Received 28 November 2017; revised 21 February 2018; accepted 23 February 2018.
Date of publication 15 March 2018; date of current version 22 June 2018.

Digital Object Identifier 10.1109/JPETS.2018.2810783

A Dual-Heat-Pump Residential Heating System for Shaping Electric Utility Load

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This work was supported by Portland General Electric.

ABSTRACT We present a residential heating system designed specifically to provide electric utilities with a means for achieving intra-hour load shifting. The system couples two heat pumps with a large water tank, which serves as a thermal storage unit. One heat pump uses the water tank as a source and serves residential heating load. The other uses the outdoor air as a source and heats the water tank during favorable weather conditions, thereby improving the efficiency of the first. By aggregating large numbers of such systems distributed throughout its balancing area, a utility may use this resource as a means for achieving load shaping, the intra-hour shifting of generation resources from periods of peak demand to periods of low demand. This paper focuses on the potential for using such an aggregated resource as a technological solution for realizing winter peak demand reduction and overall lower energy consumption, thereby displacing the need for capital investment in peak generation plants. The results of this paper are promising, demonstrating potential savings for both the utility and residential customers.

INDEX TERMS Distributed resources, load shaping, aggregated resources, thermal storage, demand-side management.

I. INTRODUCTION

Increases in peak power demand and the greater penetration of non-dispatchable generating plants on the electric power system present growing challenges to electric power providers. As the gap between peak and average electricity demand widens, the average utilization levels for generating plants decreases, increasing the cost associated with maintaining the capacity required to serve peak demand periods. The addition of more renewable energy plants such as wind power onto the grid introduces an increasing amount of intermittent generation. This creates a need for the ability to shift demand both to decrease peak periods and to better utilize non-dispatchable generating resources.

This project proposes the use of one or more water tanks as thermal storage to provide load-shifting in residential space conditioning applications. By using two heat pumps (HP), one to heat the water tank and the other to heat the home, the electricity provider can shift energy consumption to times of lower wholesale cost and reduce consumption during extreme conditions. Customer savings are achieved through better performance of the HP conditioning the home and by the purchase of therms stored in the thermal mass at a rate below what they would pay for retail electricity.

Upfront capital costs for this system are greater than that for a single-unit HP system. We consider four revenue streams, which when combined justify the higher upfront costs: 1) reduced investment in natural gas peaking plants, of 2.5 kW per unit, for the utility; 2) storage value at an annual average of 12 kWh/day for ancillary services such as load following, frequency regulation, and energy arbitrage, for the utility; 3) customer energy savings through improved coefficients of performance; and, 4) customer energy efficiency savings, around 7%, related to displaced transmission and distribution that would otherwise be billed to the customer. The customer would be responsible for the upfront costs for the customer side HP, on par with a standard single HP system. The utility would invest in the larger HP, storage tank and balance of plant. Our detailed analysis is beyond the scope of this study, but we estimate a benefit to cost ratio of 1.28. This paper focuses on the operational savings of the system, specifically peak demand and energy consumption reduction.

II. RELATED RESEARCH

In a high-renewable energy future, there will be greater demand for compensation services that can help mitigate the stochastic production of energy from wind and solar power plants. Such services are generalized as either *shaping* or *firming*, referring to *intra-hour* and *inter-hour* services, respectively. The services that a storage unit can provide depend on both its power capacity and its energy capacity, which dictate the length of time that it can supply its full power rating. Short duration storage units, on the order of 3 to 20 minutes, can provide *firming* services, compensating for wind resource fluxuations and intermittent reduction in photovoltaic production due to passing cloud-cover. However, to provide intra-hour *shaping* services, a full-power duration of at least two hours is generally required [1]. Within the context of a water tank coupled with a heat pump, the power capacity is dictated by the size of the heat pump while the energy capacity depends on the size and state of the water tank.

Research on the use of distributed storage as a mechanism for providing *firming* and *shaping* services has generated a lot of interest, particularly as a mechanism for better integration of renewable energy resources, load shifting, and mitigating minimum generation conditions [2]–[4]. Since at least the 1960's, pilot programs have used electric water heaters as distributed storage systems to provide load control services [5]. Pilot programs have been designed to provide demand response services as well as load shaping services, where water tanks are preheated during low demand times and then the tank serves as a thermal energy storage device to ride through higher demand periods [6], [7].

The use of hot water tanks with electric water heaters as thermal storage units has been widely explored as a mechanism for providing demand response services [8]–[10]. Research is often motivated by the desire to mitigate the stochastic nature of renewable generation [11]–[13]. Because of the inertia inherent to a sufficiently large thermal mass, heating applications in general, and heat pump systems in particular, have been looked to as a mechanism for providing short term control of power consumption [14]–[16].

The dual-heat pump scheme presented in this manuscript is unique [17], but the concept rests on a foundation of prior art such as [18]–[20], which share a similar aim of using heat pumps to manage a thermal energy storage reservoir for load control. Other related art focus on ice-reservoir cooling systems [21], and hybrid cooling & heating thermal storage systems [22], [23]. Aggregation of thermal storage systems has been described as a means for providing on-site power generation using thermal energy storage that can be controlled by a utility [24].

The concept of using multiple heat pumps connected to a shared reservoir is a central feature of some district heating schemes, whereby a single large reservoir is shared by multiple heat pumps [25], [26]. And given the large size of such loads, the use of thermal energy storage in district heating systems has been investigated as a means for reducing imbalance between thermal load and supply [27], and for achieving utility energy balancing [28].

III. PROJECT DESCRIPTION

The purpose of the system under investigation is to provide greater load flexibility in residential heating and cooling applications using two heat pumps coupled through a water tank, which operates as a thermal storage unit. An air-to-water heat pump is used to heat and cool the water tank. This utility-side heat pump (UHP) would be owned and operated by the electric utility and used to charge the thermal storage unit during off-peak, over-generation or low-cost times. Additionally, the UHP can charge the thermal storage unit during times of relatively warm outdoor temperature in anticipation of colder evening hours when heat pump efficiency would drop. This allows the system overall to make efficiency gains.

A second, smaller water-to-air heat pump heats and cools the home within a range of temperatures set by the resident and would be owned by the customer. The customer-side heat pump (CHP) sees significant performance gains by using the conditioned water as its source. This means that during times of high system demand when outdoor temperatures are low, which would degrade the performance of an air-to-air heat pump, the CHP maintains a high efficiency, reducing the electric power required to serve the same heating load. Thermal energy extracted from the tank by the CHP would be billed to the customer by the utility by the therm at a flat rate based on the average wholesale cost of electricity, the average coefficient of performance (COP) of the air-to-water heat pump, and standard transmission and distribution fees. Figure 1 illustrates the two-heat-pump system.

The operation of the heat pumps is projected over a sliding twenty-hour period using forecasted weather and wholesale pricing information. The anticipated behavior of the CHP is modeled based on the thermal characteristics of the home, weather conditions and the heat transfer characteristics of the heat pump itself. Operational set points for the UHP are determined by an algorithm within the controller to maintain the tank temperature within the desired temperature range and operate at specific times during the day to meet system operational objective. The CHP then operates autonomously, using the thermal storage unit as a source.

The system can be operated for different objectives, including minimizing utility cost, minimizing customer cost, minimizing the combined cost to both entities, minimizing energy consumption, or reducing demand during peak times. Two of these objectives were investigated for this study. The primary objective was to determine the savings that could be achieved in the daily operation of the system for both the utility and the customer. The secondary objective was to determine the potential of the system to provide peak shaving during extreme weather conditions. For this study, we focus on operational objectives related to heating.

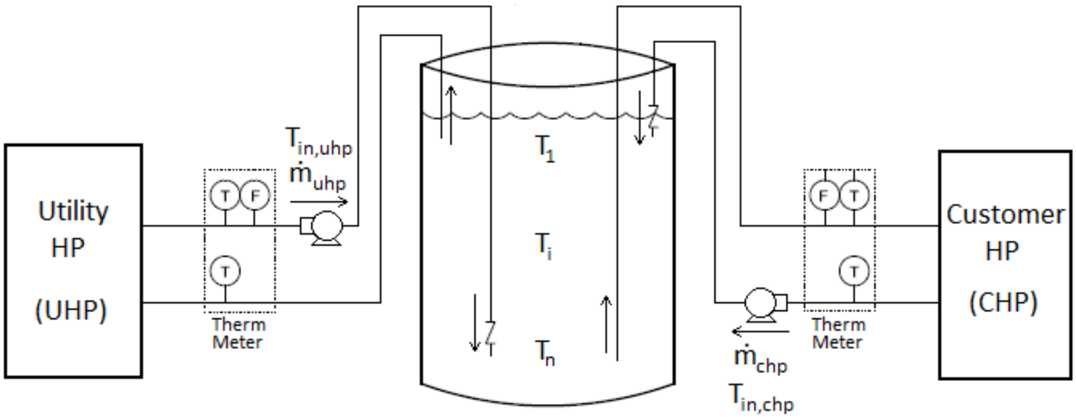

FIGURE 1. **Two-heat-pump system notional diagram.**

IV. THERMODYNAMIC MODELS

The proposed system model is comprised of an insulated, rectangular volume of air representing the home, an insulated water tank as a thermal storage unit, and two heat pumps: an air-to-water heat pump (the UHP) and a water-to-air heat pump (the CHP), each coupled to the thermal storage unit. Equations 1 and 2 describe the modeled thermal behavior of the home and a thermal storage unit, respectively.

$$\dot{T_H} = \frac{\dot{Q}_{chp} + U_h A_h (T_o - T_H)}{m_a c_a} \tag{1}$$

In Equation 1, T_H is the indoor temperature, T_o is the outdoor temperature. A_h is the surface area of the home, and U_h is the heat transfer coefficient of the home insulation. The mass and specific heat of the air in the home are m_a and c_a. $\dot{Q}_{chp}$ is the heat transfer rate of the CHP, representing the heat transferred to or from the home by the CHP.

In Equations 2a through 2e, T_1, T_i and T_n represent the water temperature at different layers in the tank where there are n layers of stratification. The first layer is at the top, the nth layer is at the bottom of the tank, and i ranges from 2 through $n-1$. $T_{in,chp}$ and $T_{in,uhp}$ are the temperatures of the water entering at the top and bottom layers respectively. These are dependent on the heat transfer of the two heat pumps, the mass flow rate through the heat pumps and the water temperature entering the heat pumps. In heating mode, where the CHP is heating the home, $T_{in,uhp}$ represents the water temperature leaving the UHP, and $T_{in,chp}$ is the water temperature leaving the CHP. $\dot{Q}_{uhp}$ is the heat transfer rate for the UHP. In cooling mode, this is reversed; when thermal energy is extracted from the tank by the UHP, the corresponding $\dot{Q}_{HP}$ expressions become negative.

A_C is the cross-sectional area of the tank, and therefore the area between stratified layers. A_S is the surface area of the side of the tank for each stratified layer, each with thickness Δx_i. U_t is the heat transfer coefficient of the tank insulation. U_l and U_b are the heat transfer coefficients of the tank lid and bottom, respectively. Because the thickness of the stratified layers are assumed to be the equal, A_S is $\frac{1}{n}$ of the total surface area of the tank side. The mass, specific heat and thermal conductivity of the water are given by m_w, c_w and k_w respectively.

The mass flow rates from the heat pumps are represented as $\dot{m}_{uhp}$ and $\dot{m}_{chp}$. In heating mode, $\dot{m}_{uhp}$ is injected at the bottom of the tank and removed at the top, while $\dot{m}_{chp}$ is injected at the top of the tank and removed from the bottom. In Equations 2a–2e, mass flow through the tank is represented as moving downward. In the case where mass flow is reversed, these equations are adjusted such that flow is represented between the appropriate layers. The inter-layer mass flow $\dot{m}$ is equal to $\dot{m}_{uhp} - \dot{m}_{chp}$; if $\dot{m}_{uhp}$ and $\dot{m}_{chp}$ happen to be equal, then there is no mass flow through the center of the tank and $\dot{m}$ is zero.

V. CASE STUDY

We evaluated the dual-heat-pump system based on implementation in the U.S. city of Portland, Oregon. Simulations were performed assuming a 2,000 square foot, single family home with a thermal admittance (UA) of 400 kBtu/hr°F (or 210 W/K) and 15% duct loss. These values were supplied by the local utility, Portland General Electric (PGE), as typical of single family homes in Portland, where the system might ultimately be implemented. The thermal mass was taken to be a 600 gallon (or 2.3 m^3) cylindrical tank of water with inlets and outlets at the top and bottom for each of the heat pumps. The UHP and thermal mass were assumed to be installed outdoors, with the water tank having an insulation of R-16. Thermal stratification was assumed to exist in the tank and was modeled in four layers.

The heat pumps were modeled using the coefficient of performance (COP) specifications of an air-to-water heat pump (the UHP) and a water-to-air heat pump (the CHP). Each heat pump therefore has its own set of COP curves, which depend on different operating conditions. The first COP curve set described here represents a surface that was derived from the manufacturer specifications for a 3.2 ton Daikin air-to-water heat pump and is used to model the performance of the UHP.

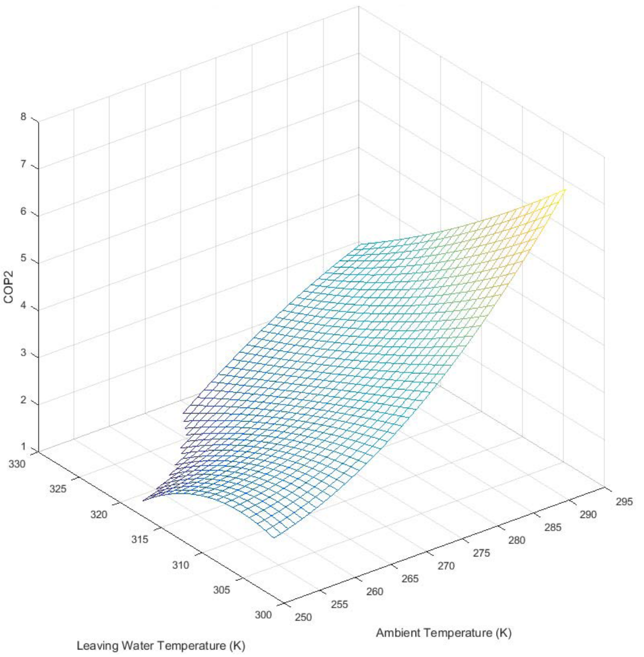

FIGURE 2. Utility-side heat pump (UHP) COP surface, heating mode.

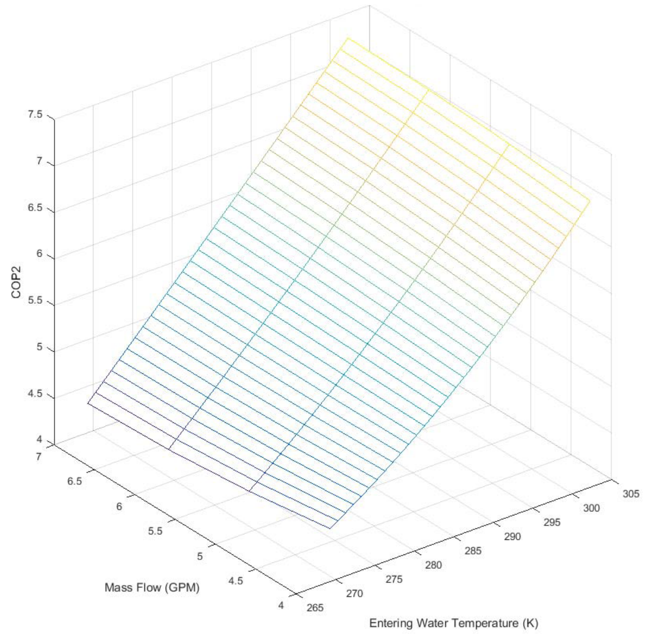

FIGURE 3. Customer-side heat pump (CHP) COP surface, heating mode.

The COP of this heat pump is a function of both the ambient air temperature and the water temperature. The surface was defined using a quadratic polynomial fitting function across the ambient air temperature and then across the water temperature. The COP surface is given in Figure 2.

The water-to-air CHP was modeled using the specifications of a 2 ton Trane water-to-air heat pump, and its COP is a function of the entering water temperature and the flow rate of the water. Again, the COP surface was derived by using a quadratic polynomial curve fitting function, this time across the water temperatures and the flow rate. The COP surface for the CHP is given in Figure 3. Water flow through both of the heat pumps was assumed to be 1.5 gpm.

A. WEATHER DATA

Ambient temperature data typical for Portland, Oregon were taken from the typical meteorological year version 3 (TMY3) data set, developed by the National Renewable Energy Laboratory (NREL).[1] When evaluating the annual cost savings based on the daily performance of the system, each season was set up using the hourly temperature data typical for each day. For the peak demand reduction study, actual weather data were used from the days that saw the highest annual peak demand in 2013-2014, which was December 9th, 2013 for the winter peak. Load values for this study were provided by PGE and corresponding weather data were taken from Weather Underground's historical database.[2]

B. PRICING INFORMATION

The wholesale cost of electricity was derived from the locational marginal price (LMP) of electricity at the Malin hub of the California Independent System Operator (CAISO). The average hourly data from 2012-2014 were used to run three years of daily performance simulations. The retail cost of

[1]National Solar Radiation Database, rredc.nrel.gov/solar/old_data/nsrdb/ (Online; accessed May-2016)

[2]Weather Underground, www.wunderground.com (Online; accessed May-2016)

$$\dot{T}_1 = (m_w c_w)^{-1}\left(\frac{k_w}{\Delta x_1} A_C(T_2 - T_1) + U_l A_C(T_o - T_1) + U_t A_S(T_o - T_1) + \dot{m} c_w(T_2 - T_1) + \dot{Q}_{chp}\right) \tag{2a}$$

$$\dot{T}_i = (m_w c_w)^{-1}\left(\frac{k_w}{\Delta x_i} A_C(T_{i+1} + T_{i-1} - 2T_i) + U_t A_S(T_o - T_i) + \dot{m} c_w(T_{i-1} - T_i) + \dot{m} c_w(T_{i+1} - T_i)\right) \tag{2b}$$

$$\dot{T}_n = (m_w c_w)^{-1}\left(\frac{k_w}{\Delta x_n} A_C(T_{n-1} - T_n) + U_b A_C(T_o - T_n) + U_t A_S(T_o - T_n) + \dot{m} c_w(T_{n-1} - T_n) + \dot{Q}_{uhp}\right) \tag{2c}$$

$$\dot{Q}_{uhp} = \dot{m}_{uhp} c_w(T_{in,uhp} - T_1) \tag{2d}$$

$$\dot{Q}_{chp} = \dot{m}_{chp} c_w(T_{in,chp} - T_n) \tag{2e}$$

electricity to the customer is defined by PGE's 2015 residential service flat rate schedule. The price of the thermal energy extracted by the CHP from the thermal storage unit is based on the average wholesale cost of electricity at the Mid-Columbia hub in 2014, plus PGE's retail transmission and distribution rates, divided by the average COP of an air-to-water heat pump in the Portland area. This was calculated to be $0.022 per kWh.

C. COST ASSESSMENT

The cost of electricity for the utility is determined by the wholesale cost of all the electrical energy used by the heating system. For the two heat pump system, this is calculated by summing the power consumed by both heat pumps multiplied by the time interval and by the wholesale cost of electricity for each time interval over the 24-hour period, as described in Equations 3, 4 and 5. For this work, this is done in intervals of 5 minutes.

$$J_{util} = \sum_{n=1}^{288} (P_{uhp,n} + P_{chp,n})\Delta t C_{wh,n} \tag{3}$$

$$P_{uhp} = \frac{\dot{Q}_{uhp}}{COP_{uhp}(T_{w1}, T_{env})} \tag{4}$$

$$P_{chp} = \frac{\dot{Q}_{chp}}{COP_{chp}(T_{w2}, \dot{m}_{chp})} \tag{5}$$

J_{util} is the wholesale cost of electricity required to operate the system in dollars, n is each time interval, P_{uhp} is the power consumed by the UHP in kilowatts, P_{chp} is the power consumed by the CHP in kilowatts, Δt is the time in each interval in seconds, C_{wh} is the wholesale cost of electricity during each time interval in dollars, Q_{uhp} is the rate of heat transfer in kilowatts from the UHP into the water tank, and Q_{chp} is the rate of heat transfer from the CHP into the home. COP_{uhp} and COP_{chp} are the COPs of the UHP and CHP respectively.

A five-minute step time was used in this work because that is the shortest interval over which the heat pumps can be operated given their cycling limitations. Ten-minute intervals were tested using data from the month of January 2012 and using an iterative method, as described below in Section V-E. Results were compared to system performance when using five-minute intervals. Using the ten-minute intervals yielded a 9.7% increase in cost over the month, confirming that higher granularity of control made available by using five-minute intervals improves system performance. However, decreasing the number of heat pump cycles would increase the life-expectancy of the equipment and so there is an implicit trade-off between the desired level of control and operating the system in such a way that minimizes operations and maintenance costs.

In the two heat pump system under investigation, customer cost is based on the cost of the energy consumed by the CHP. This includes both the retail cost of the electrical energy required to run the CHP as well as the thermal energy extracted from the thermal storage unit and purchased as therms by the customer. Equation 6 describes how this is calculated.

$$J_{cust} = \sum_{n=1}^{288} (P_{chp,n})\Delta t C_{ret} + \Delta T \dot{m}_{chp} c_w \Delta t C_{th} \tag{6}$$

Where J_{cust} is the cost of energy to the customer, C_{ret} is the per-unit retail cost of electricity, ΔT is temperature differential created across the inlet and outlet of the water loop passing through the CHP as it operates in Kelvin, $\dot{m_{chp}}$ is the mass flow rate of water through the CHP in kilograms per second, c_w is the specific heat of water in joules per kilogram-Kelvin, and C_{th} is the per-unit cost of thermal energy.

D. CUSTOMER HEAT PUMP OPERATION

The heating needs of the customer are served by the CHP using the thermal storage unit as a source. The temperature set point for the customer residence comes from ASHRAE Standard 55 - Thermal Environmental Conditions for Human Occupancy, which recommends 71 °F for the heating season with a swing tolerance of ±2 °F. Additionally, a 2 °F set back is assumed from the hours of 9 am to 4 pm and from 11 pm to 6 am. Heating in the home is simulated using the thermal home model described in Equation 1, where the operation of the CHP dictates $\dot{Q}_{chp}$ in each time interval. CHP operation is simulated in 5-minute increments. The pump operates if the temperature falls to 2 °F below the set point and reheats the home up to but no higher than 2 °F above the set point. In this way, an operational vector for the CHP is generated for the entire 24-hour period. Results are based on simulations run for 2012-2014. When the simulations are initialized for the first day in the first month, the home is assumed to be exactly at the temperature set point.

E. UTILITY HEAT PUMP OPERATION

An iterative method was developed to generate a vector of operational set points for the UHP that maintains system constraints while setting the periods of operation to times of lower cost. This method is used in this study. Operation of these heat pumps has significant potential for optimization, which we will address in future publications.

The routine assumes that the CHP will operate autonomously to heat the home within the prescribed comfort band agnostic of the wholesale cost of electricity. The UHP is then set to operate during the lowest cost period of time as long as the total amount of energy extracted by the CHP is not less than the amount injected by the UHP. The predicted temperatures for the water tank throughout the day are then found using these two preliminary operational vectors. The temperature of the water at the inlet to the utility heat pump is tested and the utility-side heat pump is triggered to operate based on a threshold temperature relative to the system's temperature constraints. This process is then iterated, as follows.

The temperature threshold is varied between -10 and 10 K in steps of 1 K, whereby the utility-side heat pump will be operated if the inlet water temperature is within this threshold

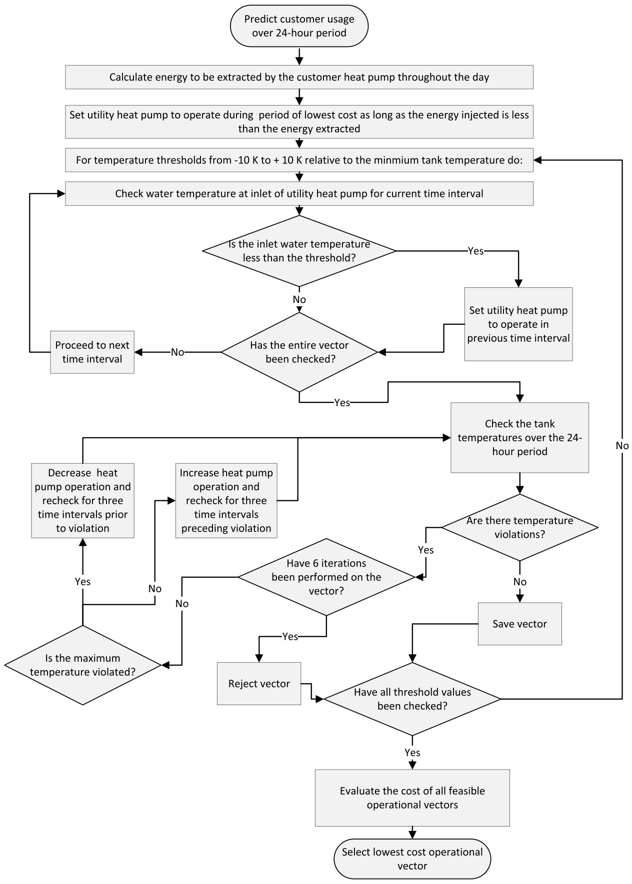

FIGURE 4. Flow diagram describing the iterative method to obtain the heat pump operational vector.

of the minimum allowable tank temperature. System constraints are tested, and the routine goes through six iterations of adjusting the operational vector to correct any violations. If there are still constraint violations, the vector is rejected. Otherwise, the operational vector is saved. Six iterations has proved sufficient to ensure sufficient adjustment. Once all 21 threshold temperatures have been tested, the cost of each of the resulting feasible solutions is found and then lowest cost solution is selected. Figure 4 illustrates this iterative method in a flow diagram.

The minimum tank temperature is set as 40 °F and the maximum as 100 °F, or 278 K and 311 K respectively, to accommodate the desired operational region of the heat pumps. When the simulations are initialized for the first day in the first month, the tank temperature is assumed to be 4 K above the minimum allowable tank temperature, or 282 K. This method is sufficient for converging on a solution within the five minute step interval between HP setpoint adjustments.

VI. BENCHMARK

Simulations were run to provide a baseline with which to compare the results of the system study, a single air-to-air heat pump. For the benchmark, the same weather and price data were used. Equations 7-8 give the calculations performed to determine the cost to utility and customer respectively to operate the system.

$$J_{utilbench} = \sum_{n=1}^{288} P_{B,n} \Delta t C_{wh,n} \tag{7}$$

$$J_{custbench} = \sum_{n=1}^{288} P_{B,n} \Delta t C_{ret} \tag{8}$$

$$P_b = \frac{\dot{Q}_b}{COP_b(T_{env})} \tag{9}$$

Where $J_{utilbench}$ is the cost to the utility of operating the benchmark system in dollars, P_{BHP} is the power consumed by the benchmark air-to-air heat pump (BHP) in kilowatts, $J_{custbench}$ is the cost to the customer for heating in dollars, $\dot{Q}_{BHP}$ is the rate of heat transfer from the BHP into the home and COP_{BHP} is the COP of the BHP, as described below.

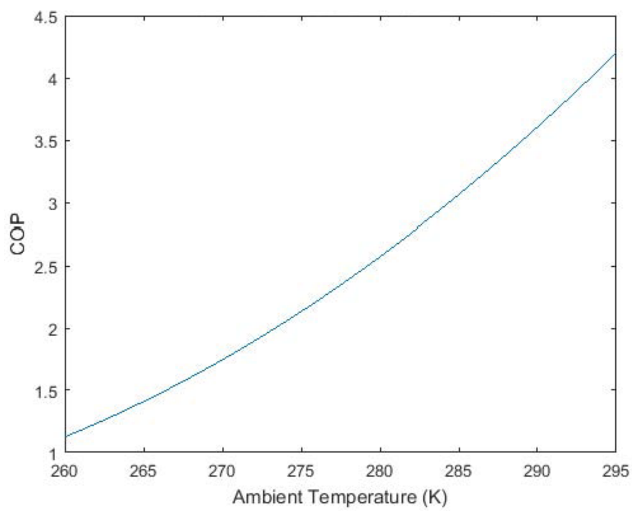

FIGURE 5. Benchmark air-to-air heat pump (BHP) COP curve, in heating mode.

The same thermal model for the home was used, and the operation of the BHP serves the same heating load as the CHP in the two-heat pump system based on the same losses and the same temperature set point of 71 °F, with a swing tolerance of ±2 °F. Heat pump operation is again simulated in 5-minute increments over a 24-hour period. For the BHP, heating is provided by a 2 ton air-to-air heat pump, which is modeled based on the specifications of a Carrier Home Comfort heat pump with a HSPF of 8.5. The COP curve used to model the BHP's performance is given in Figure 5. Results per household for the BHP simulations are presented in Table 1.

VII. RESULTS

This manuscript focuses on using aggregated dual-heat-pump systems to achieve two utility operational objectives: winter peak demand reduction, and lower overall energy consumption. We present results from these studies in this section, followed by a discussion of findings in Section VIII.

TABLE 1. Benchmark case: heating cost and energy usage, by season.

Season	Customer Cost	Utility Cost	Energy Usage (MWh)
Winter	$380	$120	3.3
Spring	$170	$38	1.5
Autumn	$250	$77	2.2
Total	$800	$235	7.0

A. PEAK DEMAND REDUCTION

According to data provided by PGE, the highest annual peak winter demand from 2013-2014 in their service territory occurred on December 9th, 2013 from 6pm-8pm. Ambient air temperatures were taken from Weather Underground, which gives the air temperature readings at the Portland International Airport (PDX) as 29 °F, or 271 K, for each hourly reading from 2:53 pm until 8:53 pm. Operating at this outdoor temperature, the BHP would have a COP of 1.8.

The ability of the two-heat-pump system to provide shaping depends on the difference in the maximum power consumption between this benchmark system and the water-to-air CHP operating with the pre-charged water tank as a source over that peak demand period. This was determined by simulating the behavior of the UHP to meet heating demand based on the ambient air temperature during the peak period and assuming that the water tank had been preheated to the maximum allowable tank temperature, 100 °F. This analysis was then redone based on the manufacturer specifications for the CHP at the highest rated input water temperature, which is 85 °F. This case study was performed to address the uncertainty introduced by extrapolating the manufacturer data beyond the empirical data provided in the unit's specifications. Results, shown in Table 2, show that the system is capable of achieving peak demand reduction during a winter peaking event under both scenarios, in comparison to the baseline case.

TABLE 2. Peak reduction, considering 10,000 homes. Aggregated dual-HP systems require less power during peak demand periods than the benchmark alternative.

Season	Benchmark (MW)	Optimized (MW) with 100 °F Water	Optimized (MW) with 85 °F Water
Winter	26	9.6	12

B. HEATING COSTS AND ENERGY USAGE

Using the iterative method, we ran the algorithm on a month's worth of data during autumn, winter and spring time periods. Doing so provided an opportunity to run the algorithm against numerous temperature and pricing profiles, and to see the cumulative affects of running the algorithm consecutively for many days in a row.

The algorithm was applied using data from the month of January, (2012) when peak demand times tend to occur more often; the U.S. Pacific Northwest has a winter peak heating load. It was also subjected to conditions from the month of April (2014) to evaluate performance during spring. Load shifting could play a critical role in increasing the utilization factor of wind generation resources in spring. This is of particular concern in the Pacific Northwest of the U.S., due to over-generation of hydroelectric power and an abundance of off-peak wind energy, resulting in frequent low or negative electricity prices in spring months. Tests were also conducted using data from October (2014), as a representative of autumn conditions. However, gains are not as pronounced in autumn as they are in winter and spring. During autumn months, wholesale electricity prices are neither driven up by peak heating demand periods as occurs in winter months or driven down due to periods of high wind and hydro production as seen in spring months. When evaluating the annual cost savings based on the daily performance of the system, each season was set up using the hourly temperature data typical for each day.

A summary of results from seasonal energy use studies is given in Table 3, which demonstrates the potential for meaningful reduction in the cost of electricity to both the utility and to the customer as well as an overall reduction in energy consumption.

TABLE 3. Dual heat pump case: heating cost and energy usage, by season. Percent reductions from the benchmark case in parenthesis.

Season	Customer Cost	Utility Cost	Energy Usage (MWh)
Winter	$270 (29%)	$100 (17%)	2.9 (12%)
Spring	$129 (24%)	$33 (13%)	1.4 (7%)
Autumn	$190 (24%)	$70 (9%)	2.1 (5%)
Total	$592 (26%)	$202 (14%)	6.4 (9%)

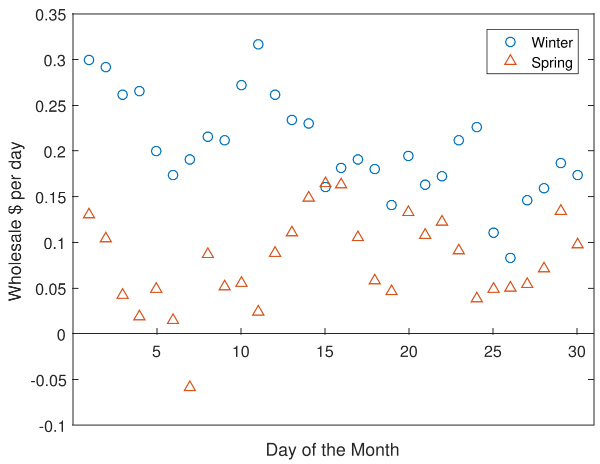

FIGURE 6. Costs per residence in wholesale electricity costs during spring and autumn time periods.

Results are also presented graphically in Figure 6, demonstrating the day-to-day variability in energy consumption. As shown, greater savings are realized in winter. Volatility is experienced in spring, with very low costs on some days, and even a negative price on one day. These results reflect the previously-discussed expected price and weather data characteristics for these seasons.

VIII. DISCUSSION

This dual-heat pump system provides efficiency gains for both the utility-side and customer-side heat pumps through improved COP due to the use of the thermal storage unit as a sink and source. Gains are also realized through reduction in consumption during peak demand times, which occurs when the thermal storage units are pre-charged in anticipation of peak demand periods during time periods when the ambient outside temperature is higher. This reduces the power consumption for both the UHP and the CHP while providing the same space conditioning. Customers realize additional costs savings through the purchase of therms from the utility based on the wholesale costs of electricity.

Using the desired operating temperature range for the water tank and heat pumps, results showed a 63% reduction in power consumption during winter power peaking times over the benchmark heating system. This savings may be a low estimate; on very cold days, many heat pumps revert to electric resistance heating in order to maintain the home temperature setpoint, which would increase power consumption for the benchmark model. We chose not to model this option, so our benchmark model remains in heat pump mode for all temperatures.

This analysis relies on the extrapolation of the heat pump manufacturer specifications to inlet water temperatures for which the unit is not rated and where empirical characterization data are not available. This analysis was therefore also performed using COP information at the highest inlet temperature available, which yielded a lower but still significant peak power reduction of roughly 53%. In addition to the potential for providing shaping, utility costs are reduced through the reduction of energy consumption and the shift of electricity consumption to times of lower wholesale electricity costs.

IX. CONCLUSION AND FUTURE WORK

This study was performed as an evaluation of a technological solution to realize winter peak demand reduction and overall lower energy consumption within a utility service territory. The technology proposes the use of water tanks as thermal storage units to increase the efficiency of a water-to-air heat pump serving residential heating load. By operating the utility-side heat pump in response to wholesale electricity prices, a utility would realize costs savings through purchase of electricity during periods of low-price energy. Simulations demonstrated that peak shaving could be accomplished using the two-heat pump system while providing customers with reduced electrical bills, assuming that the system replaced an air-to-air heat pump of equivalent size.

Four revenue streams provide financial justification for utility investment in these systems: reduced investment in peak generation plants; value of large-scale energy storage for providing ancillary services; energy savings through higher COP; and, displacement of capital investment and electrical losses associated with transmission and distribution. These revenue streams justify the upfront investment in an aggregated system of these units, which can address the challenges of peak power demand and greater penetration of non-dispatchable generation.

Results show savings through both winter peak demand reduction and overall lower energy consumption. Energy consumption results were demonstrated in winter when heating demand is greatest and in spring, when over production of hydro and off-peak wind generation can drive prices low or negative. The system's ability to perform load shifting to realize these savings demonstrates the potential for such as system as an alternative solution to investment in natural gas peaking plants.

In future, our work will consider optimized dispatch of the dual-HP system, given projections of weather and electricity prices. An objective function can be defined for wholesale costs, customer costs, or CO_2 reduction, for example. Optimization techniques have been been explored as a means for optimally controlling building heating and cooling systems coupled with large thermal masses [29]–[31]. Optimization solutions for energy management system dispatch make use of forecasted weather, load demand, and renewable generation supply data to inform the optimization algorithms [32], [33]. Using techniques such as linear programming, swarm optimization or differential evolution, and subject to constraints, an objective function for the dual-heat pump system may be defined and then minimized to realize additional savings beyond that demonstrate in this work.

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