state or cycling stage*: This stage is classified as where the combo system delivers water and air at a usable or stable temperature. In this stage, the combo may be unable to deliver air at a stable temperature and may provide cycling air temperatures. This is known as short-cycling in the industry, and is characterized by the TWH burner cycling on and off. In the case of a combo system, short-cycling is observed as the outlet temperatures fluctuating due to the TWH burner cycling on and off. The energy consumption is dominated in this stage by the water flow rate, inlet water temperature to the TWH and water temperature setting.
4. *Environmental decay stage*: The environmental decay stage is classified as the period when the combo system shuts off, after the space heating requirements have been satisfied. The water and air cool to equilibrate with the ambient temperature of the external space. The TWH combustion fan will operate and purge any remaining combustion products from the combustion chamber. Additionally, any fuel gas that was not combusted will be purged in this stage. Energy stored in the system due to thermal capacitance of the piping network and water will be transferred to the environment.

Few studies have been undertaken to determine the thermodynamic efficiency and performance of a combo system with a TWH as the heating plant for space heating applications. In particular, the space heating modality of combo operation has not been well studied and in terms of performance, the steady-state or cycling state of the combo system was of interest.

For simultaneous operation, the combo system provides both space and water heating. In this configuration, the system is considered “open” since cool feed water is entering the system and hot water for DHW use is exiting the system. As in the space heating case, the combo system can be considered either an open-direct system or open-indirect system. In a direct system, the heated water is used for both hydronic heat distribution and DHW. Conversely in an indirect system, the AHU system has separate water loop plumbing and heated water produced by the TWH is kept separate. The combo system configuration tested in this study for simultaneous operation is open direct. As well, cold feed water is mixed with the AHU outlet water line before entering the TWH water inlet in this case. When considering simultaneous operation, there are two activation scenarios worthy of consideration; (1) combo system is providing DHW and a call for space heating is initiated, or (2) combo system is providing space heating and a DHW draw is initiated. When considering annual usage, the second case is the most likely case and will be the case studied. A similar stage developed for the purpose of analysis of the performance of the combo system occurs for simultaneous operation.

1. *Ramp up stage*: This stage is identical to the previous defined stage for space heating only operation.
2. *Steady-state or cycling stage*: This stage is identical to the previous defined stage for space heating only operation.

3. *Interrupt Stage*: This stage is classified as when either water heating activation occurs when the system is already providing space heating in the steady-state or cycling stage or when space heating is initiated when the system is already providing water heating.
4. *Combined operation stage*: This stage is classified as where the combo system delivers water and air at a usable or stable temperature for providing both space and water heating. The energy consumption in this stage depends on DHW water flow rate, inlet water temperature to the TWH and water temperature setting. This stage represents the unique ability of the combo system to provide both space and water heating simultaneously in the same system. This stage ends when either the space or water heating request ends and the system can either resume operation for DHW or space heating only or conversely proceed to the environmental decay stage.
5. *Environmental decay stage*: The environmental decay stage is classified as the period when the combo system shuts off, after the space or water heating requirements have been met. The combo system behaves similar to the space heating only decay stage.

The stage of interest to be studied was the combined operation stage. As with the space heating modality, simultaneous combo operation has not been well studied and few studies address performance of a combo system for space and water heating.

3. EXPERIMENTAL METHODOLOGY

This experimental method describes the procedure and equipment used acquires data in this study and for assessment of the performance of the combo system. Due to laboratory space restrictions, the combo system was designed to be portable so as to be able to be moved in and out of the testing space.

3.1. COMBO SYSTEM APPLIANCES

The TWH and AHU equipment used in this study were a Rinnai RL75i Direct Vent Tankless Water Heater (Atlanta, GA, USA) and a Rinnai 37AH045 hydronic AHU (Atlanta, GA, USA), respectively. The Rinnai AHU unit in this study is designed by the manufacturer to be used with the TWH. Manufacturer ratings for the TWH are shown in Table 3-1. The TWH unit selected was a temperature controlled, continuous flow natural gas water heater with forced combustion with direct ventilation. The TWH unit tested had a DHW priority setting. This control strategy diverts all heating capacity of the TWH unit for DHW use during simultaneous air and water heating events.

Table 3-1: Specifications for the TWH.

Parameter	Specification
Natural Gas Rate (Input)	3 – 53 kW (10,300 – 180,000 Btu/hr)
Electrical Consumption	Normal: 76 W Standby: 2 W
Hot Water Flow Capacity	Minimum flow rate: 0.98 lpm (0.26 gpm) Minimum activation flow rate: 1.51 lpm (0.40 gpm) Maximum flow rate: 28.4 lpm (7.5 gpm)
Temperature Controller	37°C – 60°C (98°F – 140°F)
Energy Factor	Natural Gas: 0.82
Water Pressure Range	138 – 1034 kPa (20 – 150 psi)

The TWH prioritizes set point temperature to the user over flow rate. Based on the temperature set point and the TWH inlet water temperature the unit will regulate water flow rate to provide the set point temperature. The TWH unit used in this study is capable of providing up to 7.5 GPM of water with up to a 4.5°C increase water temperature from the inlet water temperature. After this threshold, the unit will begin to throttle water flow rate if a larger increase in water temperature is required. Flue gas emissions were vented directly to a fume hood and exhausted outdoors. The venting component used was the manufacturer recommend product of a concentric venting system. In this system, the inner diameter of the vent is for flue gases produced from the TWH. The outer annulus is for the intake draft of fresh air for combustion. This type of venting serves as a basic energy recovery device for preheating combustion air. However, given that the length of venting in this study is less than 0.5 m, the temperature of combustion air is considered to be the same as that of ambient air. Manufacturer ratings for the combo system AHU used in the experimental study are shown in Table 3-2.

Table 3-2: Specifications for the AHU.

Parameter	Specification
Nominal Output Capacity	13 kW (45,000 BTU/hr)
External Static Pressure	124 Pa (0.50 in. W.C.)
Nominal Airflow @ 0.5 ESP & 20°C	Heating: 22,653 lpm (800 cfm) Cooling: 18406 – 22,653 lpm (650 – 800 cfm)
Heat Blower On Delay	25 s
Indoor Blower Motor	Direct Drive Motor: 248 W (1/3 HP)
Circulating Pump	Wet Rotor: 93W (1/8 HP)
Hydronic Heating Coil Construction	9.5 mm (3/8 in) OD copper tubes, aluminum fins
Rows / Fins per inch	2 / 16
Total Face Area	0.21 m ² (2.3 ft ²)
Approximate internal volume	1.8 L (.47 gal)

The AHU was installed in a side orientation to maximize space for testing instrumentation. The AHU was wired with a simple toggle switch, which simulates the activation signal from a thermostat to activate the AHU for space heating. For AHU operation, ambient air was drawn from the testing space and supply air was exhausted to the atmosphere through a venting hood. The supply air duct for the heated air was fabricated from 1.76 RSI (10 R-value) rated insulation board to minimize natural convection losses from the duct surface. For steady operating conditions and fully developed turbulent flow, the heat loss across the length of the channel was estimated to be approximately 91 W and was considered negligible due to being significantly less than the nominal output rating for AHU output. The duct was constructed such that the length is ten times the hydraulic diameter to ensure that fully developed flow and temperature of the fluid occur by the exit of the duct. The blower motor speed was left at the manufacturer setting. In practice, the blower motor speed can be adjusted based on the ducting external static pressure requirements for the residence. However, during operation the blower is only capable of operating at this set speed and blower speed cannot be adjusted during operation. The plumbing network was constructed of 19 mm (3/4 inch) nominal diameter type L copper tubing with soldered joints. To reduce heat losses from the surface of the copper tubing, the tubing was wrapped with 0.4 RSI (2.187 R-value) rated insulation. A heat loss analysis estimating natural convective losses to the room was performed and the maximum estimated heat losses from the piping was 7 W. The plumbing network used NSF 61 certified 19 mm (3/4 inch) brass ball valves to control the flow of water through the system. The ball valves allowed for on/off control of the simulated DHW event. An ANSI Class 125 inline globe valve allowed for flow control of the volumetric flow rate of water through the system as well as control of the DHW flow rate. The experimental apparatus can be seen in Fig. 3-1.

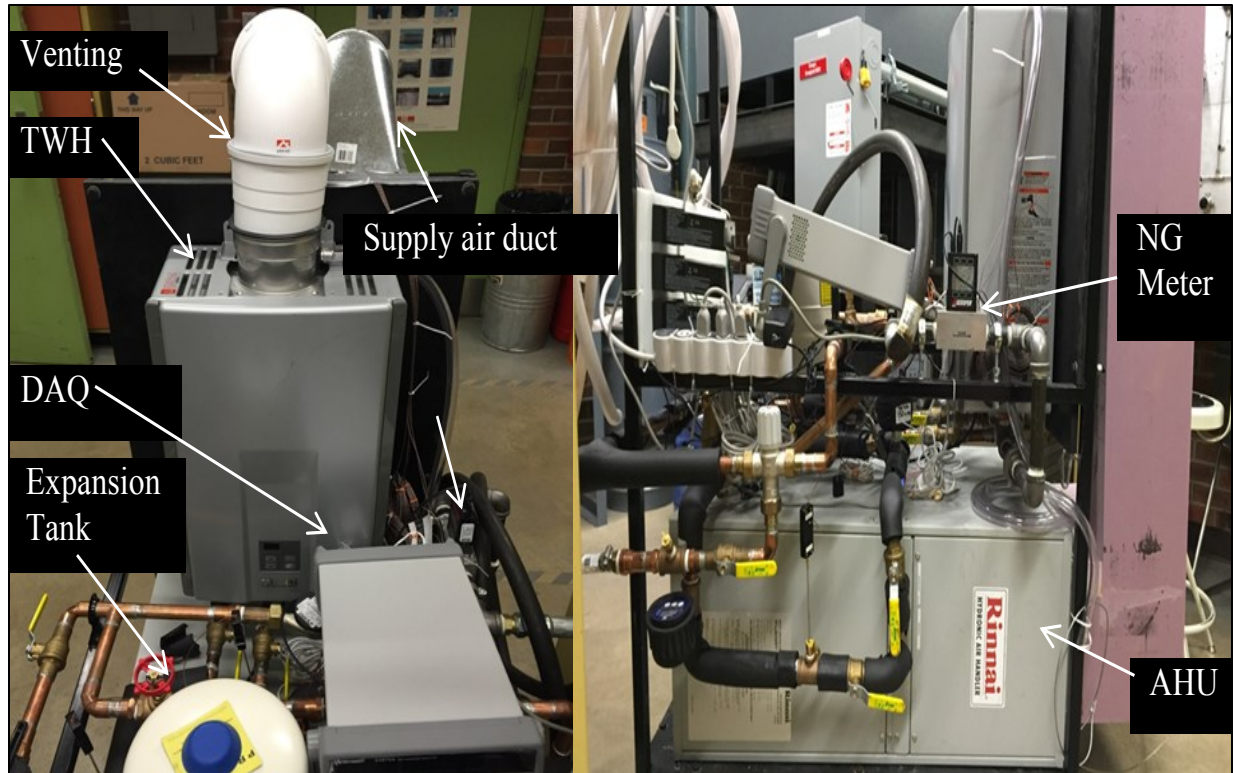


Figure 3-1: Experimental apparatus used for analysis of combo system performance. (L) Front view. (R) Side view.

3.2. COMBO SYSTEM TESTING

The combo system was operated at several different TWH temperature controller levels to study the impact of the water temperature produced from the TWH on the performance of the system during space heating only. The TWH unit temperature controller full range varied between 37°C (98°F) and 60°C (140°F). Space heating only and simultaneous operation tests were performed by varying the TWH controller temperature settings at the following levels: 38°C (100°F), 43°C (110°F), 46°C (115°F), 49°C (120°F), 52°C (125°F), 54°C (130°F), 57°C (135°F) and 60°C (140°F).

For simultaneous operation tests, in addition to the variation of TWH temperature controller as listed previously, the DHW flow rate request to the combo system was varied at two levels for each temperature setting level. DHW flow rate was set using a globe valve installed in the system DHW line. The DHW flow rate selected were 7.6 L/min (2 GPM) and 15.1 L/min (4GPM) to simulate a hot water tap, and both a hot water tap and a shower, respectively. For simplicity, the 15.1 L/min draw event was considered to occur simultaneous as opposed to having two separate draws occurring one after the other.

The following experimental variables were measured: ambient air temperature, heated air temperature, TWH inlet and outlet water temperature and AHU inlet and outlet temperature. As well, volumetric flow rate of natural gas, air and water will be recorded. For the case of flue gas analysis, flue gas temperature, flow rate and composition were measured.

Water temperature was measured using custom iron-constantan Type J immersion probe thermocouples (ICSS-18U-6, OMEGA Engineering, Laval, QC, Canada). The water thermocouples were inserted into a measurement plug located approximately 200 mm (7.85 in) at the inlet and outlet water lines of the TWH and AHU appliances to ensure fully developed temperature profiles. Ambient air temperature was measured using two Type-J thermocouples (GG-J-24-SLE, OMEGA Engineering, Laval, QC, Canada) placed at the inlet of the AHU. Supply or heated air temperature from the AHU was measured at the exit of the supply air duct. To measure heated air temperature at the exit of the supply air duct, a six-point thermocouple array was constructed. The measurements from each thermocouple were averaged to determine the heated air exit temperature. Due to the negligible maximum calculated losses across the duct

channel, the temperature at the exit of the supply air duct is considered to be approximately the same as the air exit temperature from the AHU. All temperature data was sampled at a rate of 1 Hz (1 datum point per second) and collected with a data acquisition system (34970A, Keysight Technologies, Mississauga, ON, Canada). The data was then exported into a Microsoft Excel file for subsequent analysis.

Volumetric flow rates of water, air and natural gas were also measured. Volumetric flow rate of air was measured using a pitot tube (Series 475 Mark III Handheld, Dwyer Instruments International, Michigan City, IN, USA) located at the exit of the supply air duct. The volumetric flow rate of water was measured by impulse turbine flowmeters (FTB4607, OMEGA Engineering, Laval, QC, CAN). Flowmeters were located approximately 225 mm (8.85 in) downstream and upstream of the TWH and AHU to ensure fully developed flow at these locations for measurement. Air and water flow rate measurements were recorded once the combo system established a steady-state or cycling air and water temperature profile. The natural gas volumetric flow rate was measured by using a laminar flow element sensor (MW-100SLPM-D, Alicat Scientific Inc., Tucson, AZ, USA) specifically designed for low natural gas flow rates. Data of the natural gas volumetric flow rate, temperature and pressure was collected with an internal logger system and exported to Microsoft Excel for further analysis.

For analysis of flue gas emissions and combustion a flue gas analyser was used (330-2-LL, Testo, Lenzkirch, Germany). The analyser measured flue gas temperature, flow rate and exhaust gas composition which were recorded once the combo system established a steady-state or cycling air and water temperature profile

3.3. PROCEDURES

3.3.1. Space Heating Only - Experimental Procedure

The following sequence of steps was followed for testing the combo system for space heating only.

- a. The cold feed water supply valve and DHW valve was opened and fresh water was used to flush any heated water from the system. The cold feed water supply valve and DHW valve were then closed.
- b. The TWH temperature controller was set to the desired temperature test level.
- c. A simulated request for heat was sent to the AHU to initiate AHU operation.
- d. Water pressure reading from the pressure transducer was recorded and logged in an Excel spreadsheet.
- e. The water and air temperatures and volumetric natural gas flow rate were logged and recorded for six-minute duration.
- f. The water and air volumetric flow rates were recorded once the system established a consistent temperature profile.
- g. The flue gas temperature, flow rate and composition were recorded once the system established a consistent temperature profile.
- h. At the conclusion of each test, the water was purged from the piping system and replaced with fresh water.

For each set of the testing conditions that were explored, three replicate runs were performed. The quantitative results presented in this study are the average values of the variables that were measured.

3.3.2. *Simultaneous Operating - Experimental Procedure*

The following sequence of steps was followed for testing the combo system for simultaneous operation.

- a. The cold feed water supply valve and DHW valve was opened and fresh water was run through the system and to flush any heated water. The cold feed water supply valve and DHW valve were then closed.
- b. The DHW globe valve was set at the required DHW flow rate and the DHW line was closed off from the rest of the system by closing the DHW ball valve.
- c. The TWH temperature controller was set to the desired temperature test level.
- d. A simulated request for heat was sent to the AHU to initiate AHU operation.
- e. Water pressure reading from the pressure transducer was recorded and logged in an Excel spreadsheet.
- f. The water and air temperatures and volumetric natural gas flow rate were logged and recorded for six-minute duration.
- g. The combo system operated until steady-state or cyclical stage was reached. At this point, the DHW ball valve was opened to simulate a DHW from the combo system to the residence.
- h. The volumetric flow rate of air, water in the AHU loop, and DHW draw were recorded once the system established a consistent temperature profile.
- i. At the conclusion of each test, the water was purged from the piping system and replaced with fresh water.

For each set of the testing conditions that were explored, three replicate runs were performed. The quantitative results presented in this study are the average values of the variables that were measured.

4. THERORETICAL ANALYSIS

4.1. SPACE HEATING MODE ONLY

4.1.1. *First Law and Second Law Efficiencies*

The theoretical analysis in this thesis aims to apply thermodynamic parameters to quantify combo system performance. The performance of an energy conversion device can be assessed based on the quantity and quality of the energy used to achieve a given objective and allow for commentary on the efficiency and efficacy use of energy resources [36]. To do so, the mass, energy and exergy balance equations were utilized to develop expressions to evaluate combo system performance. For the analysis of the system, the following assumptions were made: (1) all processes had negligible changes in kinetic and potential energy of the fluid streams and (2) the system pressure drop in the piping network due to friction was negligible due to the relatively short pipe lengths and the absence of fittings or pipe accessories.

In the case of a combo system, the energy input to the system is by way of combustion of natural gas and the given objective was twofold: space and water heating. In this regard, the quantity of energy can be assessed by first law efficiency (alternatively known as thermal efficiency). First law efficiency is a thermal engineering quantity that is commonly defined as the ratio of useful energy delivered to total energy input into a system for a given process and is based on the first law of thermodynamics. However, a more rigorous approach to analyze the combo system was to consider the first law of thermodynamics for a control volume (CV) [37]. From this approach a suitable analytical model for evaluation of operation of the combo system for the operation modes of space heating only and simultaneous operation. Figure 4-1 shows the

control volume selected for evaluation of the combo system operating for space heating only. By utilizing the control volume as shown in Figure 4-1, this approach allows for the inclusion of heat losses from the piping network of the combination system in the analysis.

Entering the control volume in Fig. 4-1 was: natural gas to the TWH, air for combustion to the TWH, ambient supply air to the AHU and electrical work to the TWH and AHU appliances. Exiting the control volume in Fig. 4-1 was: combustion exhaust gases, supply (heated) air from the AHU and surface heat losses. The mass balance applied to this CV can be expressed as

$$\frac{dm_{CV}}{dt} = \dot{m}_{in} - \dot{m}_{out} \quad (4-1)$$

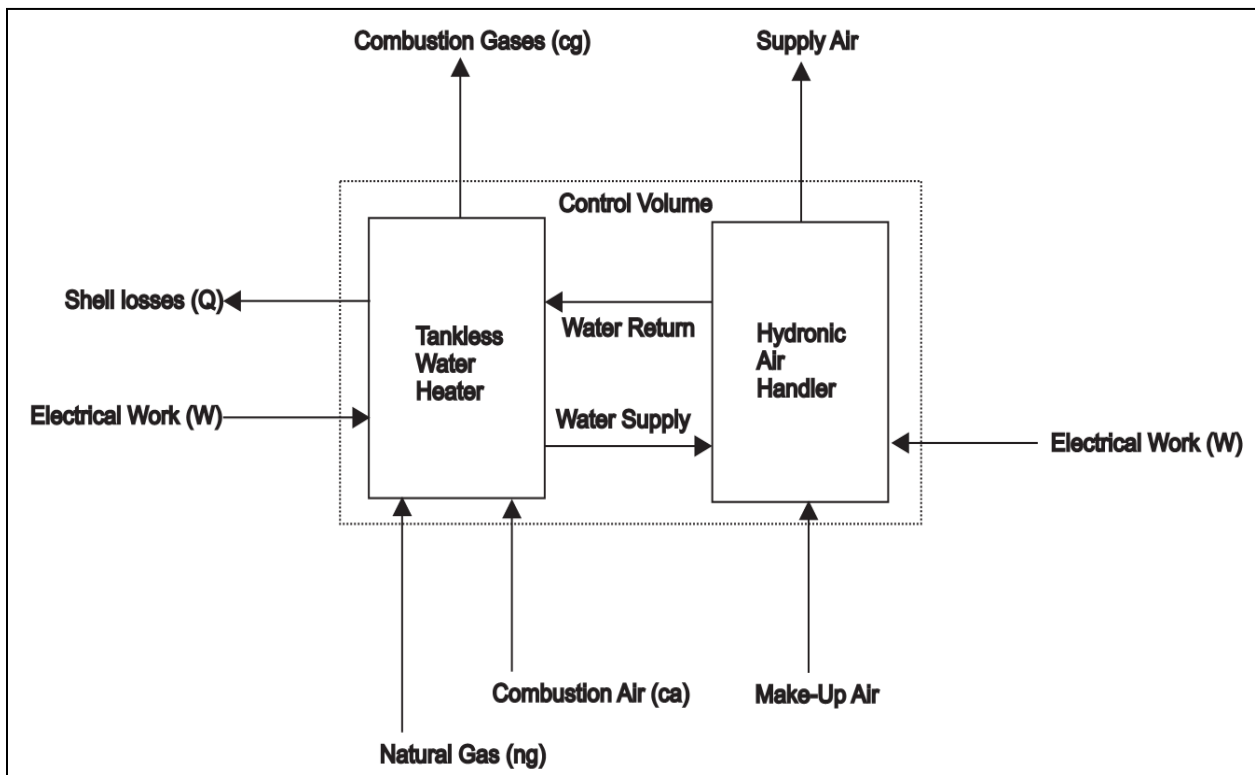


Figure 4-1: Control volume for analysis of combo system operating in space heating mode.

Eq. (4-1) can be applied to the control volume in Fig. 4-1

$$\frac{dm_{CV}}{dt} = [\dot{m}_{ng} + \dot{m}_{ca} + \dot{m}_{ma}] - [\dot{m}_{cg} + \dot{m}_{sa}]. \quad (4-2)$$

The first law of thermodynamics can be expressed in terms of energy streams entering, exiting and being generated in a control volume as:

$$\frac{dE_{CV}}{dt} = \dot{E}_{in} - \dot{E}_{out} + \dot{E}_{gen} \quad (4-3)$$

Eq. (4-3) can be applied to the control volume in Fig. 8. In Eq. (4-3), the energy generation term is zero and Eq. (4-3) expressed in terms of enthalpies, work, and heat transferred as:

$$\frac{dE_{CV}}{dt} = [\dot{H}_{ng} + \dot{H}_{ca} + \dot{H}_{ma} + \dot{W}] - [\dot{H}_{cg} + \dot{H}_{sa} + \dot{Q}]. \quad (4-4)$$

In this analysis, the maximum surface heat losses of the copper piping network were calculated to be 7 W due to insulated piping surfaces and relatively short piping lengths and were consequently considered negligible. As well, Burch *et al.* [9] studied TWH performance and concluded that shell losses from the TWH unit are negligible. Furthermore, electrical power to the system is typically a negligible portion of the total system energy [5]. For constant control volume mass as in Eq. (4-1), assuming constant specific heat capacities and defining quantity enthalpy Eq. (4-4) can be resolved as

$$\frac{dE_{CV}}{dt} = [\dot{m}_{ng} h_{f,ng}^{\circ}] - \left[(\dot{m}_{ng} + \dot{m}_{ca}) \sum_{i=1}^N X_i (h_{f,i}^{\circ} + c_{p,cg} (T_{cg} - T_{ca})) + \dot{m}_a c_{p,a} (T_{sa} - T_{ra}) \right]. \quad (4-5)$$

For the combo system operating in space heating only, cyclic behaviour of the system outlet air temperature was observed. As such, a steady-state assumption cannot be applied to Eq. (4-5) to develop an expression for first law efficiency. For cycling operation, a single cycle was defined as the period from when the burner fired, shut off to the instant before the burner fired again. Considering a single cycle ensures that the continued heat transfer from the AHU is captured within the first law efficiency even when the TWH burner was not operating. As such, Eq. (4-6) is integrated across the period of one cycle, defined as time t^* to give the expression

$$\oint_{t^*} \left(\frac{dE_{CV}}{dt} \right) dt = \oint_{t^*} \left[[\dot{m}_{ng} h_{f,ng}^{\circ}] - \left[(\dot{m}_{ng} + \dot{m}_{ca}) \sum_{i=1}^N X_i (h_{f,i}^{\circ} + c_{p,cg} (T_{cg} - T_{ca})) + \dot{m}_a c_{p,a} (T_{sa} - T_{ra}) \right] \right] dt. \quad (4-6)$$

Across one complete cycle the change in energy of the control volume is zero in Eq. (4-6). As well, the chemical enthalpy of the natural gas stream was estimated as the product of the volumetric flow rate and the lower heating value of natural gas estimated for the Edmonton area [38] where the study took place. The flue gas analysis will be considered in a separate analysis. Eq. (4-6) can be rearranged into an expression for first law efficiency,

$$\eta_1 = \frac{\oint_{t^*} \dot{m}_a c_{p,a} (T_{sa} - T_{ra}) dt}{\oint_{t^*} [\dot{V}_{ng} LHV_{ng}] dt}. \quad (4-7)$$

For the case where the combo system for space heating only operated in steady-state, steady flow (SSSF) conditions, the integrands become constants and Eq. (4-7) resolves into a simple expression for first law efficiency as

$$\eta_1 = \frac{\dot{m}_a c_{p,a} (T_{sa} - T_{ra})}{\dot{V}_{ng} LHV_{ng}}. \quad (4-8)$$

In Eqns. (4-7) and (4-8), the specific heat capacity of the air was based on the average temperature of the supply and return air temperature as per CSA P.9-11 [26].

The quality of energy used towards a given objective can be described by second law efficiency (alternatively known as exergy efficiency). Exergy analysis has been shown to be a useful tool in analysis of heating systems relative to performance and cost effectiveness [36, 39, 40]. Previous studies on the benefits of exergy analysis have shown exergy analysis results in improvements for systems not necessarily attainable using energy analysis, including: increased efficiency, reduce fuel usage, reduced environmental emissions and cost savings [41]. In this way, exergy analysis aids in thermodynamic analysis by identifying improvement to efficiency and reductions in thermodynamic losses to develop sustainable technology [41]. More specifically, exergy analysis can allow for improvement in energy-resource utilization, enabling identification of energy wastes and losses and quantifying how much an energy system or process can be improved by [36]. Additionally, especially for energy conversion systems with complex interactions, an exergy analysis can aid in increased performance and cost effectiveness by highlighting specific components within the system that contribute to significant energy

losses [42]. This can be especially useful when considering fossil fuel powered heating systems. Fossil fuels are a high grade energy source used for a relatively modest low-grade energy needs. Through use of exergy analysis, components in the system with the highest losses in work availability from the perspectives of losses in availability of energy can be identified. In part, the justification for this methodology of analysis was motivated by the results presented by the *Energy Efficiency Trends in Canada 1990 to 2010* [3], which showed that despite the introduction of higher efficiency appliances into the space and water heating market, the overall usage of energy for space- and water heating increased. In this observation, an argument for use of exergy analysis in the design and implementation of these appliances can be made given that focus on higher efficiency has not necessarily reduced national energy usage (and associated GHG emissions). An exergy analysis will allow for calculation of the exergy efficiency of the heat exchanger system in the AHU and allows for quantification of the effectiveness of the energy system in utilizing thermal energy contained in the heated water to heat air. In order to perform an exergy analysis, the second law of thermodynamics is applied and is:

$$d\dot{S} = \frac{\delta\dot{Q}}{T_0} + d\dot{S}_{\text{gen}}. \quad (4-9)$$

Integrating between an arbitrary state 1 and state 2 and solving for \dot{Q} , Eqn. (4-9) becomes

$$\dot{Q} = T_0(\dot{S}_2 - \dot{S}_1 + \dot{S}_{\text{gen}}). \quad (4-10)$$

State 2 in Eq. (4-10) can be defined as a “dead” state such that the maximum work available (*i.e.*, the exergy) will be when no entropy is generated. Eq. (4-10) can be substituted directly into the first law of thermodynamics of Eq. (4-3) and the resulting equation can be solved directly for the rate of work term to develop an expression for the exergy rate for a specific fluid stream as

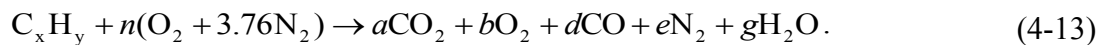
$$E\dot{x} = \dot{m}[(h - T_0s) - (h_0 - T_0s_0)]. \quad (4-11)$$

The exergy efficiency of a heat exchanger is the ratio of the exergy rate of the cold stream to the exergy rate of the hot stream [43]. For the case of the AHU heat exchanger, the cold stream is the air stream and the hot stream is the water stream

$$\eta_{II,HE} = \frac{E\dot{x}_{out,cold} - E\dot{x}_{in,cold}}{E\dot{x}_{in,hot} - E\dot{x}_{out,hot}}. \quad (4-12)$$

4.1.2. Energy Losses by Combustion Products

To model the combustion process in the TWH for the purpose of analysis of energy losses in the flue gases, the following combustion chemical reaction was used:



In Eq. (12), the products are assumed to be ideal gases. The TWH used in this study was not a condensing model, and thus any water in the product gases was assumed to be in the vapour phase. The product composition of the flue gas was determined by measurement of carbon dioxide, oxygen and carbon monoxide content by a flue gas analyzer and the remaining products were determined by an atom balance. For determination of the total energy loss from the flue gases, the sensible heat of the flue gases were determined, assuming the specific heat capacity was constant [44]. The sensible energy loss from the flue gases was calculated by

$$Q_f = \dot{m}_f c_{p,f} (T_{cg} - T_a). \quad (4-14)$$

4.1.3. Appliance Duty Cycle

To quantify the cyclical air temperature case where the TWH exhibited cycling behaviour by turn on and off the TWH burner, the duty cycle parameter was used. A duty cycle is defined as the percentage of elapsed time in which the device is operating [35]. In this study, the device is represented by the TWH burner and the operating time is represented by the burner firing. Therefore, a duty cycle represented the percentage of time that the TWH burner fires compared to the time for one complete burner cycle. For the purpose of this study, a complete burner cycle was defined as the time in which the TWH burner fires and shuts off to when the TWH burner fires again. Therefore, a burner duty cycle of 100% will indicate the TWH burner fires continuously and as duty cycle decreases, the frequency of the burner cycling will increase. The duty cycle can be simply calculated by the following expression:

$$D = \frac{t_{\text{on}}}{t_{\text{on}} + t_{\text{off}}} \times 100\% . \quad (4-15)$$

4.2. SIMULTANEOUS OPERATION (SPACE AND WATER HEATING)

4.2.1. First and Second Law Efficiencies

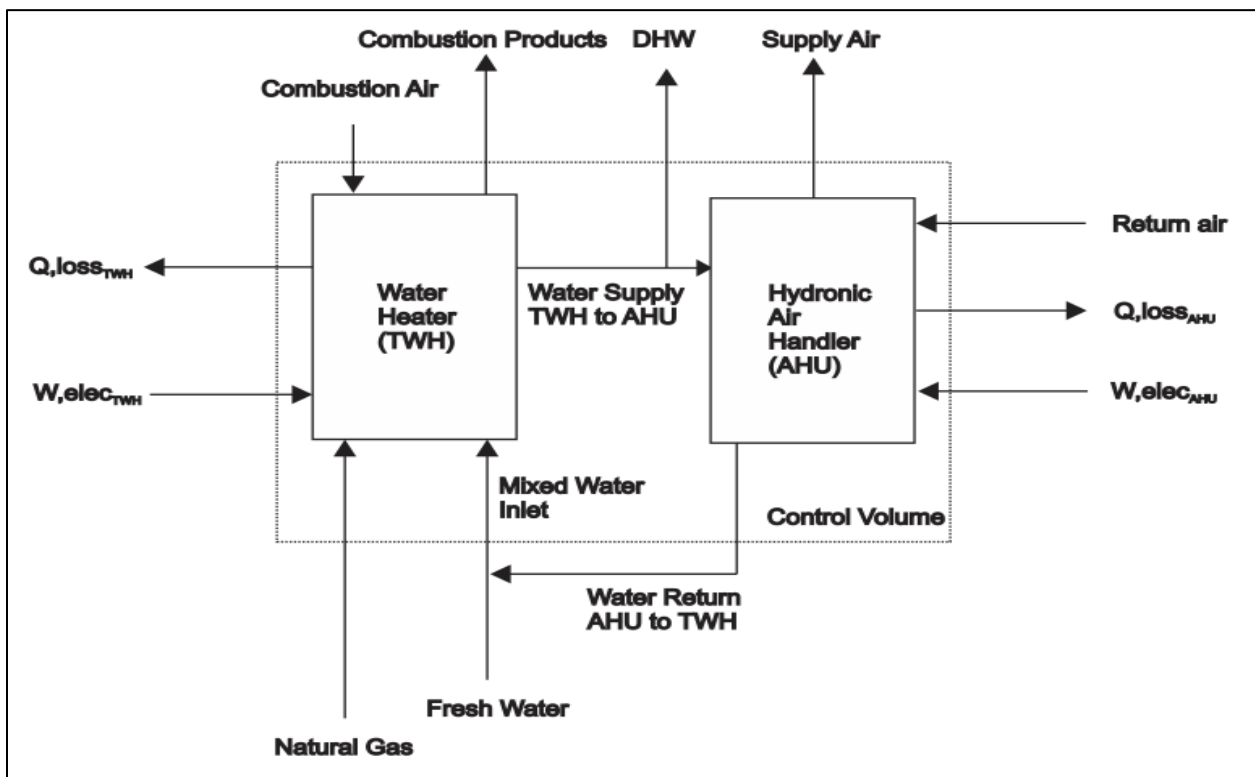


Figure 4-2: Control volume for analysis of combo system operating for space and air heating.

As in the previous section, a control volume approach was utilized to develop an expression for first law efficiency for the combo system providing both space and water heating simultaneously. Figure 4-2 shows the control volume selected for evaluation of the combo system operating in this fashion. By using the control volume shown in Figure 4-2, this approach allows for the inclusion of heat losses from the piping network of the combination system in the analysis.

Entering the control volume in Fig. 4-2 was: natural gas to the TWH, air for combustion to the TWH, ambient supply air to the AHU, mixed inlet water to the TWH and electrical work to the appliances. Exiting the control volume in Fig. 4-2 was: combustion exhaust gases, supply (heated) air from the AHU, water return from the AHU, water exiting for domestic use and surface heat losses. Eq. (4-3) can be applied to the control volume in Fig. 4-1 and expressed in terms of enthalpies, work, and heat transferred as:

$$\frac{dE_{CV}}{dt} = [\dot{H}_{mw} + \dot{H}_{ng} + \dot{H}_{ca} + \dot{H}_{ra} + \dot{W}] - [\dot{H}_{DHW} + \dot{H}_{rw} + \dot{H}_{sa} + \dot{H}_{cg} + \dot{Q}]. \quad (4-16)$$

Considering shell heat losses and electrical work negligible similar to the space heating only case, Eq. (4-16) becomes

$$\frac{dE_{CV}}{dt} = [\dot{m}_{ng} h_{f,ng}^{\circ}] - \left[(\dot{m}_{ng} + \dot{m}_{ca}) \sum_{i=1}^N X_i (h_{f,i}^{\circ} + c_{p,cg} (T_{cg} - T_{ca})) + \dot{m}_{sa} c_{p,a} (T_{sa} - T_{ra}) \right] + \dot{m}_{DHW} c_{p,w} (T_{DHW} - T_{mw}) + \dot{m}_{rw} c_{p,w} (T_{rw} - T_{mw}) \quad (4-17)$$

To account for the cyclical temperature fluctuations noted during operation of the combo system, Eq. (4-17) must consider a single complete cycle. For simultaneous operation, a single cycle is considered the time from which the mass flow rate of water is restricted, relaxed back to normal operation, and flow is restricted again. Integrating over the period of one cycle, defined as time t^* , yields the following expression

$$\oint_{t^*} \left(\frac{dE_{CV}}{dt} \right) dt = \oint_{t^*} \left[\dot{m}_{ng} h_{f,ng}^\circ - \left[\left(\dot{m}_{ng} + \dot{m}_{ca} \right) \sum_{i=1}^N X_i \left(h_{f,i}^\circ + c_{p,cg} (T_{cg} - T_{ca}) \right) + \dot{m}_{sa} c_{p,a} (T_{sa} - T_{ra}) \right] + \dot{m}_{DHW} c_{p,w} (T_{DHW} - T_{mw}) + \dot{m}_{rw} c_{p,w} (T_{rw} - T_{mw}) \right] dt. \quad (4-18)$$

Across one complete cycle the change in energy of the control volume is zero in Eq. (4-18). As well, the chemical enthalpy of the natural gas stream was estimated as the product of the volumetric flow rate and the lower heating value of natural gas estimated for the Edmonton area [38] where the study took place. Equation 4-18 becomes:

$$\eta_1 = \frac{\oint_{t^*} \left[\dot{m}_{sa} c_{p,a} (T_{sa} - T_{ra}) + \dot{m}_{DHW} c_{p,w} (T_{DHW} - T_{mw}) + \dot{m}_{rw} c_{p,w} (T_{rw} - T_{mw}) \right] dt}{\oint_{t^*} \left[\dot{V}_{ng} LHV_{ng} \right] dt}. \quad (4-19)$$

For steady-state and steady flow (SSSF) conditions, the integrands are constant and Eq. (4-19) resolves into

$$\eta_1 = \frac{\dot{m}_{sa} c_{p,a} (T_{sa} - T_{ra}) + \dot{m}_{DHW} c_{p,w} (T_{DHW} - T_{mw}) + \dot{m}_{rw} c_{p,w} (T_{rw} - T_{mw})}{\dot{V}_{ng} LHV_{ng}}. \quad (4-20)$$

Similar to Eqs. (4-7) and (4-8), the specific heat capacity of the water and air is the average specific heat capacity of the fluid at the inlet and outlet temperatures for Eqs. (4-19) and (4-20) as required by the Canadian standard, CSA P.9-2011[26].

The AHU exergy efficiency can be examined in the simultaneous case the same as in the space heating only case by utilizing Eq. (11) where the cold stream is the air stream and the hot stream is the water stream for the AHU heat exchanger.

5. RESULTS AND DISCUSSION

5.1. SPACE HEATING MODE ONLY

5.1.1. Time-Temperature Trace Curves

For the combo system operating only as a space heating system, time-temperature trace profiles were developed from the measured data for each TWH outlet water temperature setting tested. Figures 5-1 and 5-2 showed the time-temperature traces for the combo system operating with TWH water settings of 60°C and 52°C, respectively. Time temperature trace profiles for TWH water outlet settings of 57°C and 54°C can be found in Appendix A.

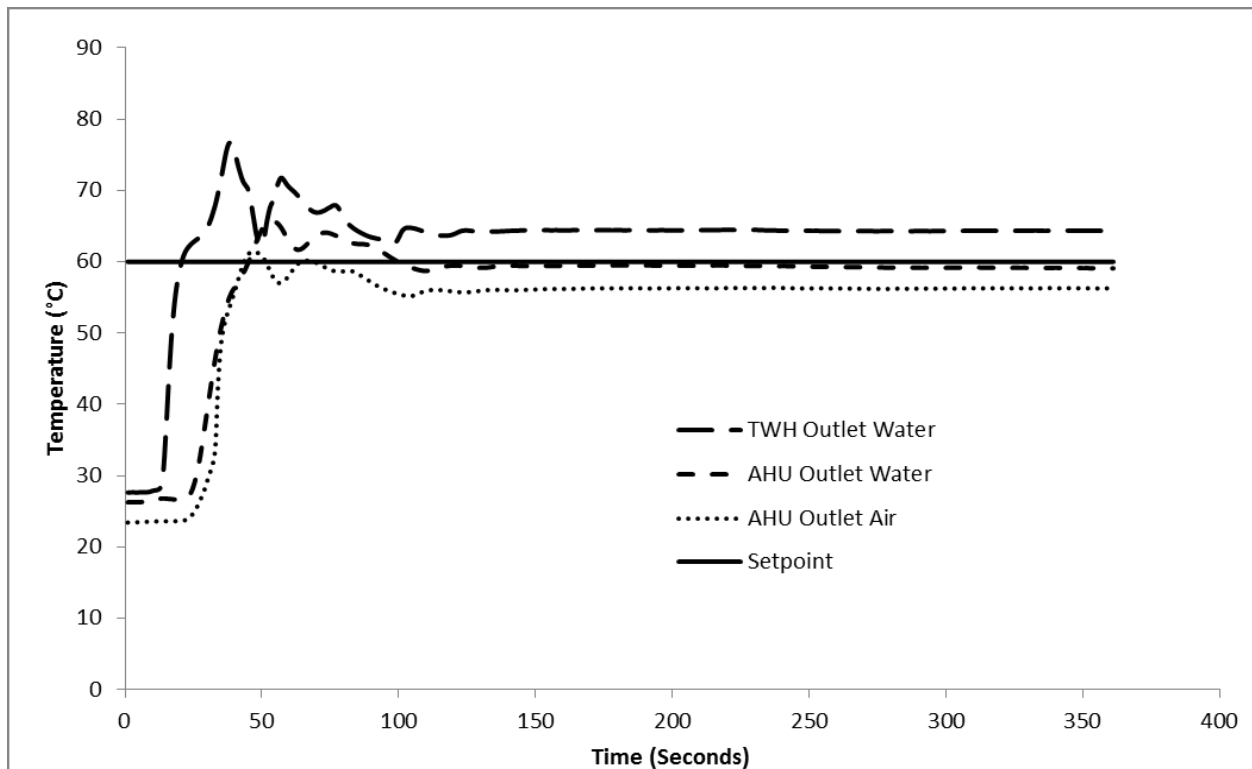


Figure 5-1: Temperature trace for the combo heating system operating in space heating mode with a tankless heater water outlet temperature setting of 60°C.

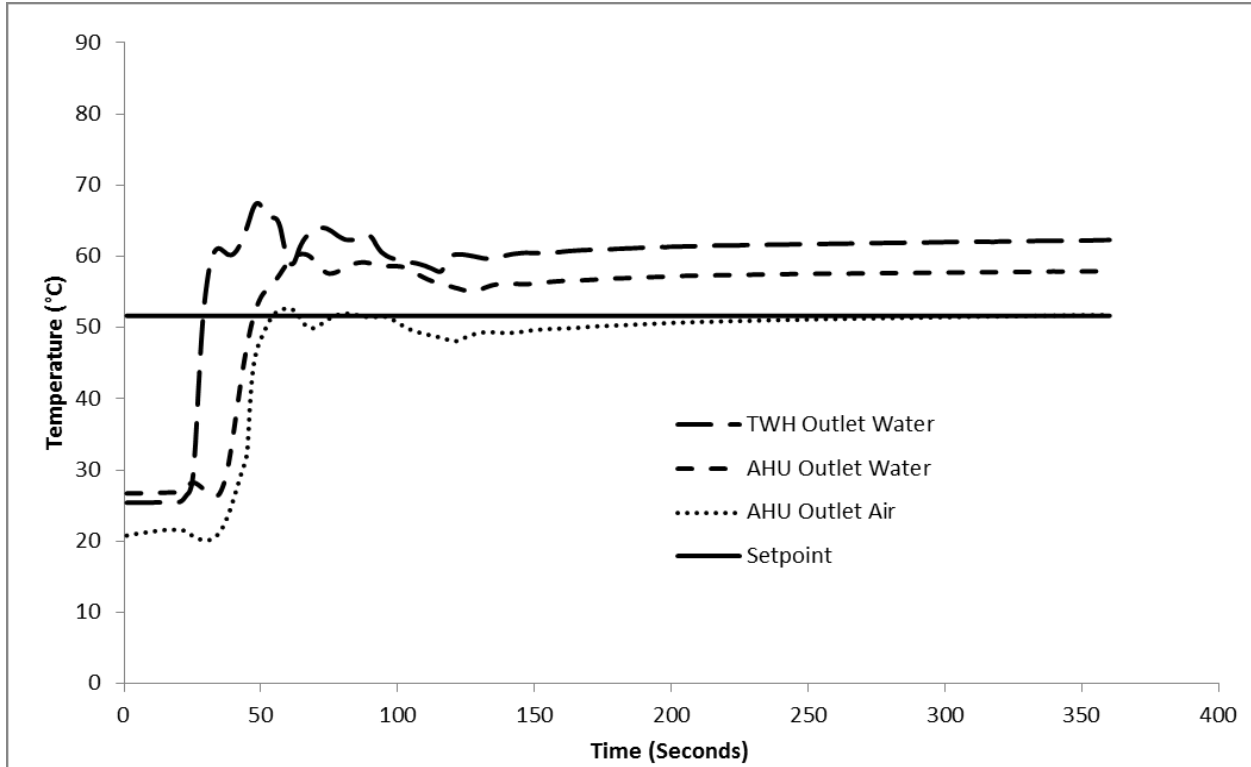


Figure 5-2: Temperature trace for the combo heating system operating in space heating mode with a tankless heater water outlet temperature setting of 52°C.

Observations of Figs. 5-1 and 5-2 showed the combo system operated in steady-state for providing space heating. The system operation for this range of TWH outlet water temperature settings (52°C, 54°C, 57°C and 60°C) was observed an initial period of transience before reaching a steady-state profile characterized by a stable supply air temperature being provided by the AHU. Observations of the steady-state traces in Figs. 5-1 and 5-2 showed, unsurprisingly, that the AHU outlet air temperature (supply air) for space heating was strongly influenced by the temperature of the water supplied by the TWH to the AHU inlet. For temperature settings below 52°C, the combo system was observed to have a fluctuating supply air temperature trace profiles. At these settings, the TWH was found to cycle on and off which resulted in cyclical supply air temperature, as shown in Figs. 5-3 and 5-4 for a TWH water temperature setting of 49°C and

38°C, respectively. This undesirable behaviour, which caused variable supply air temperatures to be delivered for space heating, is known as short-cycling [35] and will be discussed in Section 5.1.2. Time temperature trace profiles for TWH water outlet settings of 46°C and 43°C can be found in Appendix A.

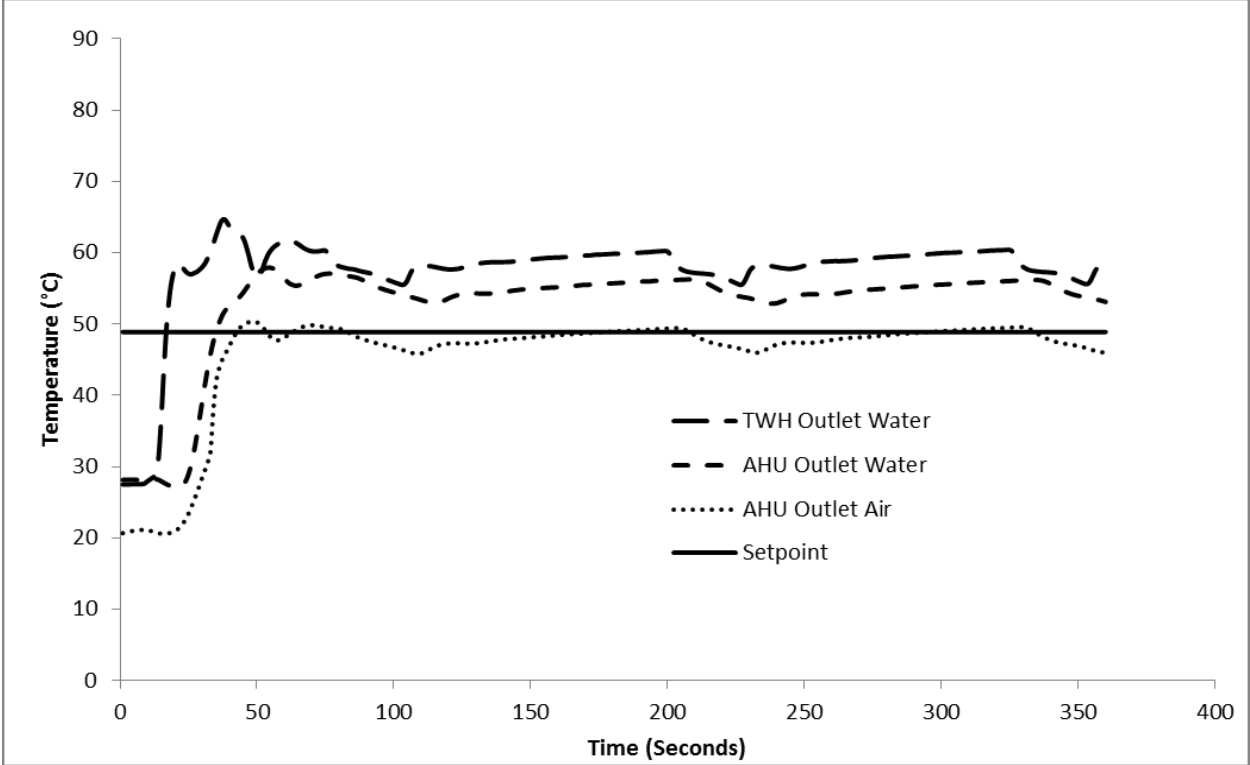


Figure 5-3: Temperature trace for the combo heating system operating in space heating mode with a tankless heater water outlet temperature setting of 49°C.

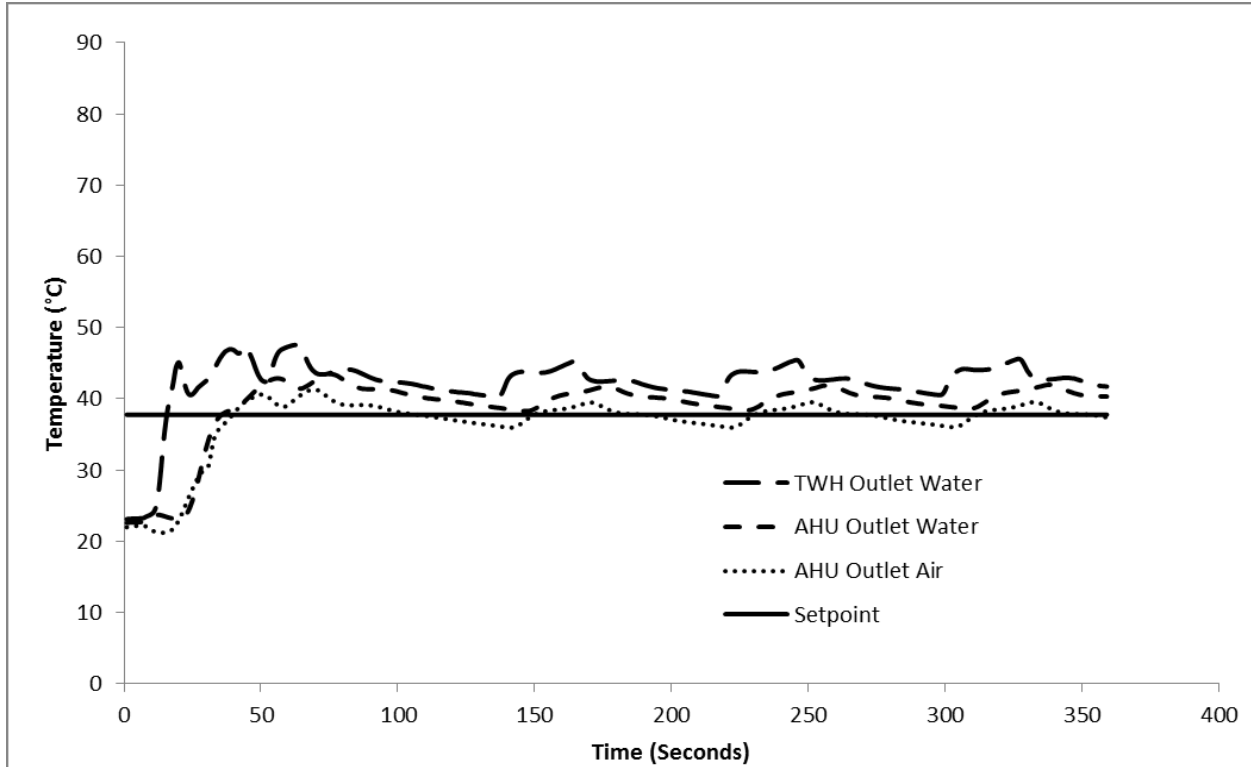


Figure 5-4: Temperature trace for the combo heating system operating in space heating mode with a tankless heater water outlet temperature setting of 38°C.

The average supply air temperature produced by the combo system for space heating can be seen in Table 5-1. To prevent occupant discomfort, delivered (supply) air temperature should be 43°C (110°F) or higher [21]. Table 5-1 showed the system provides comfortable supply air temperatures from TWH outlet water settings of 43°C to 60°C. Additionally, Figs. 5-1 to 5-4 clearly showed that the temperature of water from the outlet of the TWH exceeded the TWH water outlet setting temperature between 4°C to 10°C, as shown in Table 5-1. This characteristic was likely the result of the on/off control strategy of the TWH where the controller operated based on a temperature differential that was centered on the set point value [35]. Impacts from the deviation of actual water outlet temperatures from the TWH compared to the TWH water outlet setting temperature will be discussed in Section 5.1.3.

Table 5-1: Performance data for the combo system for space heating for various TWH settings.

TWH Water Outlet Setting Temperature (°C)	Average Supply Air Temperature (°C)	Actual TWH Water Outlet Temperature (°C)
60	56.0	63.7
57	54.4	63.3
54	52.5	62.8
52	51.0	61.5
49	47.9	58.7
46	45.0	55.9
43	42.4	51.2
38	37.1	43.7

5.1.2. Short-Cycling Behaviour

Given that the characteristic behaviour of a device or system that short-cycles is the operation of the system in short but frequent cycles, the concept of a duty cycle (previously defined in Section 4.1.3) was applied to the combo system operating for space heating only where short-cycling was found to occur. For application for a combo system, the duty cycle represented the percentage of time that the TWH burner fired for relative the time for one complete burner cycle. A complete burner cycle was defined as the time from when the burner fires, shut off to the instant before the burner fired again. For TWH outlet water temperature settings of 52°C to 60°C where short-cycling was not observed, the duty cycle would be considered to be 100% as the burner was continuously firing with no interruptions. For the case of 100% duty cycle, the TWH was operating at steady-state without short-cycling. For all other cases where the duty cycle was less than 100%, the system can be considered to be short-cycling. For TWH outlet water temperature settings below 52°C, the frequency of the cycling behaviour was observed to increase as outlet water temperature decreased. To quantify the change in frequency, the duty cycles for each outlet water temperature were calculated and the results are presented Fig. 5-5.

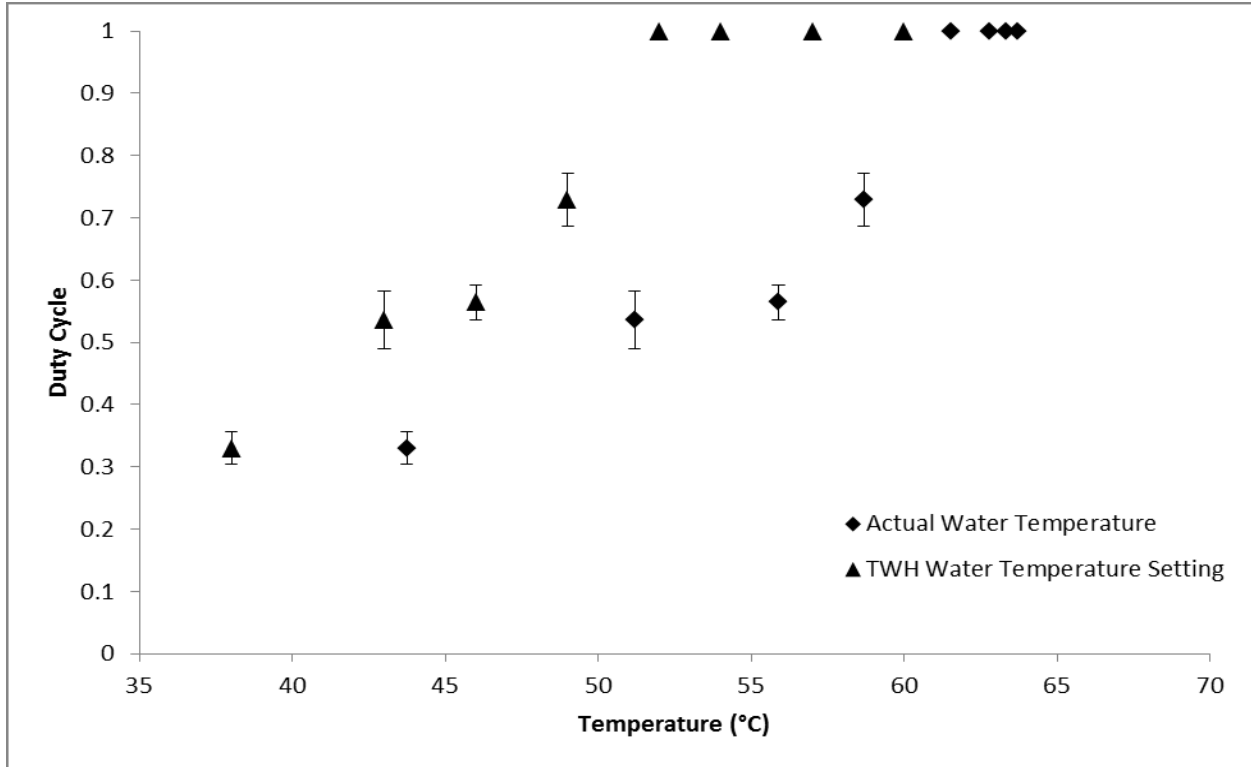


Figure 5-5: Duty cycle for TWH outlet water temperature settings and measured outlet water temperature where short-cycling was observed.

Fig. 5-5 shows as TWH temperature setting decreases that duty cycle also decreases and confirms the observation that the frequency of cycling increased as temperature setting decreased. The combo system duty cycle decreased from 73% at TWH outlet water temperatures settings of 49°C to 33% at TWH outlet water temperatures of 38°C. Therefore at the TWH outlet water setting of 38°C, there was a higher frequency of supply air temperature fluctuations in heated air for space heating than compared at 49°C settings.

From a space heating perspective considering, the tendency of the combo system to deliver air at a fluctuating temperature at these lower temperature settings is a highly undesirable trait, given that conventional space heating systems are able to reliably provide heated air at a stable

temperature. Additionally, this behaviour characteristic would likely negatively impact consumer comfort. Generally, the life expectancy of fossil fuel powered appliances are significantly reduced due to operational short-cycling [35]. Increased frequency of operation will lead to premature failure or unscheduled maintenance of appliance components such as gas burners, gas valves and ignition components. This would negatively affect consumers in the form of increased maintenance and repair costs over the lifetime of the combo system given that TWH maintenance costs are typically greater than that of those of conventional water tanks systems [9].

Short-cycling in hydronic systems occurs when heat transfer from the hydronic unit is outpaced by heat production of the heating plant of the hydronic system [35]. From a design perspective, short-cycling behaviour in a combo system indicates that the heating source was oversized for the hydronic heater [35, 45]. At these lower operating points along the heat dissipation curve, the water outlet temperature from the TWH increases as the system operates. Once the temperature set point is reached and TWH burner shuts down. The system water temperature begins to decrease as heat is dissipated across the AHU. The water temperature decreases until the lower water temperature set point for the TWH is reached and triggers the TWH to activate. This short-cycle operating mode is common for hydronic systems where the heat source is oversized for the hydronic heater [35].

Mitigation or elimination of short-cycling behaviour could be done by having a TWH appliance with an increased burner turndown ratio [35, 45]. The larger the turndown ratio, the greater the ability of the burner to adjust to required heat input and respond to system loading. A turndown ratio can be defined as the ratio of maximum heat output from a modulating heat source to the minimum stable heat output from the heat source [35]. Alternatively, heat transfer from the

AHU can be improved to reduce return water temperatures from the AHU to the TWH. This can be achieved by either allowing for modulating water pumps or blowers with a system control strategy that adjusts pump or blower speed to optimize AHU heat transfer.

5.1.3. Temperature Set Point Overshoot

As seen in Table 5-1, the combo system TWH heating plant produced outlet water temperatures in excess of the set point temperature. Figure 5-6 shows the calculated ratio of thermal energy input to the system over the theoretical thermal energy input based on the TWH controller set point. From Fig. 5-6 the TWH can be seen to have exceeded the required thermal energy input to the system by 7% to 21% and consequently overheats the water temperature to be greater than that of the set point temperature. This overshoot of the temperature set point is an attributed design characteristic of TWH systems [34] as TWH units will have a cold water bypass loop that will mix with the heated water outlet of the TWH. The purpose of this design is to dampen the transient temperature fluctuations that are typically observed in TWH units. This feature is advantageous in a combo system since it must provide domestic hot water (DHW) at temperatures above 50°C (122°F) [46] in order to effectively prevent the growth of bacteria and avoid burning the end user at faucets [7, 8]; however, the upper limit on water temperature is unnecessary when the unit is operating is for space heating only. Furthermore, for space heating, this performance feature of exceeding the set water temperature can be considered as a negative performance characteristic for operation of the combo system. However, this water temperature overshoot is an intentional design feature and ultimately as the thermal energy produced was still used for space heating, the propensity for the TWH to overproduce the required thermal energy input is not factored into the definition of thermal efficiency in this study. The TWH tested in this study has

a staged burner, and the decrease in the overheating ratio at the 38°C temperature setting may be due to the TWH controller being able to more accurately meet the required heat input due to the burner turndown ratio. As discussed previously in Section 5.1.1., the TWH temperature control was based on a water temperature differential based on the set point value, and this control strategy also contributed to the TWH overheating based on the set point temperature.

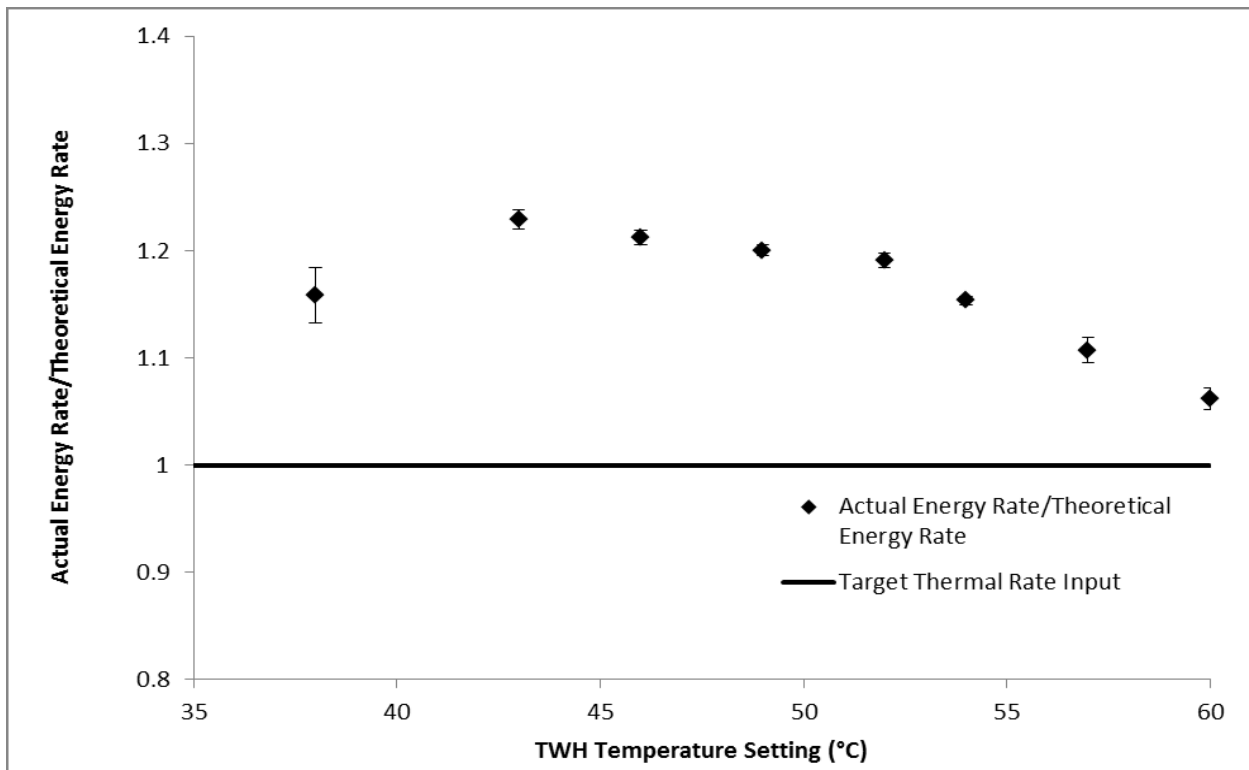


Figure 5-6: Ratio of thermal energy input to the system and the theoretical thermal energy required for the TWH unit at various TWH water temperature outlet settings.

5.1.4. First Law Combo System Efficiency

From Eqns. (4-7) and (4-8), the first law efficiency of the combo system can be calculated for the cyclical and steady-state stage, respectively. For TWH water outlet temperature settings of 52°C to 60°C where the system achieved steady-state, the first law efficiency was computed using Eqn. (4-8) as the average efficiency across the steady-state stage. For TWH water outlet temperature settings of 38°C to 49°C where short-cycling was observed, the first law efficiency was calculated using Eqn. (4-7). The calculated first law efficiencies for each TWH water outlet temperature setting can be seen in Fig. 5-7. Overall, first law efficiency can be seen to have a direct relationship with AHU inlet water temperature. As TWH outlet water temperature decreased, there was a corresponding decrease in first law efficiency. For TWH water outlet temperature settings of 52°C to 60°C where the combo system was observed to reach steady-state (see Figs. 5-1 and 5-2), first law efficiency was found to range between 88% and 95%, respectively. A laboratory study on TWH combo systems found the designs to have steady-state space heating efficiencies in excess of 85% [21]. The efficiencies for this range of temperature settings of 52°C to 60°C suggest that combination systems have comparable performance to mid- to high efficiency furnaces. However, for TWH water outlet temperature settings of 38°C to 49°C where the combo system was observed exhibit short-cycling behaviour, first law efficiency was found to range between 39% and 76%, respectively. This observation of Fig. 5-7 clearly showed the decrease in efficiency in the combo system was related to the decreased duty cycle (see Fig. 5-5) and resulting short-cycling of the TWH unit (see Figs. 5-3 and 5-4). The observed decrease in first law efficiency likely occurred due pre- and post-purge energy losses from the combustion chamber of the TWH. For pre-purging and post-purging, the combustion fan in the TWH operated to remove any uncombusted fuel or combustion product gases that may have

accumulated in the chamber as a safety feature. This pre- and post-operation purging likely removed thermal energy from the system.

Possible sources of efficiency losses in the combo system include thermal energy removed by the exhaust gases and thermal energy losses via the walls of the TWH. However, previous work has found that thermal losses from the TWH unit walls are negligible [9]. Efficiency degradation due to thermal losses from combustion products have not been studied previously for combo systems operating for space heating, and will be investigated in Section 5.1.5.

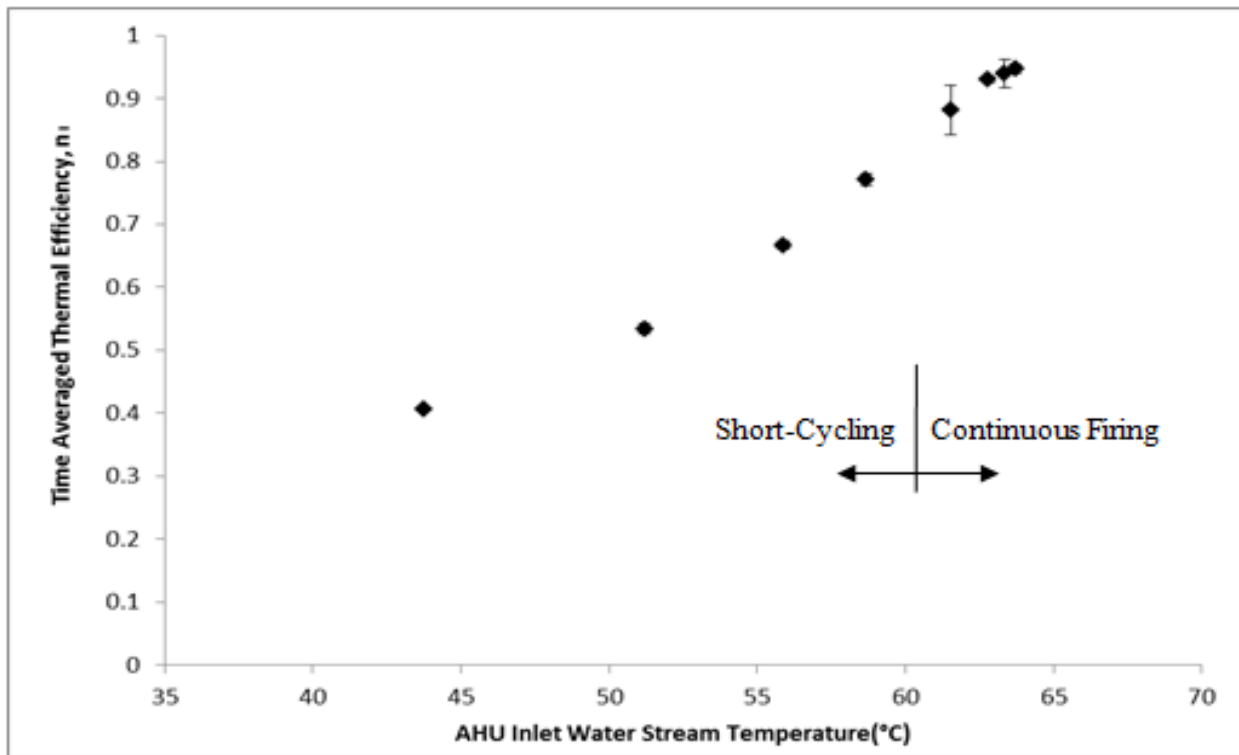


Figure 5-71: First law efficiency for combo system operating in space heating mode in the operational stage as a function of inlet water temperatures to the hydronic AHU.

5.1.5. Energy Losses Through Flue Gases

Assuming combustion products for the combustion process in the TWH as in Eqn. (4-13); Eqn. (4-14) was used to estimate the energy lost to the flue gases to the external environment. Figure 5-8 shows the calculated flue gas energy losses as a percentage of total energy loss as a function of the duty cycle for each TWH outlet water temperature setting.

When considered total energy input from the natural gas fuel to the combo system, Fig. 18 showed that flue gas energy losses represented 3% to 47% of the total energy losses for TWH outlet water temperature settings of 60°C and 38°C, respectively. Previous estimates from literature for stack losses of typical TWH systems to be approximately 25% [12].

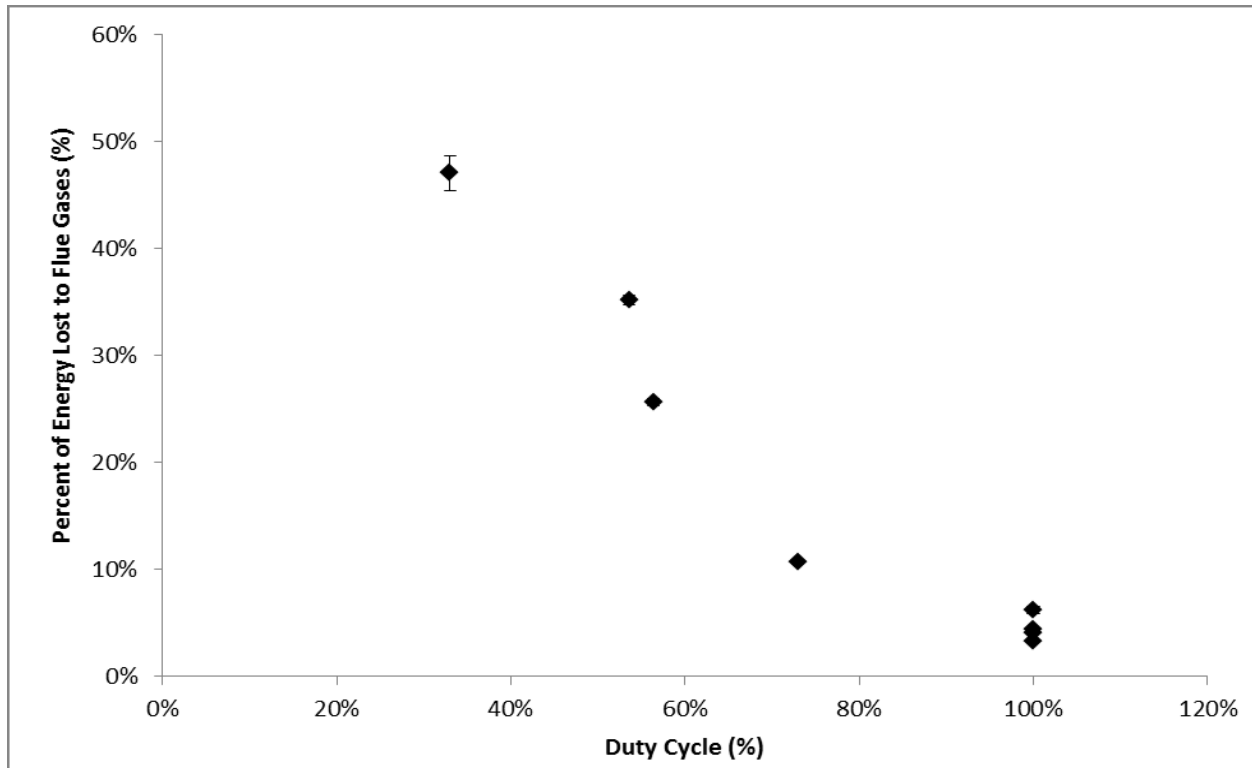


Figure 5-8: Ratio of energy losses to exhaust gases for combo systems operating in space heating mode in the operating stage for varying duty cycles.

Observation of Fig. 5-8 clearly showed that when the combo system reached steady-state operation for duty cycles of 100%, the percentage of energy lost in combustion exhaust products was decreased compared to energy lost through the flue gases when the system short-cycled. Referring to Fig. 5-7, the decrease in first law efficiency can be found to be correlated with a corresponding increase in percent of energy lost to flue gases, suggesting that energy losses through the TWH stack was the primary source of energy loss. Elimination of operational short-cycling may lead to the reduction of energy losses through the TWH stack. This may be achieved by using a TWH with a burner with a high turndown ratio which will allow the system to better accommodate the changes in output heating required. From a control perspective, increasing the temperature control differential may allow for the elimination of short cycling by increasing the allowable water temperature variance during TWH operation [35]. Alternatively, a heat recovery system could be employed to recover energy losses to the flue gases. Although the concentric venting system of the TWH already acts as a basic heat recovery system, Fig. 5-8 suggests that a heat recovery system with increased capacity for heat recovery would be necessary to reduce energy losses by way of combustion product gases through the flue, especially for cases where the system will short-cycle.

5.1.6. AHU Heat Exchanger Second Law Efficiency

The second law efficiency of the heat exchanger within the AHU calculated using Eqn. (4-12) and results are presented in Fig. 5-9. For evaluation of exergy in relation to the fluid streams, the system dead state selected was 25°C and 1 atm [47]. Observation of Fig. 5-9 showed exergy efficiency of the AHU heat exchanger was found to peak at 36% for 60°C TWH outlet water

temperature setting and was found to be as low as 15% for a TWH outlet water temperature setting of 38°C. Fig. 5-9 demonstrates a clear relationship of a decrease in exergy efficiency of the AHU heat exchanger for a decrease in inlet water temperature to the AHU from the TWH. This trend is similar to the relationship observed in Fig. 5-7 for first law efficiency. For cases where the duty cycle was less than 100%, the system was seen to short-cycle and this behaviour can be directly linked to not only to decreased first law efficiency and increased energy losses through exhaust gases but decreased second law efficiency of the AHU as well. Previous studies focusing on exergy efficiency of hydronic space heating systems found exergy efficiency of space heating radiators ranged between 17.3% to 27.6% [48]. The results suggest that the combination system operating for space heating only was able to perform comparably to other hydronic space heating methods. However, previous studies relating to the exergy efficiency of electric heat distribution systems found exergy efficiency of electric baseboard systems to range between 53% to 57% [16]. Thus, combination heating systems for space heating have comparable performance to other hydronic space heating systems, but are outperformed by electric baseboard heater systems. Furthermore, the results suggest for TWH outlet water temperature settings of 52°C to 60°C, the combo system AHU exergy efficiency surpassed the exergy efficiency of conventional radiators for space heating.

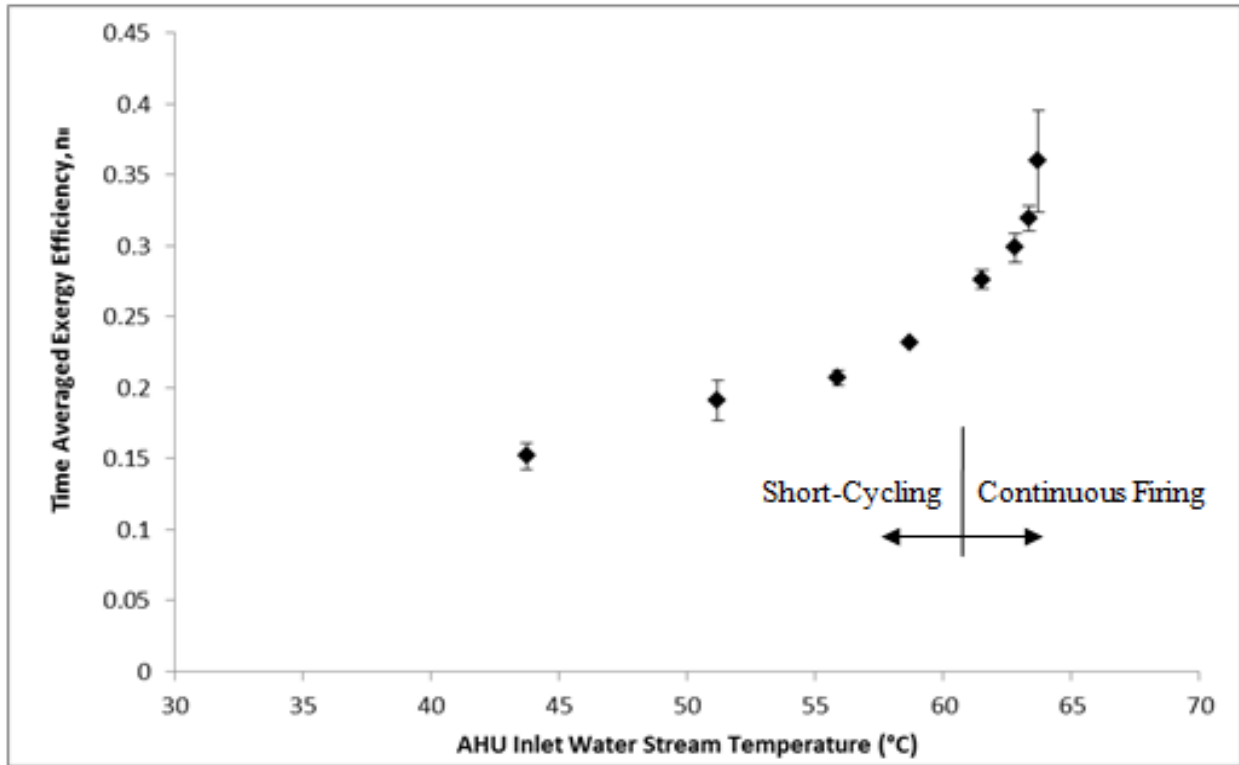


Figure 5-9: AHU heat exchanger exergy efficiency for combo system operating in space heating mode in the operational stage for varying inlet water stream temperatures to hydronic AHU.

5.2. SIMULTANEOUS OPERATION (SPACE- AND WATER HEATING)

5.2.1. Time-Temperature Traces – 7.6 L/min (2 GPM) DHW draw

To analyze combo system performance for simultaneous operation, time-temperature trace profiles were developed from the measured data for each TWH outlet water temperature setting tested and for a DHW draw rate of 7.6 L/min (2 GPM). The DHW draw event was initiated when the combo system operating for space heating only reached steady-state or the cyclical state stage. Practically, a 7.6 L/min draw rate represents a low-usage hot water event such as a hot

water faucet draw or shower hot water draw. The combo system was found to be able to deliver heated water for DHW application at a rate of 7.6 L/min at TWH outlet water temperature settings of 38°C and 54°C. However, at TWH outlet water temperature settings of 57°C and 60°C, the DHW flow rate provided was limited to 4.9 L/min (1.3 GPM) and 2.3 L/min (0.6 GPM), respectively. This reduction in flow rate is caused by an internal feature in the TWH which reduces flow rate to allow for the combo system to be able to heat water to the water temperature set point. The TWH unit has a maximum water flow rate that the unit will operate under based on the required water temperature rise.

For each temperature setting tested, the combo system was observed to have a transient response to the DHW event before achieving a steady state delivery temperature for both DHW and supply air. At TWH outlet water temperature settings of 38°C and 54°C, the combo system was observed to have an transient response that ranged from 62 s and 88 s, respectively, before achieving a steady-state DHW and supply air temperature profile after a 7.6 L/min DHW draw was introduced to the combo system initially operating for space heating only. Conversely, for TWH outlet water temperature settings of 57°C and 60°C where the flow rate of water was throttled, the combo system was observed to have a transient response that ranged from 153 s and 170 s, respectively, before steady-state air and water delivery temperatures were achieved. This can be seen when comparing the time-temperature trace profile for the TWH water outlet temperature of 38°C in Fig. 5-10 and for water outlet temperatures of 57°C and 60°C as seen in Figs. 5-11 and 5-12, respectively. The increased transient response observed at the 57°C and 60°C TWH outlet water temperature setting was likely related to the throttled water flow rate observed at these settings.

From Section 5.1.1 that dealt with the combo system operating in space heating mode only, the combo system was observed to short-cycle for TWH outlet temperature settings of 38°C to 49°C (see Figs. 5-3 and 5-4). However, at TWH outlet temperature settings of 38°C to 49°C, after a 7.6 L/min DHW event was introduced to the system, the combo system was observed to reach steady-state DHW and supply air temperature delivery. As discussed in Section 4.1.2, short-cycling in hydronic systems occurs when heat transfer from the hydronic unit is outpaced by heat production of the heating plant of the hydronic system. AHU outlet water was mixed with cold feed water which in turn reduced the temperature of the water entering the TWH system. The return water with elevated temperature to the TWH was the cause of the short-cycling of the system for TWH outlet water temperature settings less than 52°C. This observation shows that elimination of short cycling can be possible in the space heating only case if increased heat transfer from the AHU (which in turn would result in lowered AHU outlet water temperatures) can be achieved. Possible methods to improve air handler performance included control strategy that adjusted blower speed to better adjust to system loading to allow for increase heat transfer and consequently lower AHU outlet water temperatures being fed to the TWH water inlet.

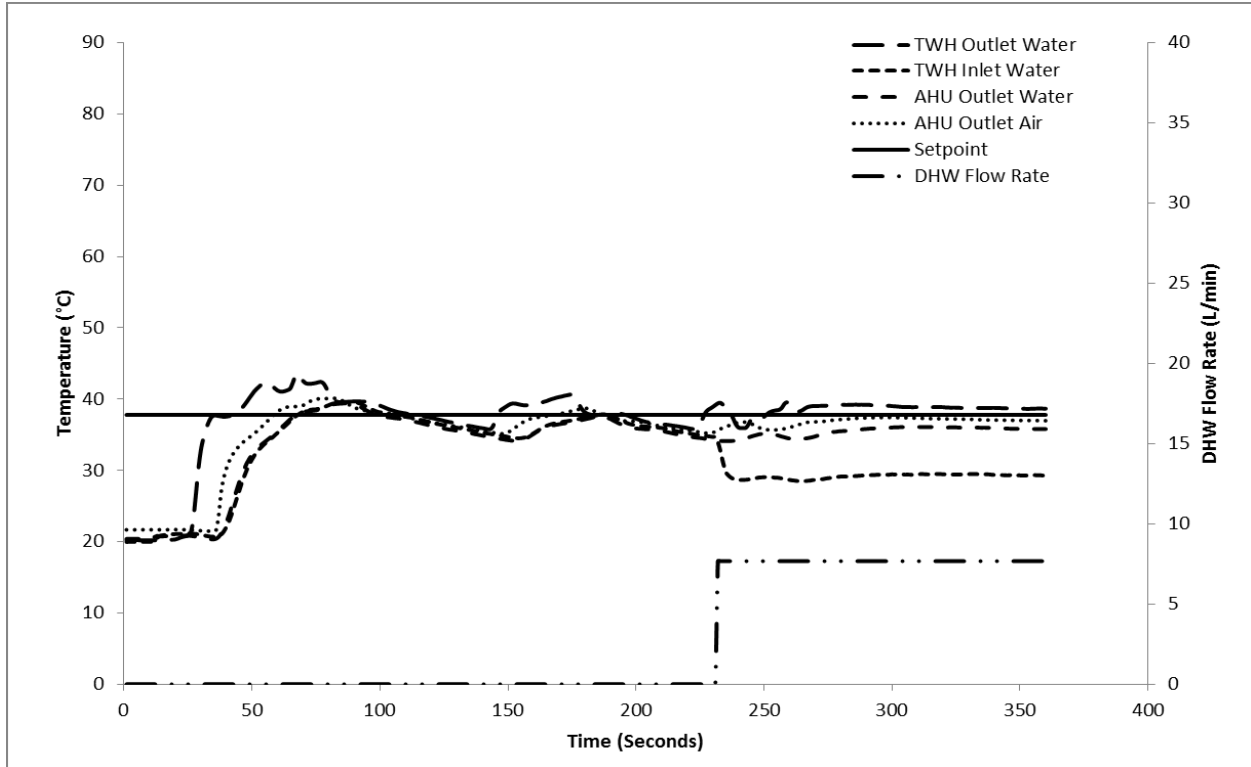


Figure 5-10: Temperature trace for a combo heating system operating for space and water heating with a tankless heater water outlet temperature setting of 38°C and requested 7.6 L/min DHW draw.

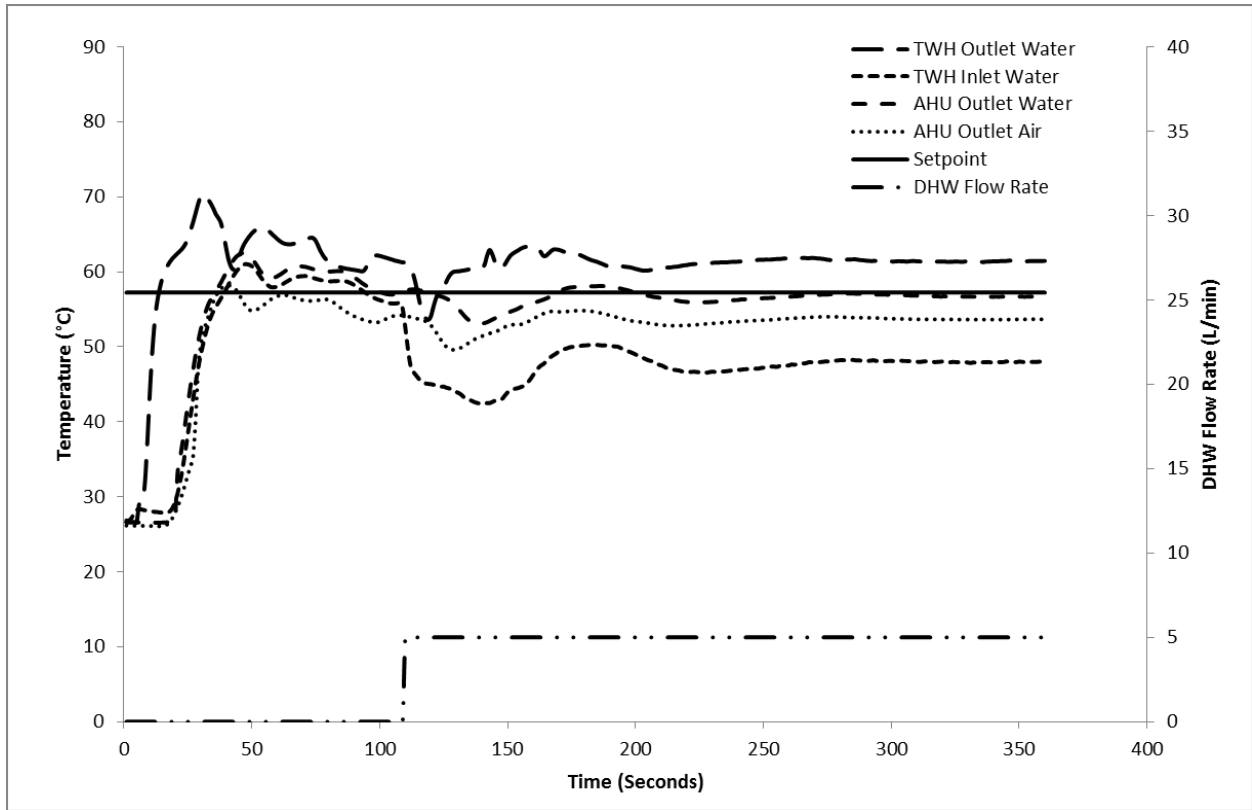


Figure 5-11: Temperature trace for a combo heating system operating for space and water heating with a tankless water heater outlet temperature setting of 57°C and requested 7.6 L/min DHW draw, but the system only provided 4.9 L/min.

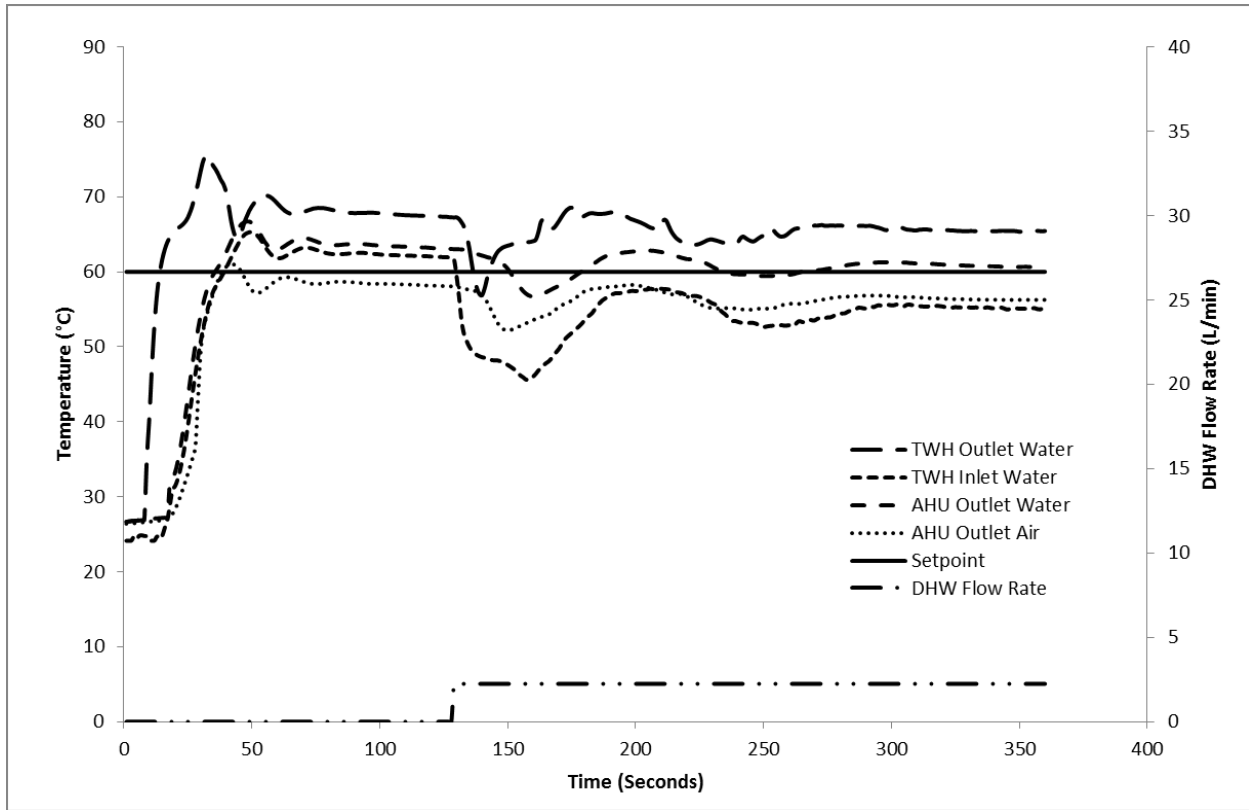


Figure 5-12: Temperature trace for a combo heating system operating for space and water heating with a tankless water heater outlet temperature setting of 60°C and requested 7.6 L/min DHW draw, but the system only provided 2.3 L/min.

Table 5-2 shows the average measured value of supply air for space heated delivered by the combo system for when the combo system operated for space heating only and during space- and water heating for each TWH outlet water temperature setting tested. To prevent occupant discomfort, delivered (supply) air temperature should be 43°C (110°F) or higher [21]. As in the space heating only case, the combo system was observed to provide acceptable supply air temperatures at TWH outlet water temperature settings of 43°C and above. Table 5-2 also showed that supply air temperatures provided by the combo system during space heating only operation and simultaneous operation for a 7.6 L/min DHW draw event. The combo system

responded to the introduction of the DHW water flow rate by adjusting water flow rate throughout the system and achieved similar supply air temperatures for space heating. Time-temperature trace profiles from the experimental data for TWH outlet water temperature settings of 43°C, 46°C, 49°C, 52°C and 54°C can be found in Appendix B.

Table 5-2: Heated Air Temperature for Space Heating Mode Only and Combined Operation 7.6

L/min requested draw

TWH Water Outlet Setting Temperature (°C)	Space Heating Only-Supply Air Temperature (°C)	Combined Operation - Supply Air Temperature (°C)	DHW Flow Rate (L/min)	AHU Water Flow Rate (L/min)
60	56.0	56.2	2.3	13.5
57	54.4	53.5	4.8	12.7
54	52.5	50.8	7.6	10.9
52	51.0	48.3	7.6	11.0
49	47.9	46.0	7.4	11.2
46	45.0	43.4	7.5	11.0
43	42.4	41.0	8.2	10.4
38	37.1	37.2	7.4	10.9

5.2.2. *Time-Temperature Traces – 15.1 L/min (4 GPM) DHW draw*

To analyze combo system performance for simultaneous operation, time-temperature trace profiles were developed from the measured data for each TWH outlet water temperature setting tested and for a DHW draw rate of 15.1 L/min (4 GPM). From a design standpoint a 15.1 L/min draw rate represents dual low-usage hot water event such as a hot water faucet draw and shower draw. For testing, the events were assumed to initiate and end at the same instant. The combo system was found to be able to deliver heated water for DHW application at a rate of 15.1 L/min at TWH outlet water temperature settings of 38°C and 54°C. However, as observed in the 7.6 L/min case for TWH outlet water temperature settings above 54°C, the water flow rate through the TWH was throttled at TWH outlet water temperature settings of 57°C and 60°C. At TWH outlet water temperature settings 57°C, the DHW water flow rate varied between 2.3 L/min (0.6 GPM) and 15.5 L/min (4.1 GPM). This behaviour has not been observed previously in literature and was classified as flow cycling. At TWH outlet water temperature settings of 60°C, the DHW flow rate provided was limited to 2.7 L/min (0.7 GPM). This reduction in flow rate was caused by an internal feature in the TWH which reduces flow rate to allow for the combo system to be able to heat water to the water temperature set point.

For each temperature setting tested, the combo system was observed to have a transient response to the 15.1 L/min DHW event before achieving a steady state delivery temperature for both DHW and supply air. At TWH outlet water temperature settings of 38°C and 54°C, the combo system was observed to have an transient response that ranged from 78 s and 83 s, respectively, before achieving a steady-state DHW and supply air temperature profile after a 15.1

L/min DHW draw was introduced to the combo system that was initially operating for space heating only (see Fig. 5-13).

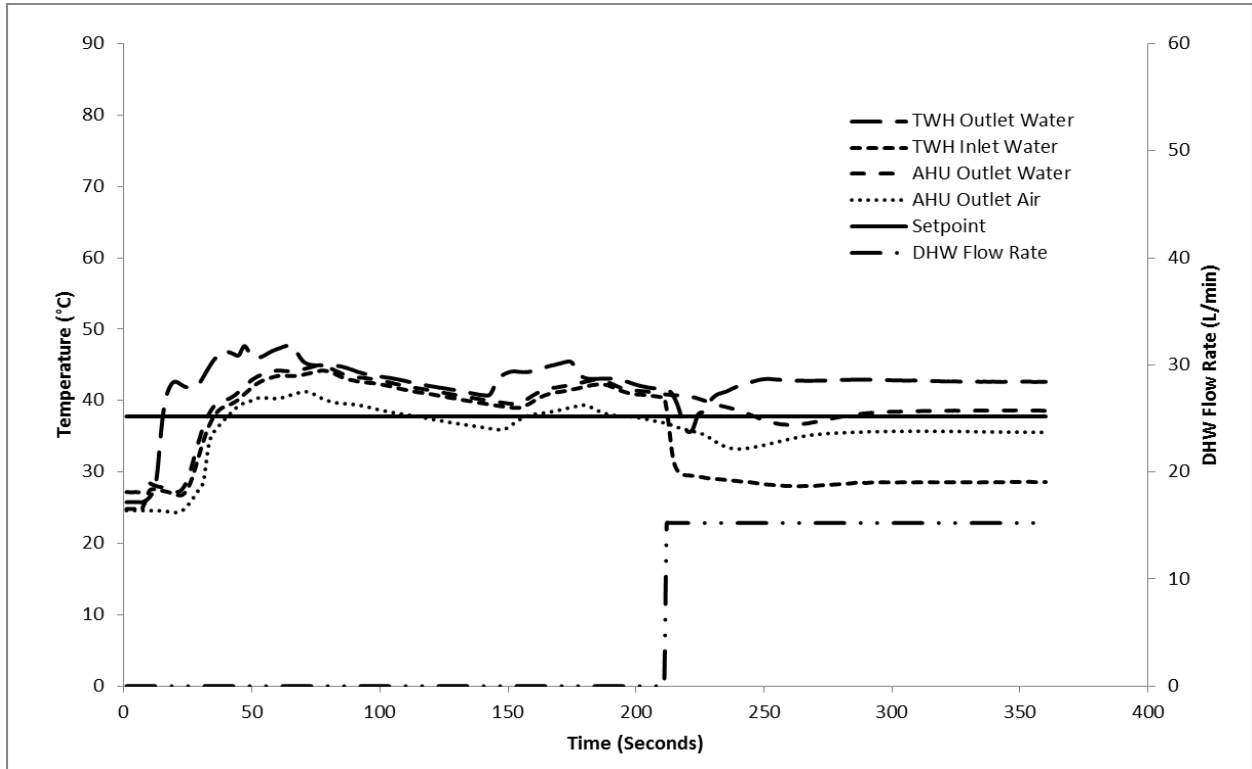


Figure 5-13: Temperature trace for a combo heating system operating for space and water heating with a tankless water heater outlet temperature setting of 38°C and requested 15.1 L/min DHW draw.

This period of transient response to the 15.1 L/min DHW event is similar to the period of transient response measured for the 7.6 L/min DHW event. However, the behaviour of the combo system diverges in the 15.1 L/min case from the 7.6 L/min case at a TWH outlet water temperature setting of 57°C.

For the 57°C TWH setting, the flow rate water through the system was observed to vary in a cyclical pattern as shown in Fig. 5-14. This mass cycling behaviour resulted in a severe fluctuations of both air and water temperature. This behaviour was likely due to the TWH unit being unable to meet the requisite energy input to the water stream in the TWH based on the water temperature set point. In Fig. 5-14, the outlet water temperature from the TWH can be directly correlated to the water flow rate through the internal compact heat exchanger of the TWH. After the 15.1 L/min DHW draw was introduced to the combo system, the inlet and outlet water temperature of the TWH initially decreased due to the mixing of the return water from the AHU with the cold feed water supply. In response, the TWH throttled the water flow rate to a minimum flow rate of 2.3 L/min, which allowed for the TWH unit to heat the outlet water temperature to the temperature set point. However, once the temperature set point was reached, the flow control valve in the TWH was opened to return flow rate to 15.5 L/min in an attempt to meet the DHW demand, which resulted in a decline in TWH inlet and outlet water line temperature. This decline caused the TWH to again throttle the water flow rate to a minimum to reach the water temperature set point and resulting in the observed system behaviour as in Fig. 5-14. Therefore, the time-temperature trace profile of Fig. 5-14 was the result of the TWH attempting to meet both the DHW flow rate demand and the water temperature setting requirements. This mass cycle behaviour was related to the control strategy of the TWH to

throttle mass flow rate to ensure that set point temperature can be achieved. In this case, where mass is observed to be throttled is due to the increase in water temperature between the inlet and the outlet of the TWH to exceed the heating capability of the unit.

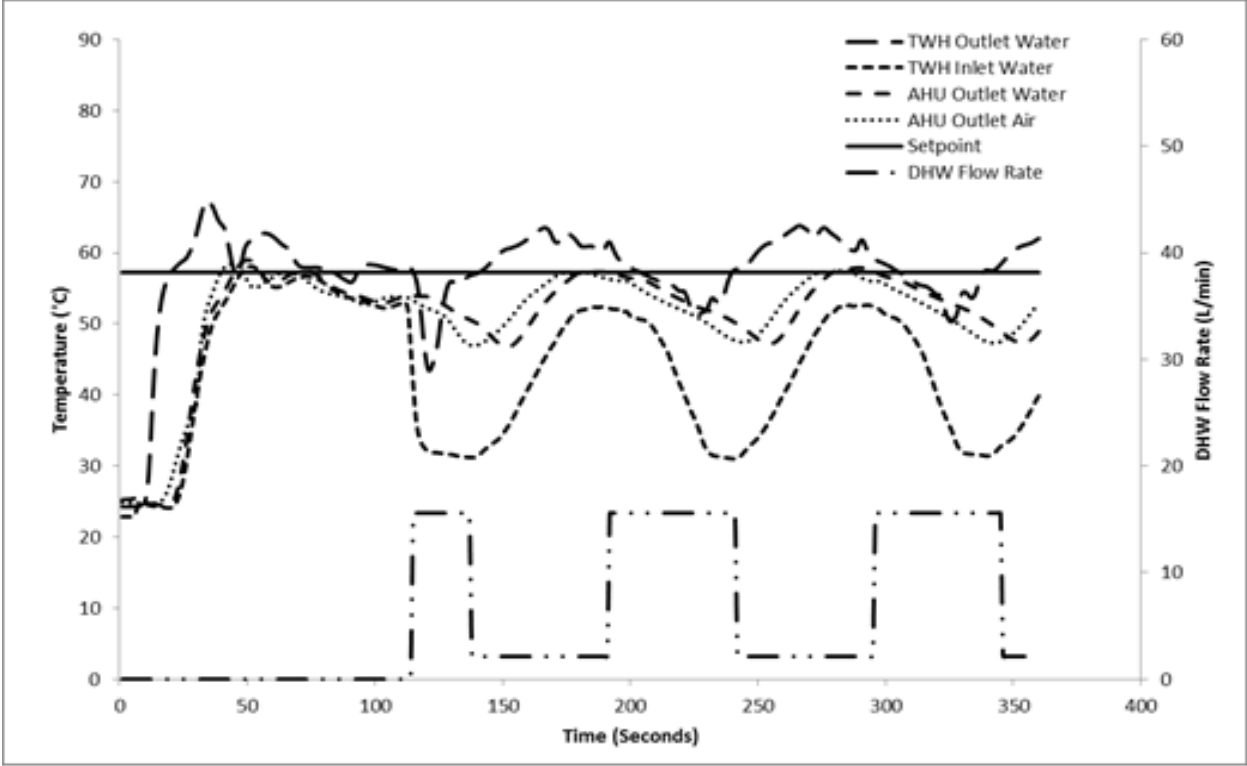


Figure 5-14: Temperature trace for a combo heating system operating for space and water heating with a tankless water heater outlet temperature setting of 57°C and 15.1 L/min DHW draw, but the system provided flows as low as 2.3 L/min.

Each temperature setting likely has a “tipping-point” DHW flow rate, in which the combo system prioritizes DHW flow rate such that the space heating water loop flow rate is minimal or zero until the DHW demand is satisfied, after which the combo system can return to providing space heating operation. This behaviour observed at the 57°C TWH outlet water temperature setting in Fig. 5-14 is not observed in the time-temperature trace profile for the 60°C TWH outlet water temperature setting in Fig. 5-15. However, the water flow rate through the system was throttled to 2.7 L/min and no mass cycling was observed here due to the required heating at this stage to exceed the TWH water heating capabilities. The increased DHW draw flow rate was observed to introduce air and water temperature perturbations in the 15.1 L/min case as shown in Fig. 5-15, resulting in a transient period of 143 s.

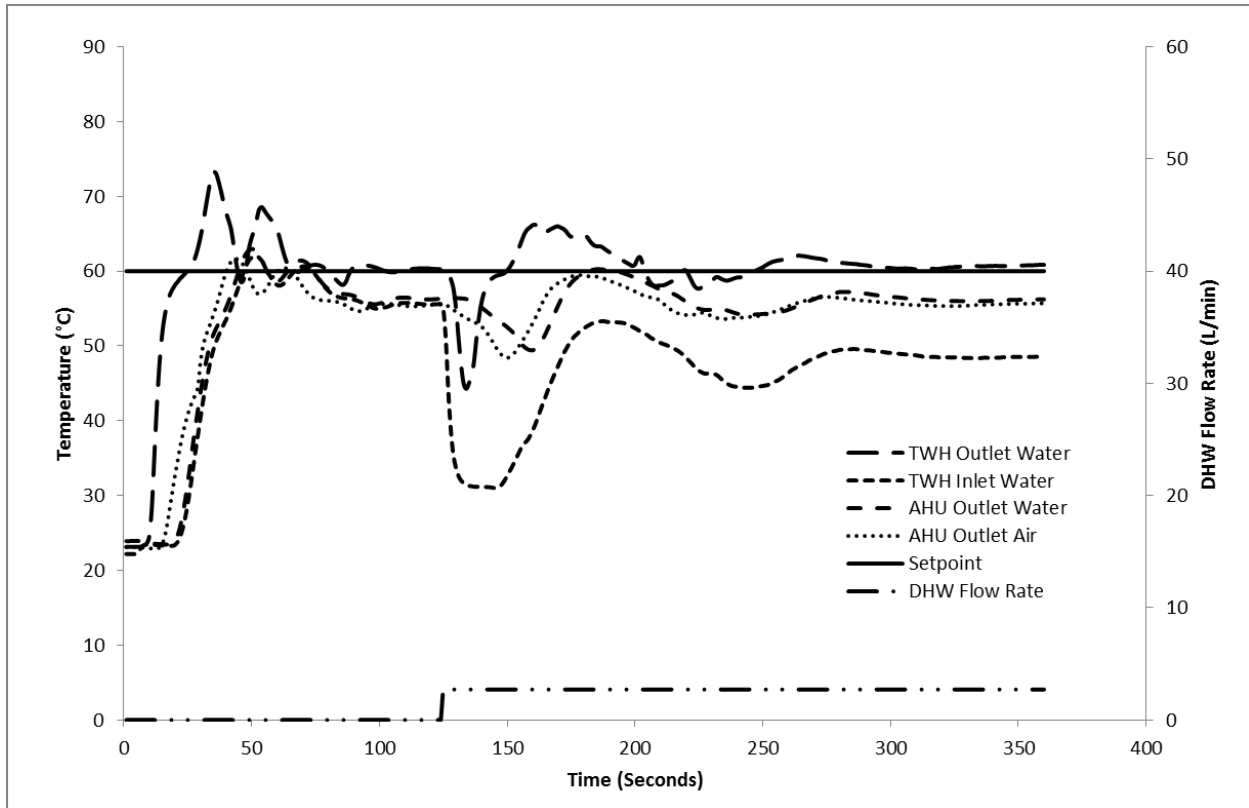


Figure 5-15: Temperature trace for a combo heating system operating for space and water heating with a tankless water heater outlet temperature setting of 60°C and requested 15.1 L/min DHW draw, but the system only provided 2.7 L/min.

This behaviour suggests that finding optimal TWH and AHU systems that work synergistically is an important design challenge, as undesirable cycling behaviour can severely impact performance. Potential solutions to alleviate mass cycling concerns may be a small buffer tank incorporated into the system may help reduce the impact of flow rate fluctuations on the DHW. Alternatively, designers may consider upsizing the combo system to a TWH model with a higher heating capacity if high DHW flow rates are anticipated. However, upsizing the TWH will likely introduce short-cycling across a wider range of TWH outlet water temperature settings when the combo system operates for space heating only. Other solutions include a TWH burner with a higher turndown ratio will allow for increased control of heating by allowing the

TWH to meet the required water temperature precisely. From Section 5.1.1 that dealt with the combo system operating in space heating mode only, the combo system was observed to short-cycle for TWH outlet temperature settings of 38°C to 49°C (see Figs. 5-3 and 5-4). However, at TWH outlet temperature settings of 38°C to 49°C, after a 15.1 L/min DHW event was introduced to the system, the combo system was observed to reach steady-state DHW and supply air temperature delivery as in the 7.6 L/min DHW case. This observation likewise suggests that elimination of short cycling can be possible in the space heating only by improving heat transfer from the water stream to the air stream in the AHU.

Table 5-3 shows the average measured value of supply air for space heating delivered by the combo system for when the combo system operated for space heating only and during space- and water heating with a requested DHW rate of 15.1 L/min. The combo system was found to produce relatively comparable supply air temperatures during simultaneous operation with a 15.1 L/min DHW draw rate to when the combo system was operating for space heating only. However, to prevent occupant discomfort, delivered (supply) air temperature should be 43°C (110°F) [21]. In the case of the 15.1 L/min DHW draw, the combo system was observed to provide acceptable supply air temperatures at TWH outlet water temperature settings of 49°C and above as opposed to the space heating only and 7.6 L/min DHW case, where temperature settings of 43°C and above produced acceptable supply temperatures. This result suggests that as DHW rate increases, the combo system is not able to provide acceptable supply air temperatures at lower TWH outlet water temperature settings. Therefore, DHW draw rate has a significant impact on the combo system ability to provide space heating. Time-temperature trace profiles

from the experimental data for TWH outlet water temperature settings of 43°C, 46°C, 49°C, 52°C and 54°C can be found in Appendix B.

Table 5-3: Heated Air Temperature for Space Heating Mode Only and Combined Operation 15.1

L/min requested draw. System water flow cycling was observed for the 57°C setting,

TWH Water Outlet Setting Temperature (°C)	Space Heating Only-Supply Air Temperature (°C)	Combined Operation 15.1 L/min requested draw-Supply Air Temperature (°C)	DHW Flow Rate (L/min)	AHU Flow Rate (L/min)
60	56.0	56.4	2.7	13.8
57	54.4	52.8	8.8	9.6
54	52.5	49.3	14.8	6.0
52	51.0	45.6	15.5	5.0
49	47.9	42.5	14.5	5.3
46	45.0	40.9	15.8	4.9
43	42.4	38.8	15.8	5.0
38	37.1	35.6	15.0	5.4

During simultaneous operation for space and air heating, the combo system was found to have variability in water flow rate response of the DHW draw and AHU loop to system loading. When the DHW event was introduced into the system, the AHU water flow rate was found to decrease. As well, as DHW flow was throttled to meet the requisite water temperature rise, the AHU water flow rate was found to increase. This relationship between units shows a potential issue for combo system in handling a large dynamic range of requested input energies and illustrates the need for an overall combo control scheme.

5.2.3. First Law Efficiency – 7.6 L/min and 15.1 L/min DHW draws

From Eqs. (4-19) and (4-20), the first law efficiency for each TWH outlet water setting tested with DHW draw rates of 7.6 L/min and 15.1 L/min were calculated as shown in Fig. 5-16.

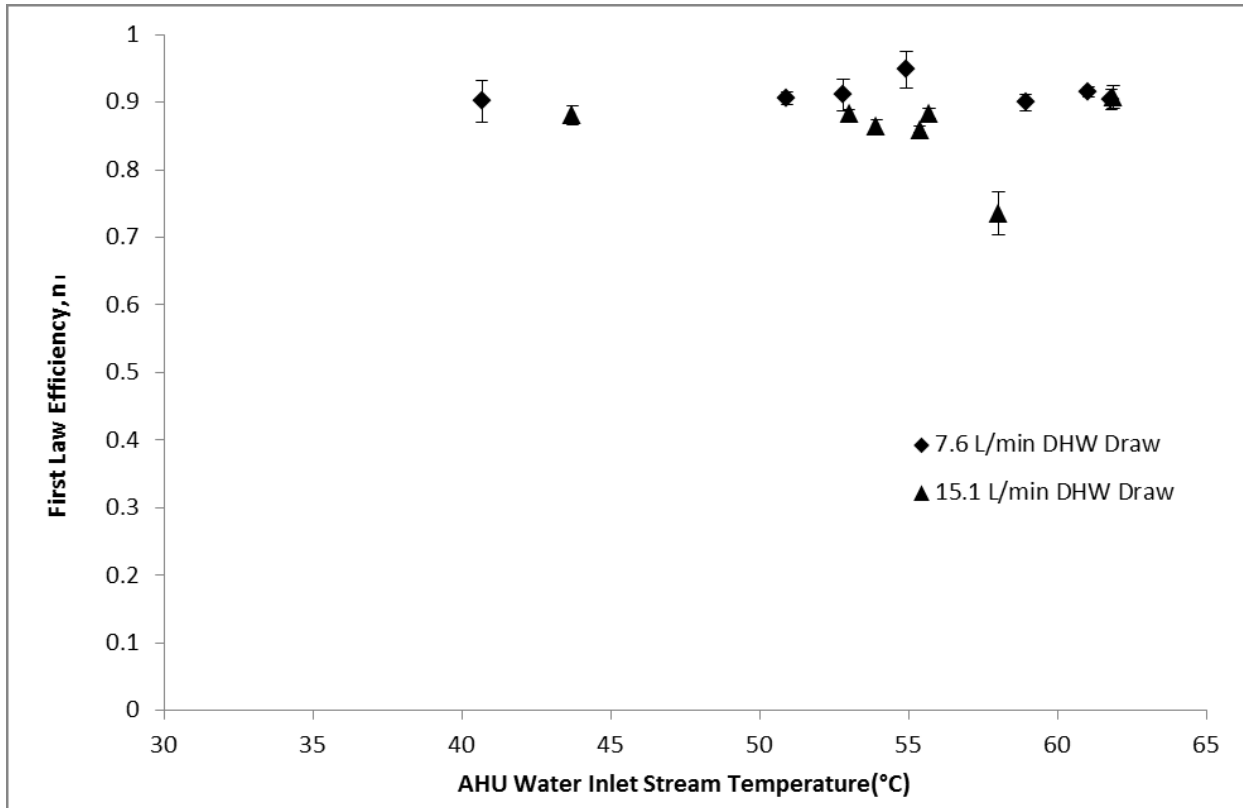


Figure 5-16: First law efficiency for combo system operating for space and water heating as a function of TWH outlet water temperatures for a 7.6 L/min and a 15.1 L/min DHW draw.

As seen in Fig. 5-16, the results suggests for simultaneous operation, the influence of the TWH outlet water temperature setting on first law efficiency was minimal, given that calculated efficiencies for each TWH outlet water setting are relatively similar ranging between 88% and 94% for simultaneous operation with a 7.6 L/min DHW draw. Additionally, for the space heating only operation of the combo system, the combo system was found to short-cycle at TWH outlet water temperature settings of 49°C and below (see Figs. 5-3 to 5-5). The short-cycling behaviour was found to result in a decrease in first law efficiency as discussed in Section 4.1.4. However, observation of the time-trace profiles as in Fig. 5-10 showed that the combo system

was able to reach a steady-state water and air temperature delivery profile after the introduction of a 7.6 L/min DHW draw. The elimination of short-cycling in combo system operation was observed to result in improved the first-law efficiency of the system as seen in Fig. 5-16.

Figure 5-16 also shows the first law efficiency calculated for each TWH outlet water temperature setting for the combo system operating simultaneously. For TWH outlet water settings where the time-temperature traces showed the combo system operating for both space and water heating reaching steady-state, first law efficiency was found to range between 87% and 94%,. As in the 7.6 L/min case, the impact of TWH temperature setting on first law efficiency was found to be relatively minimal. However, for the 57°C setting, where flow cycling was observed, the first law efficiency was found to be 74%. These results suggest that, similar to short-cycling, flow cycling has an adverse impact on combo system thermal efficiency. A possible control strategy to eliminate the system behaviour to modulate mass flow rate is to install a water buffer tank and may help to mitigate the decreased efficiency, especially in the case of high flow rate (greater than 15.1 L/min). However, a previous study concluded that a combo system with a buffer tank will result in a significant decrease in the annual system efficiency due to water storage heat losses from the buffer tank [11].

Previously literature found that for TWHs steady-state operation providing DHW only, the steady-state efficiency was not impact by DHW flow rate or TWH setting temperature [6]. Generally, the combo system operating for simultaneous operation for both a 7.6 L/min and 15.1 L/min DHW draws behaved relatively similar to this noted behaviour for DHW only production,

as steady-state efficiencies were observed to be minimally impacted by temperature setting as seen in Fig. 5-16. However, at the 57°C setting with a 15.1 L/min DHW draw the system was observed to flow cycle and efficiency was calculated to be 74%. Therefore, the first-law efficiency of the combo system providing space and water heating simultaneously was observed to be related to the DHW flow rate drawn from the system in the current combo system configuration. However, if the tendency of the system to cycle water flow rate can be eliminated by use of a small buffer tank or an increased burner turndown ratio, the combo system will be able to provide both space and water heating simultaneously at comparable efficiencies observed in Fig. 5-16 and not be influenced by DHW draw rate. It should be noted that the conclusion by *Grant et al.* [6] that TWH steady-state efficiency is independent of requested DHW flow rate should be further clarified. The system will provide water heating at similar steady-state efficiencies for any requested DHW draw rate but will do so at the cost of requested DHW draw rate, as the system will throttle the water flow rate in order the system to meet the water temperature rise required. In the context of the combo system operating for space and water heating simultaneously, a DHW priority control strategy observed in the TWH prioritized DHW production over space heating operation. In this scenario, the potential exists for the flow rate of water through the space heating loop to stop entirely and the AHU cease operation due increased heating load placed on the TWH. The combo system will cease space heating operation and solely provide DHW heating until the DHW heating requirement has been met or decreased such that the combo system can resume space heating operation. Therefore, the DHW water flow rate may not influence simultaneous operation efficiency, but creates other design issues that adversely impact the ability of the combo system to provide space heating.

Figure 5-16 provided evidence to support combo systems as a high efficiency air and water heating system for residential applications. Simultaneous operation of the combo system was found to produce heated water for DHW applications at first law efficiencies that were comparable to SWH units as well as provide heated supply air for space heating at thermal efficiencies comparable to those of mid- to high-efficiency furnaces. However, these results highlight the important of proper design of the combo system in order to function as a high efficiency system. Selection of the proper size of water heater for the residence is important. If the water heater is undersized, the combo system will not be able to provide simultaneous operation if DHW demand exceeds the TWH heating capacity.

5.2.4. AHU Heat Exchanger Second Law Efficiency – 7.6 and 15.1 L/min DHW draw

The second law efficiency of the heat exchanger within the AHU calculated using Eqn. (11) and results are presented in Fig. 5-17. For evaluation of exergy in relation to the fluid streams, the system dead state selected was 25°C and 1 atm.

In this study, for the combo system operating simultaneously with a requested DHW flow rate of 7.6 L/min, exergy efficiency was observed to increase with TWH outlet water temperature setting. Simultaneous operation with a DHW draw of 7.6 L/min showed the AHU exergy efficiency for the TWH outlet water temperature setting of 38°C and 60°C to vary between 40 to 47%, respectively (see Fig. 5-17). As well, similar to the trend observed for first law efficiency in Section 5.1.4, AHU exergy efficiency was found to be improved for TWH outlet water temperature settings where short-cycling was observed compared to the space heating only case seen in Fig. 5-9.

AHU exergy efficiencies for the combo system operating simultaneously with a requested DHW flow rate of 15.1 L/min are shown in Fig. 5-17. For the case of TWH outlet water temperature settings from 38°C to 49°C, where short-cycling was observed for space heating only operation, exergy efficiencies for these TWH outlet water temperatures settings was found to be improved in simultaneous combo operation and ranged between 31% and 35%. For TWH outlet water temperature settings of 52°C, 54°C, and 60°C, where short-cycling did not occur in the space heating only mode and no flow-cycling was observed in simultaneous operation, the exergy efficiencies were found to range between 42% and 45%. For the case where flow-cycling was observed, the average exergy efficiency was found to be 58%. In this case, the increased exergy efficiency for the flow-cycling case was due to the decreased average mass flow rate of water through the AHU during operation compared to the average mass flow rate of water through the AHU at other TWH settings in simultaneous operation.

For simultaneous operation with requested DHW draw rates of 7.6 L/min and 15.1 L/min, the exergy efficiency was found to be greater than exergy efficiency in literature for space heating radiators, where exergy efficiency ranged between 17% and 28%, depending on the type of heating plant [48]. The results suggest that during the simultaneous operation, the space heating performance of the combo system was comparable to conventional space heating devices.

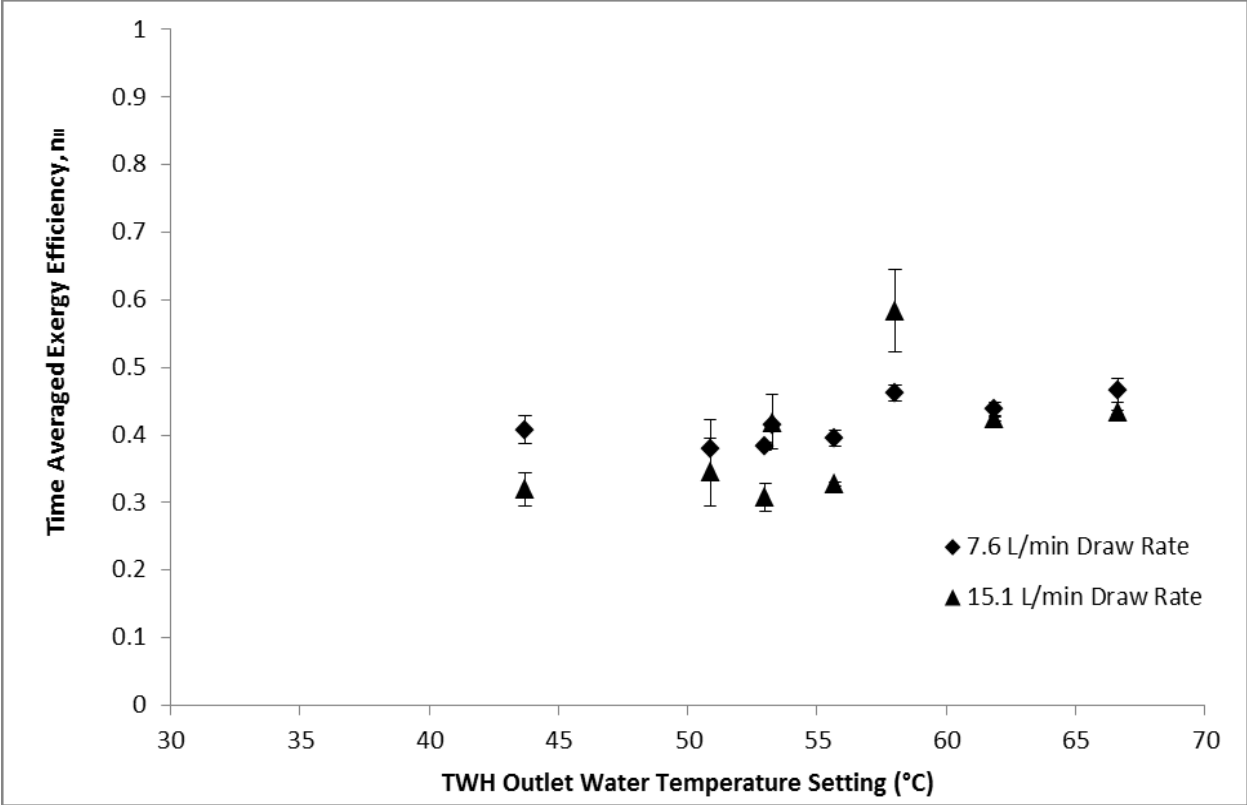


Figure 5-17: AHU heat exchanger exergy efficiency for space and water heating as a function of TWH outlet water temperatures for a 7.6 L/min and a 15.1 L/min DHW draw.

6. CONCLUSIONS

In this study, the performance of a combination heating system with a TWH heating source for use in residential building applications was investigated. Given recent advancements in TWH technology, the performance of a tankless water heater combination heating system is not well developed in previous studies. In particular, the impact of the produced water temperature on performance in space heating only and simultaneous space heating and DHW draw rate modes has not been well studied. To study and quantify expected combination system performance for residential usage, an experimental methodology and theoretical analysis were developed. This study presented first-law efficiencies, and exergy efficiencies of the AHU, during space heating operating as quantifying performance parameters for assessment of the space heating only and simultaneous (space and water heating) operation modalities. Consideration was also given to the dynamic response of water and air temperatures during the onset of demands for heating, or for DHW when the system was already providing heating.

For space heating only, the combo system was found to reach steady-state behavior at TWH outlet water temperature settings of 52°C to 60°C. However, for TWH outlet water temperature settings below 52°C, the system was observed to short-cycle. The short-cycling behaviour was a result of heat production of the TWH being greater than the heat transfer rate of the AHU to the space. To quantify this short-cycling behaviour, the concept of a duty cycle was applied to the TWH burner. Duty cycle was found to range from 73% at TWH outlet water temperatures settings of 49°C to 33% at TWH outlet water temperatures of 38°C, indicating that the frequency of short-cycling increased as TWH outlet water temperature decreased. Short-cycling was found to have an adverse effect on the combo system space heating only performance. First law efficiency for outlet

water temperatures where the system reached steady-state ranged between 88% and 94% and are comparable to thermal efficiencies for mid- to high efficiency furnaces. However, for outlet water temperatures where the system short-cycled, first law efficiency decreased to 76% to 39% at TWH outlet water temperature settings of 38°C to 49°C, respectively. Flue gas analysis showed that a majority of energy losses in the system are due to hot exhaust products being emitted. This was found to be especially true for short-cycling behaviour, where the pre- and post-purge combustion fan operation resulted in 11% to 47% loss of energy. Exergy analysis of the AHU heat exchanger showed an average exergy efficiency of 15% to 36% for TWH outlet water temperature settings of 38°C to 60°C with the decrease in exergy efficiency attributed to short-cycling. The observed exergy efficiencies for the combo system operating for space heating only were similar to exergy efficiencies found for other space heating options such as space radiators.

Short-cycling was found to occur due to heat production of the TWH being greater than the heat transfer rate of the AHU to the space and was indicative of the TWH being oversized for the AHU unit. Short-cycling has several negative implications to combo system performance including: fluctuation supply air temperatures for space heating, restriction to useable temperature settings for TWH operation, increased maintenance costs and premature component and unit failure. Elimination of short-cycling can be done by introduction of small water storage tank, increasing temperature control differential of the system or incorporating a burner in the TWH with a higher turndown ratio. However, these control strategies likely come with increased economic costs.

For simultaneous operation, the combo system was found to output relatively similar heated supply air temperatures compared to the combo system operating for space heating only at a given

water temperature setting for DHW flow rates of 7.6 L/min and 15.1 L/min, where the largest variance in supply air temperature between space heating only operation supply air temperature was 3°C and 5°C, respectively. For simultaneous operation with a requested DHW flow rate of 7.6 L/min, the time-temperature trace profiles showed the system reached steady-state for each water temperature setting. TWH outlet water temperature settings where short-cycling was observed for space heating only operation did not exhibit cycling behaviour under simultaneous operation at the same TWH outlet water temperature settings. This observation was also true in the 15.1 L/min, with exception to the 57°C TWH outlet water temperature setting which deviated from steady-state behaviour. The TWH unit modulated the flow rate of water between 2.3 L/min and 15.5 L/min to meet both the water temperature set point and DHW flow rate requirement. Similar to short-cycling, this flow-cycling behaviour was found to adversely impact first law efficiency. First law efficiency for outlet water temperatures settings where the system reached steady-state for both DHW flow rates tested ranged between 87% and 94% and are comparable to thermal efficiency for mid- and high efficiency furnaces and conventional SWH units. However, for the 57°C TWH temperature setting for a 15.1 L/min draw where flow cycling was observed, the calculated first law efficiency was found to be 74%. For the 7.6 L/min DHW draw case, exergy analysis of the AHU heat exchanger showed an average exergy efficiency of 40% and 47%, for TWH outlet water temperature settings of 38°C to 60°C, respectively. For the 15.1 L/min DHW draw case, exergy analysis of the AHU heat exchanger showed an average exergy efficiency of 42% and 45%, for TWH outlet water temperature settings of 38°C to 60°C, respectively. For the special case of the 57°C setting, where flow-cycling was observed, the average exergy efficiency was calculated to be 58% due to the decreased average flow rate of water through the AHU compared to the other water temperature settings. As in the space heating only case, the observed exergy efficiencies for

the combo system operating for space and air heating were similar to exergy efficiencies found for other space heating options such as space radiators.

The results suggested that short cycling and flow cycling have critical TWH outlet water temperature set points that should be avoided to prevent operational inefficiencies. For short-cycling behaviour for space heating only operation, the oversized TWH heating plant resulted in short-cycling at lower water temperature set points. In simultaneous operation, for increasing DHW draw rates, flow-cycling behaviour will occur at higher water temperature set points. For efficient operation for the combo system that was tested, short-cycling behaviour observed for space heating only and flow-cycling behaviour during simultaneous operation restricted the usable TWH outlet water temperature settings. The TWH outlet water temperature settings selection was restricted to 52°C, 54°C, or 60°C to prevent the occurrence of the adverse cycling behaviour.

Overall, the results demonstrated that combo systems operating for space heating only and simultaneously for space and water heating can provide space and water heating for residential applications at relatively comparable thermal efficiencies to conventional mid- to high-efficiency furnace systems and storage water heater systems. The combo system was also found to have comparable and superior heat exchanger exergy efficiencies compared to typical exergy efficiencies of hydronic space heating options. Furthermore, the results suggest that combo systems require additional control strategies to manage system operation more effectively, as several adverse behaviours were observed and discovered to have significant negative implications on system performance. As well, these adverse behaviours will likely have substantial economic impact on the combo system operating costs and maintenance costs.

Combination systems designed, rated, and installed as a packaged system with built-in controls are a key advancement in combination system for wide-scale implementation for residential applications.

7. FUTURE WORK

Future work and extensions of this study will be necessary in order to analyze the performance of a tankless water heater combo system further. The transient stage should be characterized from a first law and second law standpoint to aid in the analysis of the combo system. As well, simultaneous operation was studied based on an introduction of a DHW event during steady-state or cyclical stage operation. Introduction of a DHW during the transient stage of combo operation may induce undesirable temperature fluctuations or increased periods of transient perturbation.

Results and observations from this study suggest that control strategies for combo systems are necessary to reduce the impact of undesirable behaviour such as short-cycling or flow-cycling. Possible control strategies to examine are modulation of the air blower speed in the AHU and modulating burners of the TWH that would prevent short-cycling or flow-cycling. Segregation of the AHU heat exchanger fluid loop from the TWH system may allow for increased performance by allowing for heat transfer fluids to be used instead of water as well as allow for better control of the various mass flow rates of the fluids within the system. As well, the development of control strategies that optimize system performance will be necessary to improve combo system operation. A control system that incorporates variable feedback from the AHU to the TWH to allow for optimized operation would greatly influence combo system performance. Additionally, the impact of using renewable energy systems such as a solar thermal system for preheating the supply water could be studied.

A techno-economic analysis should be performed to quantify potential costs savings and greenhouse gas (GHG) emission abatements for consumers adopting this technology for residential domestic hot water and space heating use. This techno-economic analysis may be useful for helping to increase the market penetration rate of the technology by quantifying potential cost savings to consumers.

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APPENDICES

9. APPENDIX A

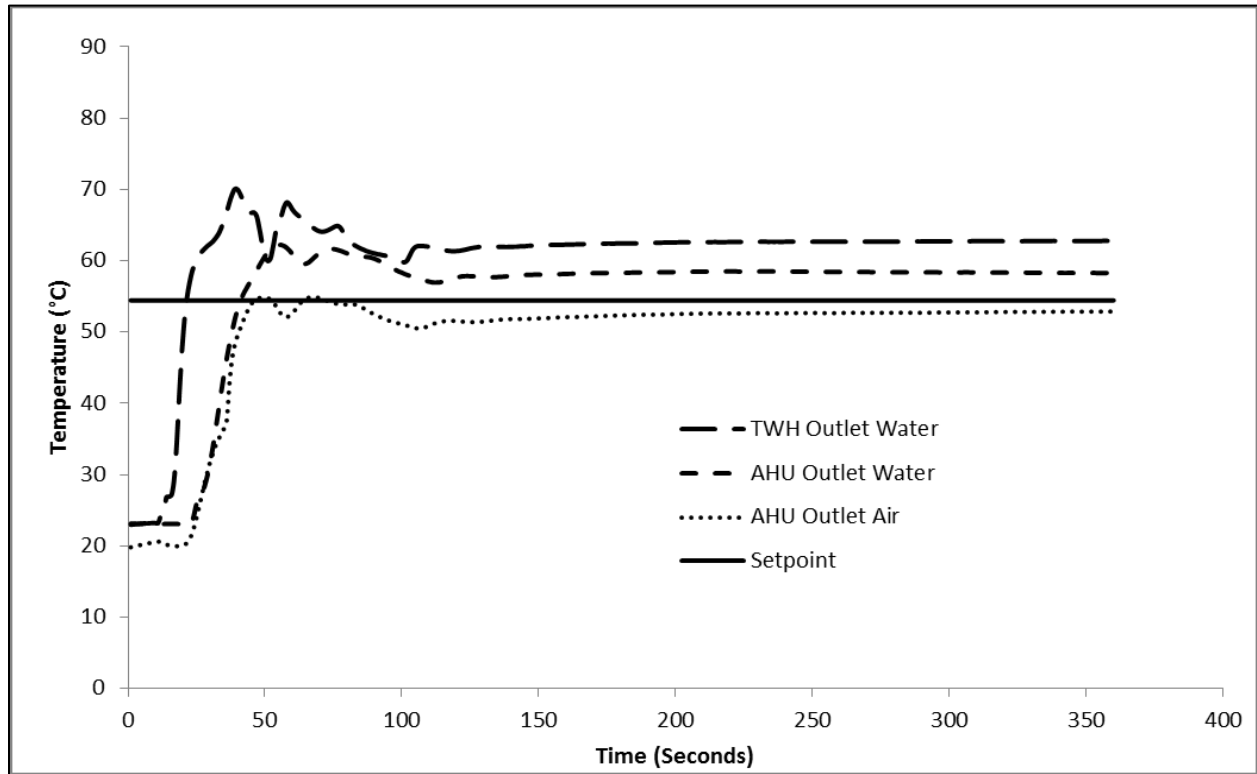


Figure A-1: Temperature trace for the combo heating system operating in space heating mode with a tankless heater water outlet temperature setting of 54°C.

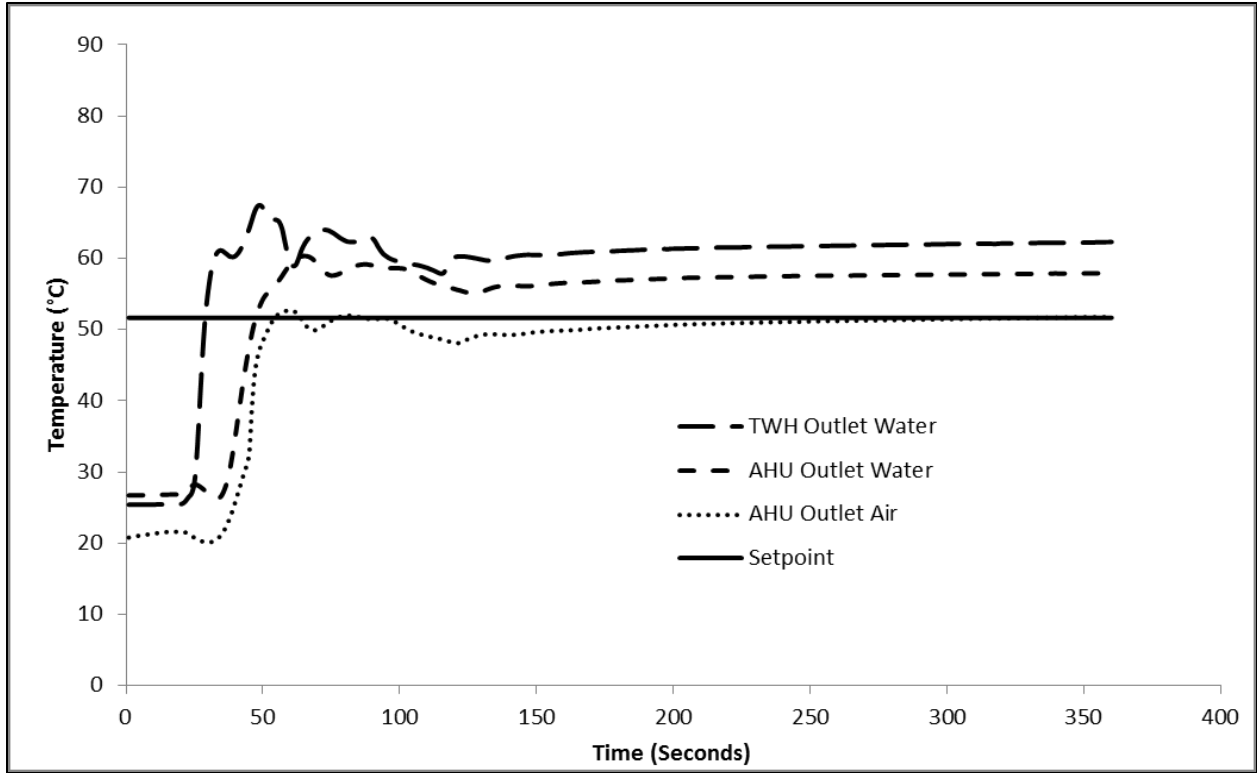


Figure A-2: Temperature trace for the combo heating system operating in space heating mode with a tankless heater water outlet temperature setting of 52°C.

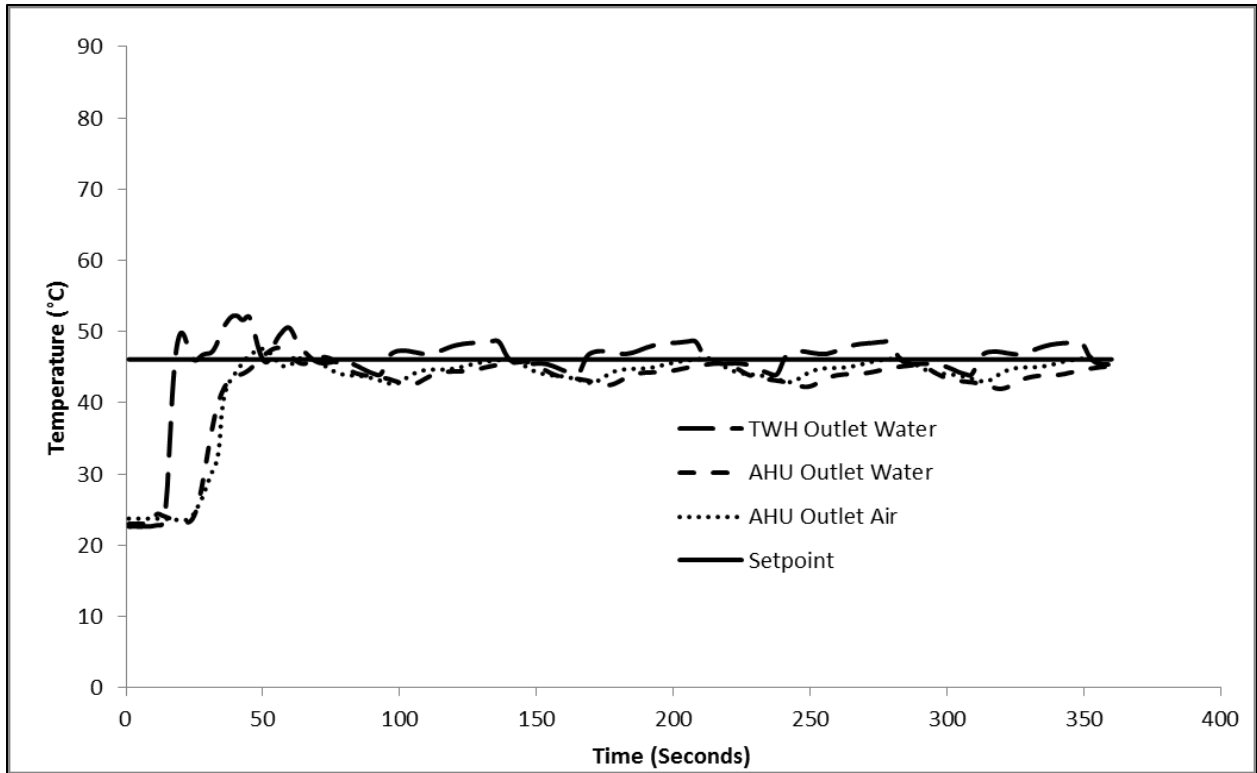


Figure A-3: Temperature trace for the combo heating system operating in space heating mode with a tankless heater water outlet temperature setting of 46°C.

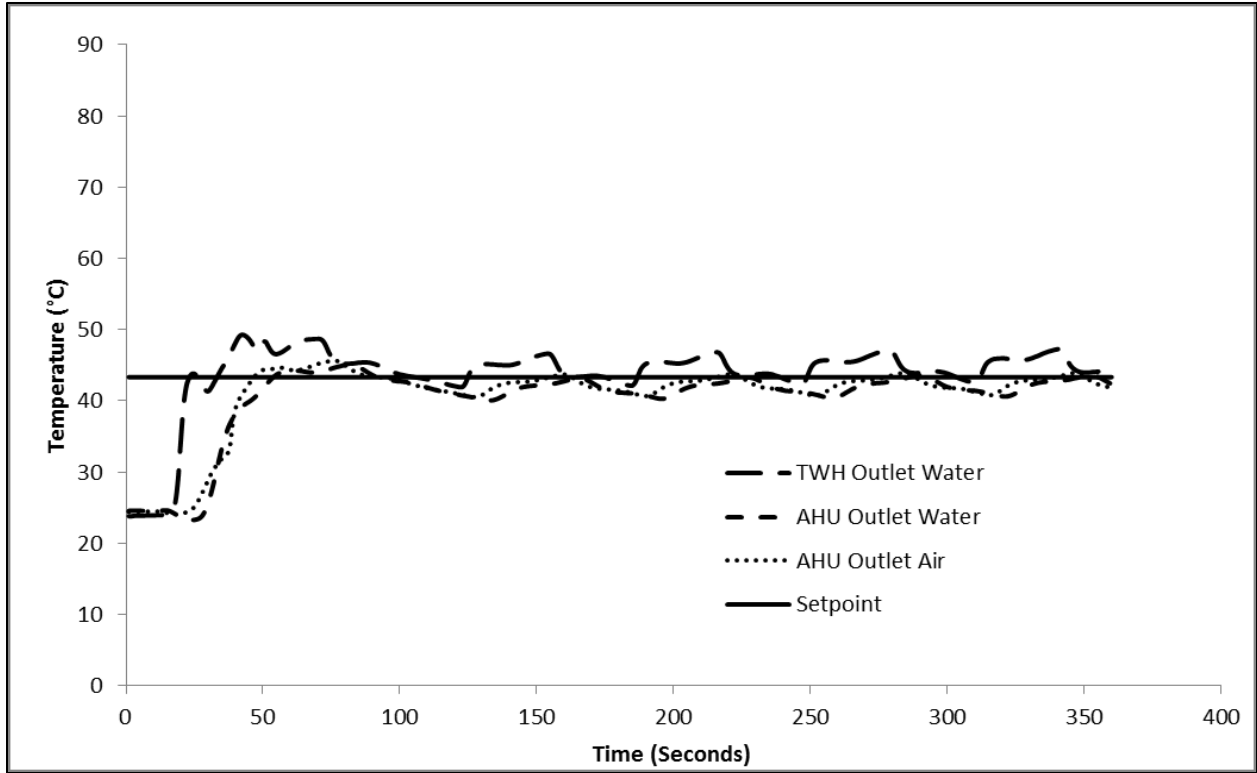


Figure A-4: Temperature trace for the combo heating system operating in space heating mode with a tankless heater water outlet temperature setting of 43°C.

10. APPENDIX B

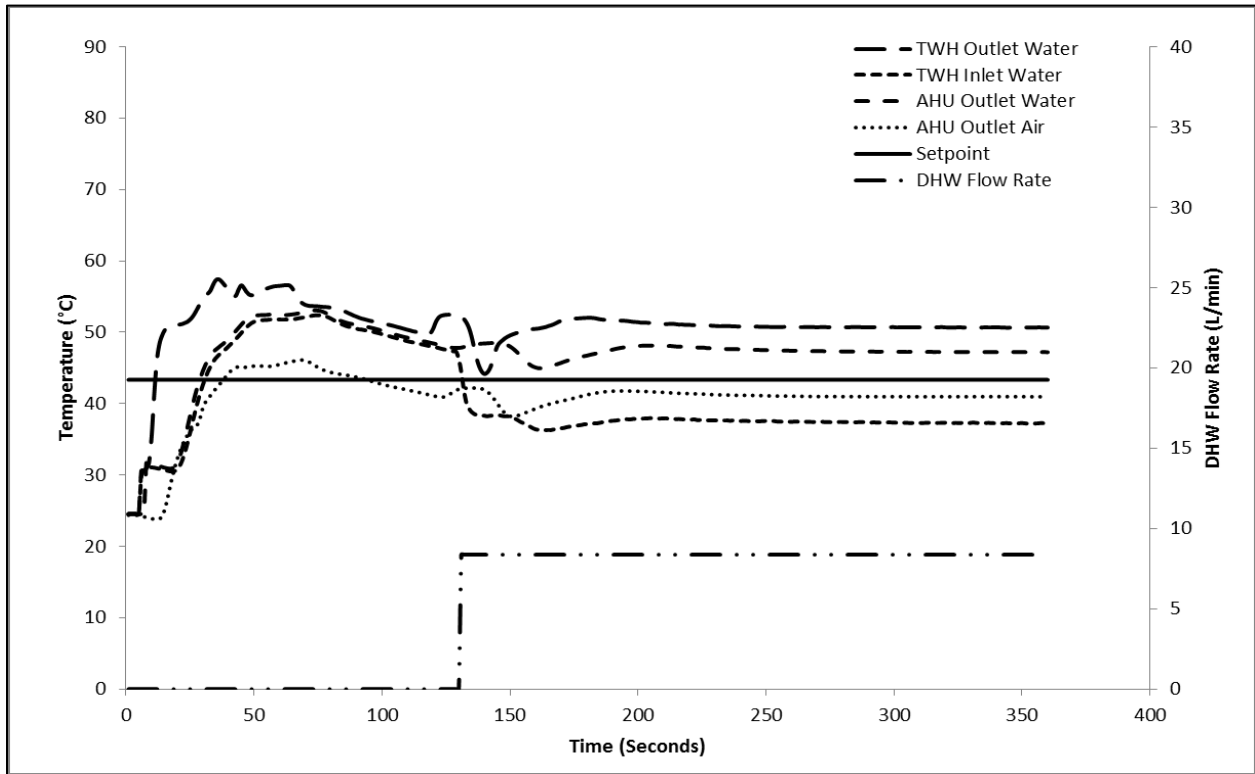


Figure B-1: Temperature trace for a combo heating system operating for space and water heating with a tankless heater water outlet temperature setting of 43°C and requested 7.6 L/min DHW draw.

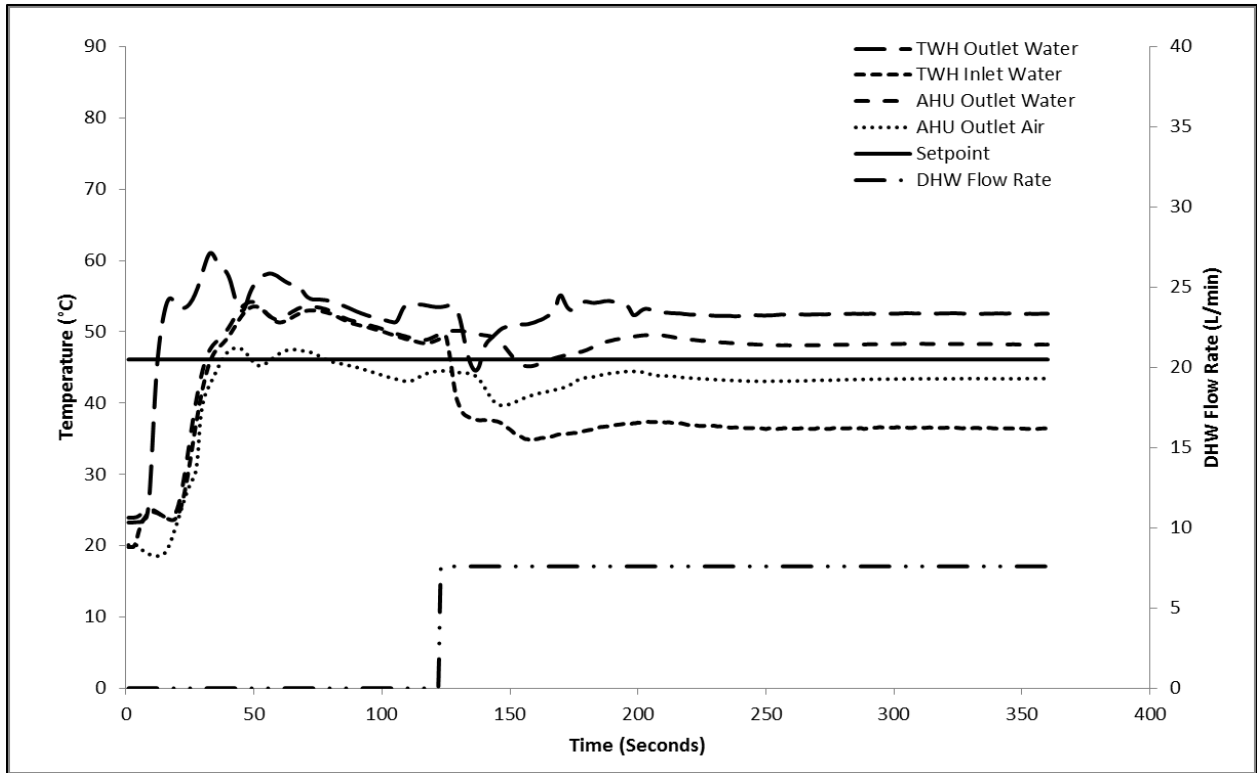


Figure B-2: Temperature trace for a combo heating system operating for space and water heating with a tankless heater water outlet temperature setting of 46°C and requested 7.6 L/min DHW draw.

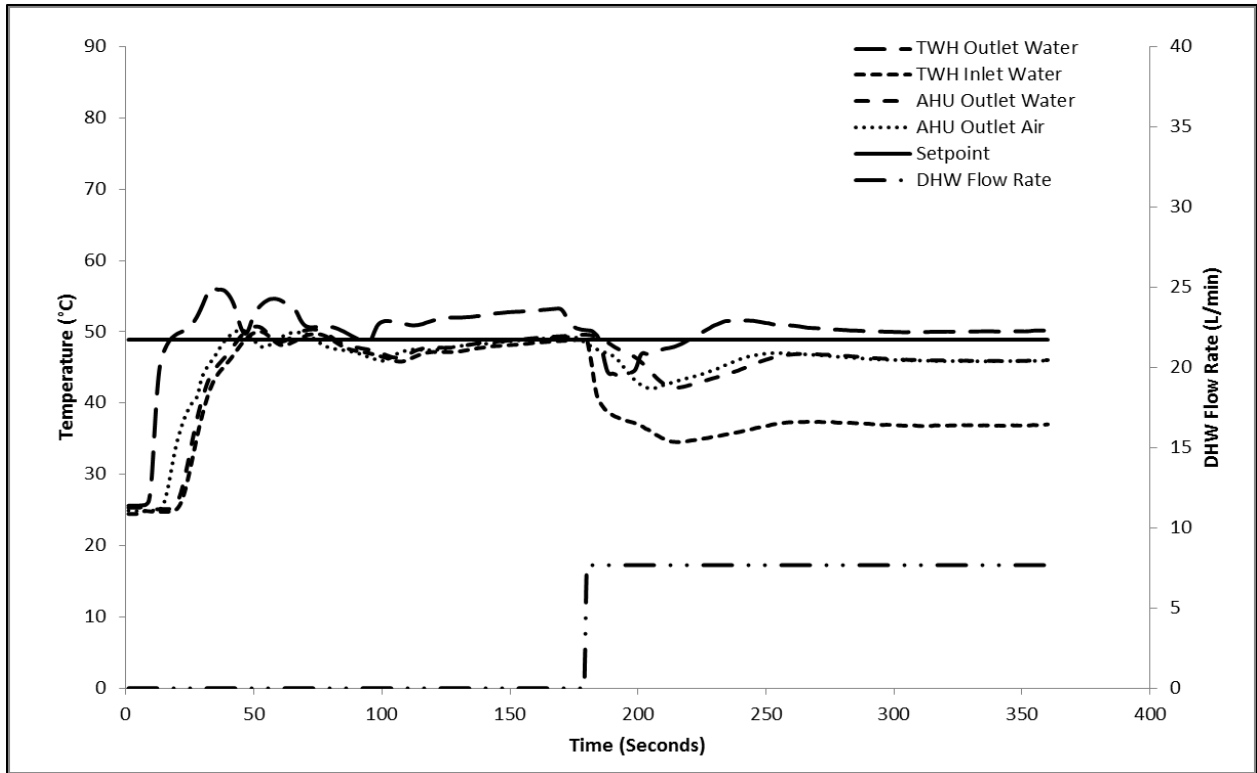


Figure B-3: Temperature trace for a combo heating system operating for space and water heating with a tankless heater water outlet temperature setting of 49°C and requested 7.6 L/min DHW draw.

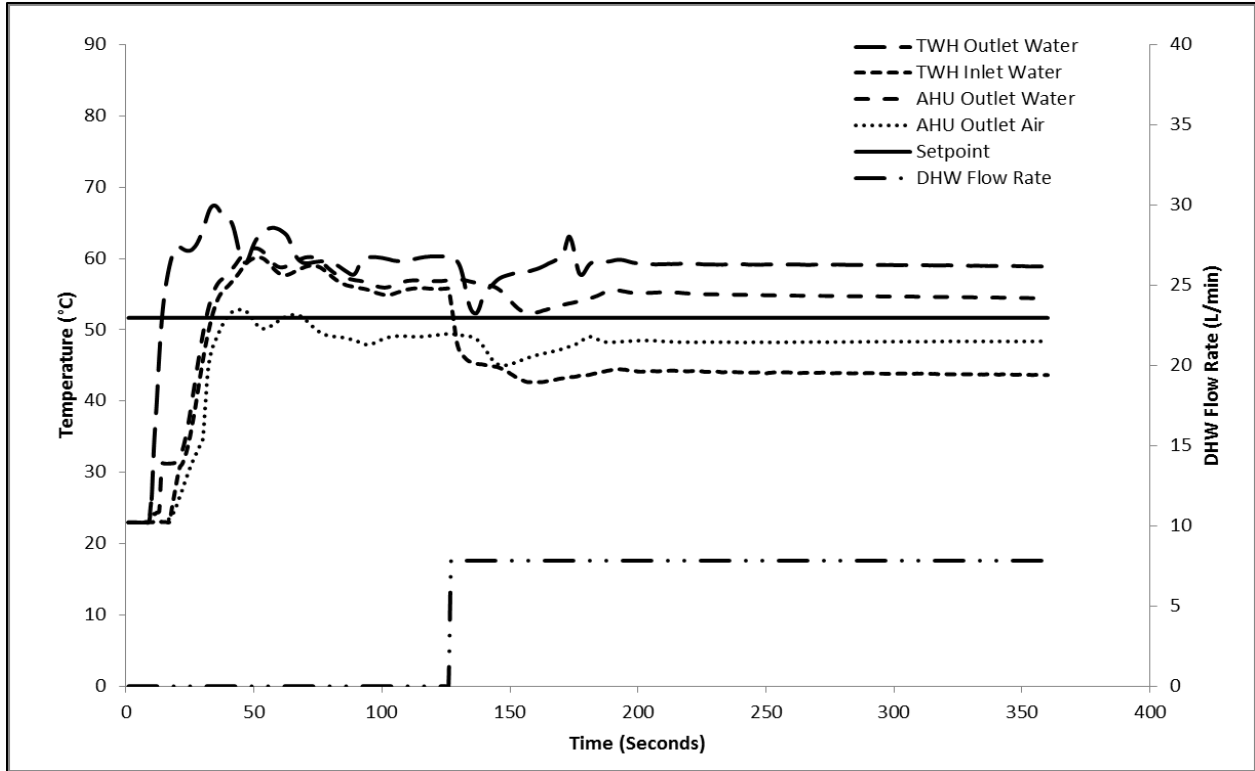


Figure B-4: Temperature trace for a combo heating system operating for space and water heating with a tankless heater water outlet temperature setting of 52°C and requested 7.6 L/min DHW draw.

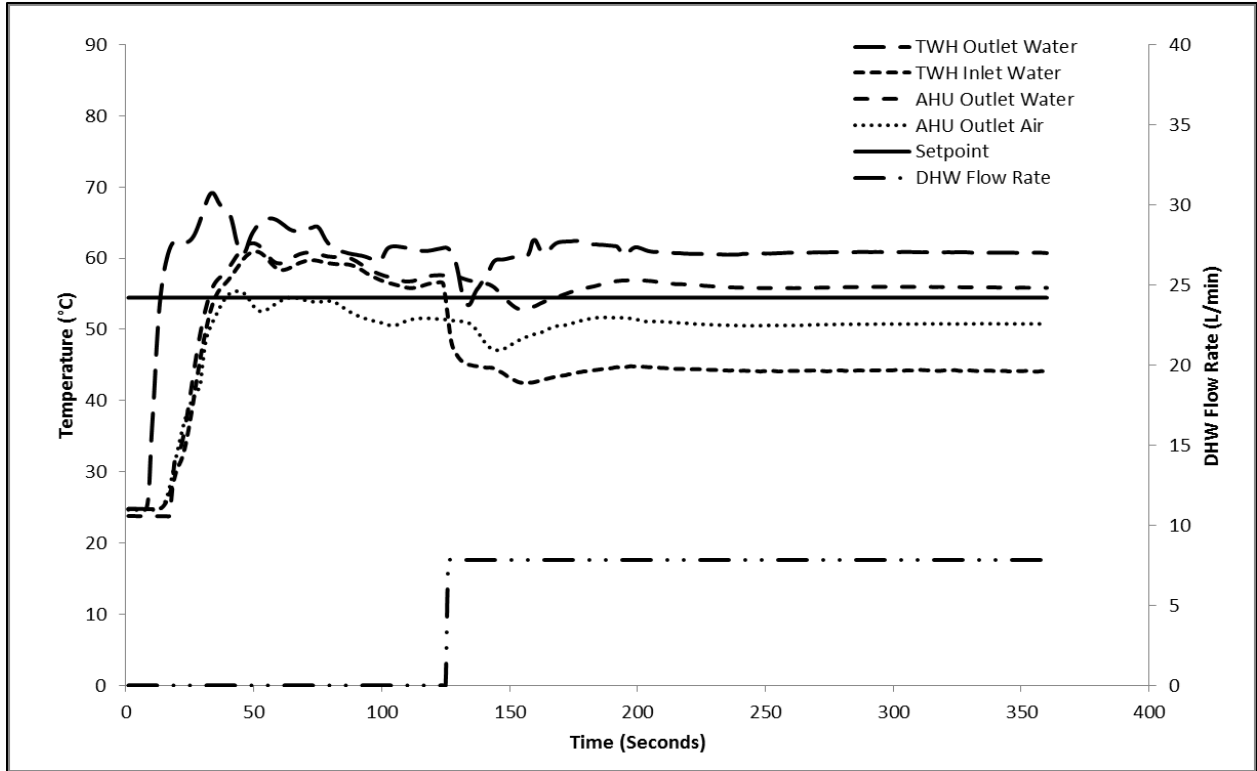


Figure B-5: Temperature trace for a combo heating system operating for space and water heating with a tankless heater water outlet temperature setting of 56°C and requested 7.6 L/min DHW draw.

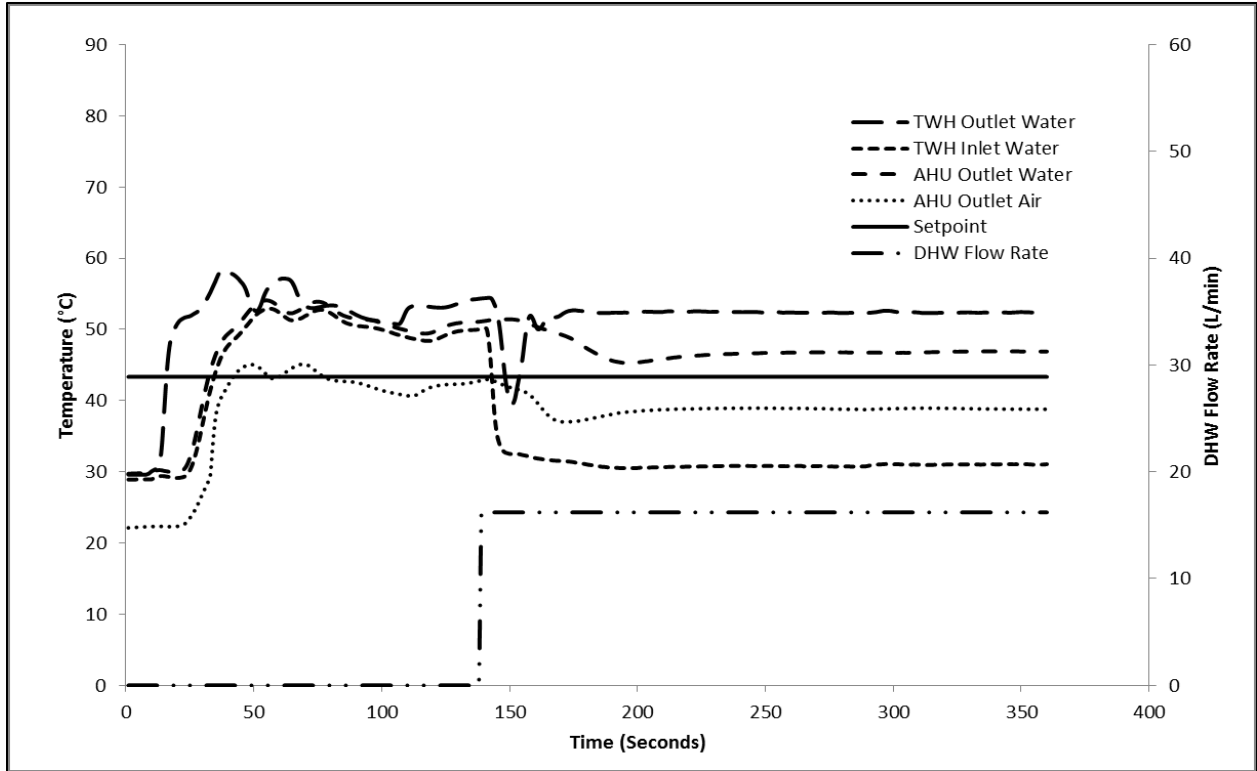


Figure B-6: Temperature trace for a combo heating system operating for space and water heating with a tankless heater water outlet temperature setting of 43°C and requested 15.1 L/min DHW draw.

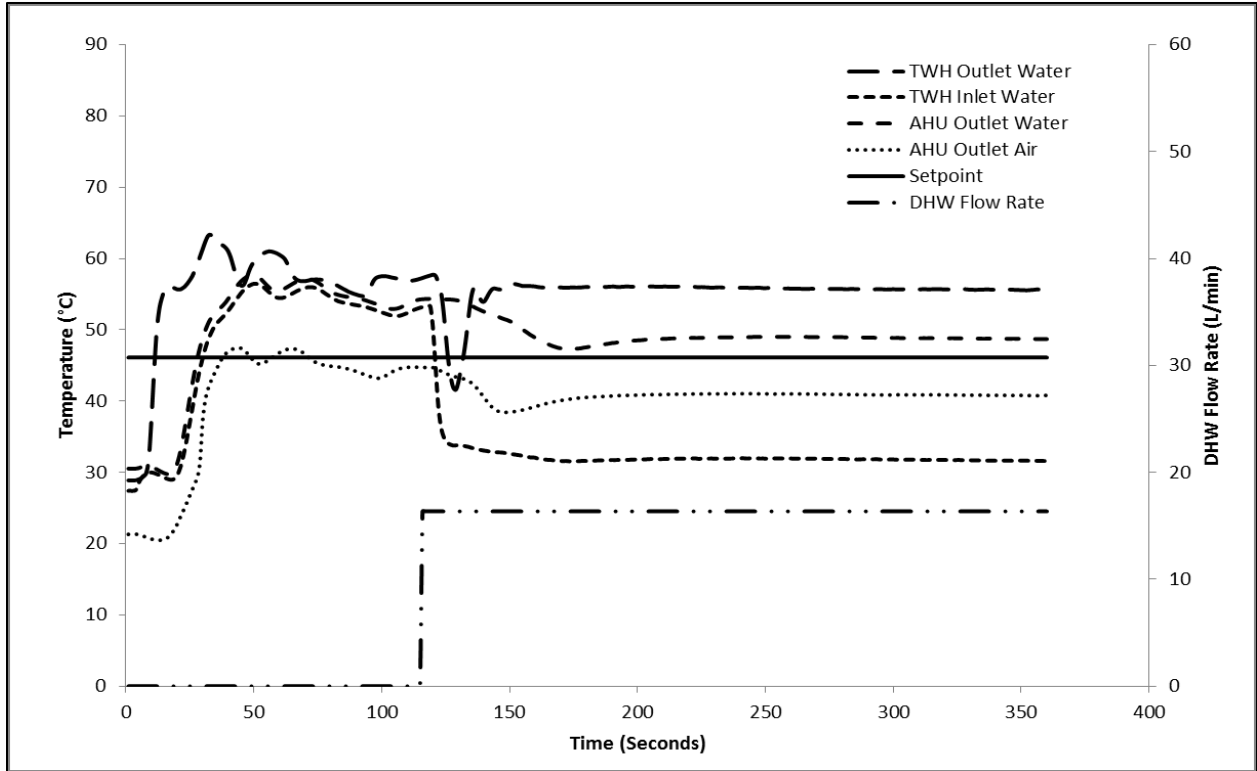


Figure B-7: Temperature trace for a combo heating system operating for space and water heating with a tankless heater water outlet temperature setting of 46°C and requested 15.1 L/min DHW draw.

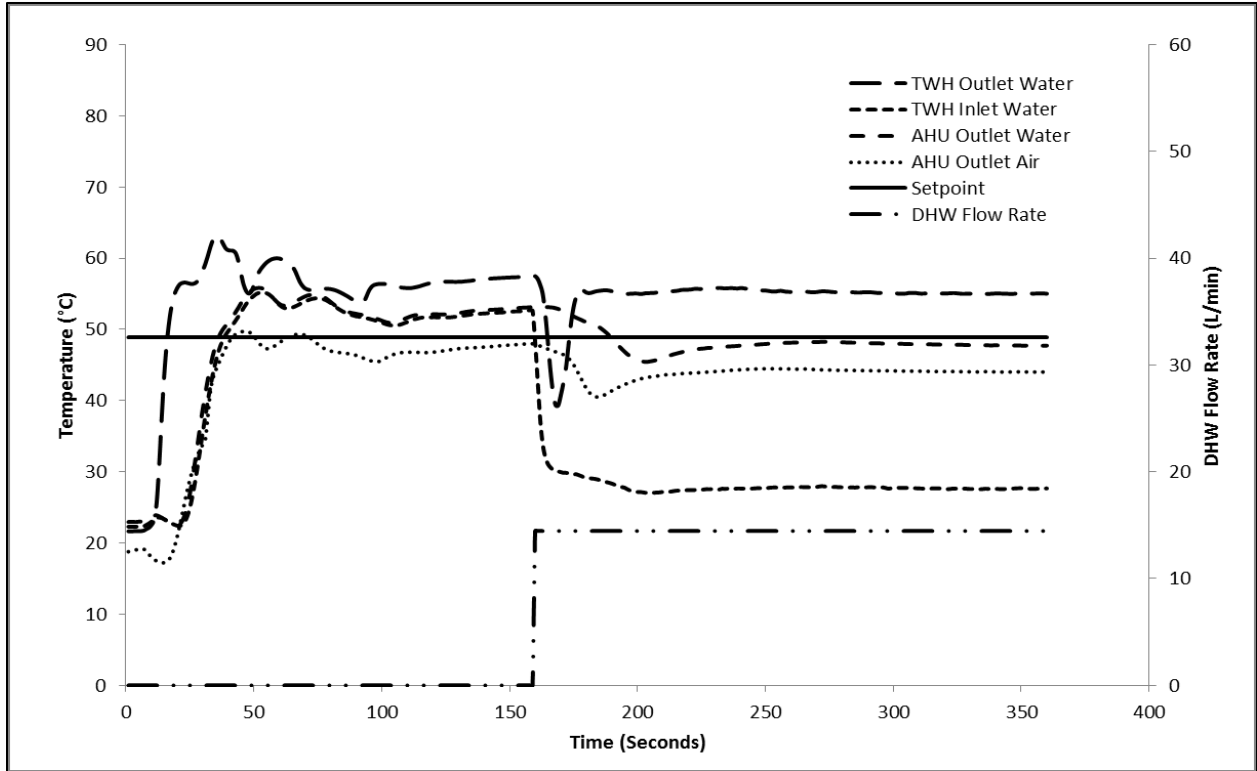


Figure B-8: Temperature trace for a combo heating system operating for space and water heating with a tankless heater water outlet temperature setting of 49°C and requested 15.1 L/min DHW draw.

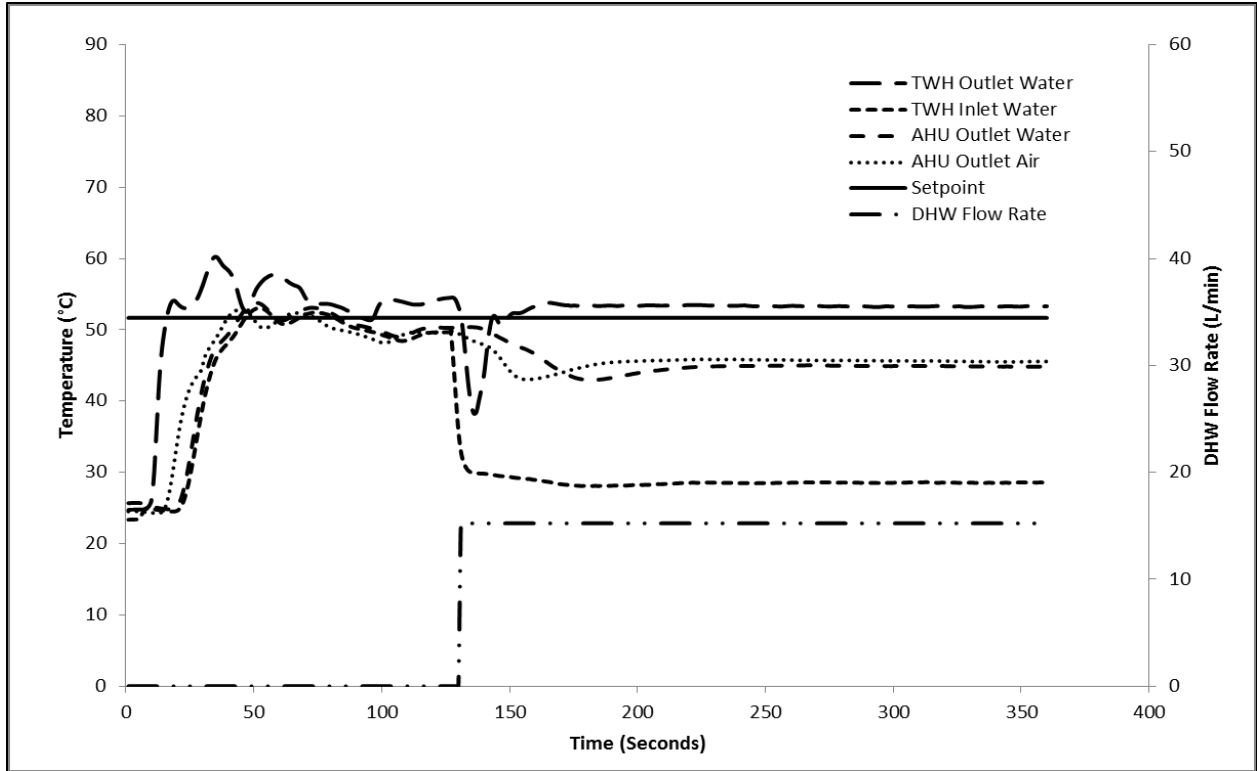


Figure B-9: Temperature trace for a combo heating system operating for space and water heating with a tankless heater water outlet temperature setting of 52°C and requested 15.1 L/min DHW draw.

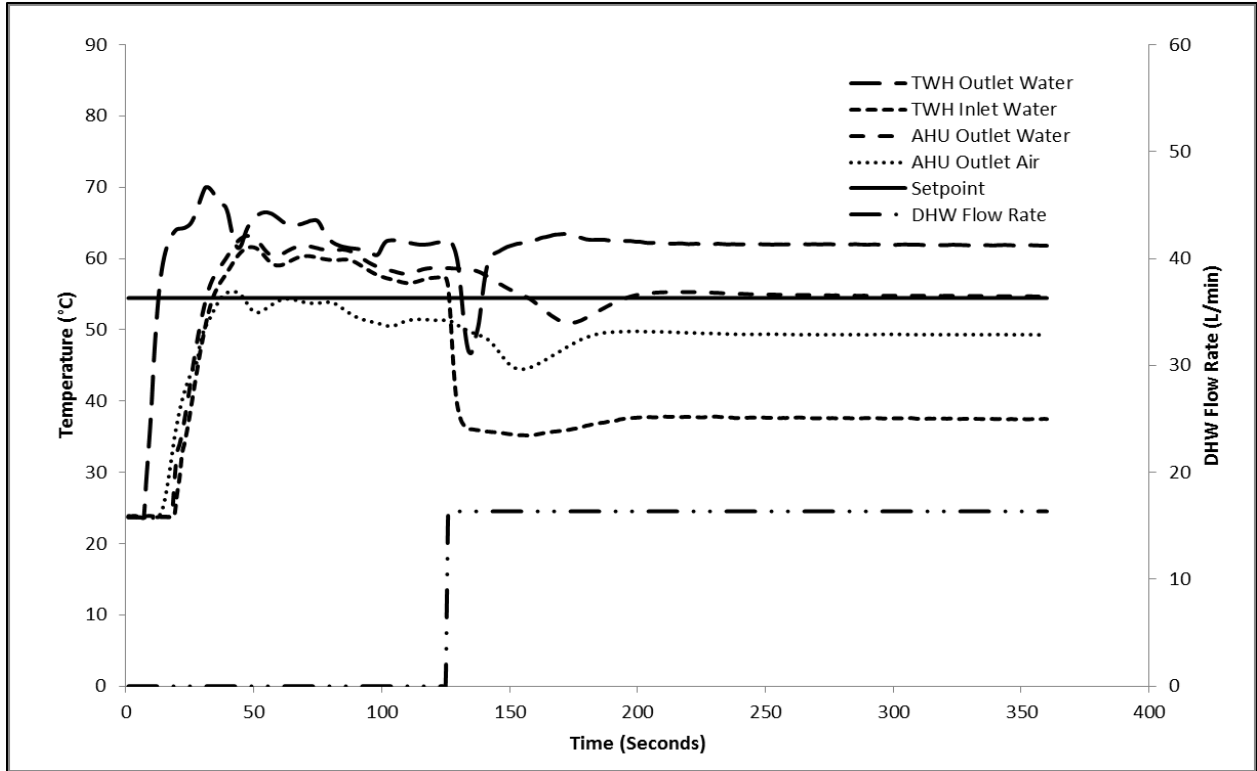


Figure B-10: Temperature trace for a combo heating system operating for space and water heating with a tankless heater water outlet temperature setting of 54°C and requested 15.1 L/min DHW draw.