Measured Effect of Air Flow and Refrigerant Charge on a High-Performance Heat Pump

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ABSTRACT

In recent years there has been increased interest in the impacts of refrigerant charge and indoor coil airflow on heat pump performance in heating mode in the Pacific Northwest. The published literature contains almost no studies of this nature. As a first step to remedy this problem, extensive laboratory measurements were made on three heat pump models at Herrick Laboratories at Purdue University.

The heat pump discussed in this paper was a 3-ton "high performance model" with a rated Seasonal Energy Efficiency Ratio (SEER) of 14.5 and a Heating Seasonal Performance Factor (HSPF) of 9.0. Both heating and cooling mode measurements were made and included capacity, power, airflow, refrigerant mass flow rate, and coefficient of performance. Tests were done at each of three outdoor temperatures in heating mode (17 F, 35 F, and 47 F) and in cooling mode (82 F, 95 F, and 125 F). In both modes tests were done at refrigerant charges and airflows varying independently from approximately 25% above to 25% below the manufacturer's nominal recommended values. In addition, cycling and defrost tests were also performed to allow estimation of the HSPF and SEER. This paper summarizes the results of the laboratory tests and also the HSPF values calculated for six climate zones using ARI Standard 210/240.

Introduction and Background

Domingorena (1980) measured the effect on heat pump heating performance of varying the charge from 25% above the nominal value to 15% below. However, there was no variation in airflow. Furthermore, HSPFs cannot be calculated because measurements were not made at the necessary temperatures, nor were sufficient defrost and cycling tests performed. To the knowledge of the authors, the present study is the only set of measurements made with varying charge and airflow levels, at the appropriate temperatures and with the defrost and coefficient of degradation tests needed to calculate the HSPF.

Heat pumps have enjoyed a significant increase in popularity in recent years in the Pacific Northwest, both with the public and with utility program designers. In 2004, a consortium of agencies in the Pacific Northwest funded an in-depth study of heat pump performance in the region (Baylon et al. 2005). As part of this study, Purdue University was contracted to conduct laboratory tests on three heat pumps: an “economy model” with a SEER of 10 and HSPF of 7.2 using R-22 refrigerant (with suction-line accumulator), a “high-performance model” with a SEER of 14.5 and HSPF of 9.0 using R-22, and a “medium performance model” with a SEER of 13.0 and HSPF of 7.9 using R-410a. The goal of these tests was to determine the performance impacts of variations from manufacturer-recommended refrigerant charge and airflow on system capacity and efficiency.

The heating mode results from the “economy model” test have been fully discussed in a previous paper (Kruse and Palmiter 2006). A presentation summary “HPlabtest_with Cooling” of the cooling mode results for the “economy model” is available from the authors. The discussion in this paper is restricted to the high-performance model with a few comparisons with the "economy model" results.

In heating mode, capacity, power, fan power and Coefficient of Performance (COP) were measured in all combinations of each of three outdoor temperatures (17 F, 35 F, 47 F), at refrigerant charges of 75%, 100%, and 125% of the nominal values, and at airflows of 800, 1050, and 1300 standard cubic feet per minute (SCFM). The results also provide measures of the part-load performance, as characterized by the coefficient of degradation (C_d), and the defrost penalty factor at the various combinations of airflow and refrigerant charge. These are essential to the calculation of the HSPF rating used to establish the relative performance of heat pumps. The C_d measurement is made only at a temperature of 47 F and the defrost penalty is measured at 35 F.

All laboratory tests were done in accordance with the Air-Conditioning and Refrigeration Institute (ARI) Standard 210/240 (ARI 20062006) and the official US Department of Energy Test Standard CFR430. The laboratory results are summarized in this paper and the combined effects of all the variables are then demonstrated through HSPF ratings calculated for all six HSPF Climate Zones.

In cooling mode, the same laboratory tests plus latent cooling were made at 3 different outdoor temperatures (82 F, 95 F, and 125F). Cycling tests were done at an outdoor temperature of 82 F with a dry indoor coil to determine the C_d and SEER. The tests were done for all nine combinations of charge and airflow.

Heating Mode Results

This section graphically summarizes the effects of charge and airflow on capacity, COP, C_d, and the defrost penalty. All values plotted in Figs. 1-3 are normalized to the capacity or COP measured at the recommended charge and airflow rate for each temperature. Table 1 shows the capacities and COPs used to normalize each point. The nominal conditions to which the data is normalized are at an airflow rate of 1050 SCFM (the rating value), and 100% of recommended charge. As expected, the capacity and COP increase with increased outdoor temperature.

Table 1. Normalization Values of Capacity and COP at Nominal Charge and Airflow Rate

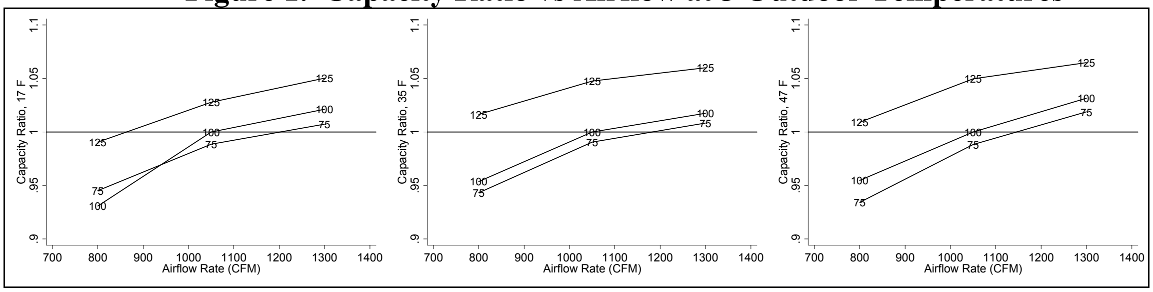
	Capacity (kBtu/hr)	COP
47 F	36.01	3.75
35 F	30.42	3.40
17 F	22.95	2.70

Capacity

Fig. 1 shows the capacity ratio versus airflow with points labeled by charge level; each line represents a common charge. The capacity increases about 7% from the lowest flow to the highest at each charge level. A charge level of 75% reduces capacity by 1 to 2% and a charge level of 125% increases capacity by about 7%. A similarly small impact of reduced charge in heating mode was also noted for the "economy model" previously tested (Kruse and Palmiter 2006).

Domingorena (1980) notes a smaller effect of charge on capacity: "The experimental results indicate that the performance of this heat pump in the heating mode is insensitive to increases of the refrigerant charge by as much as 25% above the nominal value 6 lb, 5 oz and is essentially insensitive to a charge reduction of 15% below the rated amount." However, he also notes the importance of the suction-line accumulator on these results, "This insensitivity is in contrast to the behavior of the low-first-cost unit previously tested, which has no suction-line accumulator and showed an almost linear reduction of heating capacity and COP with reduction of refrigerant charge."

Figure 1. Capacity Ratio vs Airflow at 3 Outdoor Temperatures



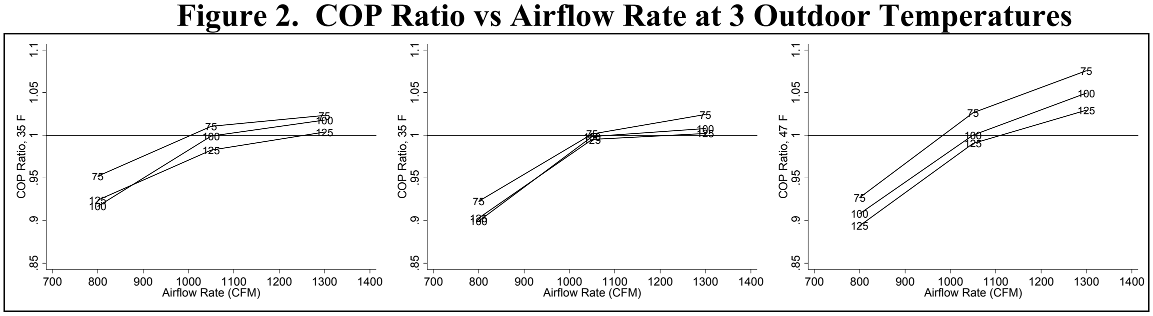
Coefficient of Performance

The coefficient of performance (COP) is also calculated using ARI Standard 210/240, which includes the indoor blower and outdoor fan power as well as that of the compressor in the denominator. The heat generated by the indoor fan is included in the capacity in the numerator. Each of these values was measured in the laboratory.

ARI mandates that for test units that do not include a specific indoor air handler, a default value for the indoor blower power of 365 W per 1000 CFM must be used. By comparison, the measured indoor blower power for the previously tested “economy model” with a permanent split-capacitor (PSC) blower motor averaged about 400 W per 1000 CFM. The “high performance model” has an electronically commutated motor (ECM) blower. The fan power was 135 W at 800 SFCM, 183 W at 1100 SCFM and 237 W at 1300 SCFM. The outdoor fan measurements for the current tests averaged approximately 200 W versus 266 W for the “economy model”.

Fig. 2 shows the normalized COP at each temperature versus airflow. The individual curves represent charge levels with the points labeled by charge level. There is an increase in COP of about 10% from the lowest airflow to the highest at each charge level and temperature. The COP is 1 to 2% higher at 75% charge and about 1% lower at 125% charge. The largest effects of charge on COP occur at 47 F where the corresponding reduction in capacity noted above occurs.

Figure 2. COP Ratio vs Airflow Rate at 3 Outdoor Temperatures



Part-Load and Defrost Penalty Factors

Two other factors have a large effect on heat pump performance: part-load operation and the defrost cycle. Under part-load conditions, the heat pump cycles off and on. For a short period during the start-up the heat pump draws nearly full power, but there is no output while the appropriate equilibrium conditions are being established throughout the refrigerant side. The indoor fan is off during this period. Additionally, each time the unit cycles off there is heat loss in the system. The net effect of these losses is an increasing loss of efficiency as the unit runs for a smaller fraction of time. The coefficient of degradation (C_d) is the percentage loss that occurs as the load approaches zero. The Standard (ARI 2006) assumes a linear percentage loss between zero load and full load (where the efficiency reaches the steady-state value). For instance, with a C_d of 0.25 the efficiency at zero load is reduced to 75% of the steady-state value, and at 50% load the efficiency is reduced to 87.5% of the steady-state value.

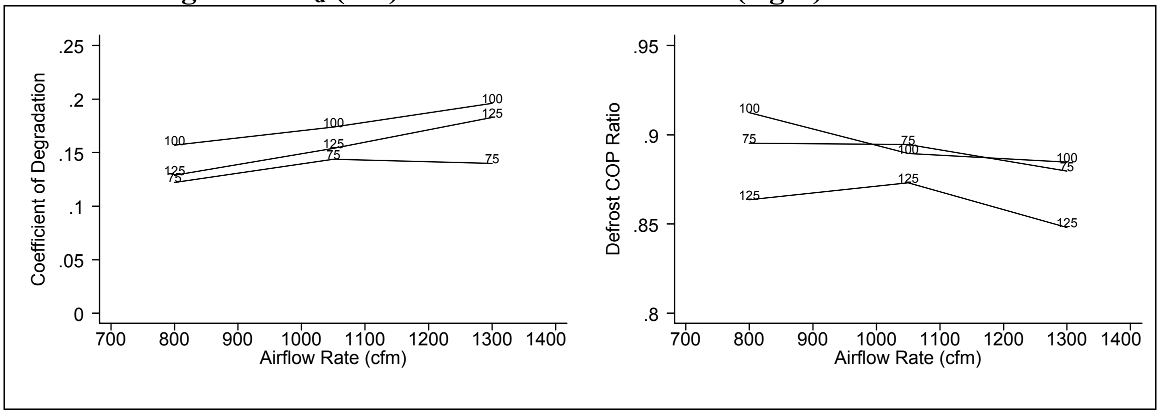
An additional performance loss occurs under outdoor conditions that lead to ice buildup on the outdoor coil. During the defrost cycle the outdoor fan is off and the heat pump operates as an air-conditioner to warm the outdoor coils. Typically the indoor fan and backup heat run during this cycle to warm the air coming off the cold, indoor coil. The defrost penalty can be stated as a multiplier of the steady-state efficiency at 35 F.

The Standard requires the measurement of C_d and the defrost penalty, but neither heat pump manufacturers nor ARI publish these measured values. Although unpublished, they are used by the manufacturer to calculate the Heating Seasonal Performance Factor (HSPF). In this study, one of the goals was to perform all of the required measurements needed to calculate an HSPF for various combinations of airflow and charge level.

The left column of Fig. 3 illustrates the effect of airflow rate and charge on the C_d, while the right column shows the effects on the defrost COP ratio. At the nominal charge level, low airflow results in a decrease in C_d of about 2.5% and high airflow increases the C_d by the same amount. These changes have the opposite sign to those seen in the "economy model".

The right column of Fig. 3 illustrates the effects of the various parameters on the defrost COP multipliers. This multiplier is simply the ratio of the measured COP with frost buildup to the measured steady-state COP with a dry outdoor coil, both at 35 F. At the rating point of 1050 CFM and 100% charge the COP multiplier is about 0.9. At a charge level of 125% the defrost multiplier drops to about 0.87. The effect of airflow is small. ARI gives a default defrost multiplier for variable speed compressors only; single speed compressors are not allowed to use a default and must perform the test. The variable speed compressor default defrost multiplier is 0.914. The manufacturer's catalog data suggests a defrost multiplier at 35 F of approximately 0.9.

Figure 3. C_d (left) and Defrost COP Ratio (right) versus Airflow Rate



Heating Seasonal Performance Factors

It is difficult to estimate the seasonal performance of a heat pump because both the capacity and the COP depend strongly on outdoor temperature. In addition, the capacity at low outdoor temperatures is usually not adequate to meet the heating load, thus requiring the use of backup heat. Also, the effects of part-load operation and defrost must be taken into account. In the late 1970's a bin-hour calculation method was developed that accounts, to some extent, for all of these effects. This method is called the heating seasonal performance factor or HSPF. Heat pump manufacturers are required to calculate the HSPF for given bin-hour profiles for six different climate zones. However, the U.S. government only allows the HSPF for Climate Zone 4 to appear on the label. This is often misleading because the HSPF varies strongly with climate zone and heating load.

We used the laboratory test data to calculate the HSPF in accordance with the ARI standard (ARI 2006) for all 6 climate zones.

Calculation Method

ARI Standard 210/240 defines the test methods used to measure the HSPF of a heat pump, as well as the equations required for calculation. The necessary variables for the calculation are airflow rates, capacities and electrical power consumption from steady-state tests at 17, 35 and 47 F, a cyclic test at 47 F, and a defrost test at 35 F. The capacity and input energy at any outdoor temperature is estimated as follows. Looking at the capacity or input curves, the slope of the line below 17 F and above 45 F is equal to that of a line connecting the 17 F and 47 F test point values. The central portions of the curves are defined by connecting the 17 F and 35 F test values and the newly defined point at 45 F.

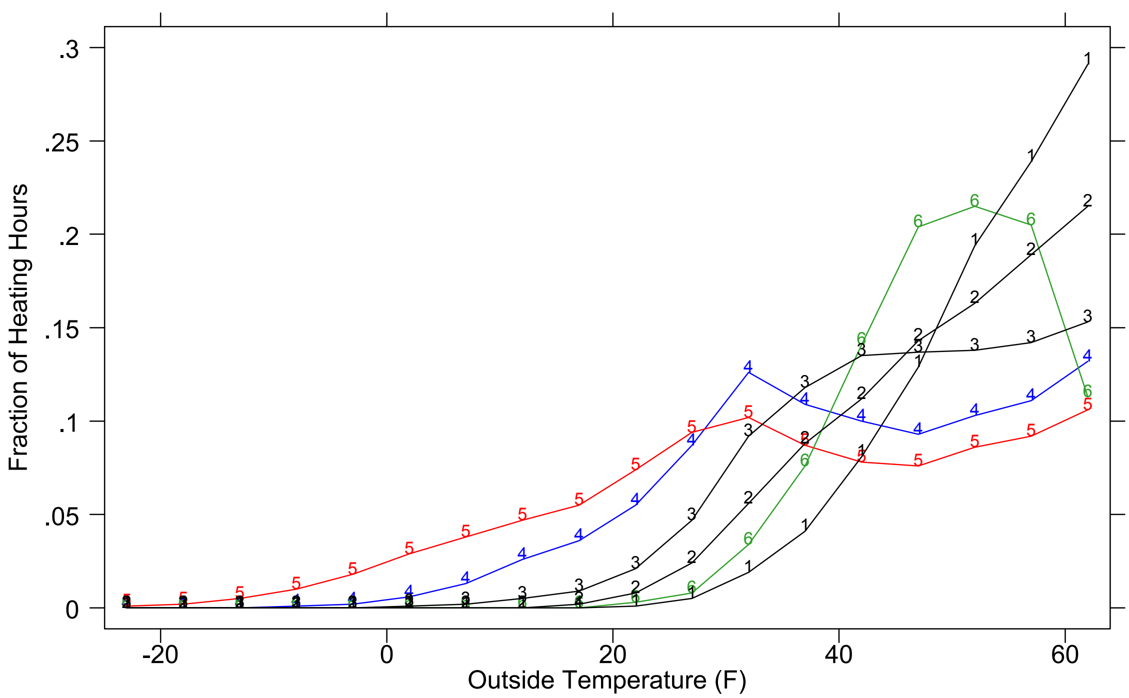
The load, capacity, compressor input, and auxiliary heat (assumed all-electric) are then calculated for each temperature bin, applying part-load corrections as needed. For details on the actual calculation method, see the Standard (ARI 2006). There are some unfortunate assumptions implicit in this method. It assumes there is no defrost penalty below 17 F; however with time-temperature defrost control there will be significant defrost penalties at all temperatures below about 40 F. There is also a large discontinuity in the performance curves at 45 F, an effect that does not occur in the laboratory. Additionally, the house load assumes a heating balance point of 65 F, which is a bit on the high side for well-insulated homes. This is compensated somewhat by multiplication of the load at each outdoor temperature by a load factor of 0.77. The load factor procedure is no longer sanctioned by ASHRAE.

Although imperfect, HSPF serves as a single point rating for heat pump performance that attempts to account for all of the major factors affecting performance. The largest error lies in failing to publish the rating for all zones (although at least one major manufacturer publishes HSPF ratings for Climate Zone 5 in addition to Zone 4).

HSPF Bin Data

Fig. 4 shows the fractional temperature bin data used for the six HSPF Climate Zones. The zones pertinent to the Pacific Northwest are 4, 5 and 6. The climates of Boise, ID and Spokane, WA are well represented by Climate Zone 4 (the label HSPF), while Missoula, MT is in Climate Zone 5. Zone 6 is representative of the climate west of the Cascade Mountains in Oregon and Washington, e.g., Seattle, WA and Portland, OR. It has a peak in the relative bin temperature distribution at about 50 F and very few hours at cold temperatures. The Zone map of the US provided in (ARI 2006) was incorrectly drawn in the first publication of the Standard and, unfortunately has never been corrected. This error results in the assignment of completely incorrect Climate Zones for the region west of the Cascade Mountains in Oregon, Washington, and Northern California. This is actually Climate Zone 6 which has a level of heat pump performance in heating mode comparable with that in Climate Zone 1.

Figure 4. Fractional Bin Hour Data for HSPF Calculations



Effect of Charge and Airflow on HSPF

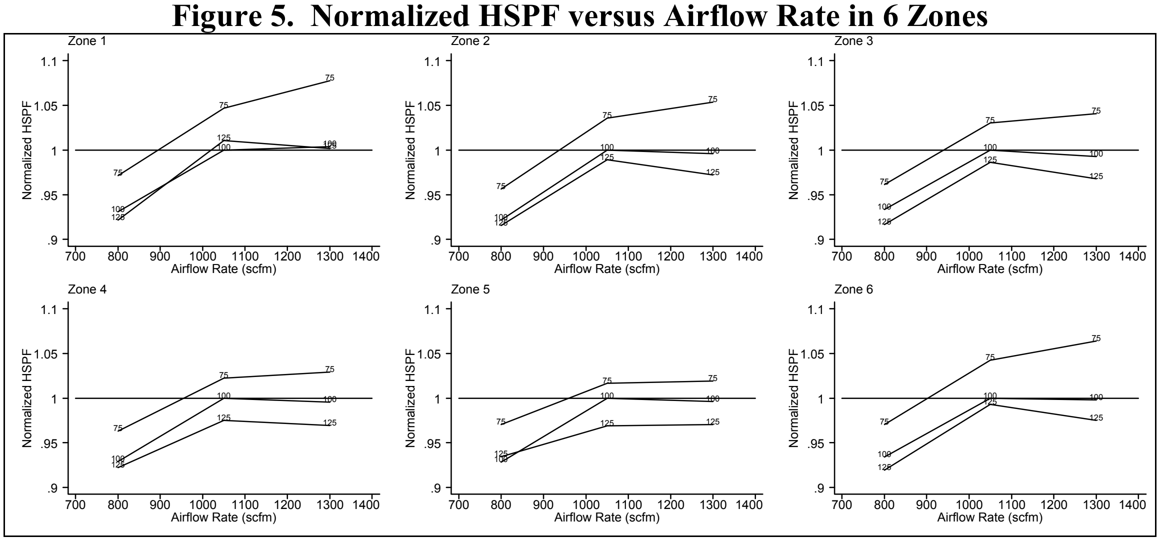
Table 2 shows HSPF values calculated from the laboratory test data for nominal charge and airflow. These values were used to normalize the HSPF plots shown below. Notice the large variation in HSPF across climate zones from 7.79 to 10.44. The measured value of 8.88 Btu/Wh in Zone 4 shows excellent agreement with the published HSPF for this heat pump of 9.0 Btu/Wh.

Fig. 5 shows the effect of airflow and charge on HSPF for each of the six climate zones. Reducing the airflow from the nominal value by 25% results in a loss in HSPF of about 7%, and increasing the airflow 25% has a relatively small effect except at low charge. Surprisingly, reducing the charge to 75% produces an increase in HSPF of about 5% in the warmer climates and 2 or 3% colder climates. The HSPF values at 100% and 125% charge are very close in the warmer climates. In the colder climates 125% charge results in a loss of about 5% in HSPF.

Table 2. HSPF at Nominal Charge and Airflow

Zone	HSPF (Btu/W hr)
1	10.44
2	10.06
3	9.59
4	8.88
5	7.79
6	10.15

Figure 5. Normalized HSPF versus Airflow Rate in 6 Zones



Cooling Mode Results

In cooling mode the standard tests required by CFR430 were performed. These tests are made at outdoor temperatures of 82 F and 95 F with wet indoor coil (entering wet bulb of 67 F and dry bulb of 80 F. In addition, we performed a set of tests at an outdoor temperature of 125 F with a dry indoor coil to examine the performance under hot dry conditions. Cycling tests to determine the C_d and subsequently the SEER were made at an outdoor temperature of 82 and an indoor temperature of 80 F with a dry indoor coil. All of these tests were performed at each of the nine combinations of airflow and charge.

Table 3 gives the measured capacity and COP at 1100 SCFM and 100% charge. These values were used to normalize the performance graphs. As expected, the capacity and COP decrease rapidly with outdoor temperature. The measured SEER for nominal flow and charge was 13.57 Btu/Wh which is about 7% less than the manufacturer's value of 14.5 Btu/Wh.

Table 3. Cooling Normalization Values of Capacity and COP at Nominal Charge and Airflow Rate

	Capacity (kBtu/hr)	COP
82 F	36.67	4.04
95 F	34.88	3.38
125 F	27.10	1.98

Fig. 6 shows the normalized capacity ratios as a function of airflow and charge level at each outdoor temperature. The capacity increases by 10-12% when the airflow increases from 800 to 1300 SCFM for each outdoor temperature and charge level. At 125% charge the capacity is increased by about 2%. At 75% charge the capacity is reduced by 5 to 7%, and a little more at an outdoor temperature of 125 F.

Figure 6. Capacity Ratio vs Air Flow Rate at 3 Outdoor Temperatures

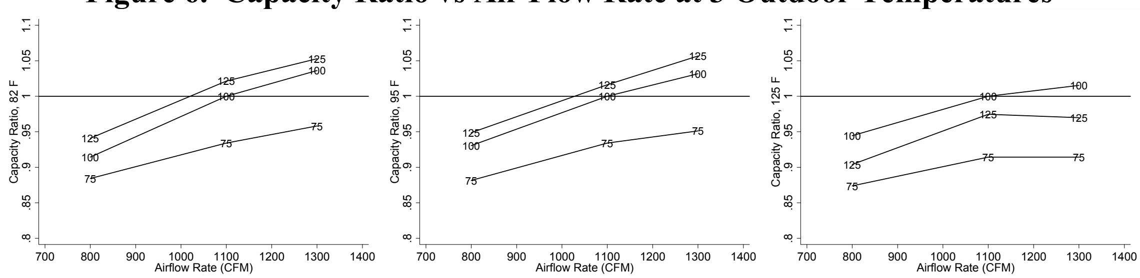


Fig. 7 shows the corresponding ratios for cooling mode COP. At charges of 100 and 125% there is a loss of COP of about 5% at a flow of 800 SCFM. At the lower outdoor temperatures, increasing the charge to 125% has very little effect and reducing the charge lowers the COP by about 7%. At an outdoor temperature of 125 F, the effects of charge are larger: 75% charge reduces COP by about 12% and 125% charge reduces COP by 5%.

The left panel of Fig. 8 shows the impact of charge and flow on the calculated SEER. At the higher charge levels (100 and 125%) reducing the airflow by 25% lowers the SEER by about 6% and increasing the airflow 25% reduces SEER by 3-4%. Reducing the charge to 75% results in about a 7% percent loss in SEER except at low airflow where the effect is smaller.

The right panel of Fig. 8 shows the impact of charge and flow on the measured $C_{d.}$ At nominal flow and charge the C_d is about 0.035. For the two higher charge levels it rises to about 0.04 to 0.06 at low airflow and also at high airflow. A charge level of 75% results in a C_d of almost 0.08 at nominal airflow and 0.10 at high airflow.

Figure 7. COP Ratio versus Air Flow Rate at 3 Outdoor Temperatures

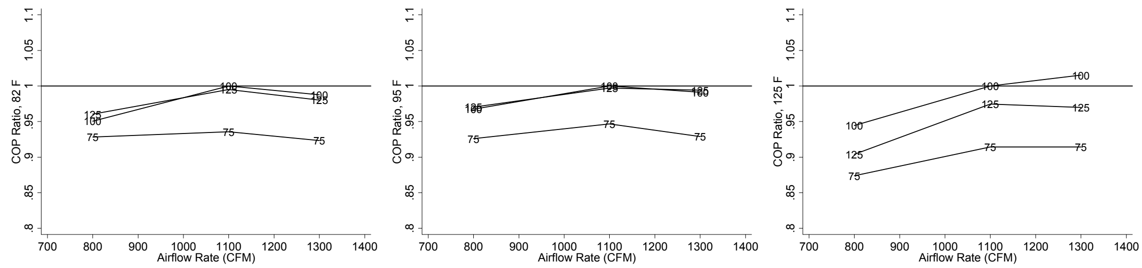
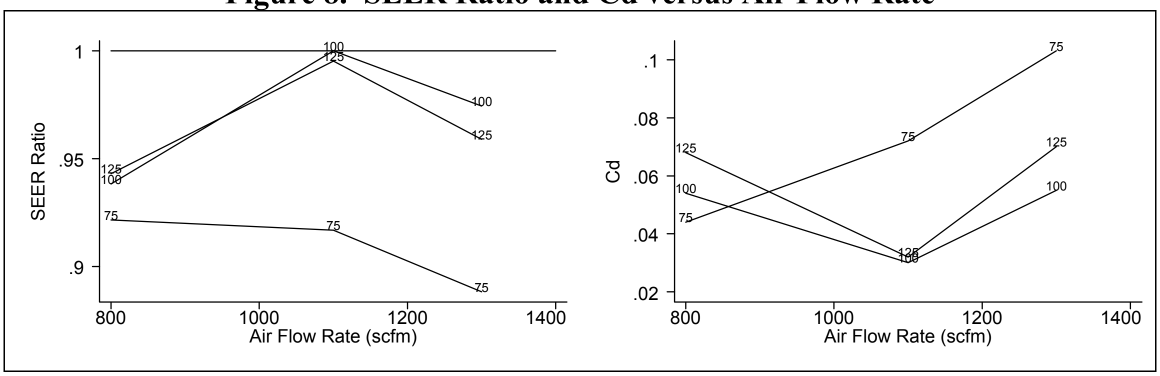


Figure 8. SEER Ratio and Cd versus Air Flow Rate



Heating Mode Conclusions

It is important to note that only two models of heat pump have been tested thus far and the results suggest caution in making any statements about the effects of charge and airflow on heat pump performance in general. In particular, it is expected that a heat pump without a suction-line accumulator would be much more sensitive to variations in charge level. It should also be noted that both of the tested units used R-22 refrigerant. These effects could also differ for an alternative refrigerant, such as R-410a. These results cannot currently be generalized across all heat pump models, even those using the same refrigerant. Additional testing is needed to identify the range of results possible across all available heat pumps.

The major findings of the heating mode laboratory tests and the HSPF calculations are summarized below. The term "nominal" in this discussion refers to values at 100% charge and an airflow rate of 1050 SCFM.

- Heat pump capacity falls within ±6% of the nominal capacity, for all evaluated charge levels, air flow rates and outdoor test temperatures.
- Heat pump COP is more sensitive to air flow. At low flow the COP is reduced by about 10% at the warmer temperatures (35 F and 47 F). At 47 F the COP increases by 3 to 5% at high airflow. The effect of charge is about ±1 or 2% across all of the tests.
- At nominal values, the coefficient of degradation (C_d) is 0.17. Various levels of charge and airflow result in values of C_d from about 0.12 to 0.20. Part-load performance is therefore relatively sensitive to charges and flow.
- At nominal values of charge and flow the defrost COP multiplier at 35 F was 0.88. These values compare closely with the ARI default value of 0.914 and the catalog data of approximately 0.9.

- At 125% charge the defrost multiplier falls to 0.85 to 0.87. The effect of air flow is 1 to 2%.
- The calculated HSPF at nominal conditions varies across climate zones from 7.79 in Zone 5 to 10.44 in Zone 1. The value of 8.88 in calculated for Zone 4 shows excellent agreement with the published HSPF for this heat pump of 9.0
- Reducing the airflow by 25% results in a loss in HSPF of about 7%, and increasing the airflow 25% has a relatively small effect except at low charge. Surprisingly, reducing the charge to 75% produces an increase in HSPF of about 5% in the warmer climates and 2 or 3% colder climates.

Cooling Mode Conclusions

The major findings for cooling mode are summarized below. In cooling mode the performance is more strongly affected by both charge and air flow.

- The capacity increases by 10-12% when the airflow increases from 800 to 1300 SCFM for each outdoor temperature and charge level. At 125% charge the capacity is increased by about 2%. At 75% charge the capacity is reduced by 5 to 7%, and a little more at an outdoor temperature of 125 F.
- At charges of 100 and 125% there is a loss of COP of about 5% at a flow of 800 SCFM. At the lower outdoor temperatures, increasing the charge to 125% has very little effect and reducing the charge lowers the COP by about 7%. At an outdoor temperature of 125 F, the effects of charge are larger: 75% charge reduces COP by about 12% and 125% charge reduces COP by 5%.
- At nominal flow and charge the C_d is about 0.035. For the two higher charge levels it rises to about 0.04 to 0.06 at low airflow and also at high airflow. A charge level of 75% results in a C_d of almost 0.08 at nominal airflow and 0.10 at high airflow.
- At the higher charge levels (100 and 125%) reducing the airflow by 25% lowers the SEER by about 6% and increasing the airflow 25% reduces SEER by 3-4%. Reducing the charge to 75% results in about a 7% percent loss in SEER except at low airflow where the effect is smaller.

Acknowledgements

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