Inverter Drive Control and Seasonal Performance Analysis of a Single Speed Unitary Air-Source Split-System Heat Pump

Nicholas P. Salts[a*], Eckhard A. Groll[a]

[a]Purdue University, Ray W. Herrick Laboratories 177 S. Russel St., West Lafayette 47907, USA
*Corresponding Author, email: nsalts@purdue.edu

Abstract

Heat pumps utilizing variable-speed compressors and other capacity control techniques offer significant seasonal efficiency improvements over traditional single speed systems by reducing system cycling and consequent system inefficiencies. Yet, the majority of residential heat pumps operated in the United States utilize single-speed compressors and offer few capacity control options. This situation presents an opportunity for existing single speed units to be modified or retrofitted with variable speed technology. The main focus of this research is to optimize the seasonal cooling performance of a 5 ton residential split-system single-speed heat pump that has been modified with a variable-speed drive. The variable-speed drive is an inverter drive coupled with a controller that allows the user to control the compressor and fan motor speeds. The purpose of the variable-speed drive is to improve the seasonal performance of a system by operating traditionally single-speed components in variable-speed mode. Initial tests of the inverter drive have achieved SEER improvements of up to 10% compared to the baseline heat pump. A method has been developed to optimize the SEER rating of the modified heat pump as a function of compressor speed, outdoor fan speed, and indoor airflow rate. Results of this optimization indicate that improvements in SEER of greater than 10% can still be achieved by modifying the single-speed heat pump with the variable-speed drive.

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Keywords: inverter drive, air-source heat pump, controls, seasonal performance rating

1. Introduction

Residential split-system heat pumps are typically designed to be able to meet the maximum load that a home will see during the cooling season. Throughout the cooling season though, a heat pump will have to meet the demands of a varying load that are lower than the design point. Conventional single-speed heat pumps (SSHPs) meet these part load conditions by cycling on and off at full speed. A more efficient approach to meeting the part load conditions is offered by variable speed heat pumps (VSHPs). VSHPs are equipped with compressors and fans that can adjust the capacity delivered by adjusting the speeds of their motors. By doing this, VHSPs can meet part load conditions, and reduce the losses associated with cycling. Additionally, at certain part load conditions, the heat exchangers or compressor will operate in modes of higher efficiency due to better heat transfer properties or favorable flow characteristics, respectively. The benefits of variable speed technology described here have been well established over many years of research. For example, in 1983 Tassou [1] showed that a speed-controlled compressor can offer efficiency improvements of up to 30% compared to the same heat pump using a single-speed compressor. In 2014, Bush [2] demonstrated that for the same cooling load, a VSHP resulted in a 44% energy savings compared to a SSHP installed in the same home. The U.S. DOE [3] also recommends that homeowners could see a 20-50% reduction in energy use for air conditioning by replacing an older SSHP with a high-efficiency VSHP on the market today.

Although the benefits of VSHPs are well documented, the market is still dominated by SSHPs. In 2014, the maximum available seasonal energy efficiency rating (SEER) of a new air-source heat pump was 26, while the average available SEER was 15.5 (2.5 SEER above the minimum of 13 allowed by the DOE at that time) [4]. Additionally, the expected useful lifespan of heat pumps is estimated between 9 and 15 years [4]. This will contribute to the number of SSHPs in operation due to the fact that VSHPs have only gained popularity in the last few years. This situation presents an opportunity for existing single-speed systems to be modified or retrofitted with variable-speed technology. The main focus of this research is to present the evaluation of an inverter drive that is specifically designed to be used as a modification or a retrofit to existing SSHPs. The inverter drive has been connected to an off-the-shelf unitary split-system SSHP with a SEER rating of 14.0. The inverter modified VSHP allows for the control of system capacity as a function of three variables; compressor speed, indoor fan speed, and outdoor fan speed. To compare the baseline SSHP to the inverter modified VSHP, the SEER has been calculated and compared. A method to optimize SEER as a function of the three variables controlled by the inverter drive has also been developed. The results of this method are shown for the SEER rating as a function of minimum declared compressor speeds. The optimization for SEER as a function of indoor fan speed and outdoor fan speed is still underway.

2. Methods

2.1 Baseline Heat Pump

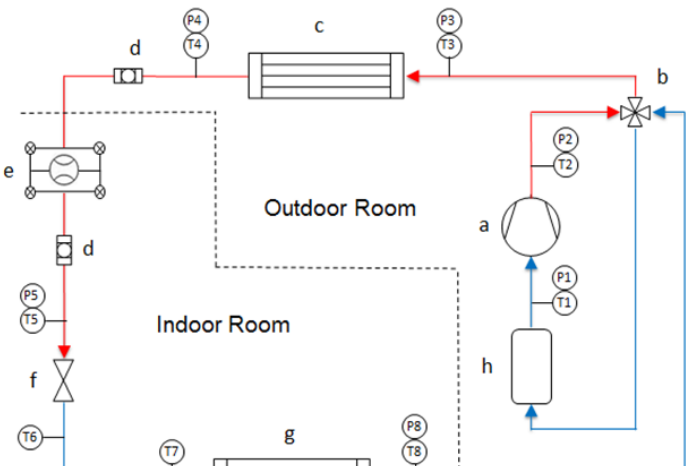
The baseline heat pump has been installed and instrumented in a pair of temperature and humidity controlled psychometric chambers at the Ray W. Herrick Laboratories at Purdue University. The heat pump consists of an indoor unit and an outdoor unit. The indoor unit contains the evaporator, thermostatic expansion valve (TXV), and indoor fan. The outdoor unit contains the compressor, condenser, outdoor fan, four-way valve, and accumulator. The baseline system is a 5 ton heat pump with a 14.0 SEER rating. The nominal indoor airflow rate is 2970 m^3/hr, the nominal compressor speed is 3600 RPM, and the system is charged with 4.7 kg of R410A. The compressor used is a fixed speed scroll compressor. The indoor fan is a constant volume flow rate fan using an ECM motor. The outdoor fan uses a fixed speed motor.

2.2 Inverter Drive Modified Heat Pump

The purpose of the inverter drive is to allow for the variable speed operation of the compressor, indoor fan, and outdoor fan motors on existing SSHPs. The applications for such a device include retrofitting existing systems that are in service or modifying systems that are being newly manufactured. The drive is installed by connecting it in-line between the power source and the components to be speed controlled. Control of the motor speeds during the laboratory experiments is performed manually, and the motor speeds can be adjusted independently of one another. The only modification made to the baseline heat pump is the addition of the inverter drive; all fan and compressor motors are the original baseline components.

2.3 Data Collection and Analysis

The cooling capacity of the heat pump has been determined using both the refrigerant and air enthalpy methods. The power consumptions of the indoor and outdoor unit are measured separately using Watt transducers. Data is collected through a National Instruments cRIO-9074 using a VI written in LabVIEW. All refrigerant and air properties have been determined using the software Engineering Equation Solver (EES.) A schematic of the instrumented refrigerant loop of the heat pump in cooling mode is shown in Figure 1.



a. Compressor
b. Four-way valve
c. Condenser
d. Sight glass
e. Mass flow meter
f. Thermostatic expansion valve
g. Evaporator
h. Accumulator

Fig. 1. Heat Pump Schematic

State points 1 to 5 and 8 in Figure 3.1.1 are determined by measuring the temperature and pressure of the refrigerant. Surface temperature measurements of the refrigerant piping are taken at state points 6 and 7. The surface temperatures are used for observation during testing, but are not used in any calculations. The mass flow rate of the refrigerant is determined using a Coriolis-type mass flow meter. Isenthalpic expansion is assumed across the TXV from state points 5 to 6. Because the pressure of the refrigerant was not able to be measured between the TXV and evaporator, the assumption of no pressure drop is made for the section between the outlet of the TXV and the outlet of the evaporator.

Using the measured state points along with the listed assumptions, it is possible to calculate the refrigerant-side cooling capacity of the evaporator from a first-law energy balance resulting in Equation1.

$$\dot{Q}_{evap} = \dot{m}\Delta h_{evap} = \dot{m}(h_8 - h_7) \quad (1)$$

In order to calculate the net capacity of the system, the power consumption of the indoor unit is assumed to be equal to the heat added to the air by the indoor fan. This heat can be subtracted from the evaporator capacity to give the net capacity of the system shown in Equation 2.

$$\dot{Q}_{net} = \dot{Q}_{evap} - \dot{W}_{indoor} \quad (2)$$

The air-side capacity of the system is calculated from a first-law energy balance across the indoor unit as shown in Equation 3. This equation accounts for the change in enthalpy of the air stream as well as the small amount of water that is condensed out of the air stream.

$$\dot{Q}_{net} = \dot{m}_{air}\Delta h_{air} - \dot{m}_{water} h_{water} \quad (3)$$

The volume flow rate of the air is determined using a nozzle box connected to the duct work of the indoor unit. The nozzle box is designed as outlined in ASHRAE Standard 37-2009, and the volume flow rate is calculated iteratively according to Equation 4, where the expansion factor (Y) and coefficient of discharge (C_i) are also established in ASHRAE Standard 37-2009 [5.]

$$V_{air} = Y\sqrt{\frac{2\Delta p_{nozzle}}{\rho_{air}}}\sum_{0}^{k} C_i A_i \quad (4)$$

The mass flow rate of air is determined by Equation 5.

$$\dot{m}_{air} = \rho_{air} V_{air} \quad (5)$$

The dry-bulb temperature of the air is measured at the inlet and outlet of the unit using a 3x3 grid of thermocouples. The relative-humidity of the inlet air is measured using a relative-humidity sensor. The outlet air is sampled at a rate of 1 L/min and passed through a chilled mirror dew point sensor to determine the dew point of the outlet air. Using these values, the inlet air enthalpy, outlet air enthalpy, and outlet water enthalpy are determined using EES. The mass flow rate of the condensed water is evaluated according to Equation 6.

$$\dot{m}_{water} = \dot{m}_{air}(\omega_2 - \omega_1) \tag{6}$$

The coefficient of performance (COP) and the energy efficiency ratio (EER) are defined as the ratio of cooling capacity delivered to power consumed by the unit. The COP is a unit-less value while the EER has units of Btu/W-hr.

$$COP = \frac{\dot{Q}_{net}}{\dot{W}_{total}}[-] \tag{7}$$

$$EER = \frac{\dot{Q}_{net}}{\dot{W}_{total}}\left[\frac{\text{Btu}}{W \cdot hr}\right] \tag{8}$$

3.3 Test Matrix

Both the baseline and inverter modified heat pumps have been tested according to the ANSI/AHRI Standard 210-240 [6.] The baseline heat pump has been tested according to the guidelines for a SSHP, while the inverter modified heat pump has been tested according to the guidelines for a VSHP. The test matrices outlined in this standard are used for the purpose of determining the Seasonal Energy Efficiency Rating (SEER) of a system. The SEER is a standard method used to rate the cooling performance of heat pumps and air conditioners in the United States. Because the benefits of VSHPs come in the form of their ability to match part load conditions, a seasonal analysis is necessary to characterize the actual benefits of a VSHP versus a SSHP. This paper will therefore use SEER to quantify the improvement in performance of the baseline SSHP to the inverter modified VSHP.

The test matrix shown in Table 1 contains the ratings tests for a SSHP, and the test matrix in Table 2 contains the ratings tests for a VSHP. Each test matrix contains a number of required tests as well as optional tests to determine the coefficient of cyclic degradation (C_D^C). The C_D^C is a factor used to account for the loss in efficiency of the heat pump during periods of cycling on and off. If the optional tests are not performed, the standard calls for a value of 0.25 to be assumed for the C_D^C. Because the SEER calculation requires input temperatures with units of [°F], temperature values have been co-listed in both English and SI units.

Table 1. ANSI/AHRI Test Matrix for a Heat Pump Having a Single Speed Compressor and Constant Volume Indoor Fan

Test description	Air entering indoor unit °F/°C		Air entering outdoor unit °F/°C		Air volume rate
	Dry–bulb	Wet–bulb	Dry–bulb	Wet–bulb	
A (required)	80/26.7	67/19.4	95/35	75/23.9	Full–load
B (required)	80/26.7	67/19.4	82/27.8	65/18.3	Full–load
C (optional)	80/26.7	< 57/13.9	82/27.8	---	Full–load
D (optional)	80/26.7	< 57/13.9	82/27.8	---	Full–load

Table 2. ANSI/AHRI Test Matrix for a Heat Pump Having a Variable Speed Compressor and Indoor Fan

Test description	Air entering indoor unit °F/°C		Air entering outdoor unit °F/°C		Compressor speed	Air volume rate
	Dry–bulb	Wet–bulb	Dry–bulb	Wet–bulb		
A_2 (required)	80/26.7	67/19.4	95/35	75/23.9	Maximum	Full–load
B_2 (required)	80/26.7	67/19.4	82/27.8	65/18.3	Maximum	Full–load
E_V (required)	80/26.7	67/19.4	87/30.6	69/20.6	Intermediate	Intermediate

B_1 (required)	80/26.7	67/19.4	82/27.8	65/18.3	Minimum	Minimum
F_1 (required)	80/26.7	67/19.4	67/19.4	53.5/11.9	Minimum	Minimum
G_1 (optional)	80/26.7	< 57/13.9	67/19.4	---	Minimum	Minimum
I_1 (optional)	80/26.7	< 57/13.9	67/19.4	---	Minimum	

For the testing of the baseline and inverter modified heat pumps, only the required tests were performed. This was done due to the significant time required to carry out the optional tests, as well as having the ability to make a reasonable assumption for the C_D^C. Instead of assuming a value of 0.25, a value was assumed that resulted in the calculated SEER of the baseline SSHP to equal the SEER declared by the manufacturer. This value was found to be 0.11. The C_D^C for the VSHP was assumed to be the same as that calculated for the SSHP.

2.4 SEER Analysis

A brief review of the SEER calculation is presented here in order to provide the reader with a better understanding of the results section of this paper. A full description of the SEER calculation can be found in the ANSI/AHRI Standard 210-240 [6].

The SEER calculation for the baseline SSHP is defined as a function of the EER at Test B conditions and the part-load performance factor (PLF). The PLF accounts for the losses due to cycling and is a function of the coefficient of cyclic degradation (C_D^C). Equations 9-11 outline the SEER calculation for a SSHP.

$$SEER = PLF(0.5) \cdot \mathrm{EER}_{test\ B} \tag{9}$$

$$\mathrm{EER}_{test\ B} = \frac{\dot{Q}_{test\ B}}{\dot{E}_{test\ B}} \tag{10}$$

$$PLF(0.5) = 1 - 0.5 \cdot C_D^{\,c} \tag{11}$$

The SEER calculation for the inverter modified VSHP is defined as the sum of the cooling delivered by the unit divided by the sum of the power consumed by the unit over the entire cooling season as shown in Equation 12. To perform the calculation, the cooling season is divided into 8 bins as shown in Table 3. Each bin is defined by a representative outdoor temperature and fraction of total bin hours for which the heat pump operates at the given bin temperature.

$$SEER_{VSHP} = \frac{\sum_{j=1}^{8} q\left(T_j\right)}{\sum_{j=1}^{8} e\left(T_j\right)} \tag{12}$$

Table 3. Distribution of Fractional Hours within Cooling Season Temperature Bins

Bin Number, j	Bin Temperature Range ºF/ºC	Representative Bin Temperature ºF/ ºC	Fraction of Total Temperature Bin Hours
1………………….	(65-69)/(18.3-20.6)	67/19.4	0.214
2………………….	(70-74)/(21.1-23.33)	72/22.2	0.231
3………………….	(75-79)/(23.8-26.1)	77/25	0.216
4………………….	(80-84)/(26.7-28.9)	82/27.8	0.161

5………………….	(85-89)/(29.4-31.7)	87/30.6	0.104
6………………….	(90-94)/(32.2-34.4)	92/33.3	0.052
7………………….	(95-99)/(35-37.2)	97/36.1	0.018
8………………….	(100-104)/(37.8-40)	102/38.8	0.004

A building load for each bin is defined as a function of the representative bin temperature and the cooling delivered by the heat pump at Test A conditions as shown in Equation 13. The maximum building load is found in bin 8, and the building load decreases with decreasing bin temperatures.

$$BL\left(T_j\right) = \frac{T_j - 65}{95 - 65} \times \frac{\dot{Q}_{test\ A_2}^{\max}}{1.1} \tag{13}$$

For each bin, the cooling delivered and power consumed is then calculated for each of the possible three operating modes: minimum, intermediate, and maximum speed. The equations used for these calculations are shown below in Equations 14-19.

$$\dot{Q}_j^{\min}\left(T_j\right) = \dot{Q}_{test\ F_1} + \frac{\dot{Q}_{test\ B_1} - \dot{Q}_{test\ F_1}}{82 - 67}\left(T_j - 67\right) \tag{14}$$

$$\dot{E}_j^{\min}\left(T_j\right) = \dot{E}_{test\ F_1} + \frac{\dot{E}_{test\ B_1} - \dot{E}_{test\ F_1}}{82 - 67}\left(T_j - 67\right) \tag{15}$$

$$\dot{Q}_j^{\text{int}}\left(T_j\right) = \dot{Q}_{test\ E_V} + M_Q\left(T_j - 87\right) \tag{16}$$

$$\dot{E}_j^{\text{int}}\left(T_j\right) = \dot{E}_{test\ E_V} + M_E\left(T_j - 87\right) \tag{17}$$

$$\dot{Q}_j^{\max}\left(T_j\right) = \dot{Q}_{test\ B_2} + \frac{\dot{Q}_{test\ A_2} - \dot{Q}_{test\ B_2}}{95 - 82}\left(T_j - 82\right) \tag{18}$$

$$\dot{E}_j^{\max}\left(T_j\right) = \dot{E}_{test\ B_2} + \frac{\dot{E}_{test\ A_2} - \dot{E}_{test\ B_2}}{95 - 82}\left(T_j - 82\right) \tag{19}$$

The equations for minimum and maximum speed operation are used to create curves between the measured performance at Tests B_1 and F_1 and Tests A_2 and B_2, respectively. The equation for intermediate speed operation is used to create curves at the measured performance of Test E_v having the slopes M_Q and M_E. The exact formulation for these slopes can be found in the ANSI/AHRI Standard 210-240. To determine the actual operating mode, the building load is compared to the cooling delivered for each operating mode, and the operating mode is selected using the following logic:

- If $BL\left(T_j\right) < \dot{Q}_j^{\min}\left(T_j\right)$, the unit will cycle on and off at minimum speed to match the building load.
- If $\dot{Q}_j^{\min}\left(T_j\right) < BL\left(T_j\right) < \dot{Q}_j^{\max}\left(T_j\right)$, the unit will operate continuously at intermediate speed without cycling to match the building load.
- If $BL\left(T_j\right) < \dot{Q}_j^{\min}\left(T_j\right)$, the unit will operate constantly at maximum speed to try and match the building load.

After determining the actual cooling delivered and power consumed for each bin, the values are adjusted based on the total fractional cooling hours, and the final SEER value is determined. While additional calculation steps exist in the standard, the steps outlined in this section present the primary governing equations used to determine the SEER for a VSHP.

3. Results

Baseline test results for the SSHP are shown in Table 4. The cooling capacity matches the manufacturer declared value of 5 tons (17.6 kW), and a SEER rating of 14.0 is achieved when using a C_D^C equal to 0.11. The assumed coefficient of cyclic degradation used to reach the target SEER rating falls in the range specified by the standard of 0 to 0.25, and is therefore considered to be a reasonable assumption. During baseline testing, a percent difference in cooling capacity between the refrigerant and air enthalpy methods of 2% or less was achieved. The declared cooling capacities in this results section therefore represent those calculated using the refrigerant-enthalpy method. Because compressor speeds are manually controlled using a percent of full speed as the input signal, the compressor speeds reported are also in units of [%] where 3600 RPM represents the nominal compressor speed.

Table 4. Baseline Test Results

Test	Comp. Speed	Ref. Mass Flow Rate	Cooling Capacity	Total Power	Outdoor Power	Indoor Power	Indoor Air Flowrate	Static Head	COP	EER
[-]	[%]	[g/s]	[kW]	[kW]	[kW]	[kW]	[m³/hr]	[Pa]	[-]	[Btu/W-hr]
A2	100	100.7	16.57	4.677	4.275	0.4027	2995	70	3.54	12.08
B2	100	99.4	17.64	4.064	3.655	0.4091	2997	72	4.34	14.81

Table 5 shows the results of testing the inverter modified VSHP with only the compressor operating in variable speed mode. Table 6 shows the results of testing the inverter modified VSHP with both the compressor and indoor fan operating in variable speed mode. The first set of VSHP tests were run at a minimum speed of 55%, an intermediate speed of 70%, and an air volume rate of 1750 cfm. The second set of VSHP tests were run at the same compressor speeds, but at a reduced air volume rate of 1000 cfm.

Table 5. Variable Compressor Speed Test Results

Test	Comp. Speed	Ref. Mass Flow Rate	Cooling Capacity	Total Power	Outdoor Power	Indoor Power	Indoor Air Flowrate	Static Head	COP	EER
[-]	[%]	[g/s]	[kW]	[kW]	[kW]	[kW]	[m³/hr]	[Pa]	[-]	[Btu/W-hr]
Ev	70	75.05	12.93	3.06	2.665	0.3956	2994	70	4.23	14.42
B1	55	60.3	10.71	2.39	2.007	0.383	2985	67	4.48	15.29
F1	55	62.5	11.89	1.997	1.615	0.3821	2983	65	5.95	20.31

Table 6. Variable Compressor and Indoor Fan Speed Test Results

Test	Comp. Speed	Ref. Mass Flow Rate	Cooling Capacity	Total Power	Outdoor Power	Indoor Power	Indoor Air Flowrate	Static Head	COP	EER
[-]	[%]	[g/s]	[kW]	[kW]	[kW]	[kW]	[m³/hr]	[Pa]	[-]	[Btu/W-hr]
Ev	70	66.56	11.87	2.752	2.663	0.08921	1707	32	4.31	14.72
B1	55	55.03	10.08	2.112	2.023	0.08932	1711	32	4.77	16.28
F1	55	55.58	10.93	1.735	1.649	0.08678	1704	32	6.30	21.49

Decreasing the air volume rate had the effect of decreasing the power consumed by the indoor fan and decreasing the cooling capacity delivered at all three test conditions. The net effect of these factors contributed to an overall increase in system efficiency.

Table 7 summarizes the results of the SEER improvement provided by each test mode presented in Tables 5 and 6. A SEER improvement of 5.3% was realized by only using a variable speed compressor, and a SEER improvement of 9.8% was realized by using a combination of the variable speed compressor and indoor fan.

Table 7. SEER Comparison for First 3 Test Cases

Test Case	SEER	Improvement

[-]	[-]	[-]	[%]
Baseline..	14.0	-	-
Variable Compressor Speed............................	14.74	0.74	5.29
Variable Compressor and Indoor Fan Speed...	15.37	1.37	9.79

While the chosen compressor and fan speeds used for this test provide a significant improvement to the SEER rating, the chosen speeds still may not represent the maximum possible improvement in SEER. In order to identify the maximum SEER improvement, a SEER optimization has been carried out as a function of declared minimum compressor speed.

To determine SEER as a function of minimum compressor speed, tests at E_v, B_1, and F_1 conditions were carried out over a range of compressor speeds from 50% to 70%. Previous research indicated that a maximum value of SEER may occur within this range. The results of these tests are plotted in Figure 2, which displays the cooling capacity and total power consumption of the system as a function of compressor speed. Linear trend lines have been used to fit the data points collected from these tests. The cooling capacity in Figure 2 is reported in units of [Btu/hr] and the power consumption is reported in [W], both of which are required input units for the SEER calculation.

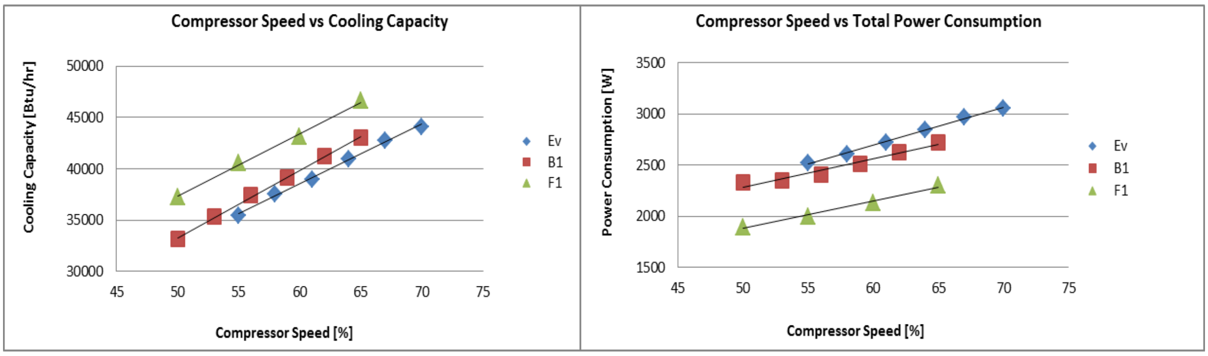


Fig. 2. Compressor Speed vs Cooling Capacity and Total Power Consumption for SEER Optimization

The equations for the trend lines are used to provide the cooling capacity and power consumption data for the SEER optimization. A parametric study of the SEER calculation as a function of the system performance was carried out, where the performance of the system is defined by the trend lines from Figure 2 as well as by the baseline performance at full load. The results of the analysis are shown in Figure 3

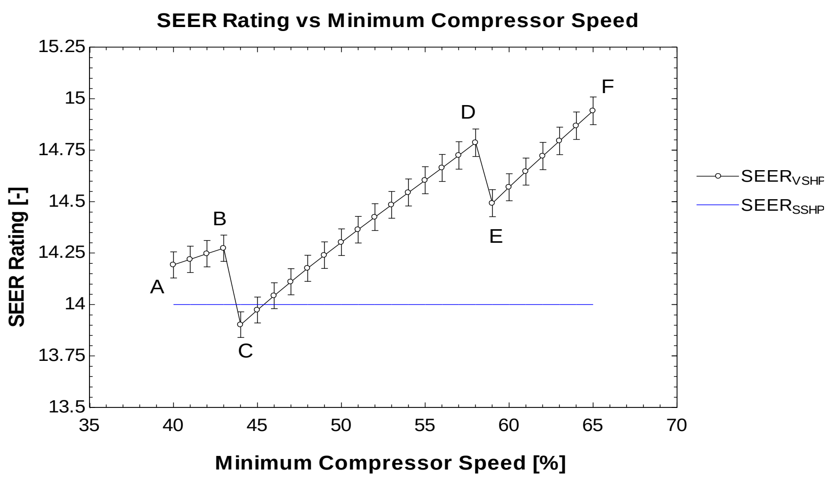


Fig. 3. SEER Optimization for Variable Compressor Speed

The blue line in Figure 3 represents the SEER of the baseline SSHP while the black line represents the SEER of the inverter modified VSHP as a function of minimum compressor speed calculated in 1.0% increments. The SEER rating of the unit reached a maximum value of 14.94 at a minimum compressor speed of 65%; this represents a 6.71% improvement from baseline. At a minimum compressor speed of 55% the SEER optimization method predicts a SEER value of 14.6; this is a 0.9% deviation from the SEER value of 14.74 determined at the same test conditions earlier in this paper. In theory, at the maximum compressor speed of 100% the unit must operate in the same manner as the baseline SSHP, and the black line would converge with the blue line. Therefore, it can be concluded that the maximum SEER for the heat pump with a variable speed compressor lies somewhere between 65% and 100% according to this optimization method.

Additionally, two discontinuities can be seen where the black line jumps from points B to C and from points D to E. The discontinuities in the curve can be explained by a change in operating mode for 1 of the 8 cooling season bins defined earlier. Table 8 shows how the smooth sections of the curve correspond to constant modes of operation across all bins, while a discontinuity occurs any time an operating mode is changed. This means that at an operating point near one of these discontinuities, a one percent change in minimum declared compressor speed could lead to large jump in SEER.

Table 8. Operating Modes at a Given Bin Number for Three Sections of the SEER Curve

Bin #	Bin Temp	Operating Mode		
		A-B	C-D	E-F
[-]	[ºF/ ºC]	[-]	[-]	[-]
1	(65-69)/(18.3-20.6)	min	min	min
2	(70-74)/(21.1-23.33)	min	min	min
3	(75-79)/(23.8-26.1)	min	min	min
4	(80-84)/(26.7-28.9)	int	min	min
5	(85-89)/(29.4-31.7)	int	int	min
6	(90-94)/(32.2-34.4)	int	int	int
7	(95-99)/(35-37.2)	int	int	int
8	(100-104)/(37.8-40)	max	max	max

In theory, as the number of bins used for the calculation increases, these discontinuities should become smaller in magnitude, and the curve should appear smoother. This method was tested by expanding the original set of 8 bins to 40 bins. Each bin was split up into 5 smaller bins and the fraction of total temperature bin hours was distributed linearly across the 5 bins. The representative temperatures were also distributed linearly across all 40 bins from 65 ºF (18.3 ºC) to 104 ºF (40 ºC). The change in the shape of the curve and the magnitude of the discontinuities can be seen in Figure 4. The overall trend did not change, but the large discontinuities no longer appear, and the curve appears to be much smoother. The validity of the method of expanding the number of bins as compared to the 8-bin-method hinges on the accuracy of the temperature data available. TMY3 data files are available for download online and provide real hourly weather data [7]. These files are used commonly in climate and building models, and could easily be incorporated into a SEER calculation to provide an accurate source of temperature data.

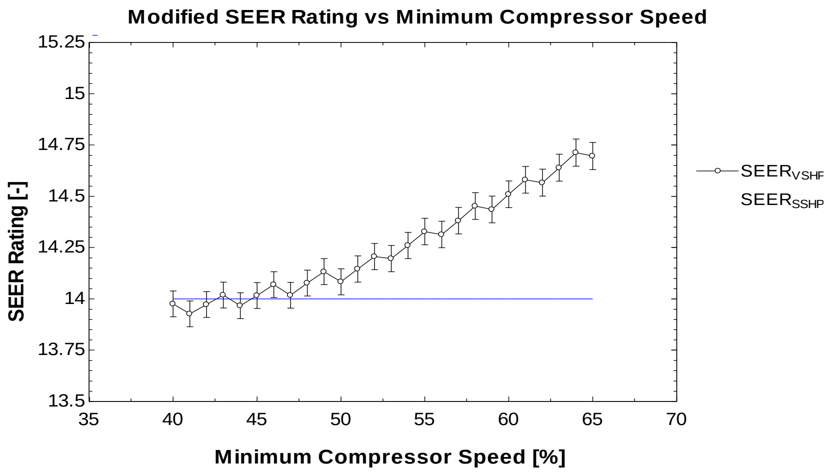


Fig. 4. Expanded Bin SEER Optimization for Variable Compressor Speed

4. Conclusions and Future Work

The results of this paper clearly indicate that using an inverter drive improves the seasonal performance of an existing single-speed heat pump without modifying any of the existing components. Based on laboratory testing of the inverter drive, a 5.3% SEER improvement was achieved by varying the speed of the compressor. An additional 4.5% SEER improvement was achieved through the use of varying the indoor fan speed. Varying the outdoor fan speed also showed some potential for improving the efficiency of the system at part load conditions, but more testing is required to determine the impact on the entire seasonal performance. The optimization method that was demonstrated in this paper indicates that a SEER improvement of greater than 6.7% is possible just by varying the compressor speed. It is the intent of this research project to finish the optimization method for both indoor and outdoor fan speeds to gather more information on how these variables impact SEER.

Additional studies have been proposed to understand how well the SEER rating predicts actual system performance. The results of the expanded-bin SEER optimization show that this would be a worthwhile pursuit. The method of expanding the number of bins and using published temperature data could provide a simple method to improve the accuracy of using the SEER to predict actual performance.

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