Residential HP Modeling and Prototyping: *Phase II Report*

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1 Introduction

With the greater penetration of renewable energy generation into the power grid, utilities are facing new challenges. Renewable energy sources by nature are non-dispatchable, and pose new difficulties to the utilities such as voltage fluctuations, lack of capacity, and intermittent generation, which in many cases can cause grid instability. The traditional method to counter these issues is to construct or recommission old plants to become dispatchable generation plants to generate the required power during peak demand times. However as the gap between peak and average electricity demand continues to widen, the average utilization levels for these generating plants continues to decrease. This results in increased generating plant operating costs. To counter this problem, utilities have been developing programs and technologies that shift demand from times of high usage to more modest times of usage in order to alleviate the need for peaking generation plants. To accomplish this demand shift, also known as "peak shaving," this project investigates leveraging the thermal mass of a water tank system to shift power usage away from peak demand periods while providing residential heating. Briefly stated, the water tank will serve as a thermal energy storage unit that can be preheated during low demand times and utilized during peak demand times.

This project proposes the use of one or more water tanks as thermal storage to provide demand response in residential heating applications. This is accomplished using two heat pumps, one utility-owned pump that would manage thermal energy from the exterior of the house and store it in the water tank, and one customer-owned pump that would serve the heating or cooling needs of the home, drawing on the thermal storage unit as a source. Thus, it supplies the home's space-conditioning load while allowing the electric utility to shift energy consumption to times of lower wholesale cost to better utilize renewable energy sources. Customer savings are achieved through the better performance of the customer-owned heat pump providing the residential space-conditioning and by the purchase of therms stored in the thermal mass at a rate below what they would pay for retail electricity.

An air-to-water heat pump is used to heat and cool the water tank. This heat pump would be owned and operated by the electric utility and used to charge the thermal storage unit during off-peak times and times of lower wholesale electricity costs. A 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. Thermal energy extracted from the tank by the customer-side heat pump 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.

The two heat pumps operate autonomously, and their operation is predictively optimized over a sliding twenty-hour period using projected weather and wholesale pricing information. Operational set points for the heat pumps would therefore be sent to the controller at some regular interval. The performance of the system can be optimized 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. A diagram of the system is included in Figure 1.

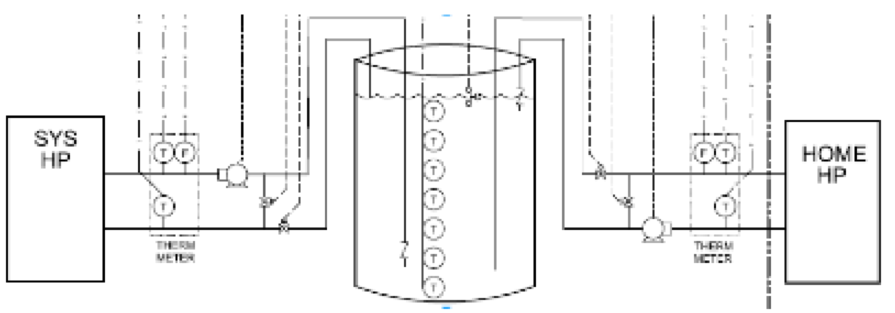

Figure 1: ResHP System Diagram

The first phase of this project saw the development of thermal models for the home and water tank, an initial structure of the optimization routine, a mechanical design for the system and the beginning of a system prototype. The second phase of the project entailed the characterization of the prototype, the further development of the optimization routine to reflect actual system performance and use-tesing.

2 System Prototype

2.1 Heat Pumps

By coupling the two heat pumps through a water tank, the system is able to partially decouple energy generation from consumption. This is done by using a larger heat pump to put energy into a thermal storage reservoir and a smaller heat pump to draw off the reservoir to heat and cool the home. The premise-side heat pump is the unit that is installed to condition air temperature in the house. It transfers energy from the thermal capacity of the water tank to increase or decrease the temperature of the house to the resident's comfort. The system-side heat pump conditions the storage tank using the outdoor environment as a source. Both heat pumps are powered by 208 VAC, are connected in parallel, and can be operated independently of each other. In heating mode, which was primarily investigated during system characterization and use-testing, the system-side heat pump can reliably operate within 34°F to 120°F water inlet temperature, while the premise-side will reliably operate from 40°F to 100°F water inlet temperature. Once thermal energy is transferred to the tank by the system-side heat pump, the premise-side heat pump may use this energy to more efficiently heat the home. In this operating scheme, the tank is pre-charged to allow greater rate of heat transfer for a given amount of power consumed.

The premise-side unit is a 3/4 ton water-to-air heat pump. During the heating season, the premise-side heat pump uses the thermal storage during desirable heating times. The COP of the premise-side heat pump is highest for heating the house when the water tank has the highest temperature of water stored.

The system-side is a one ton air-to-water heat pump. During the heating season, the system-side heat pump is used to take in outside air and heat the water that is used as thermal storage in the tank. The COP of the system-side heat pump is highest for heating the water tank when the temperature outside the house is highest because there is a greater amount of available energy per unit of air taken into the system.

2.2 Water Pumps

The 1/8 HP hot water circulator pump for the hydronic system was sized to provide 2.5 to 4 gallons per minute flow over the operable range with a hand-operated throttle valve on the premise-side. Both water pumps receive full power when the prototype is switched on. Earlier phases of prototype construction utilized variable frequency drives to adjust water pump flow. GS-1 AutomationDirect (AD) was the natural choice for the VFD since it is designed to work with the AD PLC that was selected for the control system, making integration simple. However, high-frequency noise introduced into the system by these VFDs made sensor information intermittent and highly inaccurate. This method of control was therefore removed, and a valve was added on the premise-side water loop to provide flow speed control.

2.3 Plumbing Materials

Cross Linked Polyethylene (PEX) piping was used for the hydronic system. PEX is easy to work with, flexible and inexpensive. A food-grade polyethylene barrel was used for the thermal storage tank. This tank has a volumetric capacity of 55 gallons. This size was chosen because it was estimated that a one ton heat pump would be able to heat the entire tank by 30°F in about three hours.

Superstrut was used as the structural racking for the hydronic system of the prototype. This allows for reconfiguration and is sturdy enough to hold up to any operation and testing. ASCO Red Cap valves with solenoids were selected for the hydronic valves in the prototype. They are a well-proven and reliable valve that is easily controlled with the AD PLC. WYE-strainers were added to the hydronic system to filter out large particulates that might get into the thermal storage tank. This helps to prevent flow blockage and damage to the heat exchangers in the heat pumps.

2.4 Sensors

All temperature readings for the prototype are taken using DS18B20 units. This enables the collection of many temperature readings without using up an I/O port for each sensor. Using a simple micro-controller, many temperatures can be taken and sent to the PLC or to a computer. The DS18B20 has a capacitor included in the IC so it is capable of using its own stored power. It can also take that power from the data line so it has no need for a separate power source. The temperature response and accuracy characteristics are well within the acceptable range for this proof-of-concept prototype.

The prototype is equipped with a vertical array of temperature sensors in the thermal storage tank. This is used to track the energy stored in the tank and to characterise stratification so its effects on the overall system performance can be tracked. Temperature sensors are also at the inlet and outlet of both heat pumps. These are used to track the actual energy flow through the system.

Simple paddle wheel flow meters were selected for the prototype because they interface nicely with the CTRIO high speed counter card of the PLC and have a high degree of accuracy for their price point.

2.5 Power Meter

The power meter was donated by PGE and is used to collect voltage, current and power factor drawn by the heat pumps. The system only runs on two phases and the third phase is available for any auxiliary functions if needed. The meter communicates to the computer via ModBus protocol. The data goes into the computer's MatLab program and is used to calculate the systems power status during intervals of sampling. The meter is very important because the COP of the heat pumps are measured using this device's power information data.

2.6 Control Hardware

The control system hardware consists of components from the Do-More line of PLCs and C-More line of touch screens from Automation Direct. A combination of the DL-205 base and H2-DM1E CPU were selected in conjunction with various appropriate input and output modules for creating the control system while the user interface is accomplished by an 8 inch C-More panel, which can be used to manually operate the system. The DL-205 is equipped to handle ModBus communication, allowing for a standard communication for all devices used in the prototype. The touch screen is programmed to control the basic functions of the system such as valve configuration, turn the heat pumps on and off, and to control the heat transfer direction. The system can also be operated through the optimization routine, which sends control information directly into PLC registers, locking out manual touchscreen operation.

2.7 Thermotron

The Thermotron S/SM-Series Environmental Chamber is a climate controlled thermal chamber. It has 6" vents on its side, to which ducting can be attached. Our intent for the Thermotron is to replicate a set indoor or outdoor environment for the heat pumps to source from. The intake of the heat pumps draws air through the ducting and into the heat pumps for thermal energy transfer.

The Thermotron has a touch screen interface to input desired temperature set points. It has an option for manual adjustment, where one can change the temperature inside the Thermotron by simply typing in a new one. While in manual control, the user can select a temperature ramp rate which controls how fast the Thermotron will respond to a given input. In programmable mode the user can create a program which will dictate chamber temperature over a series of time intervals.

One of the challenges presented by the Thermotron was its transient behavior. It tends to initially overshoot or undershoot a temperature set point. However, if given time to settle into a temperature range, the Thermotron is able to handle slight adjustments in temperature, which makes it possible to replicate the gradual change of temperature over a 24 hour day.

3 Thermal Models

An important part of this project was the development of thermal models of the water reservoir and the home. These models were important when doing studies on our system. They were used to predict the behavior of these systems over a full day. It is very important that these models be accurate, especially when implemented in the optimization routine. Without accurate thermal models we cannot with confidence say that our best solution is in reality the best solution.

3.1 Water Reservoir

Our water reservoir was comprised of an insulated 55 gallon tank with the heat pumps water inlets and outlets attached. Our model had to account for the energy transfer from the heat pumps, and energy lost to the air. These equations can be found in appendix A. We were unable to achieve temperature stratification within the tank so it wasn't accounted for in our model. The equations used for this model were accurate, but only for about an hour before the error becomes significant. The goal is for this model to accurately predict the water reservoir temperatures for an entire day so the optimization routine may give a proper solution. To correct this error we took 24 hour optimization test data from the prototype. Using MATLAB we recreated this test using the thermal model. We found the relationship between the error, actual tank temperature minus simulated tank temperature, with respect to time was linear. Using this relationship we were able to properly compensate the simulated tank temperatures to achieve results within 2 °F with respect to our 24 hour test data.

3.2 Home Model

The home was modeled to be a 2000 sqft house with 10 ft ceilings. The model accounts for the thermal conductivity of the home, outside temperature, and the heat transfer of the customer side heat pump. We were unable to test this model with the equipment that was available to us. For future work this model will need to be properly tested. This model is what determines when and for how long the customer side heat pump will need to be turned on. If the thermal behavior of the home can be accurately predicted, you can predict the required operation of the utility side heat pump. This type of prediction is heavily relied upon by the optimization routine.

4 System Characterization

4.1 System-Side Heat Pump

The characterization of the system-side heat pump was done by holding a constant inlet air temperature and heating the tank through the entire operable range. Tests set a stiff inlet temperature from the Thermotron to simulate an outdoor environment. Then we would operate the heat pump, ranging the temperature of the tank between 35°F up to 100°F in order to develop COP and capacity curves for the unit. Then by using the temperature sensor data of the system, and the power information of the meter for each time step, a COP and $\dot{Q}$ value was calculated.[1] Using MatLab's poly-fit function, a second-order a best fit curve through the data was fitted and plotted as shown below. This was repeated for an inlet air temperature ranging from 35°F to 74°F in intervals of approximately 5°F.

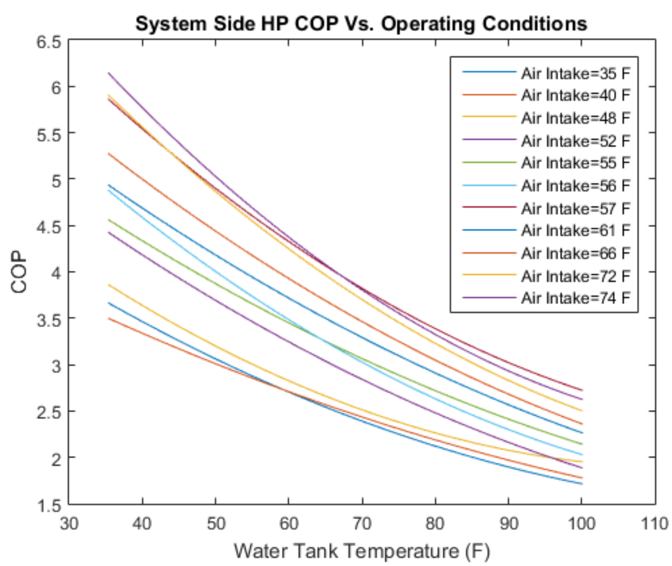

Figure 2: System-side Heat Pump Vs. Operation Conditions

The COP curves followed a similar pattern and decreased with the increase of water temperature in the tank. However, the COP of the system-side heat pump

[1] $\dot{Q}$ is the thermal energy transfer rate

is lower on a cold day with an air temperature of 35°F, versus a warm summer day. During winter months, the system-side heat pump COP would range from 3.5 to 2. It is worth noting that a lower-bound COP of 2 still outperforms an equivalent air-to-air heat pump. The benchmark air-to-air heat pump we considered has a heating seasonal performance factor (HSPF) and would have a COP of 1.8 under similar operating conditions. The system-side heat pump can also be operated during warmer periods to pre-heat the tank in anticipation of colder periods, further increasing the COP of the overall system.

The results of heat transfer characterization of the system are shown in the plot below. As the temperature of the tank increased, $\dot{Q}$ decreased. Also, the higher the inlet air temperature the greater the heat transfer was. A source of error for the test was humidity in the air. On a rainy day with high humidity, the system would behave differently than on a sunny day with low humidity in the air. The Thermotron environmental chambers were used to control only the air temperature, but not the humidity in the air hence some of our $\dot{Q}$ versus water temperature plots cross. Overall, all characterization tests resulted in the same pattern of $\dot{Q}$ decreasing as the water temperature increased.

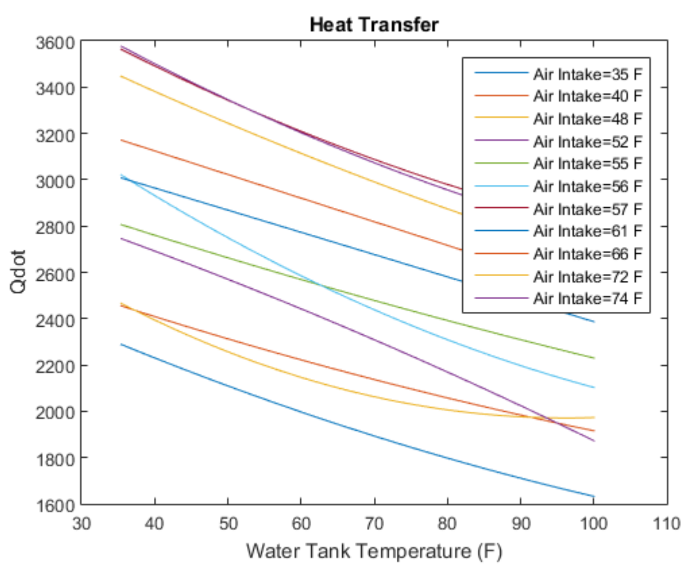

Figure 3: Heat Transfer of the System-side heat pump

Below is a surface plot that visually shows the COP value relationship for outdoor air temperature and charged water tank temperatures using interpolated values from every system-side test. COP was highest for high inlet air temperatures and low tank water temperatures. Conversely, COP was lowest for low inlet air temperatures and high water tank temperatures.

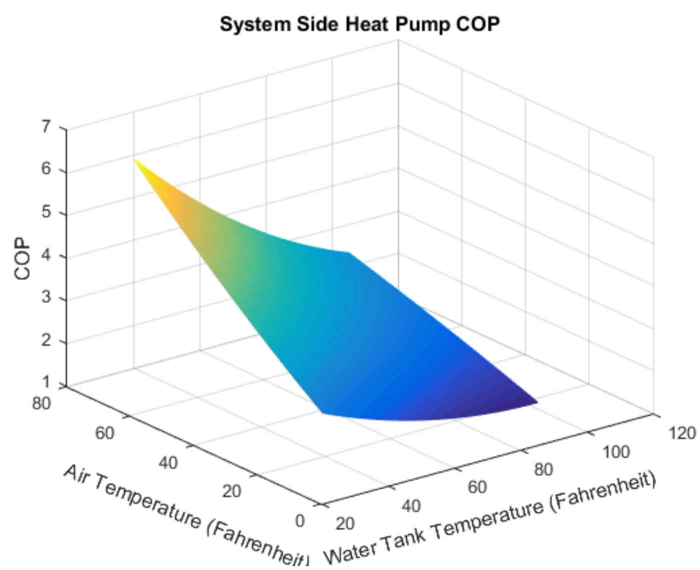

Figure 4: System-side heat pump COP

4.2 Premise-Side Heat Pump

The Premise-side heat pump was characterized by connecting the Air inlet of the heat pump to the Thermotron while maintaining the Thermotron temperature at 68°F, which is within the expected range of temperatures for most conditioned indoor spaces. With the Thermotron connected, the heat pump was operated in space heating mode until the water tank temperature was reduced from 100°F to 35°F. For each characterization tests, the mass flow rate was varied from 2.0 to 3.7 in steps of 0.5 gpm until hitting a ceiling of 3.7 gpm. 5 different test cases have been included in the formulation of the COP models for the premise-side heat pump, including data collected from the strategically placed temperature sensors and the readings from the smart power meter. Using MatLabs' polyfit function a best fit line through the data was created for each mass flow rate for future usage in modeling an optimization routine.

Before discussing the results it is worth mentioning that during testing the premise-side heat pump produced hot air at its outlet, which is expected to happen. However the flow of hot air overwhelmed the temperature in the room, at times inhibiting the Thermotron from maintaining the desired temperature of 68°F, hence creating room for error as the air inlet temperature on the heat pump rose about 2°F after running for some time. Additionally, the heat transfer is better on days with a higher humidity and worse on less humid days, creating a small discrepancy in COP curves depending on the day the test was performed. After implementing strategies to reduce errors we compiled the data in MatLab obtaining what is seen in Figure 5.

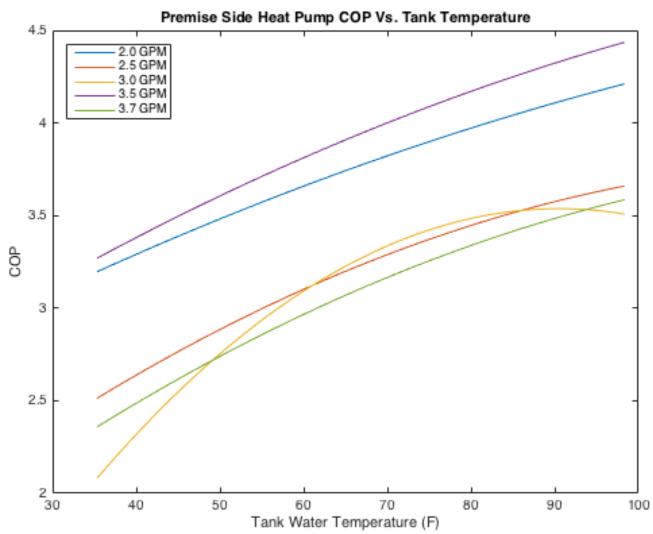

Figure 5: Premise side heat pump COP Vs. Tank temperature

As expected the COP is better for higher temperatures of the water in the tank. We can also observe that although the best COP seems to be when the premise-side system is running at 3.5 gallons per minute (GPM), which is being represented by a purple line on the plot, there is no expressive influence from the mass flow rate on the COP values obtained. The differences between the curves are caused basically by differences on air humidity and temperature. These results were obtained by comparing each test to the meteorological data for their respective testing days.

The system could also be used as a heat sink during hot days, utilizing the system-side heat pump to cool the water and then during peak hours use the premise side to sink heat from the house into the water, and provide cold air back to the residence. The water in the tank could also be used to supplement the hot water supply to the house. There still exists several possibilities of expansion in functionality and efficiency that could be implemented in later testing.

Figure 6 shows the premise heat pump capacity curve and provides insight into how much energy this system has stored for its temperature operating range. The higher the water temperature is, the more thermal energy the heat pump is able to provide to the residence. The capacity curves are very susceptible to the thermal mass size given that for a larger tank more thermal energy would be able to be stored for the same temperature range.

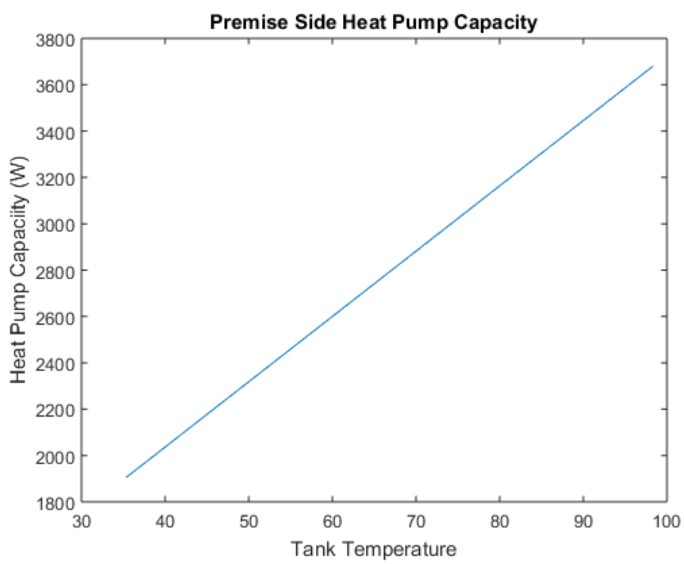

Figure 6: Premise side water-to-air heat pump capacity(Temp in units of F)

The COP curve for the customer heat pump in Figure 7 shows better values for higher tank temperatures. This means that less electrical energy is required from the pump to draw the same amount of heat out of the water for higher water temperatures. Even though the COP values decrease for lower temperatures, at its worst operation point it is still better than the than the air-to-air heat pump COP shown in Figure 8 for more than half of its operation range. Taking in to account that the system-side gets better COP values for higher outdoor temperatures as well, the system as a whole beats the benchmark COP for a air-to-air heat pump.

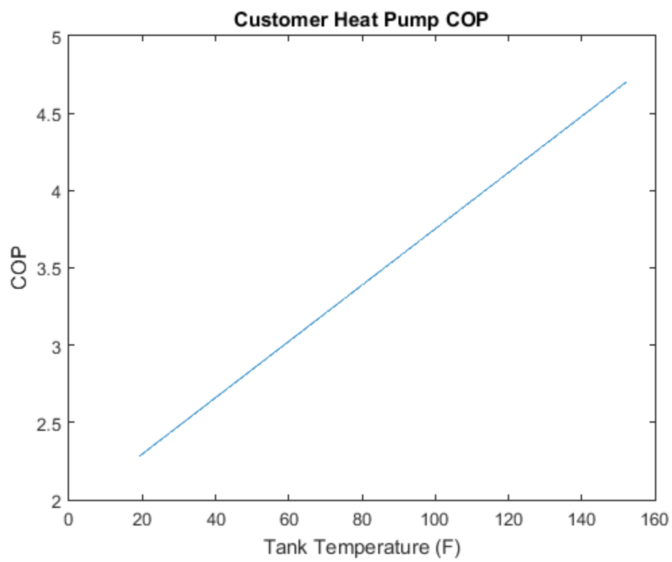

Figure 7: Premise side water-to-air heat pump COP plot

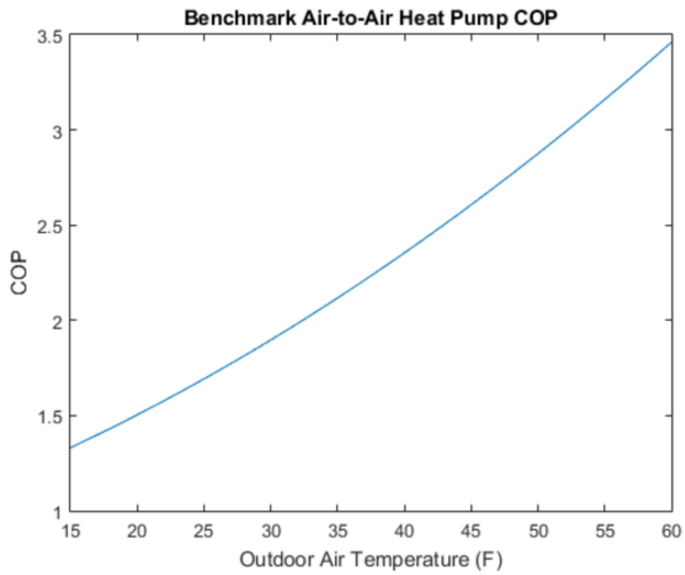

Figure 8: Premise side air-to-air heat pump benchmark COP plot

A look-up table was developed in Matlab to output the COP and heat transfer characteristics of both heat pumps given the current operating conditions. This allows the optimization routine to estimate heat pump performance given projected temperatures.

4.3 Water Tank

The water tank has the following specifications:

- 55 gallons
- 1 meter tall
- 1/2 meter diameter
- Contained only tap water
- The PVC piping and tank were insulated with 1/2 inch thick elastomeric insulation that had an R value of 3.

Given that the size of the tank is small, there was almost no stratification in temperature between the bottom and top of the tank. When developing a thermal model to describe the tank temperature and heat capacity profiles, the stratification on the water temperature was considered negligible.

5 Optimization

A routine was developed in Matlab to predictively determine a good operational schedule for the system-side heat pump to charge the thermal storage unit during times of low wholesale electricity cost such that it will maintain water temperature within the operating bounds of the premise-side heat pump and reduce demand during times of high wholesale electricity costs. This routine incorporates the models developed for the water tank as well as the heat pumps' operational behavior determined through characterization testing. Coupled with forecasted temperature and electricity cost information, it is capable of modeling the system's performance over a progressive twenty-four hour period and then sending operational instructions to the heat pumps.

A genetic algorithm was chosen as the strategy for generating this operational vector, which dictates the performance of the system-side heat pump. This decision was made because a genetic algorithm is designed to operate on binary strings, which is a convenient mechanism for encoding fixed speed operation where a 0 indicates that the heat pump is off during the time interval in question and a 1 indicates that the heat pump is operating.

An iterative method was created to generate a known feasible solution using discrete heat pump operation. This seeding routine iteratively checks for system constraint violations and then flips bits within the operational vector to correct detected violations until a feasible solution is found.

A discrete hill-climbing algorithm was also developed as a method of benchmarking the genetic algorithm during development. A population of fifty binary strings is initialized randomly with the exception of one individual, which is the known feasible solution. Each individual in the population is then perturbed one element at a time and the fittest of these "neighbor" solutions is selected. This is then repeated twenty times on each individual in the population. Within the context of this algorithm, a neighbor is defined as a binary string having a Hamming Distance of one from the original individual.

For the genetic algorithm, a population of ten binary strings is initialized randomly with the exception of one individual, which is the known feasible solution.

The fitness of each individual is then evaluated and the best individual is saved so that it is not lost during evolution. Mutation is then performed with a probability of 2% using bit-flipping as the mutation operator, meaning that an average of roughly 2% of the bits in an individual will be flipped from either a one to a zero or from a zero to a one. Once mutation has occurred, the fitness of the mutated individuals is then evaluated, replacing the elite individual if a better solution has been generated.

Two-point cross-over then occurs with a probability of 70%. During this process, two parent individuals are combined to create an offspring by randomly selecting sections of each parent binary string to build a new solution. Finally, a new population is selected probabilistically in proportion to the quality of the solution. The worst solution in the population is then replaced with the elite solution and the next generation begins. On a Intel(R) Core(RM) i7-2600 CPU Processor with 8-GB of RAM, 25 generations of the algorithm can be run in 11 to 15 minutes. A summary of the algorithm operators and parameters is given in Table 1.

Table 1: Genetic Algorithm Parameters

Population	10
Generations	25
Runs	1
Mutation Operator	Bit-Flipping
Recombination Operator	2-Point Cross-Over
Probability of Mutation	2%
Probability of Cross-Over	70%
Local Search Depth	1
Selection Method	Roulette Wheel

The genetic algorithm was first developed and tested using typical meteorological data and average hourly locational marginal pricing information from the Malin node for January 2012. The genetic algorithm showed a 6% improvement over hill-climbing. An example plot of the evolution of the best solution for a day's operation over 50 generations is given in Figure 9.

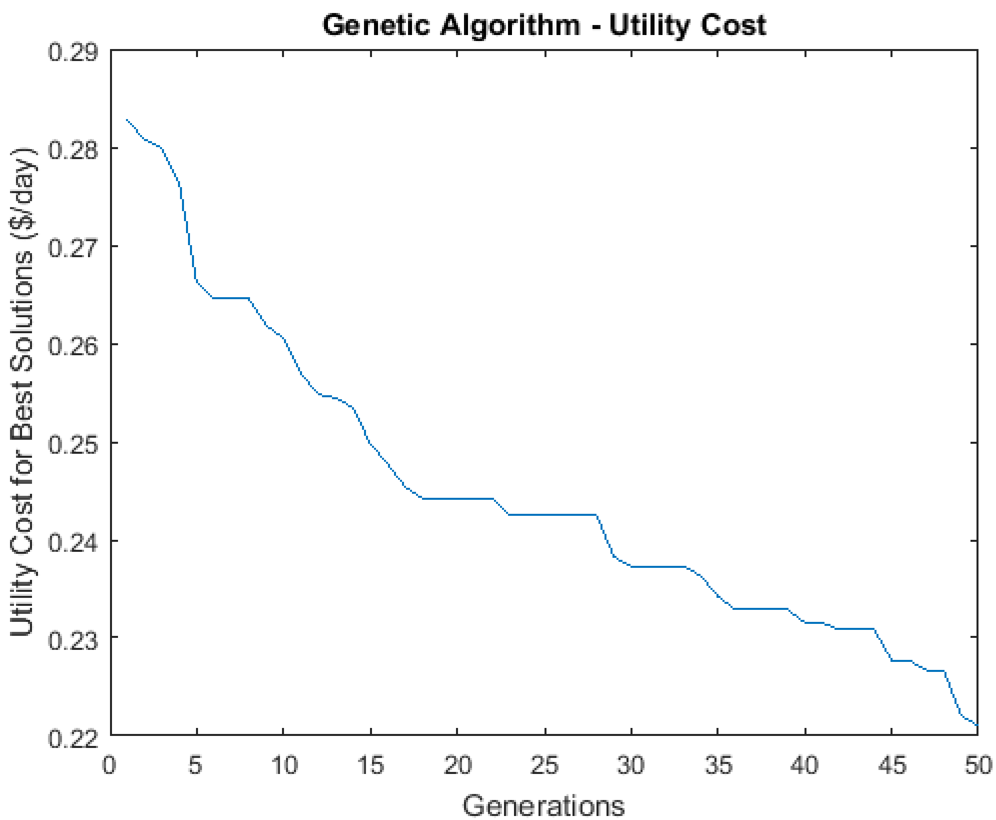

Figure 9: Example Plot of Best Solution Evolution

6 Performance Testing

6.1 Case Study

A case study was performed using the following characteristics:

Home Model

- House : 2000 ft, 10 ft Ceilings, w/ 15% Duct Losses
- Premise Side Heat Pump: 2 ton
- System Side Heat Pump: 3 ton
- 20,000 homes incorporated
- Thermal admittance (UA) of 400 kBtu/hr

Thermal Model

- Water reservoir : 210 gallons

Benchmark Heat Pump

- 2 ton air-to-air Carrier Home Comfort with a Heating Seasonal Performance Factor (HSPF) of 8.5

The system was scaled up from the prototype to represent an actual residential house. To demonstrate a real-world application of our ResHP prototype, we ran both systems through a simulation, where we simulated scaled-up operation in 20,000 homes. Each house was modeled as 2000 square feet.

The case study focuses on the highest peak loading day in the winter of 2013 based off of PGE-supplied data. PGE is a winter peaking power utility, and has the largest loads on the system during very cold days where heat pumps would be used to heat buildings. The weather data for the peak day was taken from a website called Weather Underground for the specified day. The weather during the max peak day started at around 17°F, and worked its way up to around 30°F during the time of the actual peak. The system is compared to a 2 ton air-to-air heat pump that was used as a benchmark. The goal of this study was to determine if our ResHP system could preform peak shaving and load shaping compared to the air-to-air benchmark heat pump.

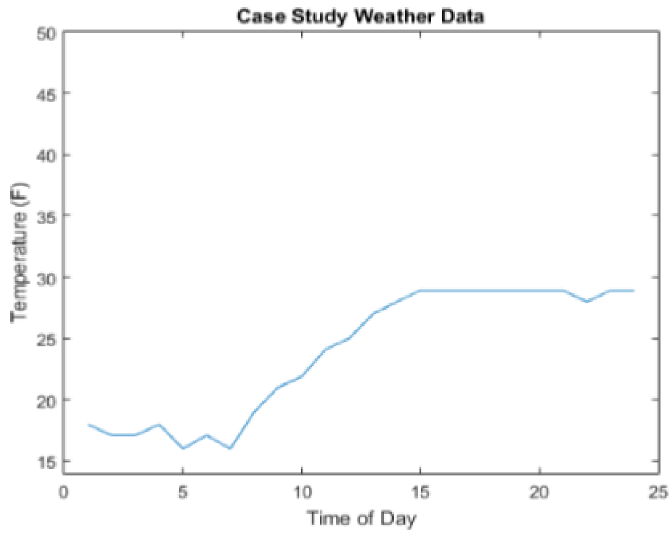

Figure 10: Weather Data, Dec. 2013 - Peak loading occurred on PGE system.

The case study looked into how the ResHP system would affect the grid during peak times, by observing what would happen if 20,000 homes were outfitted with the system in the PGE service territory. 20,000 homes is a realistic and obtainable goal to reach for the utility and will show the full capabilities of the ResHP system based on our data collected from performing tests on the system.

6.2 Peak Shaving and Load Shifting

Peak shaving, load shifting and cost minimization tests were run on the system to establish the application of the ResHP prototype. The idea for this prototype is that power can be put into the system-side heat pump to store energy in the water tank, and the premise-side heat pump can draw off of that energy at a later time. This serves the purpose of minimizing peak power usage for the utility thus saving large amounts of money that would be spent providing the extreme demands of power for the grid. The utility would need to invest a large sum of money into the construction and maintenance of the peaking power plants which would only be operated very few hours through out the year during extreme peaking periods. Figure 11 shows a typical load duration curve where 20% of the nameplate power capacity was used for only 0.5% of the total hours in a year. This means PGE could have peaking plants idle for almost the whole year and only turn them on for very brief peak demands.

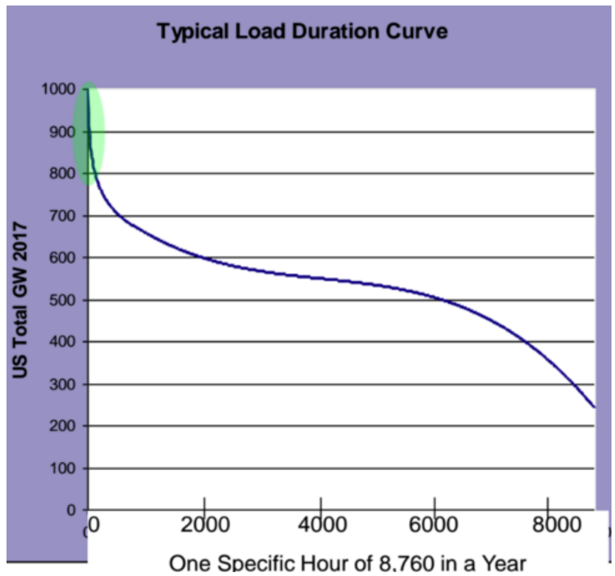

Figure 11: Amount of power required from grid vs. total yearly hours it was used

Load shifting is the mechanism of decoupling generation with the customer's power consumption. This means the utility can store energy in a thermal reservoir before extreme peaking periods, letting the customer use their own heat pump at a more efficient rate during the actual peaking time, which ultimately use less power therefor performing peak shaving.

Figure 12 shows the comparison of load profiles of the case study (Base Load), and with the ResHP system utilized (Modified Load). In the simulation of our ResHP system versus the benchmark on the worst winter day of the year the system-side heat pump precharged the tank on two separate occasions during off-peak hours (0th through 6th hour, and 13th through 16th hour). Both time periods correspond to downtimes in energy usage. The premise-side heat pump operated at a higher COP during peak hours (17th through 19th hour), as a result of system-side heat pump precharging the tank. This relationship is seen in Figure 12, where higher tank temperature corresponds to a higher running efficiency of the premise-side heat pump. The premise-side heat pump was able to meet peak energy demand, while simultaneously shaving 30 MW.

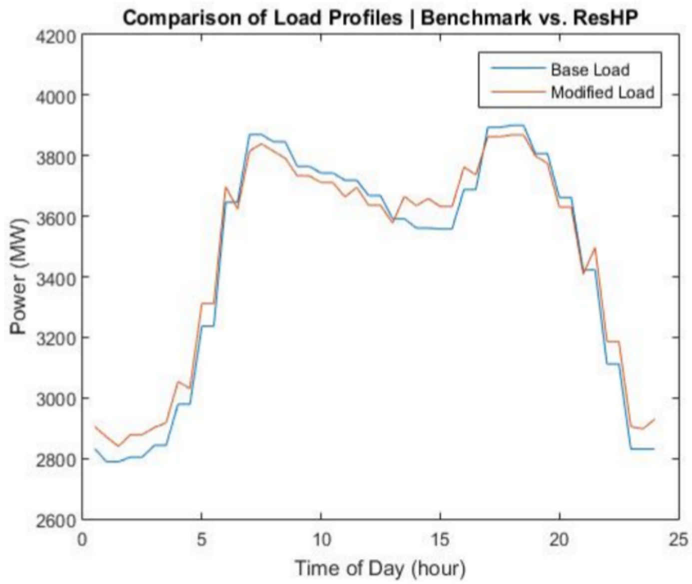

Figure 12: Peak Shaving and Load Shaping with the ResHP System

6.3 Cost minimization

Cost minimization is important for both the customer and the utility. Together the purpose of the prototype system is to provide information and data on the possible use of providing both the utility and the customer the ability to minimize costs to use power.

To provide cost reductions to the utility, the system attempts to concentrate energy consumption during times of cheaper wholesale electricity prices throughout the day. Wholesale power is cheapest for the utility when there is over generation on the grid. Power is most expensive during peak load periods when the grid is most strained for resources to provide sufficient energy to the grid. Customer savings are achieved by maximizing the COP of the premise side heat pump using energy stored in the water tank.

6.4 Genetic Algorithm use testing

The use testing of the Genetic algorithm showed that the cost of operating the system pre-optimization was found to be 16.58¢ /kWh and after optimization was found to be 15.87¢/kWh per kilo watt hour with the optimization achieving savings of 0.707¢/kWh. The system has an energy consumption of 11.34 kWh.

7 Discussion

After running multiple tests on the system at different operating temperatures and creating COP curves from the data, the results show that the ResHP system could be used to successfully provide peak shaving to the grid. The COP of the customer heat pump gets better the closer the water tank temperature is to 100°F, which is the maximum possible operating temperature for the premise-side heat pump used in this prototype work. At 100°F the COP is above 3.6 and with two hours of operation needed to provide peak shaving the COP would end at about 3.2. An air-to-air heat pump has a COP of about 1.8. This shows that the ResHP system can provide peak shaving due to the customer having to operate their heat pump for less time during the peak do to the improved COP of the air-to-water heat pump operating with the preheated water tank.

The case study simulated the effects the ResHP system would have on the actual grid if it was implemented in 20,000 homes using scaled up versions of the heat pumps and water tanks. The results were positive and showed that the system shaved PGE's maximum peak it experiences by 30 MW.

In the course of research, it was found that the system would only operate reliably over a water temperature range of 35 to 100 degrees Fahrenheit while in heating mode. The smaller, premise-side heat pump was incapable of operating with source water outside of this temperature range. Also, it was found that the COP curves were only marginally influenced by the mass flow rate. However, the humidity of the testing conditions influenced the heat pump COP values more than expected because the heat transfer capability of the heat pumps varies with the thermal conductivity of the air. In addition, it was found that the variable frequency drives (VFDs), initially implemented to control the mass flow rate gave off too much electromagnetic interference despite extensive measures taken to alleviate the issue. This corrupted the temperature sensor data so that they were eventually switched to a manual valve that directly constricted water flow. Also, it should be noted that the heat pumps used in the prototype are smaller, indoor units that lack backup resistive heating elements. They were also operated well outside of their intended operating conditions. It is reasonable to expect that larger units would perform better. In particular the system-side unit, which would be designed to tolerate outdoor conditions and lower inlet air temperatures, would almost certainly have higher COP values. Initially, we looked at the manufacturer specifications for a Daikin 3-ton air-to-water heat pump as a model for this unit. At 40 °F outdoor temperature, extrapolated manufacturer data shows a range of COP values from 3.5 at 100 °F exiting water temperature to 8 at 35 °F exiting water temperature. This is considerably better than the prototype unit, which shows COP values from 1.5 to 3.5 over the same range.

8 Conclusion

The findings of this prototype prove that this system could be implemented to reduce the peak demand on the grid. A large adoption of our system in Portland households has the potential to diminish overall power consumption during peak times. Our case study showed that if 20,000 homes were fitted with the ResHP system, the maximum peak PGE experienced in 2013 was reduced by 30 MW. This is a significant reduction and this prototype proves that energy consumption before peak hours would reduce the size of the peak, saving the utility money on expenses in association with building and maintaining peaking plants. The utilization factors of generation resources can be increased by shifting load to times of over-generation and by reducing peak demand, which reduces the need for peaking power plants. Also, savings would be made by not having to upgrade transformers and distribution lines in some cases.

The highest COP for the premise-side system is achieved when the system-side maintains highest allowable water temperature in the tank during operation. The ResHP system used a 55 gallon water tank, a larger water tank would have a longer heating time, but would also provide a higher COP for the premise side heat pump during peak hours. However, even with the small tank the results were significant and showed that the ResHP system has the potential to perform peak shaving on a large scale.

This system is a win-win for the utility and customer since, the customer will save money due to increased efficiency of their heat pump by pre-charging the thermal storage unit. The price of electricity changes throughout the day, and is cheaper during off peak hours than during the peak.

9 Future Work

As with any project, the longer you work on it the more opportunities arise to make it better. Several such opportunities have been identified that could add value to the project.

Most homeowners with heat pumps have standard air to air heat pumps. In its current configuration the project is not designed to augment such a system, rather it would be a new installation, likely replacing an old inefficient furnace. With two heat pumps and a water storage tank the footprint could be overwhelming to some potential customers. We propose using one heat pump rather than two, with ducting separated by a motor actuated damper. This would eliminate the need for the second heat pump without adding any more to the system besides a damper. Customers may find the smaller footprint more appealing and the overall cost would be dramatically decreased.

Ideally the water storage tank will be sized appropriately for the size of the home in which the system is placed, maximizing efficiency in heat transfer both to and from the tank. For a large portion of the year this will be adequate to serve the heating and cooling needs but there will exist occasions in which a backup heat source will be necessary. We could rely on heat strips in the air handler as most home units do now or we could explore ways to boost the thermal energy of the storage tank. We discussed resistive heating loads and quickly turned the conversation to existing homeowner hot water heaters. Hot water heaters are a notoriously inefficient appliance. Most of the day the hot water heater keeps a large tank of water hot when there is no demand for it. If we added plumbing from the storage tank to the hot water heater along with a pump and valve we could utilize that wasted heat energy to boost the overall heat pump system. We could even add heat to the hot water heater tank when the heat pump storage tank has reached its maximum thermal mass. The possibilities for integrating the existing hot water supply system into the heating and cooling system are very interesting and leave the door open for plenty of growth.

A Appendix: System Equations

Variables:

- A: Area
- c: Specific heat
- m: Mass
- k: Thermal conductivity
- $\dot{m}$: Mass flow rate
- T: Temperature
- $\dot{Q}$: Thermal transfer
- P: Power
- J: Total cost
- Y: Wholesale price of electricity
- t: Time
- R: Customer rate
- C: Heat pump capacity
- CR: Capacity ratio, the ratio of heat pump demand to heat pump capacity

Subscripts

- w: Water in the storage tank
- $1in$: Inlet 1
- $1out$: Outlet 1
- $2in$: Inlet 2
- $2out$: Outlet 2
- C: Cross-sectional
- $down$: Down in tank
- env: Environment
- o: Outdoor
- i: Generic node
- S: Surface
- $tank$: Storage tank
- up: Up in the tank
- $hp1$: System-side heat pump
- $hp2$: Premise-side heat pump
- H: Customer home
- a: Air in the home
- u: Utility
- c: Customer
- t: Total
- fl: Full load
- pl: Partial load

A.1 Thermal Systems Equations

Storage Tank Interior Layer

$$\dot{T}_1 = \frac{[k_w/\Delta x_1]A_C(T_2 - T_1) + U_l A_C(T_o - T_1) + U_t A_S(T_o - T_1) + \dot{m}_2 c_w(T_2 - T_1) + \dot{m}_1 c_w(T_{in1} - T_1)}{m_w c_w} \tag{1}$$

Storage Tank Top Layer

$$\dot{T}_i = \frac{[k_w/\Delta x_i]A_C(T_{i+1} + T_{i-1} - 2T_i) + U_t A_S(T_o - T_i) + \dot{m}_1 c_w(T_{i-1} - T_i) + \dot{m}_2 c_w(T_{i+1} - T_i)}{m_w c_w} \tag{2}$$

Storage Tank Bottom Layer

$$\dot{T}_n = \frac{[k_w/\Delta x_n]A_C(T_{n-1} - T_n) + U_b A_C(T_o - T_n) + U_t A_S(T_o - T_n) + \dot{m}_1 c_w(T_{n-1} - T_n) + \dot{m}_2 c_w(T_{in2} - T_n)}{m_w c_w} \tag{3}$$

Input Water Temperature

$$T_{in1} = \frac{\dot{Q}_{hp1}}{\dot{m}_1 c} + T_n \tag{4}$$

$$T_{in2} = \frac{\dot{Q}_{hp2}}{\dot{m}_2 c} + T_1 \tag{5}$$

Heat Transfer Into the Customer Premise

$$\dot{T}_H = \frac{\dot{Q}_{hp} + k_h(T_o - T_H)}{m_a c_a} \tag{6}$$

A.2 Optimization Equations

$$min\{J_u(P(\dot{Q}_{hp1}, Y)\} \tag{7}$$

subject to $\dot{Q}_{hp} \in [0, 10500]$, $T_i \in [274, 373]$

$$min\{J_c(P(\dot{Q}_{hp2}, R)\} \tag{8}$$

subject to $\dot{Q}_{hp} \in [0, 10500]$, $T_h \in [Min.Setpoint, Max.Setpoint]$

$$min\{J_t(P(\dot{Q}_{hp1}, P(\dot{Q}_{hp2}, Y, R)\} \tag{9}$$

subject to $\dot{Q}_{hp} \in [0, 10500]$, $T_i \in [274, 373]$, $T_h \in [Min.Setpoint, Max.Setpoint]$

$$min\{P_t(\dot{Q}_{hp1}, \dot{Q}_{hp2})\} \tag{10}$$

subject to $\dot{Q}_{hp} \in [0, 10500]$, $T_i \in [274, 373]$, $T_h \in [Min.Setpoint, Max.Setpoint]$

A.3 Other System Equations

System Heat Pump Coefficient of Performance

System Heat Pump Capacity

Customer Heat Pump Coefficient of Performance

Customer Heat Pump Capacity

Electrical Power

$$P = \frac{\dot{Q}_{hp}}{COP} \tag{11}$$

Cost Calculation

$$J = \frac{PYt}{3600} \tag{12}$$

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