Measured Effect of Air Flow and Refrigerant Charge on Heat Pump Performance in Heating Mode

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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 one heat pump model at Herrick Laboratories at Purdue University.

The heat pump tested was a 3-ton "economy model" with a rated Seasonal Energy Efficiency Ratio (SEER) of 10 and a Heating Seasonal Performance Factor (HSPF) of 7.2, which may be representative of the heat pumps commonly found in existing homes. The measurements included capacity, power, airflow, refrigerant mass flow rate, and coefficient of performance. At each of three outdoor temperatures (17°F, 35°F, and 47°F), tests were done at refrigerant charges and airflows varying independently from approximately 30% above to 30% below the manufacturer's nominal recommended values. In addition, cycling and defrost tests were also performed to allow estimation of the HSPF. A complete set of tests was performed for each of two metering devices: a short-tube orifice and a thermostatic expansion valve. This paper summarizes the results of the laboratory tests and the HSPF values calculated for several climate zones using Air-Conditioning and Refrigeration Institute (ARI) Standard 210/240.

Introduction and Background

Domingorena (1980) measured the effect 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 a 3-ton Carrier YKC heat pump (R-22 refrigerant, suction-line accumulator) in heating mode only. 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. Capacity and coefficient of performance (COP) were measured in all combinations of each of three outdoor temperatures (17°F, 35°F, and 47°F), at refrigerant charges of 70%, 85%, 100%, 115%, and 130% of the nominal values, and at airflows of 800, 1100, 1300, 1500, and 1700 cubic feet per minute (CFM).

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 and defrost penalty measurements were performed for a reduced number of combinations. The C_d measurement is made only at a temperature of 47°F and the defrost penalty is measured at 35°F. Each of these measurements was made at refrigerant charges of 70%, 100%, and 130% of the nominal values, and at airflows of 800, 1300, and 1700 CFM.

All of the above measurements were performed once with each of two metering devices: a short-tube orifice (STO) and a thermostatic expansion valve (TXV) on the outdoor unit (evaporator in heating mode).

All laboratory tests were done in accordance with the *ARI Standard 210/240* (ARI 2003). The laboratory results (Shen, Braun & Li 2005) are summarized in this paper and the combined effects of all the variables are then demonstrated through HSPF ratings calculated for the three HSPF Climate Zones relevant to the Pacific Northwest region.

The laboratory test results showed systematic error in the air-side performance measurements. The laboratory report contains adjusted airflow rates, capacities and COP values based on refrigerant-side measurements. For a complete discussion of this adjustment see the laboratory report (Shen, Braun & Li 2005). Only the adjusted data are used in this paper.

Measurement Results

This section graphically summarizes the effects of charge, airflow and metering device on capacity, COP, C_d, and the defrost penalty. All values plotted in Figures 1-4 are normalized to the capacity or COP 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 an airflow rate of 1300 CFM (close to the rating value of 1250 CFM), and 100% of recommended charge. Input is not explicitly discussed in this paper, but it may be calculated by dividing the capacity by the COP.

As expected, the capacity and COP increase with increased outdoor temperature. The short-tube orifice and the TXV metering devices give virtually identical capacity results. The TXV metering device results in a slight improvement in the COP of up to 4% at 17°F.

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

	Capacity (kBtu/hr)		COP	
	STO	TXV	STO	TXV
47°F	36.59	36.43	3.19	3.16
35°F	30.13	30.08	2.77	2.82
17°F	20.94	21.19	2.15	2.24

Capacity

Figure 1 shows the capacity versus charge for a heat pump with a short-tube orifice, with points labeled by nominal airflow rate; each line represents a common airflow rate. The trend with respect to charge levels is fairly flat, with the exception of the measurements at 47°F. At that temperature, 85% charge level results in a capacity reduction of about 10%, and increases to 15% at 70% charge.

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."

With the exception of the lowest flow rate, the variation with respect to flow rate falls within a band of plus or minus 3% around the nominal capacity. At the lowest flow rate, the capacity is reduced in most cases by approximately 10%. However, it is important to note that measurements at the lowest flow rate have the most uncertainty.

Figure 2 is the same as Figure 1 for the same unit with a TXV replacing the short-tube orifice. The effect of charge at 17°F and 35°F is level from 85% to 115% charge. Above and below these charge levels capacity falls off about 4%. At 47°F, the effect is similar, but with a smaller effect at the highest charge (2%) and a greater effect at the lowest charge (up to 8%).

At all but the lowest flow rate, the capacities are within plus or minus 3% of the nominal capacity, with the lowest flow rate resulting in a capacity reduction of 8 to 10%.

A comparison of Figures 1 and 2 reveal relatively little impact of metering device on capacity at 17°F and 35°F, with the capacity at low charge levels improving slightly with a TXV. At 47°F, the TXV results in a higher capacity at low charge.

With regard to the loss of capacity at high temperatures, it is important to remember that above the design balance point the heat pump is still able to meet the load without use of the backup electric resistance elements. In the Pacific Northwest heat pumps are frequently sized for a design balance point of around 30°F. With a house balance point of 65°F, the load at 47°F will be only 50% of the load at 30°F. Thus loss in capacity of 15% at 47°F discussed above will have no effect on the ability to meet the load. In fact, if the COP remained constant with changes to charge and flow, due to the increased runtime and subsequent reduction in part-load penalties, the overall performance would improve. It is the loss of COP at 47°F that will have the primary impact on performance.

It is interesting to note that the HSPF procedure calculates the heating load based on the size of the heat pump being rated (ARI 2003). The result of this calculation is an implicit heat pump balance point of 17 to 20°F, except in Zone 5 (the coldest zone) where it is about 12°F. Thus reductions in capacity at 47°F are of even less significance F in calculating the HSPF than the example above at 30°F.

Figure 1. Capacity Ratio (STO) versus Charge at 3 Outdoor Temperatures

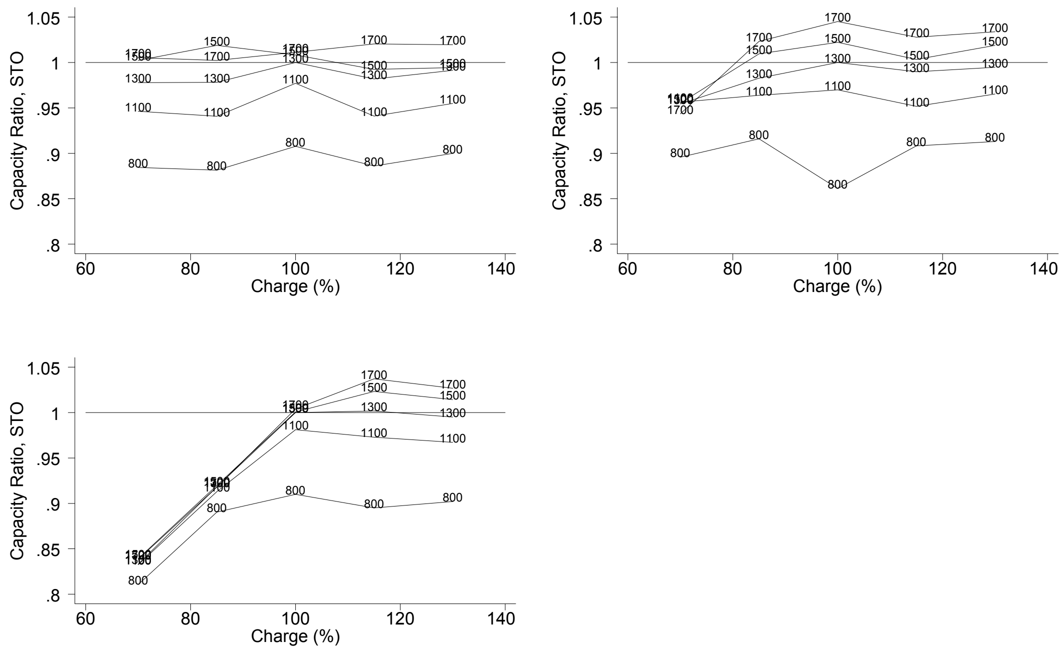
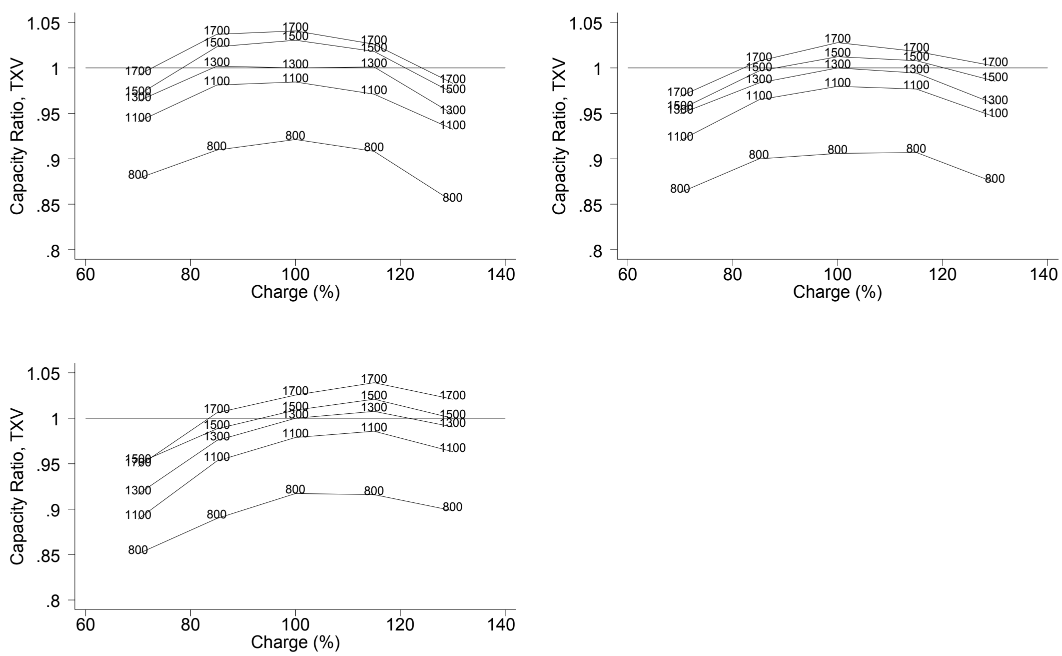


Figure 2. Capacity Ratio (TXV) versus Charge at 3 Outdoor Temperatures



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 averages about 400 W per 1000 CFM. The outdoor fan measurements averaged approximately 266 W.

Figures 3 and 4 show the normalized capacity at each temperature versus the percent charge for the short-tube orifice and the TXV respectively. The individual curves represent airflow rates with the points labeled by the nominal airflow rate in CFM. These figures show a strong resemblance to Figures 1 and 2 respectively because the input power is fairly constant with changes in airflow and charge in comparison to the capacity. Therefore, variations in capacity are also evident in the COP curves.

As with capacity, there is a 5 to 10% reduction of COP relative to the nominal case at the lowest airflow rate.

The largest effects of charge on COP occur at 47°F, where the corresponding reduction in capacity noted above occurs. The percentage reduction in COP at this point is less than that in the capacity. Compared to a 15% reduction in capacity, the COP shows a 10% drop at 70% charge. More generally, the percentage variations in COP due to charge are smaller than those noted in the capacity because the input tends to change in the same direction.

Part-Load and Defrost Penalty Factors

Two other factors have a large affect 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 are heat losses 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 *ARI Standard* (ARI 2003) 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 the defrost 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.

Figure 3. COP Ratio (STO) versus Charge at 3 Outdoor Temperatures

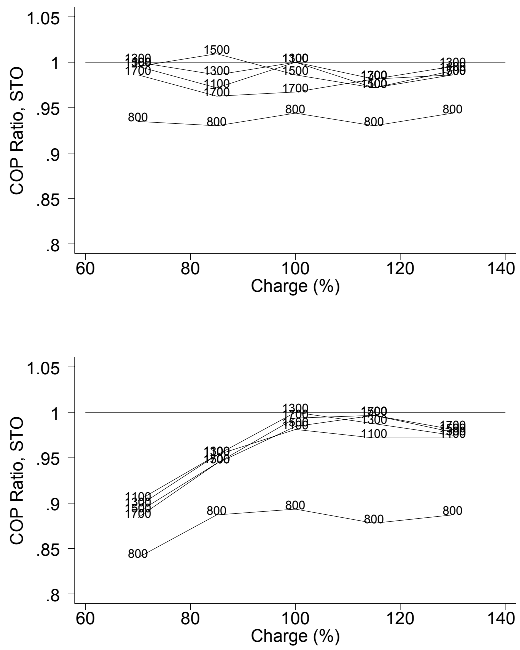
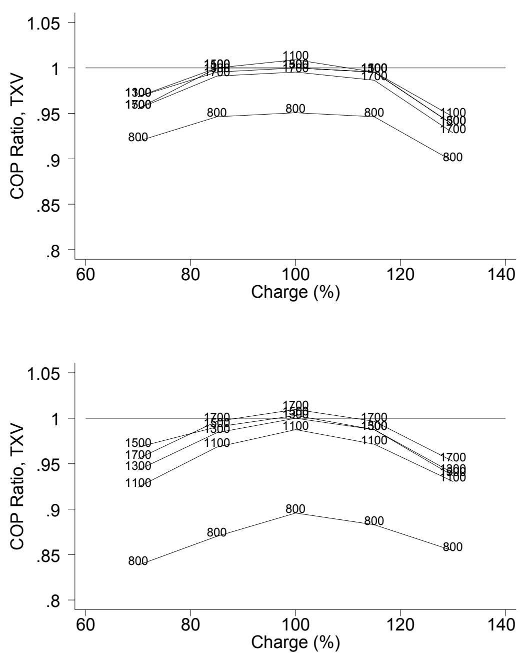


Figure 4. COP Ratio (TXV) versus Charge at 3 Outdoor Temperatures

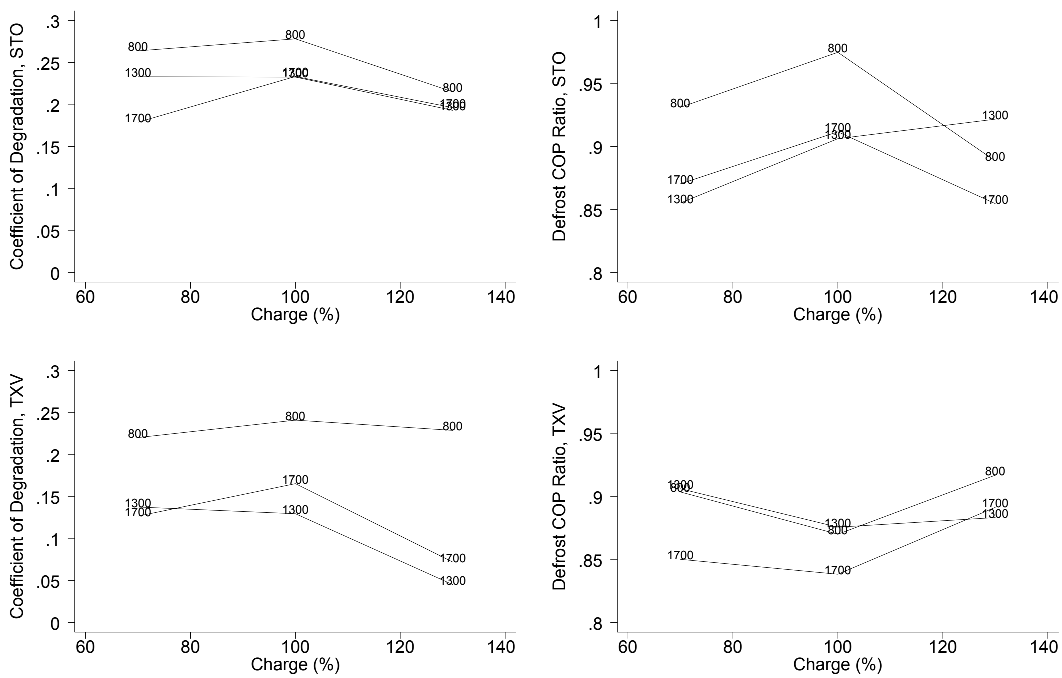


The *ARI Standard* requires the measurement of C_d and the defrost penalty, but neither heat pump manufacturers nor ARI publish these measured values. Although unpublished, they are used by the manufacturer to calculate the HSPF. In this study, one of the goals was to perform all of the required measurements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

The left column of Figure 5 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 an increase in C_d of about 17% for the STO, and 65% for TXV. Both metering devices show some improvement in C_d with 130% charge. With the TXV, a high charge level appears to improve the C_d by a factor of about 2.5.

Comparing the graphs in the top row to those in the bottom of Figure 5, the effects of the metering device are more evident than with capacity or COP. With a TXV, the C_d generally shows more sensitivity to changes in the flow rate and charge levels. The C_d is also considerably smaller than with the short-tube orifice except at the lowest airflow rates. For instance, at nominal flow and charge, the C_d with an STO is 0.23 and with a TXV is 0.13. Compared to the ARI default value of 0.25, our test value for C_d with the short-tube orifice is similar, but with a TXV is considerably smaller. The manufacturer's catalog data for this heat pump provides the values necessary to back-calculate a C_d of 0.14 for cooling. For heating, this value is comparable to the test values with a TXV.

Figure 5. C_d versus Charge (column 1) and Defrost COP Ratio (column 2) with STO Metering (row 1) and with a TXV (row 2)



Note: Points labeled by charge (%)

The right column of Figure 5 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 coil, both at 35°F. At the rating point of 1300 CFM and 100% charge, the defrost multiplier for short-tube orifice metering is 0.91 and for TXV is 0.88. Other combinations of flow and charge produce multipliers that deviate as much as 5% from the nominal values. 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 COP multiplier is 0.914. The manufacturer's catalog data suggests a defrost multiplier at 35°F of approximately 0.91.

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.

We used the laboratory test data to calculate the HSPF in accordance with the *ARI Standard* (ARI 2003) for Climate Zones 3, 4, 5, and 6, which are those pertinent for the Pacific Northwest.

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 *ARI Standard* (ARI 2003).

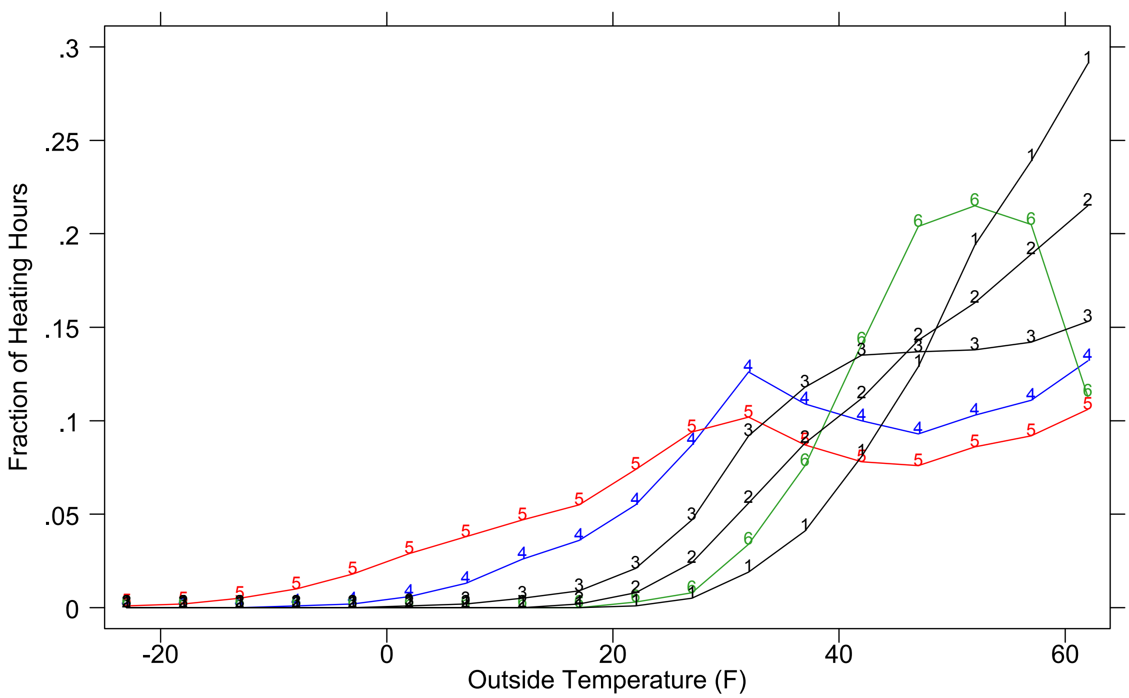
There are some unfortunate assumptions implicit in this method. It assumes there is no defrost penalty below 17°F, however with time-temperature defrost control there will be significant defrost penalties at all temperatures below about 40°F. There is also a large discontinuity in the performance curves 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

Figure 6 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. 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 few hours at cold temperatures. 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.

Figure 6. Fractional Bin Hour Data for HSPF Calculations



Effect of Charge, Airflow, and Metering Device 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 6.25 to 8.50 for the STO and 6.63 to 9.19 for the TXV. The measured value of 7.25 in Zone 4 shows excellent agreement with the published HSPF for this heat pump of 7.2. The fourth column shows the improved performance due to the TXV ranges from about 5% to 8%.

Table 2. HSPF for Nominal Charge and Airflow

Zone	HSPF with STO	HSPF with TXV	Ratio
1	8.50	9.19	1.081
2	8.29	8.80	1.062
3	7.90	8.34	1.056
4	7.25	7.63	1.052
5	6.25	6.63	1.061
6	8.33	8.93	1.072

The normalized HSPF values calculated from the test data for ARI Climate Zones 3, 4, 5 and 6 are shown in Figure 7 for the STO and Figure 8 for the TXV. Each point represents an HSPF calculated from the test data for the unique capacity, COP, defrost multiplier, and C_d for that combination of flow rate, charge and metering device.

For both metering devices, the lowest airflow results in a 10 to 17% reduction in HSPF at nominal charge relative to the HSPF at normal airflow. The impact of low charge is a 2 to 5% reduction in HSPF with the exception of Zone 5 for the STO where there appears to be a slight increase in HSPF. One should be cautious not to over-interpret some of the smaller changes because the individual measurements have errors of a few percent.

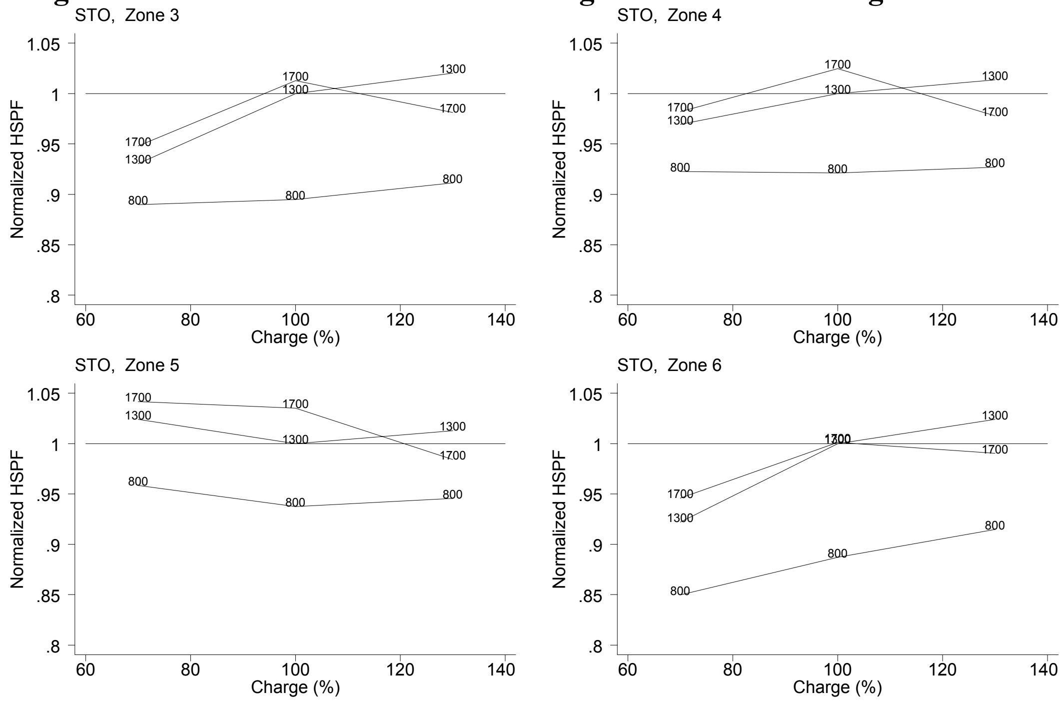
Conclusions

It is important to note that all of the testing was performed on one heat pump (3-ton Carrier YKC model) and it is unclear how greatly the results might vary with a different heat pump model. 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 this heat pump 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 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 1300 CFM.

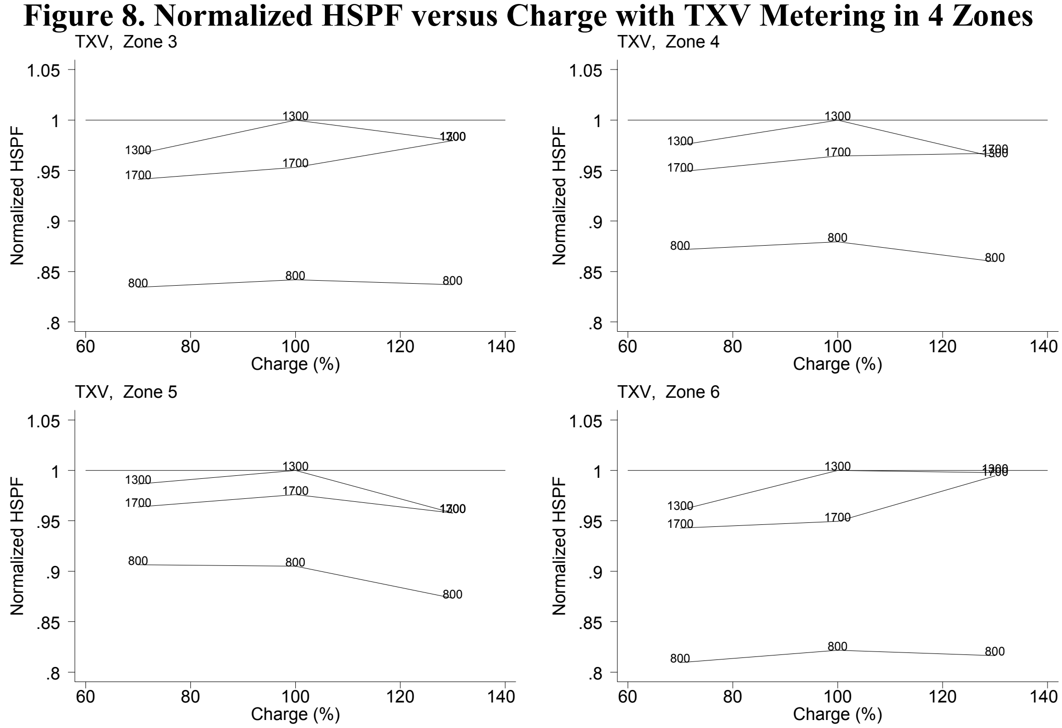
- Capacity at 17°F and 35°F with either metering device falls within plus or minus 3% of the nominal capacity except at the lowest flow rate where there is a reduction of about 10%.

Figure 7. Normalized HSPF versus Charge with STO Metering in 4 Zones



Note: Points labeled by airflow rate (CFM)

Figure 8. Normalized HSPF versus Charge with TXV Metering in 4 Zones



Note: Points labeled by airflow rate (CFM)

- At 47°F and charge levels at or above nominal, the capacity falls within plus or minus 5% of the nominal capacity except at the lowest flow rate, where there is a reduction of about 10%. At charge levels below nominal there is a reduction in capacity that reaches 15% at 70% charge for the short-tube orifice and 8% for the TXV. As noted in the text, loss of capacity at 47°F typically has no effect on the ability of the heat pump to meet the load.
- The effects of charge and airflow rate on COP are similar to those for capacity, but the magnitude of the variation is smaller.
- At nominal values, use of a TXV shows almost no effect on capacity and a very small improvement in COP.
- At nominal values, the coefficient of degradation (C_d) is 0.23 with the short-tube orifice and 0.13 with a TXV. At the lowest airflow rate, there is an increase in C_d of about 17% for the STO, and 65% for TXV.
- The defrost COP multiplier at 35°F with the STO is 0.91 and with a TXV is 0.88. These values compare closely with the ARI default value of 0.914 and the catalog data of approximately 0.9. Various combinations of flow and charge produce multipliers that deviate as much as 5% from the nominal values.
- HSPF varies across climate zones from 6.25 to 8.50 for the STO and 6.63 to 9.19 for the TXV. The measured value of 7.25 in Zone 4 shows excellent agreement with the published HSPF for this heat pump of 7.2. The TXV results in an improvement ranging from 5 to 8%.
- The lowest airflow results in a 10 to 17% reduction in HSPF at nominal charge relative to the HSPF at nominal airflow.

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