Compressor Selection and Equipment Sizing for Cold Climate Heat Pumps

Bo Shen, Omar Abdelaziz, C. Keith Rice
Building Equipment Research Group, Building Technologies Research and Integration Center, Oak Ridge National Lab, TN, USA

Abstract: An extensive array of air source heat pump design and sizing options were investigated to identify solutions to limit heating capacity degradation at -13°F (-25°C) ambient to ≤25%, compared to the nominal rating point capacity at 47°F (8.3°C). Sixteen equipment design options were evaluated in one commercial building and one residential building, respectively in seven cities. The energy simulation results were compared to three baseline cases: 100% electric resistance heating, a 9.6 Btu/Wh (2.8 W/W) heating seasonal performance factor (HSPF) single-speed air-source heat pump unit, and 90% AFUE gas heating system. The general recommendation is that variable-speed compressors and tandem compressors, sized such that their rated heating capacity at a low speed matches the building design cooling load, are able to achieve the capacity goal at low ambient temperatures by over-speeding. For example, in a home with a 3.0 ton (10.6 kW) design cooling load, a tandem heat pump could meet this cooling load running a single compressor, while running both compressors to meet heating load at low ambient temperatures in a cold climate. Energy savings and electric resistance heat reductions vary with building types, energy codes and climate zones. Oversizing a heat pump can result in larger energy saving in a less energy efficient building and colder regions due to reducing electric resistance heating. However, in a more energy-efficient building or for buildings in warmer climates, one has to consider balance between reduction of resistance heat and addition of cyclic loss.

Key Words: Heat Pump, HVAC Equipment Modeling, Building Energy Simulation, Heat Seasonal Performance Factor

1 INTRODUCTION

In the U. S., there are approximately 14.4 million dwellings that use electricity for heating in very cold and cold regions with an annual energy consumption of 0.16 quads (0.17 EJ). A high performance air-source cold climate heat pump (CCHP) would result in significant savings over current technologies (> 70% compared to strip heating). It can result in an annual primary energy savings of 0.1 Quads (0.1055 EJ) when fully deployed, which is equivalent to 5.9 million tons (5.35 million MT) of annual CO_2 emissions reduction. For cold climate heat pumps, the primary market segment consists of existing and new residential buildings in cold climate regions using electricity as primary heating source. Electric resistance furnaces are used in cold climates with limited access to natural gas to provide heating as the result of the severe capacity loss and energy performance degradation experienced by conventional heat pumps in extreme ambient conditions. Resistance heating is limited to a maximum COP of 1.0. Cold climate heat pumps have the capability to provide much higher energy efficiency, i.e. significant energy savings. Presently, current technical and economic barriers limit market penetration of heat pumps in cold climates. R&D efforts can be employed to overcome these barriers and develop high performance CCHPs that minimize, or even eliminate, the need for backup strip heating.

The major R&D challenges are to limit the heat pump heating capacity degradation and enhance the operation efficiency as much as possible at extremely low ambient temperatures. The heating capacity of a regular heat pump using a single-speed compressor decreases drastically as the ambient temperature drops, while the building heating demand increases. Consequently, a regular heat pump can't match the heating demand at extremely low ambient temperatures, and significant supplemental resistance heat has to be used, which decreases the heat pump annual operation efficiency. In addition, Bertsch et al. (2005) pointed out that due to the high pressure ratio and low suction pressure at low ambient temperatures the compressor discharge temperature could exceed maximum compressor operating limits thus limiting the working range of the heat pump. It is difficult to size a CCHP properly - if the rated heating capacity matches house load at low ambient temperatures excessive cycling and efficiency losses occur at moderate ambient temperatures.

Researchers have investigated several cycle configurations for CCHP. Bertsch et al. (2006, 2008) studied two-stage compression with three alternatives, i.e. with oil cooling between the low-stage and high stage compressors; with inter-stage economizing; with inter-stage heat exchanger (cascade cycle). These cycle configurations effectively lower the discharge temperature and maintain a good efficiency at low ambient temperatures. However, those options have not been implemented successfully on the market, due to the oil return difficulty and much higher initial cost. Therefore, our focus is on "conventional" technologies, which can be implemented on a large scale, with reasonable cost increment, to investigate the potential of single-stage compression using available compressors on the market.

Our analyses are based on equipment system modeling and building energy simulation. Heat Pump Design Model (HPDM) is a public-domain, hard-ware-based, heat pump design and modeling tool, developed by Rice et al. (2005). In order to model compressors, HPDM uses AHRI 10-coefficient compressor maps to calculate mass flow rate, power consumption; simulate energy balance from inlet to outlet using the calculated power and given heat loss ratio; it also considers the actual suction state to correct the map mass flow prediction. For heat exchanger modeling, It uses a segment-to-segment modeling approach; Each tube segment has individual air side and refrigerant side entering states, and considers possible phase transition; An ε-NTU approach is used for heat transfer calculations within each segment. Air-side fin is simplified as an equivalent annular fin. Both refrigerant and air-side heat transfer and pressure drop are considered; the coil model can simulate arbitrary tube and fin geometries and circuitries, any refrigerant side entering and exit states, misdistribution, and accept two-dimensional air side temperature, humidity and velocity local inputs. It is capable of modeling both condenser and evaporator.

EnergyPlus 7.2 (2013) is used for the building energy simulations discussed in this paper. EnergyPlus uses off-design performance curves to correct heat pump capacities and power consumptions from the design condition at one speed level (i.e. Cooling: 80°F(26.7°C) indoor dry bulb/67°F(19.4°C) indoor wet bulb, 95°F (35°C) outdoor dry bulb temperature; Heating: at the outdoor temperatures of 47°F (8.3°C) DB/ 43°F (6.1°C) WB and indoor temperature of 70°F (21.1°C)), to account for impacts of off-design temperatures. The curves are normalized to the heat pump rating point performance. To facillitate variable speed modeling capability, EnergyPlus 7.2 requires inputting normalized performance curves at individial speed levels, and linearly interpolates performance between two neighboring speed levels. The real-time operating speed is chosen by the model, matching the coil capacity to the building load.

Our analyses are based on compressor maps provided by a manufacturer, and calibrated system and heat exchanger models of HPDM. HPDM is used to generate off-design performance curves by running steady-state simulations in an extensive working range. The performance curves are used in EnergyPlus for building energy simulations.

2 Heat Pump Equipment Modeling

The base air source heat pump unit we selected for the analyses is a 5-ton, split heat pump, using a single-speed, 5-ton (17.6 kW) scroll compressor, having rated SEER (seasonal energy efficiency ratio) of 13.0 Btu/WH (cooling seasonal performance factor, CSPF, of 3.8 W/W) and HSPF of 9.6 Btu/Wh (2.8 W/W). The unit information, i.e. heat exchangers, lines, fans, etc. was provided by our industry partner. We used HPDM to model the equipment and calibrated the model against the manufacturer's published performance data. After this was done, we kept the same heat exchangers, indoor blower and outdoor fan, and refrigerant connection lines, and evaluated different compressor options including: a 2-stage 5-ton scroll compressor (the top speed provides 5-ton cooling capacity at the rated condition, and ratio of capacity between the low and high stage is 67%/100%); a 2-stage 6-ton scroll compressor (67%/100% capacity ratio); a tandem 8-ton scroll compressor (consisting of two identical 4-ton compressors); a variable speed (VS) scroll compressor (having 5-ton nominal cooling capacity at 4500 RPM); a larger size variable speed scroll compressor (20% more mass flow rate and power consumption than the 5-ton VS at each speed); a smaller size, 5-ton tandem scroll compressor (consisting of two identical 2.5-ton compressors); and a 5-ton single-speed vapor injection (VI) scroll compressor. It should be noted that the variable speed compressors investigated have speed ranges of 1800 RPM to 7200 RPM.

The refrigerant charge of a typical heat pump unit is normally optimized in cooling mode. This results in a larger than needed charge (overcharged) for heating mode with very high subcooling and discharge temperature, thus the heating performance is impaired and the working range is limited. In order to design a cold climate heat pump, we want to optimize the heating performance, rather than the cooling performance. In addition, for a modulated system with significant refrigerant mass flow rate variation, we need an adjustable expansion device. At low ambient temperature, we want to minimize the evaporator superheat degree to elevate the suction saturation temperature for better efficiency. Considering these, we use a suction line accumulator to maintain the evaporator exit at saturated state, and an electronic or thermostatic expansion valve (EXV or TXV) refrigerant flow control device to control the condenser exit subcooling to 10 R (5.6 K). These two measures improve the heating efficiency as well as keeping the discharge temperature below 270°F (130°C), with the heat pump running down to the ambient temperature of -13°F (-25°C).

The next modeling step involved studying the interaction of the building envelope and operating schedule with the equipment and ambient conditions. This involves sizing strategies, i.e. determining which speed level to match the heat pump cooling capacity to the building design cooling load. In Table 1, we list sixteen equipment design and sizing options. The table also describes hardware information and the particular sizing strategy used to set up the equipment model for each option.

Each design and sizing option is assigned a unique name using the following naming convention:

CMPTYPE_SIZINGRATIO_HEXSCALE_CMPSCALE

Where,

→CMPTYPE compressor type: 1S means single-stage compressor; 2S means two-stage compressor; 2T means tandem compressor; VS means variable-speed compressor; VI means single-speed, vapor injection compressor.

→SIZINGRATIO = ratio of "total" cooling capacity to match the building design cooling load, The "total" capacity is obtained at the high speed for the 1S, 2S, 2T and VI compressors, and at 4500 RPM for the VS compressors (allowing reserving over-speeding capability for heating mode, from 4500 RPM to 7200 RPM). For an air source heat pump unit with a 1S compressor the rated cooling capacity at standard rating conditions (i.e. 95°F/35°C outdoor; and 80°F (26.7°C) DB/ 67°F (19.4°C) WB indoor) is approximately equal to the rated heating capacity rated at outdoor temperatures of 47°F (8.3°C) DB/ 43°F (6.1°C) WB and indoor temperature of 70°F (21.1°C).

→HEXSCALE = scaling factor of indoor and outdoor heat exchangers, in comparison to the heat exchangers used in the 5-ton baseline heat pump.

→CMPSCALE = scaling factor of the compressor "total" cooling capacity, in comparison to the single-speed 5-ton scroll compressor used in the baseline heat pump.

For building energy simulations, normalized performance of the design options were scaled to match the same building design cooling load. i.e.

(Compressor Capacity@100% × SIZINGRATIO) $_{Option\text{-}n}$ *× UnitScale* $_{Option\text{-}n}$ *= Design Load* *(Eq. 1)*

Here the design load is the EnergyPlus autosized Nominal Cooling Capacity for a single-speed unit at standard rating conditions (ambient: 95°F; indoor: 80°F DB/ 67°F WB). The Design Load is determined for the unit to match the peak cooling demand in a cooling design day, specific to each city, as regulated by ASHRAE 90.1-2007 (ASHRAE 2007). The *UnitScale* is the scaling factor for a unit, which is used to uniformly multiply the compressor size and heat exchanger area, to match the building cooling demand.

Table 1: Heat Pump Design and Sizing Options

#	Design and Sizing Options	Practical Scenarios for Equipment Modeling
1	1S-1.00R-1.0HX-1.0CMP	Single speed heat pump having SEER of 13.0, HSPF of 9.6. Using a single-speed 5-ton scroll compressor, and heat exchangers in the 5-ton HP.
2	2S-1.00R-1.0HX-1.0CMP	5-ton HP, using a 2-stage, 5-ton scroll compressor, rated at high speed to match a building design cooling load of 5 ton.
3	2S-0.73R-1.0HX-1.0CMP	5-ton HP, using a 2-stage, 5-ton scroll compressor, rated at low speed to match a building design cooling load of 3.5 ton.
4	2S-0.73R-1.0HX-1.2CMP	5-ton HP, using a 2-stage, 6-ton scroll compressor, rated at low speed to match a building design cooling load of 4 ton.
5	2T-0.57R-1.0HX-1.6CMP	5-ton HP, using a tandem, 8-ton scroll compressor (2*4-ton), rated at low speed to match a building design cooling load of 4 ton.
6	VS-0.64R-1.0HX-1.2CMP	5-ton HP, using a 6-ton, variable-speed scroll compressor, rated at 2700 RPM to match a building design cooling load of 4 ton. (working speed from 1800 RPM to 7200 RPM)
7	VS-0.82R-1.0HX-1.2CMP	5-ton HP, using a 6-ton, variable-speed scroll compressor, rated at 3600 RPM to match a building design cooling load of 5 ton.
8	VS-1.00R-1.0HX-1.2CMP	5-ton HP, using a 6-ton, variable-speed scroll compressor, rated at 4500 RPM to match a building design cooling load of 6 ton.
9	VS-0.64R-1.0HX-1.0CMP	5-ton HP, using a 5-ton, variable speed scroll compressor, rated at 2700 RPM to match a building design cooling load of 3 ton.
10	VS-0.82R-1.0HX-1.0CMP	5-ton HP, using a 5-ton, variable speed scroll compressor, rated at 3600 RPM to match a building design cooling load of 4 ton.
11	VS-1.00R-1.0HX-1.0CMP	5-ton HP, using a 5-ton, variable speed scroll compressor, rated at 4500 RPM to match a building design cooling load of 5 ton.
12	2S-0.80R-1.0HX-1.0CMP	5-ton HP, using a 2-stage, 5-ton scroll compressor, to match a building design cooling load of 4 ton.
13	2T-0.57R-1.0HX-1.0CMP	5-ton HP, using a tandem, 5-ton scroll compressor (2*2.5-ton), rated at low speed to match a building design cooling load of 3 ton.
14	2S-0.59R-1.0HX-1.0CMP	5-ton HP, using a 2-stage, 5-ton scroll compressor, to match a building design cooling load of 3.0 ton.
15	VI-1.00R-1.0HX-1.0CMP	5-ton HP, using a 5-ton, single-speed VI scroll compressor, to match a building design cooling load of 5.0 ton.
16	1S-0.50R-2.0HX-2.0CMP	Two identical single speed heat pumps, having SEER of 13.0, HSPF of 9.6: only one unit used for cooling mode and one or both for heating mode based on the building heating demand.

Table 2 lists nominal heating capacities at 47°F, which are mostly obtained at low speed levels, and approximately equal to the selected nominal cooling capacity at 95°F. We also list capacity degradations at -13°F, obtained using the maximum heating capacity (at the highest speed) of the system at -13°F, divided by the nominal heating capacity at 47°F. The system heating COPs at 47°F are approximately around 4.0, and the COPs at -13°F are around 2.0 (50% reduction). We also calculated HSPFs (Heating Seasonal Performance Factor), following AHRI 210/240 standard (AHRI 2010), respectively for region IV and V. When calculating the HSPFs, the degradation coefficient (C_d) of the 1S and VI systems is assumed to be 0.1, C_d of the 2S systems is assumed to be 0.13, C_d of the 2T systems is assumed to be 0.17, and C_d of the VS systems is assumed to be 0.20, those are empirical numbers recommended by a compressor manufacturer. Equipment cyclic operation only occurs at the lowest speed, except for Option 16 (1S-0.50R-2.0HX-2.0CMP) where cyclic losses of the two identical single-speed HP units are considered for both the capacity levels. Regarding frosting/defrosting (F/D) operations, we follow the recommendation of AHRI 210/240, i.e. neglecting F/D losses when the ambient temperature is higher than 47℉ and below 17℉ (power and capacity correction factors being 1.0); assuming 0.9 capacity correction factor, 0.985 power correction factor, to the steady-state performance at 35℉ and linearly interpolating the capacity and power correction factors, respectively, from 35℉ to 47℉ and from 35℉ to 17℉. Also, we ignore the power used for the crank-case heater. The same treatment for cyclic losses and F/D operations is used in EnergyPlus building simulations. Table 2 shows 8 options having capacity degradation at -13℉ smaller than 30%, highlighted in green. They are four variable-speed systems (#6: VS-0.64R-1.0HX-1.2CMP, #7: VS-0.82R-1.0HX-1.2CMP, #9: VS-0.64R-1.0HX-1.0CMP, and #10: VS-0.82R-1.0HX-1.0CMP) sized to match building design cooling load at 2700 RPM or 3600 RPM, two tandem systems (#5: 2T-0.57R-1.0HX-1.6CMP, #13: 2T-0.57R-1.0HX-1.0CMP) sized to match building design cooling load at the lowest speed, one two-speed system to match the design cooling load even below the system capacity at the low speed (#14: 2S-0.59R-1.0HX-1.0CMP), and a system composed of two identical 9.6 HSPF heat pumps (#16: 1S-0.50R-2.0HX-2.0CMP). We call these 8 options as CCHPs.

Table 2: System Indices Predicted by HPDM and HSPF Calculations by AHRI 210/240

Design and Sizing Options	Nom COP @47 F	Nom Capacity @47 F (DHR)	Capacity Ratio @ -13 F	COP @ -13 F	HSPF - DHR_{min}	HSPF - DHR_{max}	HSPF - DHR_{min}	HSPF - DHR_{max}
	[-]	[kBtu/h]	[-]	[-]	Region IV	Region IV	Region V	Region V
1. 1S-1.00R-1.0HX-1.0CMP	3.58	62.72	40%	1.92	9.55	7.35	8.42	6.68
2. 2S-1.00R-1.0HX-1.0CMP	3.79	58.06	42%	2.09	9.96	8.08	8.66	6.72
3. 2S-0.73R-1.0HX-1.0CMP	3.78	42.55	57%	2.09	9.98	7.83	8.92	7.87
4. 2S-0.73R-1.0HX-1.2CMP	3.45	49.31	58%	1.91	9.28	8.64	8.37	7.42
5. 2T-0.57R-1.0HX-1.6CMP	3.89	47.82	77%	1.85	10.43	9.41	9.44	8.09
6. VS-0.64R-1.0HX-1.2CMP	4.06	43.70	94%	1.80	11.41	10.56	10.44	8.88
7. VS-0.82R-1.0HX-1.2CMP	3.90	55.89	74%	1.80	11.48	9.77	10.14	8.00
8. VS-1.00R-1.0HX-1.2CMP	3.51	68.01	61%	1.80	11.26	8.97	9.86	8.00
9. VS-0.64R-1.0HX-1.0CMP	4.30	37.99	94%	1.89	11.61	10.93	10.42	9.47
10. VS-0.82R-1.0HX-1.0CMP	4.14	48.37	74%	1.89	11.71	10.06	10.34	8.43
11. VS-1.00R-1.0HX-1.0CMP	3.80	59.16	61%	1.89	11.59	9.61	10.11	7.82
12. 2S-0.80R-1.0HX-1.0CMP	3.79	46.45	52%	2.09	10.05	8.97	8.88	7.54
13. 2T-0.57R-1.0HX-1.0CMP	4.38	36.53	75%	1.98	11.31	10.71	10.01	8.84
14. 2S-0.59R-1.0HX-1.0CMP	3.78	34.04	71%	2.09	9.92	9.73	8.87	8.41
15. VI-1.00R-1.0HX-1.0CMP	3.75	61.64	43%	2.12	10.09	7.59	9.00	6.95
16. 1S-0.50R-2.0HX-2.0CMP	3.58	62.72	80%	1.92	N/A	N/A	N/A	N/A

3 EnergyPlus Building Energy Simulations

To facilitate simulation of a variable speed heat pump system, the performance maps predicted by HPDM are reduced at individual compressor speeds, for the variable-speed systems, the speeds are selected at five levels of 1800 RPM (minimum), 2700 RPM, 3600 RPM, 4500 RPM (100%) and 7200 (maximum). The curves cover -25°F (-32°C) to 63°F (17.2°C) for outdoor dry bulb temperatures with 70% relative humidity, and 60°F (15.6°C) to 74°F (23.3°C) for indoor dry bulb temperatures.

We selected one small commercial lodging building, and one residential single-family detached house with heated basement for the building simulations. The input files of the small commercial lodging building were produced using the EnergyPlus Example File Generator (developed by NREL), for seven cities, i.e. Chicago, IL; Boulder, CO; Helena, MT; Minneapolis, MN; Duluth, MN; Fairbanks, AK, and Indianapolis, IN. The building is in a rectangular shape, with length of 131 feet (40 m) and width of 66 feet (20 m), and it has one floor and five zones, and each individual zone is conditioned by one heat pump unit. The building envelope characteristics, wall thickness, window sizing, etc., were chosen for each climate zone, according to ASHRAE 90.1-2007 (ASHRAE 2007). The input files of the residential, single-family house were converted from Residential Prototype Building Models (developed by PNNL), for seven cities: Minneapolis, MN; Duluth, MN; Fairbanks, AK; Helena, MT; Indianapolis, IN; Peoria, IL and Eagle County, CO. The residential house used one heat pump unit for heating and cooling. We intended the residential building case to represent a retrofit application, and thus, the building envelope characteristics were chosen to match requirements in the 2006 International Energy Conservation Code (International Code Council 2006) . For both buildings in heating season, the zone temperatures are uniformly controlled at 70.0°F. We evaluated the 16 design and sizing options, as listed in Table 1, using the two buildings and allowing EnergyPlus to auto-size the equipment at a selected modulation level to match the building design cooling load, as described in Equation 1.

The small commercial lodging building and the residential single-family house not only differ in energy codes, but also, in relative ratios between the design cooling and heating loads. The commercial building has a larger percentage of interior lighting energy and more human occupants per unit indoor area, than the residential building. Consequently, the ratio of the design cooling load to the design heating load in the commercial building is larger than the residential building. In other words, a heat pump unit, sized in cooling mode, is more capable of meeting the heating demand in the commercial building, and uses less electric resistance heat. As illustrated in Figure 1 in the commercial building CCHPs can reduce resistance heat use below 10% for Fairbanks, and below 3% for Minneapolis, in comparison to the total annual heating energy consumption (heat pump energy consumption + supplemental heating). However for the residential building used in the analyses, as shown in Figure 2, using the same options, resistance heat use can only be reduced to 30% for Fairbanks, and to 10% for Minneapolis. In Figure 3, we compare the average HSPFs of Minneapolis, simulated by EnergyPlus, to the calculated HSPFs, following AHRI 210/240 standard, for Region V. One can see that HSPFs calculated with EnergyPlus for the commercial building in Minneapolis lie approximately midway between those for the AHRI 210/240 DHRmin and DHRmax lines. On the other hand, annual HSPFs of the residential building in Minneapolis almost coincide with the DHRmax curve.

Note: DHR is a terminology defined in AHRI 210/240, which means Design Heating Requirement, representing the heating load at a design ambient condition specific to individual climate zones. DHRmin means minimum design heating load in a representative building with adequate thermal insulation; DHRmax means maximum design heating load in a representative building with inadequate thermal insulation.

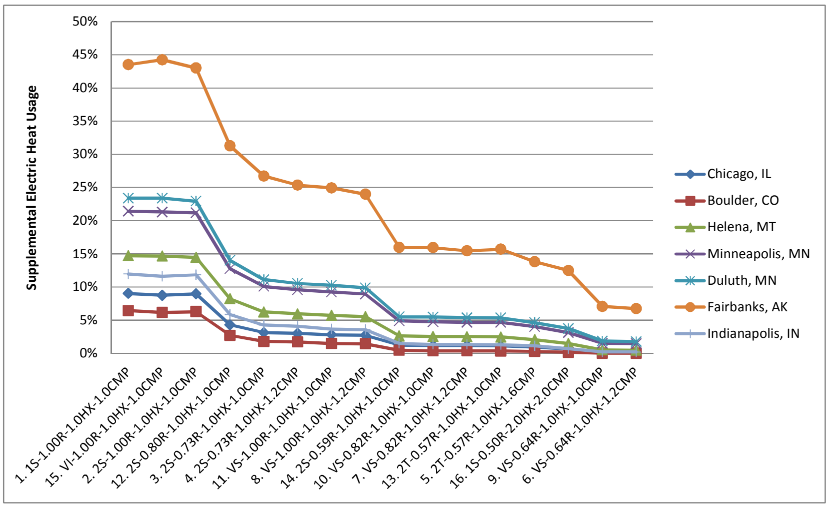

Figure 1: Supplemental Electric Resistance Heat Usages, Relative to Annual Heating Energy Consumption, in Commercial, Small Lodging Building

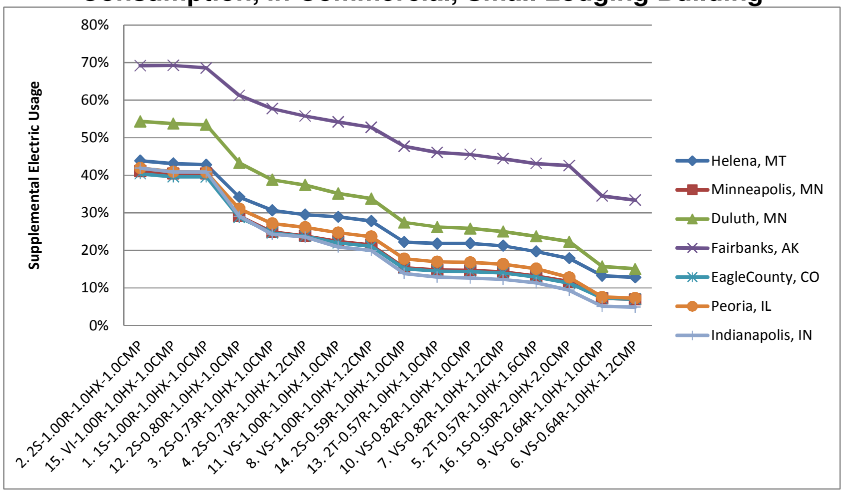

Figure 2: Supplemental Electric Resistance Heat Usages, Relative to Annual Heating Energy Consumption, in Residential, Single-Family Detached House with Heated Basement

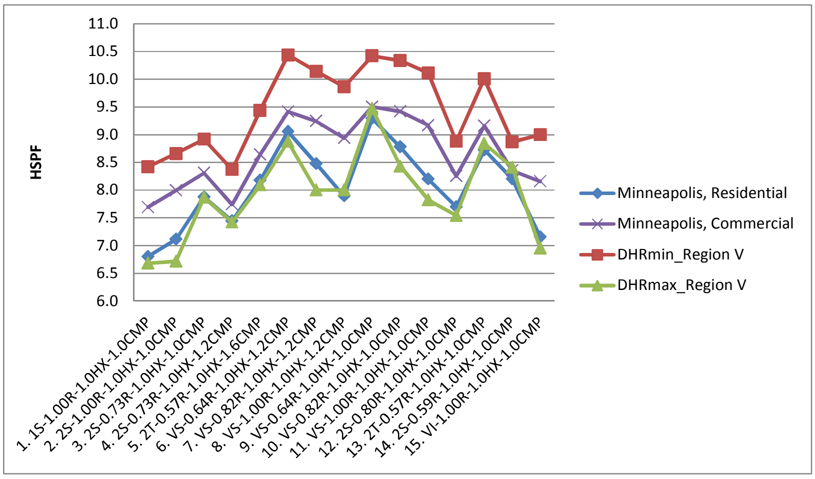

Figure 3: Comparisons of Annual, Average HSPFs (EnergyPlus Simulations) to Calculated HSPFs (AHRI 210/240) in Region V

Figures 4 and 5 illustrate ratios of annual heating energy usage compared to 100% electric resistance heating in the two buildings. Extremely cold regions like Fairbanks and Duluth have smaller percentages of annual heating energy reduction because in these locations the heat pump options require more backup heating. Figures 6 and 7 illustrate ratios of heating energy usage compared to the baseline heat pump. Using one of the CCHP design options to replace the single-speed heat pump appears more beneficial in the residential building than in the commercial building. For the commercial building, the percentages of backup heat uses are smaller, and thus, it has larger percentage of energy savings, compared to 100% electric resistance heat use, than the residential building; however, its benefit of over-speeding is less. Consequently, the relative advantage of a multiple-speed system over a single-speed system is smaller for the commercial building than the residential building.

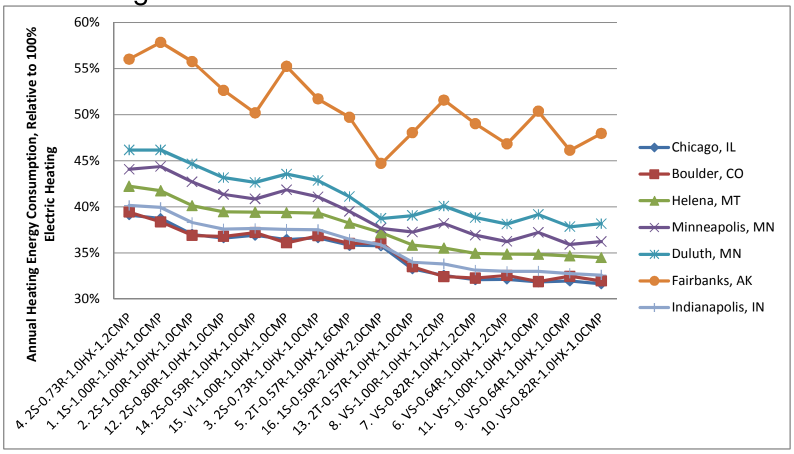

Figure 4: Ratios of Heating Energy Consumption, Compared to 100% Electric Resistance Heating in Commercial, Small Lodging Building

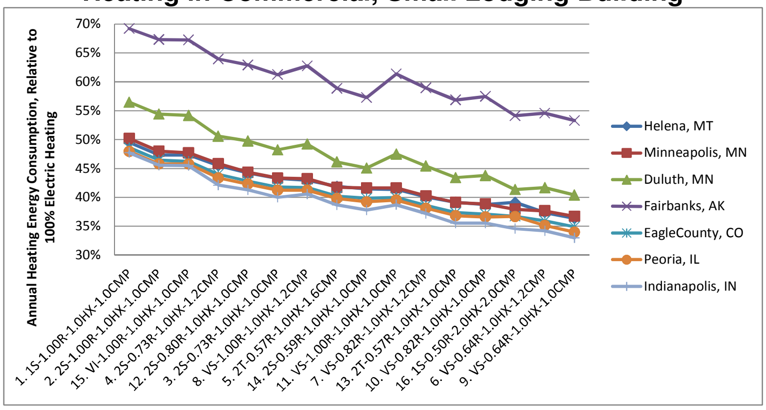

Figure 5: Ratios of Heating Energy Consumption, Compared to 100% Electric Resistance Heating in Residential, Single-Family Detached House

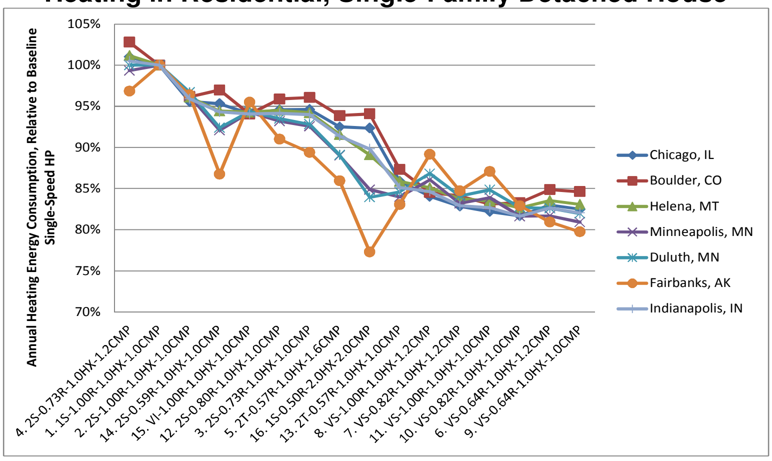

Figure 6: Ratios of Heating Energy Consumption, Compared to Baseline 9.6 HSPF, Single-Speed Heat Pump, in Commercial, Small Lodging Building

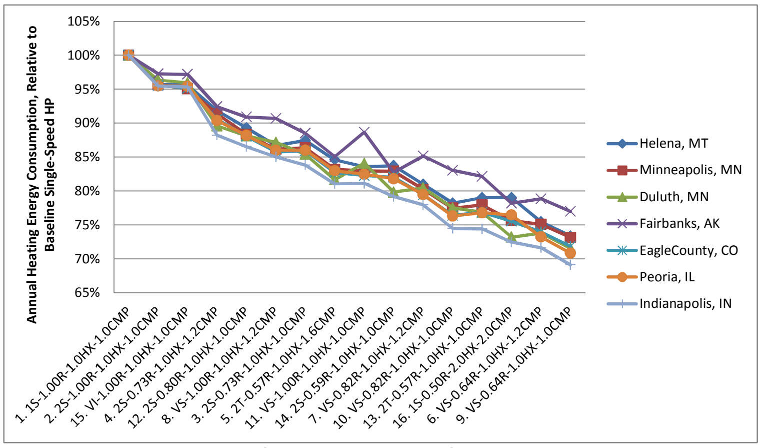

Figure 7: Ratios of Heating Energy Consumption, Compared to Baseline 9.6 HSPF, Single-Speed Heat Pump, in Residential, Single-Family Detached House

Heating energy reductions per ton of building design cooling capacity, in the commercial building, are shown in Figure 8 vs. 100% electric resistance heating. These normalized values can be used to estimate payback periods for each design and sizing option. Assuming electricity cost of \$0.10/kWh, CCHPs can lead to annual savings ranging from up to \$500 per ton (\$142/kW) in Fairbanks to around \$200 per ton (\$52/kW) for Indianapolis. Figure 9 compares heating energy use reductions between the commercial building and residential building. The residential building modeled is less energy-efficient (based on older codes) than the commercial building. In addition, its design cooling capacity is smaller relative to its design heating demand in each city. Hence, heating energy saving potential for the residential building can be almost double that for the commercial building depending upon location and heat pump sizing option.

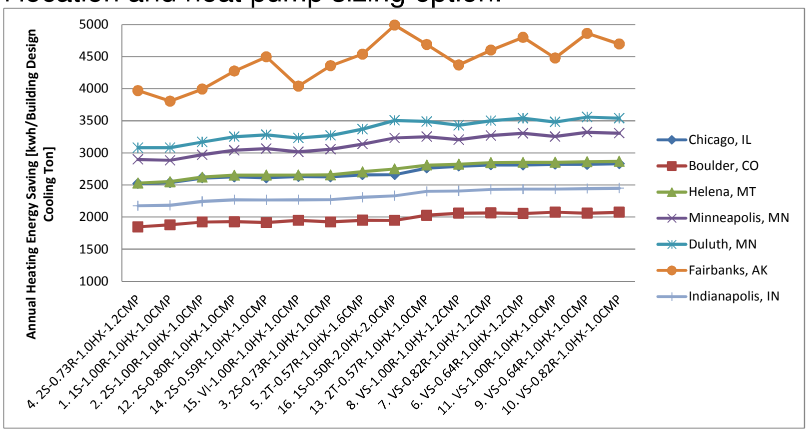

Figure 8: Heating Energy Reductions per Ton of Building Design Cooling Capacity, in Commercial, Small Lodging Building, Compared to 100% Electric Resistance Heating

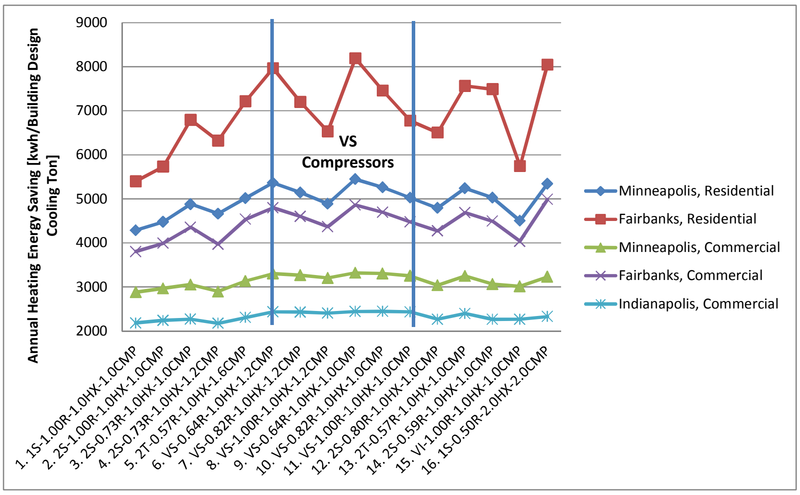

Figure 9: Comparing Heating Energy Reductions per Ton of Building Design Cooling Capacity, between Commercial, Small Lodging Building and Residential, Single-Family Detached House, Compared to 100% Electric Resistance Heating

To meet the goal (Capacity Ratio @-13°F > 75%) with conventional heat pump systems, one would need to oversize the heat pump, either the equipment or the compressor. This will result in less electric resistance heat use at low ambient temperatures but at the expense of reducing operating efficiency at moderate ambient temperatures due to oversizing (cyclic loss, reduced relative heat transfer area, etc.). The system options using VS compressors provide the greatest efficiency since they have a good balance between the oversizing loss and the reduction in the resistance heat usage. The qualified options can maximize heating energy savings in northern cities, for both building types considered. However in warmer locations or with a more energy-efficient building, the benefit of oversizing would be offset by the cyclic loss. For example, for the commercial building in Indianapolis, the trend is flat when selecting the variable-speed systems at different speed levels. Design # 4 (2S-0.73R-1.0HX-1.2CMP) and Design # 12 (2S-0.80R-1.0HX-1.0CMP) even hurt the energy performance, as compared to Design # 2 (2S-1.00R-1.0HX-1.0CMP). That means that the reduction in electric resistance heat use can't make up for the increased cycling losses and lower operation efficiency at high ambient temperature. We can also see from Design # 16 (1S-0.50R-2.0HX-2.0CMP, using two identical single-speed units), it is not a preferred choice for the commercial building in Indianapolis, IN due to increased cyclic losses; however, it stands out for Minneapolis, and Fairbanks, where reducing resistance heat becomes more important.

Figures 10 illustrates comparison to a baseline 90% AFUE gas heating system in terms of source energy (assuming electricity source energy efficiency of 32%). We can see that the only heat pump cases with lower primary energy consumption are the variable-speed heat pumps in warmer climates, e.g. Indianapolis, Peoria, and the other heat pump cases consume more primary energy in the residential building.

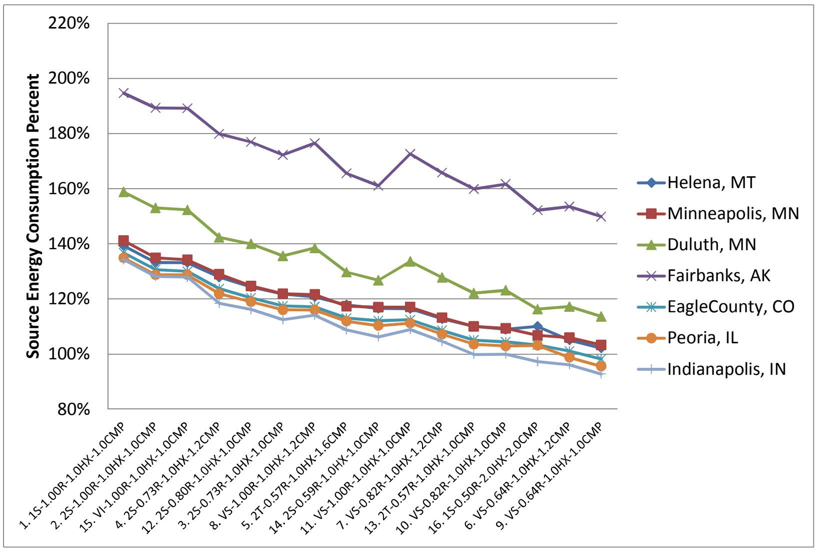

Figure 10: Total Source Energy Consumption Percentages Relative to Baseline 90% AFUE Gas Heating in Residential, Single-Family Detached House

4 CONCLUSION

In the sixteen design configurations listed in Table 1, eight can approximately meet the project heating capacity degradation target and can thus be labled “CCHP designs”. They are more advantageous in colder regions and in less energy-efficient buildings. However, in warmer regions and more energy-efficient buildings, the qualified options might not result in noticeable savings, since the reduction in electric resistance use is offset by the operation efficiency degradation at high ambient temperatures due to oversizing. If a cold climate heat pump is sized according to the building design cooling load, the payback period would be shorter for a residential building than a commercial building, since it has more heating energy reduction per ton of building design cooling capacity. The qualified options can’t totally eliminate resistance heat usage in colder regions, e.g. Minneapolis, Duluth and Fairbanks, because various building types i.e. retrofit or new, result in different percentages of supplemental heating, even in the same climate zone.

In order to develop a cold climate heat pump, using “conventional” vapor compression system technologies, we would recommend using a variable-speed compressor as the best choice. Variable-speed heat pumps can maintain good operation efficiencies at high ambient temperatures and significantly reduce backup heat usages at low ambient temperatures. Tandem compressors, with the nominal capacity rated at the low speed, would be the second best choice.

The “best” concept selection varies with location. For warmer climates like Indianapolis, Boulder, and Chicago, where energy reductions are not sensitive to the equipment sizing ratio, we want to increase the heat pump sizing ratio (as listed in Eq. 1) to match the design load, so as to reduce the equipment cost and shorten the payback period. This means moderately oversizing a heat pump unit in less cold regions (e.g. rating a VS heat pump at 4500 RPM in Indianapolis) and gradually increasing the degree of oversizing with weather getting colder (e.g. rating a VS heat pump at 3600 RPM in Minneapolis, and 2700 RPM in Fairbanks).

To seek further energy savings, one may try improving system operation efficiencies at low ambient temperatures. Recommended design configurations include coupling an ejector cycle with a variable-speed compressor; using a variable-speed vapor injection compressor; using two compressors in series rather than in parallel to reduce pressure ratio of each compressor and provide intermediate cooling for better compression efficiency. Better insulation on the compressor shell and discharge line should be considered to reduce the heat loss.

5 ACKNOWLEDGEMENTS

The authors thank Mr. Antonio Bouza, Technology Development Manager for HVAC, WH, and Appliances, Emerging Technologies Program, Buildings Technology Office at the U.S. Department of Energy for supporting this research project.

6 REFERENCES

AHRI STANDARD 210/240, 2010, "UNITARY AIR-CONDITIONING AND AIR-SOURCE HEAT PUMP EQUIPMENT," Air Conditioning, Heating, and Refrigeration Institute, Arlington, VA, USA

ANSI/ASHRAE/IES Standard 90.1-2007, "Energy Standard for Buildings Except Low-Rise Residential Buildings," American Society of Heating, Refrigeration, and Air-Copnditioning Engineers, Atlanta, GA, USA

ANSI/AHRI Standard 540-99, 2008, "Positive Displacement Refrigerant Compressors and Compressor Units", Air-Conditioning, Heating, and Refrigeration Institute, Arlington, VA

Bertsch, S. S. (2005). Theoretical and experimental investigation of a two stage heat pump cycle for nordic climates (Doctoral dissertation, Master's thesis, Mechanical Engineering, Herrick Labs 2005-13P, Report). Purdue University, West Lafayette, IN, USA

Bertsch, S. S., and Groll, E. A. (2006). Air Source Heat Pump for Northern Climates Part I: Simulation of Different Heat Pump Cycles. in Proceedings of the 2006 International Refrigeration and Air Conditioning Conference, Purdue University, West Lafayette, IN, USA, July.

Bertsch, S. S., and Groll, E. A. (2008). Two-stage air-source heat pump for residential heating and cooling applications in northern US climates. International Journal of Refrigeration, 31(7), 1282-1292.

LBNL. 2012. EnergyPlus Example File Generator, Lawrence Berkeley National Laboratory, Berkeley, CA, USA, http://apps1.eere.energy.gov/buildings/energyplus/cfm/inputs/

International Code Council (2006). "2006 International Energy Conservation Code," International Code Council, Washington, DC. http://www.iccsafe.org

PNNL. 2012. Residential Prototype Building Models, Pacific Northwest National Laboratory, Richland, WA, USA, http://www.energycodes.gov/development/residential/iecc_models

Rice, C. K.and W.L.Jackson, 2005. DOE/ORNL Heat Pump Design Model, Mark VI, Oak Ridge National Laboratory, Oak Ridge, TN, USA. http://www.ornl.gov/~wlj/hpdm/MarkVI.shtml