Development of a Variable-Speed Residential Air-Source Integrated Heat Pump

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Abstract: A residential air-source integrated heat pump (AS-IHP) is under development in partnership with a U.S. manufacturer. A nominal 10.6 kW (3-ton) cooling capacity variable-speed unit, the system provides both space conditioning and water heating. This multi-functional unit can provide domestic water heating (DWH) in either full condensing (FC) (dedicated water heating or simultaneous space cooling and water heating) or desuperheating (DS) operation modes. Laboratory test data were used to calibrate a vapor-compression simulation model for each mode of operation. The model was used to optimize the internal control options for efficiency while maintaining acceptable comfort conditions and refrigerant-side pressures and temperatures within allowable operating envelopes. Annual simulations were performed with the AS-IHP installed in a well-insulated house in five U.S. climate zones. The AS-IHP is predicted to use 45 to 60% less energy than a DOE minimum efficiency baseline system while meeting total annual space conditioning and water heating loads. Water heating energy use is lowered by 60 to 75% in cold to warmer climates, respectively. Plans are to field test the unit in Knoxville, TN.

Key Words: integrated heat pump design, dual function, water heating, efficiency

1 INTRODUCTION AND BACKGROUND

The U.S. Department of Energy's (DOE) Building Technologies Office (DOE-BTO) has a long term goal to maximize the energy efficiency of the US building stock by year 2030. To achieve this vision, a deep reduction of the energy used by the energy service equipment (equipment providing space heating and cooling, water heating, etc.) is required - 50% or more compared to today's best common practice. One approach to achieving this is to produce a single piece of equipment that provides multiple services. ORNL developed a general concept design for such an appliance, called the integrated heat pump (IHP) [Murphy, et al 2007a]. Both air-source (AS-IHP) and ground-source (GS-IHP) versions of the concept are possible - this paper focuses on the AS-IHP. The GS-IHP concept has been developed into a product as reported by Rice et al (2013).

Full details of the AS-IHP concept development can be found in the report by Murphy, et al (2007b) and are briefly summarized here. This system concept (Figure 1) uses one variable-speed (VS) modulating compressor, VS indoor blower, VS outdoor fan, and a single-speed pump for hot water circulation. A 50 gallon (~189 L) DWH tank is included. The concept development analyses reported in Murphy et al (2007b) included a dedicated dehumidification mode and a humidifier option (neither of which are included in the manufacturer partner's current prototype system design as described and analyzed later in this paper).

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Multifunction systems like the AS-IHP have the potential to make fuller use of the high-efficiency but higher cost VS components by meeting not only the sensible space conditioning loads but also the water heating and dehumidification loads. The VS capability allows the larger space conditioning design loads to be met with higher speed operation, while the smaller water heating loads are met at lower speeds. Significant energy savings are possible from the higher efficiency operation of the components, the load matching operation of the equipment (providing heat exchanger unloading benefits), and waste heat recovery in the combined space cooling and water heating mode. While the waste heat recovery is not free due to the elevated condensing temperatures required to meet domestic hot water needs in full condensing WH operation, the combined mode is quite efficient as both space cooling and full condensing water heating are delivered from one compressor power input. A conceptual packaging approach is shown in Figure 2 with the compressor and water-to-refrigerant HX located indoors in a separate module.

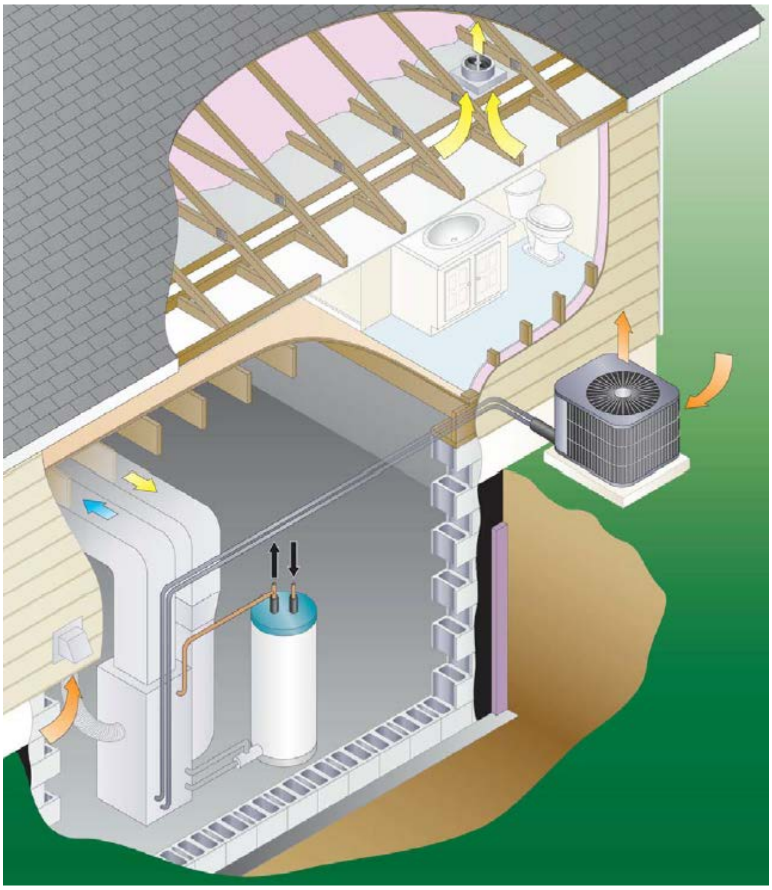

Figure. 1. Conceptual Installation of the Residential Air-Source Integrated Heat Pump

Murphy et al (2007b) conducted sub-hourly annual energy use simulations to compare the performance of the AS-IHP system concept to that of a baseline suite of individual systems; 3.8 W/W Cooling Season Performance Factor (CSPF) [13 Btu/Wh Seasonal Energy Efficiency Ratio (SEER)]) heat pump with humidifier option, 0.90 W/W energy factor (EF) electric WH, a standalone space dehumidifier, and ventilation per ASHRAE standard 62.2 (ASHRAE 2007) requirements). The TRNSYS 16 system simulation software platform (Solar Energy Laboratory, et al. 2010) was used to conduct these analyses for five US locations - representing cold (Chicago), mixed humid (Atlanta), hot humid (Houston), hot dry (Phoenix), and marine (San Francisco) climate zones. A tight, very well insulated house was used for the analyses. Results of these analyses showed that the estimated annual energy savings for the initial AS-IHP concept prototype design ranged from about 46% in Chicago to almost 70% in San Francisco. In addition estimated summer afternoon peak demand for the AS-IHP ranged from 20% to ~60% lower than that of the baseline system depending on location.

Figure 2. Conceptual AS-IHP Packaging Approach

2 AS-IHP EQUIPMENT DESIGN AND SIMULATION APPROACH

The AS-IHP concept investigation reported by Murphy, et al (2007b) led to collaboration with an air-source heat pump manufacturer to develop a design suitable for existing residential applications using R-410A refrigerant. A nominal 10.6 kW (3-ton) design cooling size was selected for development leading to the first lab prototype testing. The design uses inverter-driven variable-speed brushless permanent magnet (BPM) rotary compressor, blower, and fan. Dual electronic expansion valves (EEVs) are used to provide a wide range of refrigerant flow control. A nominal 10.6 kW (3-ton) double-walled fluted tube-in-tube heat exchanger (HX) was used for the domestic hot water with tube-and-fin HXs for the indoor and outdoor coils for the first prototype design.

Expected water heating (WH) modes of operation are 1) dedicated WH using the outdoor coil as the heat source, 2) combined space cooling (SC) and WH, and 3) desuperheating along with space cooling or heating operation. The water-to-refrigerant HX is arranged in series with the air-to-refrigerant condenser in DS mode and in parallel in FC mode. A pump capable of at least two-speed operation is required to meet both full condensing and desuperheating flow requirements.

The key design issues were to determine the optimal component operating speeds, flow controls, and refrigerant charge for the various operating modes. The compressor has overspeed capability in the space heating mode relative to the nominal cooling capacity speed. By using over-speed compressor operation in the heating mode, as proposed by Rice 1992, the heating balance point can be further reduced to minimize the need for supplemental resistance heat. The design must also keep the refrigerant operating conditions within the compressor manufacturer's allowable operating envelope of suction and discharge pressures and temperatures -- limits which vary to some degree with operating speed. An example of the condensing pressure operating limit with rotary compressors is shown in Figure 3.

One technical challenge for AS-IHP system designs is refrigerant charge management. This challenge is greater for air-source systems than for ground-source units because outdoor air coils have much larger internal volume than water-to-refrigerant HXs of similar capacity. When in combined space cooling and water heating mode, the condenser internal volume is somewhat less than in the space cooling mode. To deal with this AS-IHP issue, the manufacturer developed a proprietary design to manage charge between operating modes.

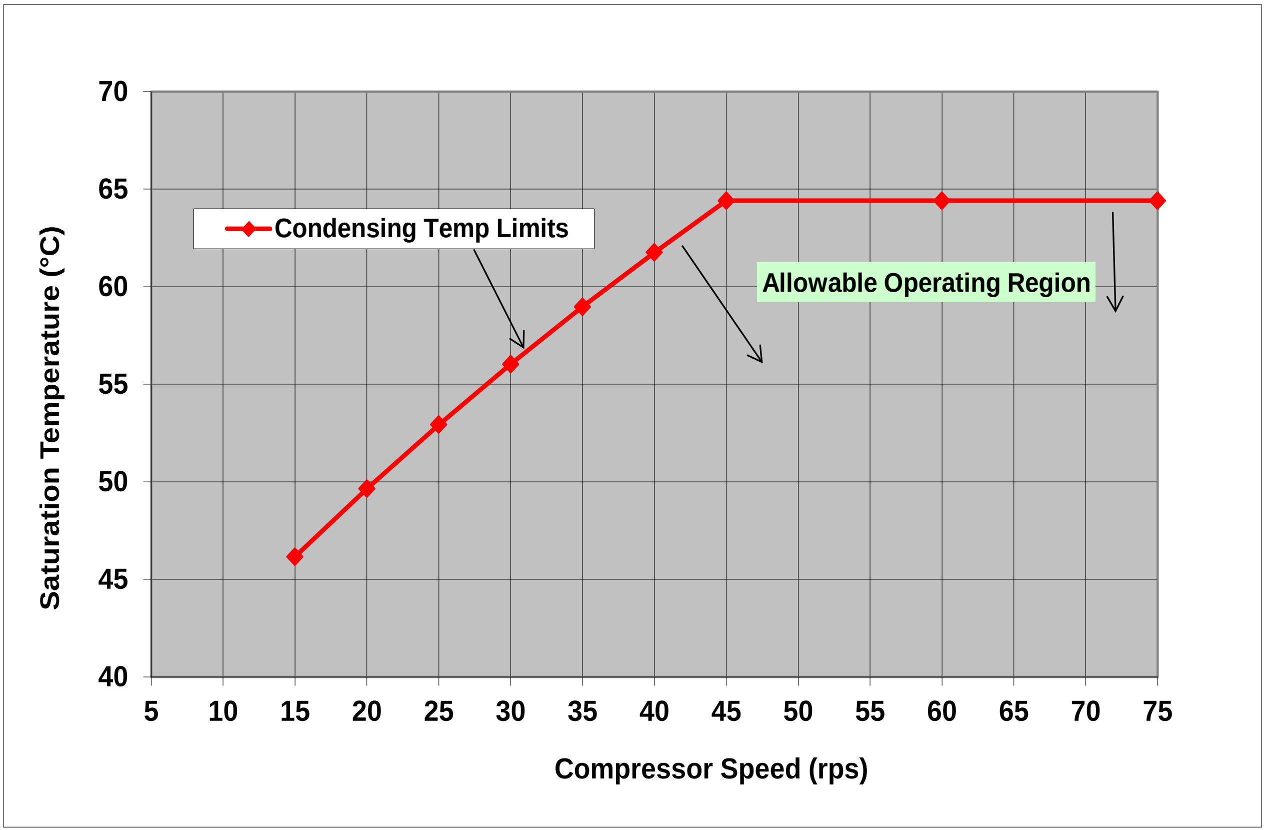

Figure 3. Condensing Temperature Limits Versus Compressor Speed for an Example Rotary Compressor

Another design challenge is in water heating. Variable-speed compressors typically can operate at maximum condensing temperatures only above a certain speed, with limits on condensing temperature dropping linearly below this speed. This constraint limits the minimum compressor speed for dedicating water heating operation. In addition, to reach maximum output water temperatures above about 50°C (122°F), higher speeds with output capacity of 10.6 kW (3 tons) or higher are required. As such, a pump capable of providing ~1.14 m^3/h (5 gpm) or higher flow is required. Operation in desuperheating-only mode can also provide temperatures above 50°C (122°F).

The first lab prototype design was assembled by the manufacturer and operationally tested in their laboratory at nominal conditions in each operating mode This unit shown in Figure 4 was then tested at ORNL in a two-room environmental chamber over a range of steady-state air-source conditions in each of the operating modes.

We used the detailed lab measurements of refrigerant and source/sink conditions to calibrate a detailed heat pump system design model (HPDM) (Rice and Jackson, 2005) in each of four operating modes: space heating, space cooling, space cooling and WH, and dedicated WH. The fluted-tube water-to-refrigerant component model in the HPDM (Rousseau 2003) requires internal geometry specifications which were obtained by direct measurements of a cutaway section as shown in Figure 5. We first obtained the refrigerant-side volume and other volume-related geometry information by successively filling the inner tube and annulus with water and comparing the weight of the assembly with that of an empty HX. Geometry details of the air-to-refrigerant HXs and compressor, blower, and pump performance maps were provided by the manufacturers.

Figure 4. First Lab Prototype AS-IHP System; (l to r) Water Heating Section with Tank, Indoor Blower and Coil Section, Compressor Section, and Outdoor Fan and Coil Section

Figure 5. Fluted Tube-in-Tube Water-to-Refrigerant HX

A software wrapper was developed to provide seamless coupling between a publically available optimization program, GenOpt® (Wetter, 2009), and the HPDM (Shen et al 2012). The GenOpt® wrapper program accepts objectives for optimizing, targeting, and bounding. It can be a flexible and powerful tool for model calibration, control strategy determination, and product configuration optimization. Also, it considers design constraints by setting bounding objectives. Furthermore, it facilitates parametric optimization runs over an extensive range, and helps achieve optimized design over an entire operation envelope. Manual calibration can be a time-consuming and error-prone practice. The GenOpt® wrapper provides an auto-calibration means by selecting targeting objectives. The auto-calibration function makes it possible to calibrate a system model against experimental data over a large range. With this approach, one can apply functional calibration curves to improve the model accuracy for wider ranges of operating conditions.

The HPDM was used with GenOpt® in this manner with the lab test data to auto-calibrate available HX adjustment factors as linear or quadratic functions of compressor speed and/or source/sink temperatures for best match to measured suction and discharge pressures. The test data were also used to determine compressor map power and mass flow corrections, compressor shell heat loss factors, line heat gains/losses and suction superheat levels as similar functions of compressor speed and/or other operating conditions, as well as the indicated active refrigerant charge in each mode. Examples of the heat transfer multipliers obtained from model calibration in combined space cooling and water heating mode

(SC+WH) are shown in Figure 6. The evaporator multipliers are usually less than 1 due to airflow mal-distribution while the condenser multipliers are usually greater than 1 for the fluted tube HXs due to the simplified model of the annular refrigerant-side heat transfer. Differences between the calibrated model and the lab data in capacity and compressor–only COP for the dedicated WH mode averaged 1.3% % with standard deviations of 3.0 and 4.6%, respectively.

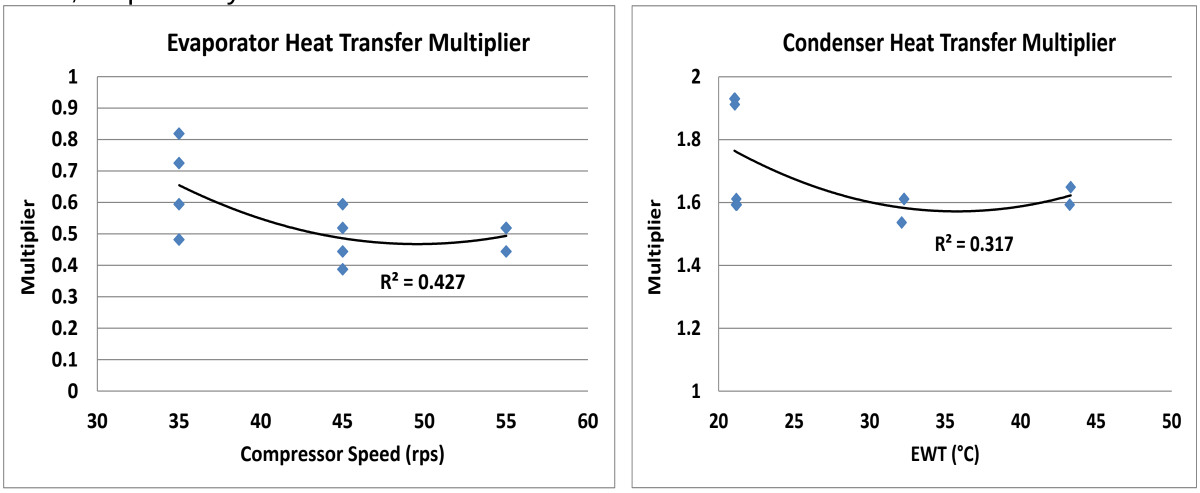

Figure 6. Example Calibration Equations for Heat Transfer Multipliers for SC+WH Mode

Power versus airflow relationships were developed for the indoor blower and outdoor fan from test data. The HPDM was again used with GenOpt® to optimize airflow rates for maximum performance, within minimum allowable delivered air temperature, over the range of appropriate compressor speeds and associated source/sink temperatures in each operating mode. One design feature specific to an integrated heat pump in combined space cooling and water heating mode is that, at EWTs above 35°C (95°F) the indoor airflow needs to be lowered relative to that for space cooling only to maintain an acceptable sensible heat ratio. An example of the required airflow reduction is shown in Figure 7, for EWTs of 45 and 55°C (113 and 131°F) over a range of compressor speeds. Also, in combined space heating and desuperheating mode, the indoor airflow needs to be lowered compared to that for the space heating only mode to maintain acceptable supply air temperatures.

This information was applied by the manufacturer in developing suitable unit control tables for the four operating modes based on the unit inlet source and sink temperatures and thermostat calls.

Once the design control approaches and calibration equations were complete, we used the HPDM to generate performance maps (i.e., tables) of capacities, powers, and mass flow rates for each mode as a function of all relevant independent variables, e.g., compressor speed, indoor and outdoor DB, indoor or outdoor RH, and entering water temperature (EWT) from the DHW loop. The desuperheating operation mode was modeled in TRNSYS as a fixed HX effectiveness based on our laboratory test data.

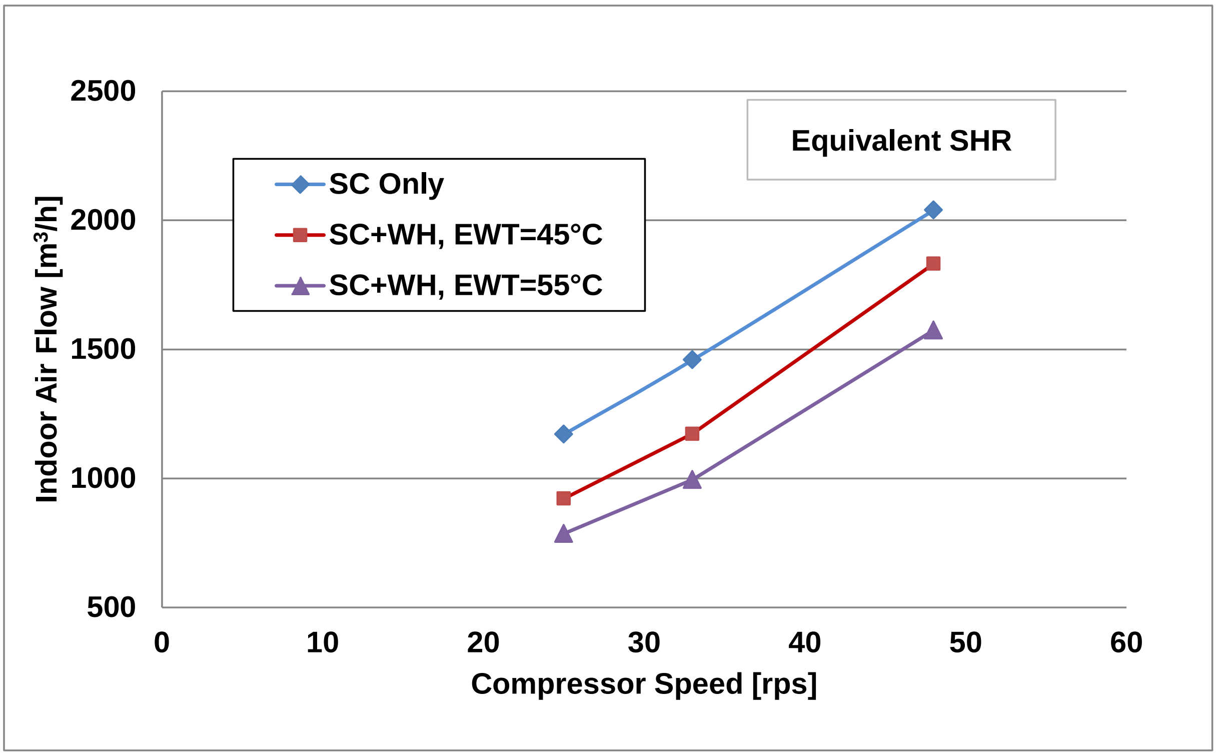

Figure 7. Required Indoor Airflows to Maintain Similar SHR Levels in SC and SC+WH Modes

3 ANNUAL ENERGY USE ANALYSIS AND SAVINGS PREDICTIONS

The HPDM performance maps were used as input to the TRNSYS model for sub-hourly annual AS-IHP performance simulation using a custom interface and thermostat control logic and linked with standard TRNSYS house, weather, and DHW tank models. The available house for the analysis was a tight-well insulated 242 m^2 (2600 ft^2) three-bedroom unit with 7 kW (~2-ton) design cooling load; as such we scaled the performance maps from 10.6 to 7 kW (3- to 2-tons) in nominal capacity size.

The DHW tank was a nominal 189 L (50 gallon) capacity. The DHW tank was modeled using a TRNSYS Type 534 module, which models a vertical cylindrical water tank. The tank is divided into 6 isothermal temperature nodes (to model stratification observed in storage tanks) where each constant-volume node is assumed to be isothermal and interacts thermally with the nodes above and below through several mechanisms; heat conduction between nodes and through fluid movement (either forced movement from inlet flow streams or natural destratification mixing due to temperature inversions in the tank). Mechanical ventilation per ASHRAE STD 62.2 (2007) was assumed to be provided by continuous operation of a bathroom ventilation fan.

DHW controls for heat pump dedicated WH operation in the analysis were set to operate until the lower tank temperature was 50°C (122°F) and the upper electric element was set to minimize electric element use while maintaining the upper tank delivery temperature above 41°C (105°F). The assumed daily use schedule shown in Figure 8 includes discrete tempered [41°C (105°F)] and untempered hot water draws totaling ~243 L/day (~64.3 gal/day), which is consistent with the Department of Energy (DOE 2010) daily hot water draw totals for electric resistance and HPWH Energy Factor testing.

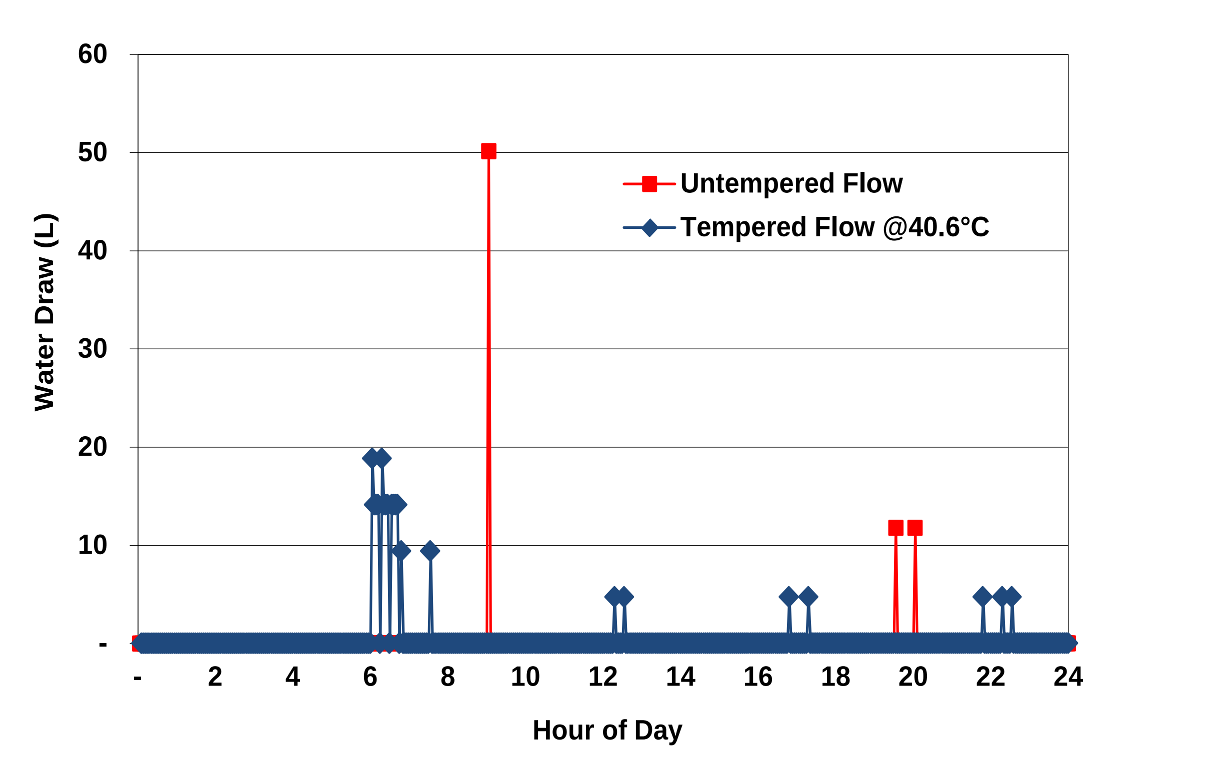

Figure 8. Assumed Daily Hot Water Draw Schedule from DHW Tank

To determine the energy savings potential of the AS-IHP design, a suitable baseline all-electric case was defined and its annual performance simulated in TRNSYS. This consisted of a 7 kW (2-ton) fixed capacity air-source heat pump (ASHP) with a 3.8 W/W CSPF (13 SEER Btu/Wh), a 2.3 W/W HSPF (7.7 HSPF Btu/Wh) in combination with a 0.90 W/W Energy Factor electric water heater. The ASHP performance was represented in TRNSYS as a function of ambient and indoor conditions based on a manufacturer's published data.

The time steps in TRNSYS for AS-IHP seasonal performance analysis were set at 3.0 minutes between thermostat call priority decisions. In our initial analysis, control logic rules were applied, as in the AS-IHP concept report by Murphy et al 2007b to give priority to water heating when both space and water heating calls were active if the indoor DB was within 1.1°C (2°F) of the heating mode set point. Dedicated WH operation was however constrained to a specified minimum ambient due to refrigerant discharge temperature limits. Simulations were run for five Building America climate regions (U.S. DOE 2013) of mixed-humid (Atlanta), hot-humid (Houston), hot-dry (Phoenix), marine (San Francisco), and cold (Chicago). The HVAC-WH energy savings predictions for the reference house in the 5 climates averaged 52%, ranging from 46 to 61% for Chicago and San Francisco, respectively. The average space conditioning savings exceeded 40% while the average water heating savings were 67%.

Following these results, work proceeded to develop and test a second lab prototype design using more compact microchannel and brazed-plate air- and water-to-refrigerant HXs, respectively, of similar performance and revised indoor compressor and HX module packaging. This unit was run through similar steady-state laboratory testing in the various operating modes and the data used to re-calibrate the HPDM. A more efficient BPM pump was also assumed in this analysis. A DHW pump power relationship as a function of water flow rate was developed based on matching manufacturer's performance curves for a brushless permanent-magnet (BPM) pump against manufacturer's system head curves for an assumed DWH loop head characteristic. The required power for the pump was lowered by ~ 60% at full flow and ~80% at reduced flow. A plot of the pump power and assumed head curve versus water flow for the two pump options is shown in Figure 9. With the slightly stronger BPM pump, the full flow level was increased from 1.2 to 1.36 m^3/h (5.3 to 6 gpm). Note that the low flows below 0.15 m^3/h (0.66 gpm) needed for desuperheater operation required an added flow restriction in the system water line as shown by the change in the system head curve below 0.75 m^3/h (3.3 gpm) .

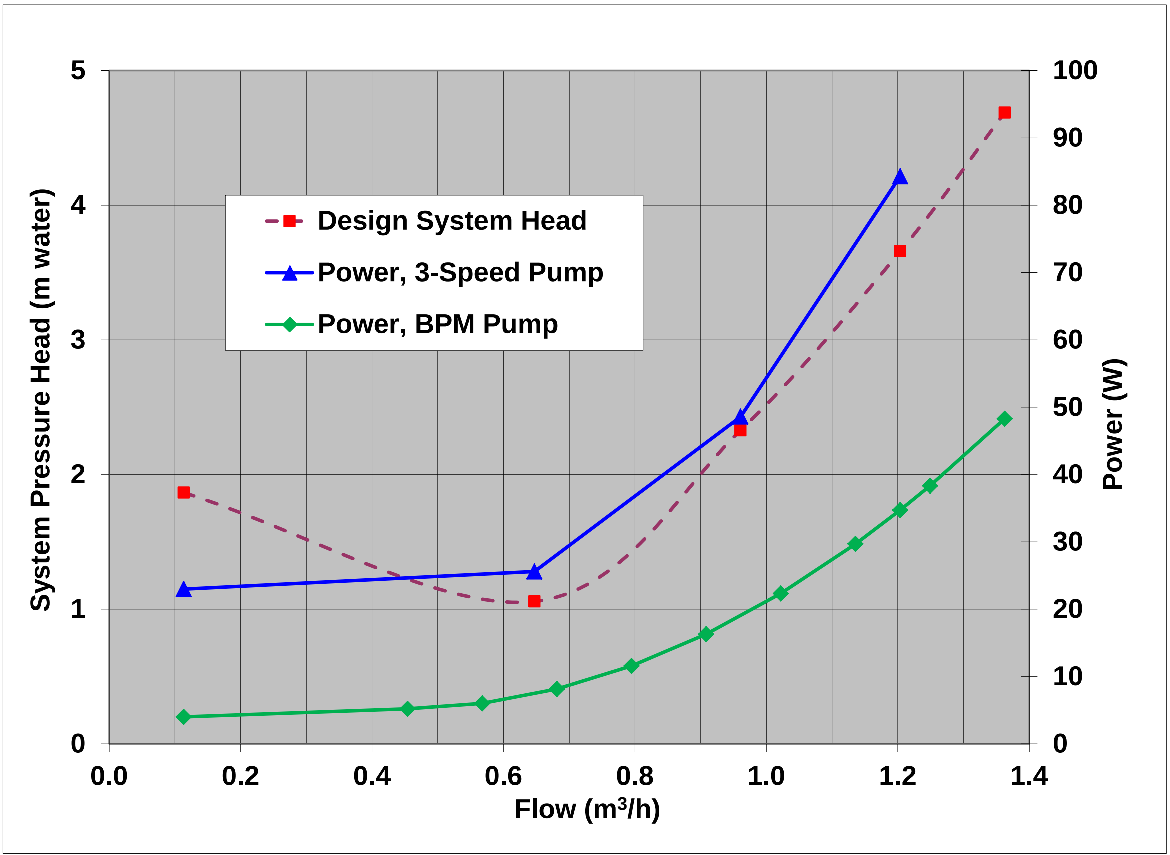

Figure 9. Power Comparisons Versus Flow Between 3-Speed Induction and BPM Pumps for Design Head Requirements

Parametric runs varying condenser subcooling and water flow with the BPM pump were used to determine the optimal levels for the full condensing water heating modes. Figure 10 shows that the combined EER in SC+WH mode (both cooling and WH outputs / input power) has a distinct peak near the maximum water flow rate available from the BPM pump, as shown by the **bold 'X'.**

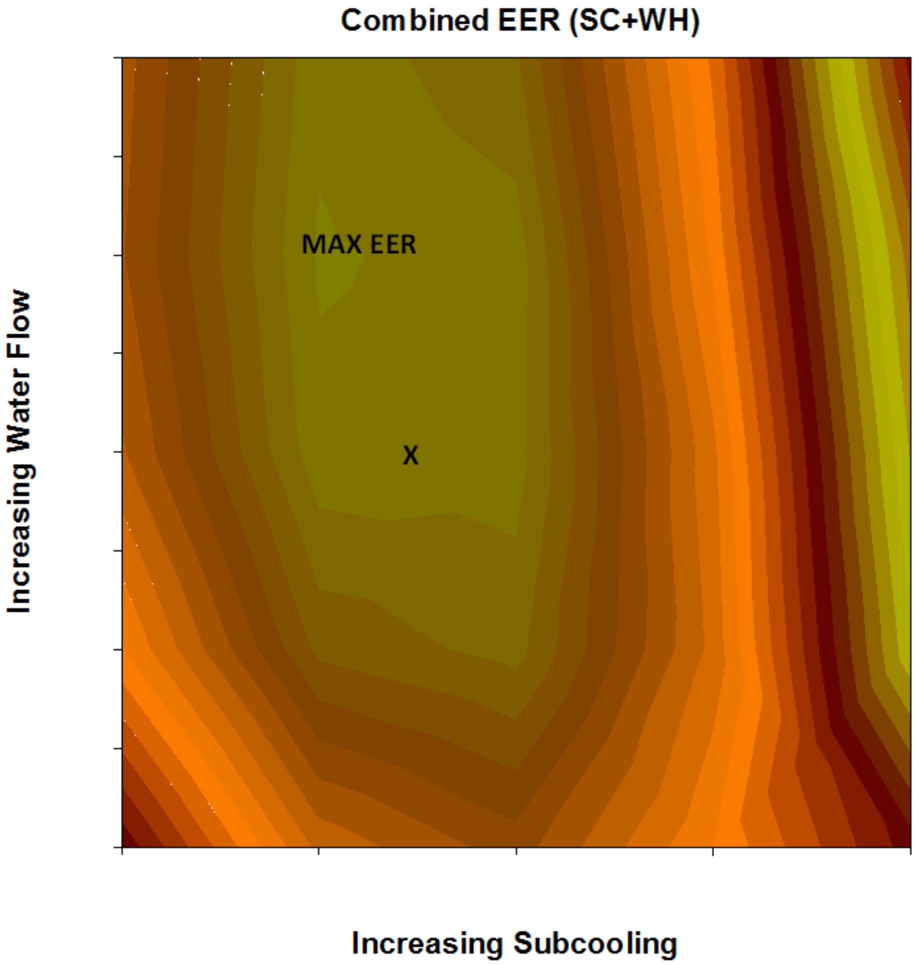

Figure 10. Optimal Water Flow Versus Condenser Subcooling for BPM Pump and Brazed Plate HX in Second Prototype Design

The re-calibrated HPDM was used to generate performance maps for the second prototype AS-IHP unit that were used for a second round of annual performance simulations with TRNSYS. As compared to the analyses for prototype 1, thermostat control priority was given in winter operation to space heating with water heating limited to desuperheating and electric elements until the space heating load is satisfied. This approach gave better control of the indoor temperature in the winter season than the previous approach with water heating priority. Dedicated water heating (using the outdoor coil as a source) is limited to operation above a specified cutoff ambient, when no space heating call is active, and in shoulder months when the ambient is below a specified cutoff. In space cooling mode, desuperheating is used first when a WH call is active, until a prescribed water draw is reached, when the unit will switch to combined SC+WH operation.

Results of these annual performance simulations for the five cities are shown in Table 1. The entries in red show the portion of the total energy use for that mode that was from resistance heat. Total HVAC/WH energy savings relative to the all-electric baseline unit again averaged 52%, with a similar range in total savings between the cold and marine climates as before. The predicted average space conditioning savings are 42% with average WH savings of 70%. The increase in predicted WH savings from the first prototype analysis is attributed mainly to the more efficient pump assumed for newer design and a lower allowed ambient limit for dedicated WH operation.

Table 1. Energy Use and Savings Predictions for AS-IHP Lab Prototype 2 Design

Energy Use by Mode, 242 m^2 Tight, Well-Insulated House			
	Equipment Performance		
	Baseline	Prototype AS-IHP	
Operation Mode	Energy Use, kWh (I^2R)	Energy Use, kWh (I^2R)	Savings from Base (%)
Atlanta			
space heating	2314	1359	41.2%
resistance heat	(42)	(0)	
space cooling	1566	905	42.2%
water heating	3293	987	70.0%
resistance heat	(3293)	(324)	
ventilation fan	189	189	
totals	7361	3440	53.3%
Houston			
space heating	1062	598	43.6%
resistance heat	(3)	(0)	
space cooling	2498	1480	40.7%
water heating	2728	664	75.7%
resistance heat	(2728)	(121)	
ventilation fan	189	189	
totals	6476	2931	54.7%
Phoenix			
space heating	724	398	45.0%
resistance heat	(1)	(0)	
space cooling	3395	2320	31.7%
water heating	2392	665	72.2%
resistance heat	(2392)	(117)	
ventilation fan	189	189	
totals	6700	3572	46.7%
San Francisco			
space heating	1304	703	46.1%
resistance heat	(1)	(0)	
space cooling	21	11	44.8%
water heating	3676	1126	69.4%
resistance heat	(3676)	(361)	
ventilation fan	189	189	
totals	5189	2030	60.9%
Chicago			
space heating	6287	3974	36.8%
resistance heat	(1037)	(474)	
space cooling	623	340	45.5%
water heating	4110	1545	62.4%
resistance heat	(4110)	(691)	
ventilation fan	189	189	
totals	11209	6048	46.0%

4 CONCLUSIONS AND RECOMMENDATIONS

A variable-speed AS-IHP prototype unit has been developed for residential HVAC and WH application. Two generations of lab prototypes have been fabricated, tested, and annual performance simulated for five U.S. locations with predicted average total energy savings of >50%, relative to a minimum efficiency all-electric baseline system. The variable capacity control maximizes efficiency for cooling and moderate space heating mode conditions (moderate outdoor temperatures). Additionally it enables overspeed operation for cold outdoor temperatures in space heating mode to reduce supplemental resistance heat requirements. By the use of desuperheating, dedicated water heating, and water heat recovery in the summer, the design can provide efficient, high capacity water heating, with average water heating savings of 70%. A third generation prototype has been fabricated based on the lab test and analysis results reported above. Field testing of this unit in Knoxville, TN is planned for the 2014 cooling and heating seasons.

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