B O N N E V I L L E   P O W E R   A D M I N I S T R A T I O N

Mini-Split Ductless Heat Pump Bench Test Results

July 31, 2009





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MEMORANDUM

31 July 2009

To: Kacie Bedney, Bonneville Power Administration
From: Bob Davis, Ecotope, Inc.
Subject: Mini-Split Ductless Heat Pump Bench Test Results—Final Progress Report

Introduction

This memorandum reports on various aspects of a Fujitsu mini-split heat pump's energy performance. Of primary interest are defrost cycles, typical heating season performance, and midsummer cooling performance. Earlier memos (June and September, 2008) reported on cooling performance and various setup and logistical issues, and a January, 2009 memorandum reported on heating performance during very cold weather (December, 2008).

The indoor unit of the tested system is partly enclosed by a plywood box; this enclosure has facilitated measurement of system airflow by use of a depressurization fan (combined with logged measurements of box pressurization during normal system operation). Since this configuration has not been shown before, it is presented below for reference:

Figure 1. Head-on View of Indoor Unit Test Enclosure

Defrost Cycle Behavior

Heating coefficient of performance (COP) during a cold snap (ambient temperatures ranging between about 0ºF and 20ºF over a week's time) was reported on in some detail in an earlier memorandum. More review of the data showed there was some defrost cycle activity during this time, which is expected given the long run times of the equipment.

Figure 2 shows the supply air temperature (red line), return (garage) air temperature (green line), and outside temperature (blue line) and whole-system energy usage (black line) during part of the cold snap. Note ambient temperature started in the high 30ºs F at the left of the graph and headed downward; there is some data interruption around noon on 12/13 that had to do with a system adjustment. Also, there are some missing power measurements for most of 12/14; this problem was corrected.).

The most interesting aspects of defrost on this mini-split have to do with the overall amount of time spent in the cycle and the range of system energy usage during and after defrost. The best data to consider are found on 12/13 in the afternoon and on 12/15 in the late afternoon/early evening.

During the former period, ambient temperature is in the 30ºs F. The system goes through about six defrost cycles over the course of an afternoon, with the first (at about noon) being the most significant. This cycle lasts for about 20 minutes. Note the supply air temperature drops to about 70ºF for part of the cycle, but just after the cycle ends, the supply air temperature jumps to about 115ºF. This behavior is congruent with other defrost cycles; the control logic appears to compensate for the relatively low delivery temperatures during the reverse cycle by boosting output temperatures just after the cycle concludes. This boost comes at a price: increased system energy usage, with power jumping from about 600 Watts (pre-defrost-cycle usage) to over 1000 Watts for the five minutes or so of "boost." After this period is over, the usage settles down. Note that during defrost, the fan airflow (measured in cubic feet per minute [CFM]) also modulates to a very low level to mitigate complaints of discomfort.

During the reverse cycle itself, power usage is modest since the system is taking heat from a mild source and dumping it to a very cold sink (the iced-up coil in the outdoor unit). Since there is no electric resistance backup heat (as would be found in a standard residential heat pump), the overall effect of the reverse cycle is negligible. Also, there are no times where the supply air temperature dips below the return air temperature.

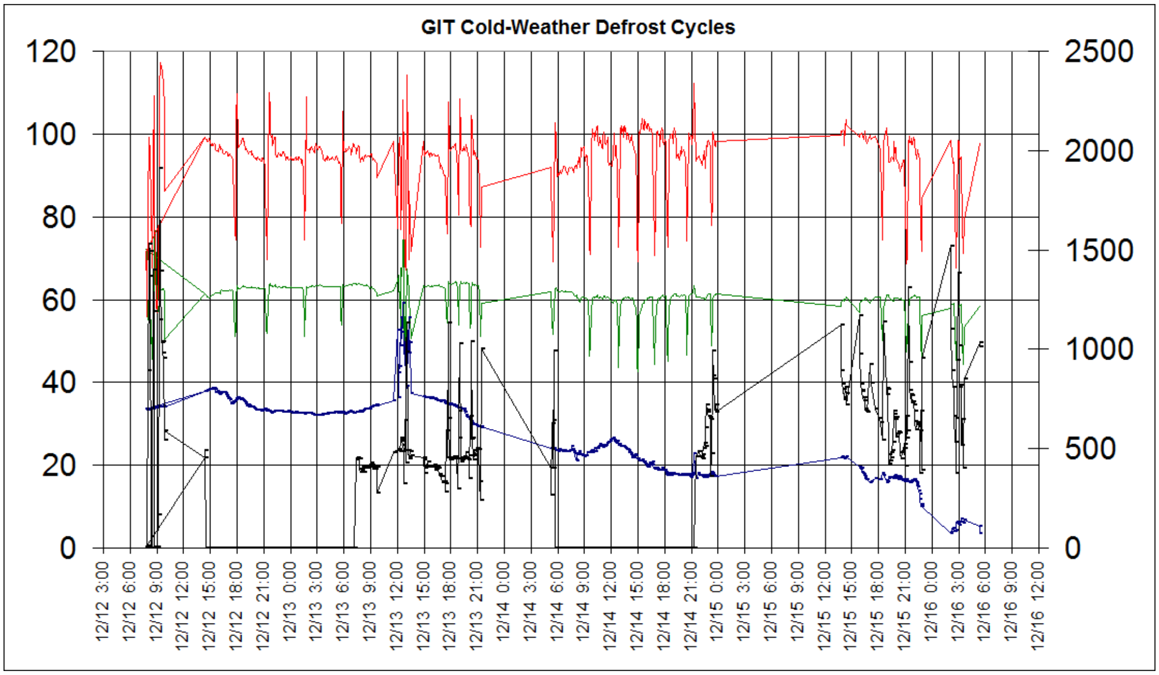

Figure 2. December 2008 system behavior (emphasis on defrost cycles)
Note left y-axis label is degrees F and right y-axis label is system power (Watts).
Red line is supply air temperature; green line is return air temperature.
Blue line is outdoor temperature; black line is DHP power.

Late Winter/Early Spring Defrost Behavior

Some technical difficulties complicated defrost analysis during the coldest weather. Data for early April show a more complete picture of defrost behavior. Ambient conditions ranged between the low 30ºs F and low 60ºs F during this period, with the earlier data (April 1-3) quite representative of west-side Pacific Northwest heating weather.

There are several defrost cycles over the course of each day, but the overall effect on system power is, again, not a net negative. Delivery temperatures drop significantly during the reverse cycle, but the average defrost cycle length is about four minutes. Also, the supply temperature usually boosts for two to four minutes after the cycle ends to overcome the effects of the colder delivery temperature. Overall, the amount of extra power needed for the boost phase is balanced out by lower power usage during the reverse cycle.

Some calculations of system COP were made before and after a defrost cycle. With the system running in part load, the COP is typically in a range of 3.0 to 4.5 (depending on outdoor temperature). Supply air temperature drops during the defrost cycle, and the indoor fan "gears down" so the thermal output is into the room is reduced greatly. (But note that during actual defrost, the goal is to remove ice, not to add heat into the room.) For the first few minutes after defrost ends, thermal output increases by about 50% relative to the period just before defrost, and COP also spikes (typically by about 25% relative to just before defrost).

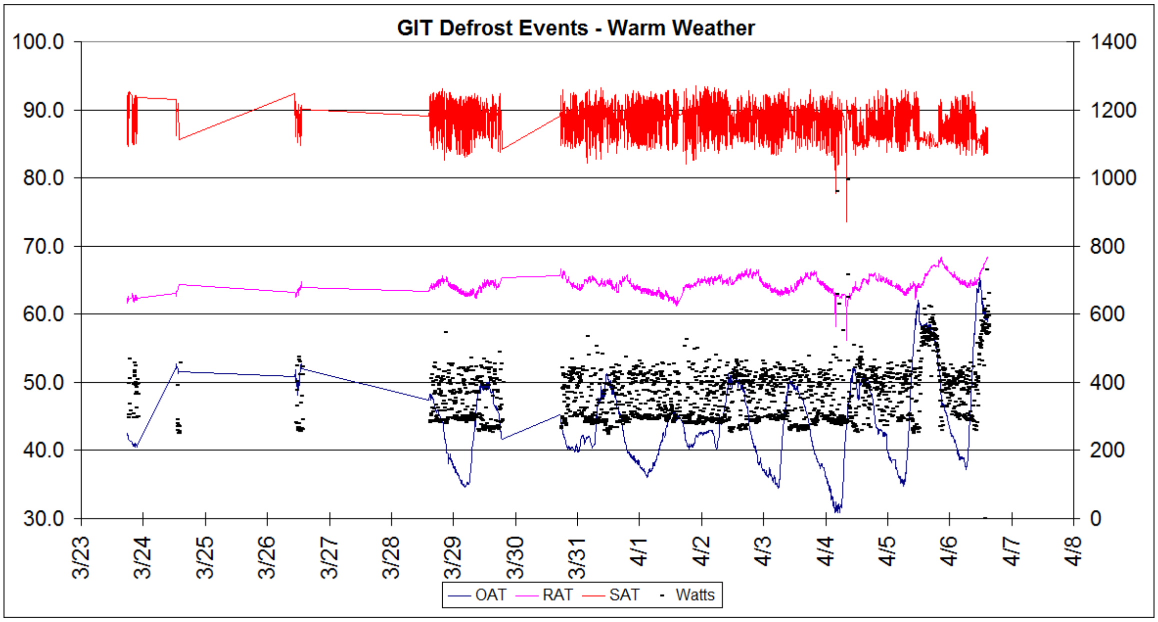

Figure 3. Late Winter System Behavior (emphasis on defrost cycles)
*Same color conventions as Figure 2 except return air temperature is now magenta***.**

Heating Coefficient of Performance (COP)

Most of the data shown in Figure 3 correspond to part-load heating operation; the system is run in the AUTO mode and is able to maintain heating setpoint under most outdoor temperature conditions. The system typically draws around 350-450 Watts to maintain a 25-35º F temperature rise at outdoor temperatures between the low 30ºs and low 50ºs F. Measured airflow for most of this operation is around 220 CFM. The raw COP using these inputs ranges from about 3.8 at 32º F to 5.6 at 55º F. These are very impressive efficiency figures. During the "boost" phases just after defrost cycles end, COPs are more in the range of 2.5, but this is still impressive given that the system is recovering from a reverse cycle.

Overall, the performance of the system is quite remarkable. When expressed as a function of outdoor temperature bins, a fuller picture emerges. (Note data for times where there were data collection problems are not included in this graphic.) Most of the system operation is under part load, so this level of performance is not unexpected.

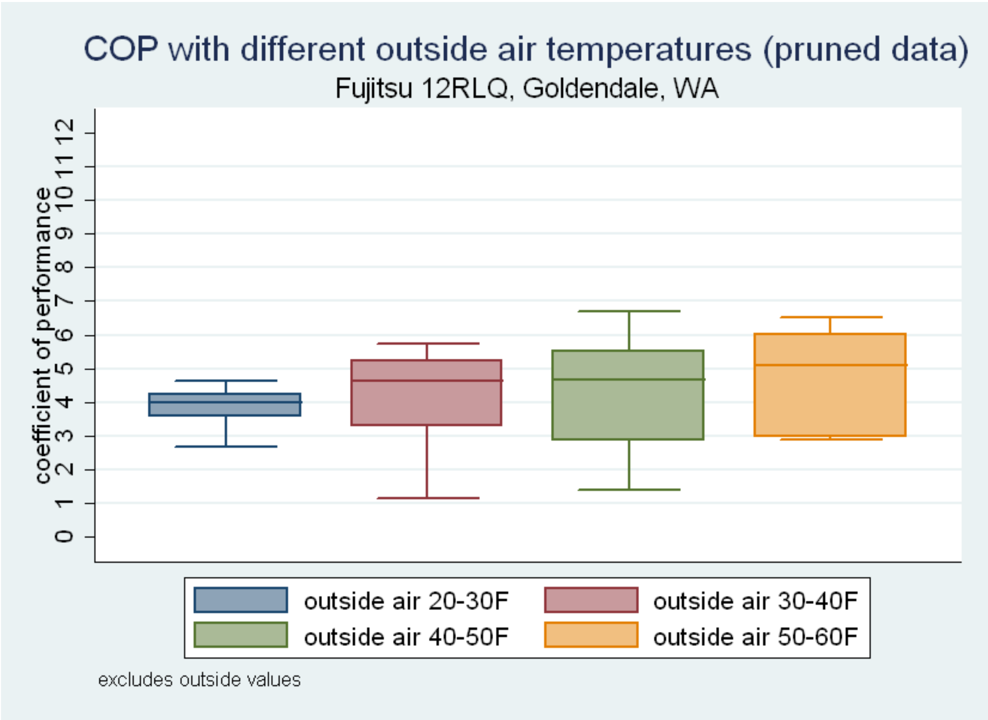

Figure 4. Summary of Heating COP at Different Outdoor Temperature Bins

Part-Load Cooling Performance – Early July

System performance in cooling was evaluated several times; a representative set of data from early July illustrates the very impressive performance of the system during part-load operation. (Note only sensible cooling was evaluated; a condensate tipping gauge had been prepared for this site but observation in summer, 2008 convinced us there was very little condensate produced.) The system was running during what could be considered a typical summer afternoon in western Washington or Oregon—outdoor temperature ranged from about 78ºF to 90ºF between about 2 pm and 8 pm, for an average of about 86ºF.

The system was running in the AUTO mode and fan was on the LO setting (which is customary when the system is maintaining a moderate Δ (delta) T and is approaching the customer's desired setpoint.) During the monitoring period, sensible cooling averaged about 4500 Btu/hr, fan flow averaged 220 CFM, temperature split across the indoor unit averaged 20.1ºF, input watts averaged 241, and COP averaged 5.6! If the system were to run predominantly in this mode, the *ad hoc* SEER (average COP x 3.413) would be on the order of 19! (Note the actual SEER test includes a range of testing conditions and assumes a significant latent load [wet coil], so is not really an appropriate comparison here, but we do see a rough agreement between rated SEER and our COP measurement.)

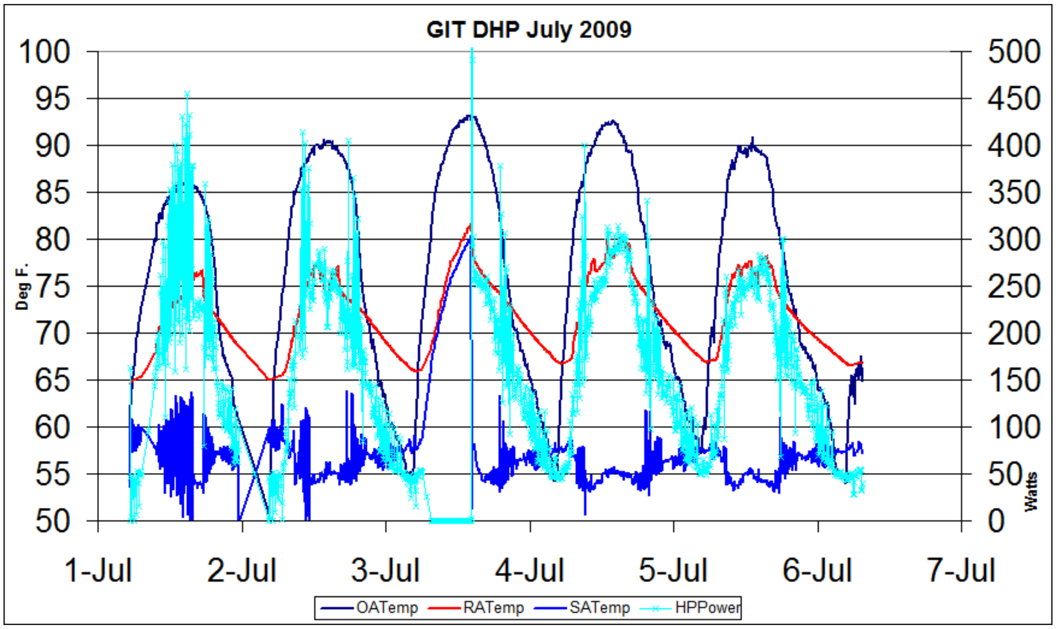

Figure 5. Early July Cooling Data

A box-and-whisker plot of cooling COP as a function of outdoor temperature (similar to Figure 4, but for cooling) shows the system generally performs better than one would expect given nominal manufacturer's ratings (EER 12.5). When COP is converted to a pseudo-EER using the 3.413 conversion factor, we find an "EER" estimate of about 17 for all operating conditions within 5°F of the outdoor temperature used in the actual DOE EER test (95ºF). Note, though, that almost all operation captured in the graphic is for part-load operation.

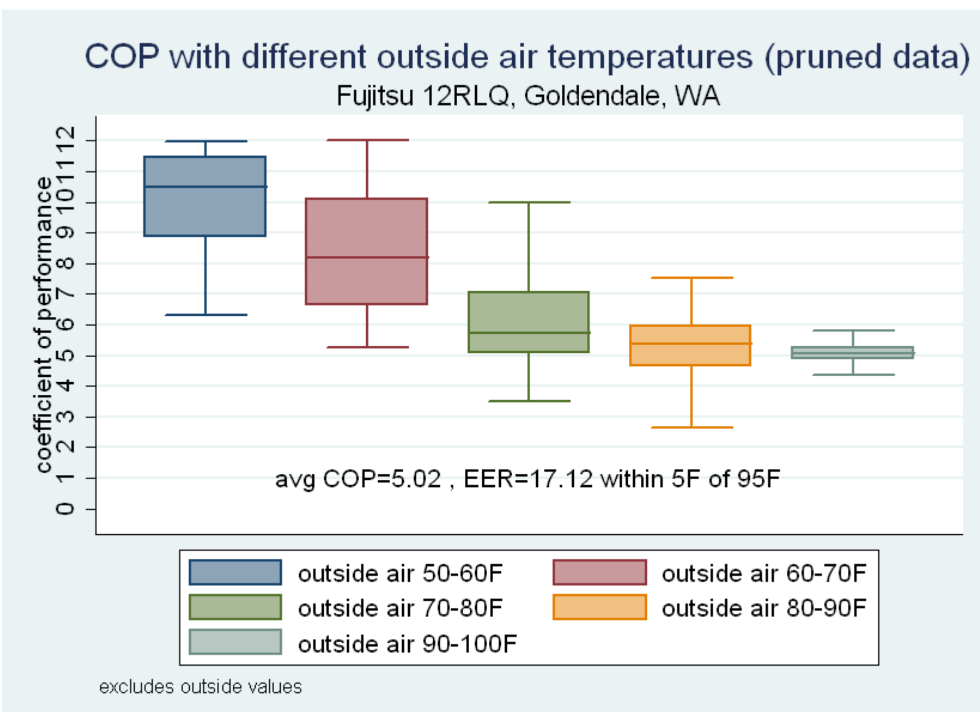

Figure 7. Summary of Cooling COP at Different Outdoor Temperature Bins Performance on a Very Hot Afternoon (Full-Capacity Forcing)

The system was run at maximum setting on a hot (mid 90ºs F) afternoon to get an idea of its performance in sort of a reverse of the conditions in late December (outdoor temperatures near 0ºF at times). That is, the cooling setpoint was put in the low 60ºs F and the fan set to run in HI mode.

Interestingly, the system maintained cooling output of about 9500 Btu/hr for much of the time and an average temperature split of almost 30ºF (so delivery temperatures range from about 50ºF to low 40°s over the course of the afternoon; the lower temperatures corresponded to return air temperatures in the low 70ºs F after about three hours of operation). Input energy was similar to that during the peak winter cooling conditions, averaging about 980 Watts. Nominal efficiency averaged about COP 2.9 over the course of the afternoon (or about EER 10). The nominal efficiency under these conditions is not nearly as impressive as was found when the system operated under part-load conditions, so apparently the type of operating scheme is important in overall efficiency. Still, a system of this size (nominal cooling output of about 12,000 Btu/hr) would keep a good-sized room comfortable.

From a peak load perspective, it is significant that the maximum load of this system is only about 1 kW (versus at least 2 kW for most newer split system air conditioners, which would start at a nominal size of 2 tons versus 1 ton for this system). This technology could be expected to keep a medium size room comfortable during peak cooling conditions in much of the Northwest.

Indoor Unit Fan Performance (Deferred Maintenance)

A few days before these data were collected, the system was checked out for summer monitoring. During the course of this checkout, we realized there was a problem with the airflow measurement; even on High setting, the system was only delivering about 265 CFM. (Measurements from before had found about 320 CFM on the High setting.) The system filter was evaluated and found to be very dirty. See Figures 4 and 5 for further details. Note the indoor fan on this system does <u>not</u> have an ECM-type motor (which will vary RPM and CFM as resistance to flow increases). This is typical of the inverter-driven DHPs and is a surprise to some.

Data from days prior to the summer checkout was examined. The system had been set to High fan as an experiment (rather than having it run on Auto fan mode, which tends to prioritize either Low or Medium fan speed in cooling mode). Use of 265 CFM as the system flow (rather than about 320 CFM) caused a significant degradation in system performance. Over the course of an afternoon of similar temperatures to those mentioned above (about 81ºF average temperature, so a bit cooler, actually), COP averaged about 2.5, or approximately 22% worse.

This finding is not terribly surprising but does reinforce the need for regular system maintenance on any HVAC appliance. We built a wire mesh barrier on top of the unit to protect the air inlets during the rest of the summer's testing.

Figure 8. Fouled Filter in Place in Unit

(note supply temperature array and side of enclosure box at rear of photo)

Figure 9. Partially Cleaned Filter and Source of Fouling

(although this cat is not the primary perpetrator)