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ACCELERATED LIFE TEST AND FIELD TEST PERFORMANCE RESULTS FOR AN INTEGRAL HEAT PUMP WATER HEATER

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ABSTRACT

A residential heat pump water heater (HPWH) of integral-type design (heat pump unit mounted directly to tank with no water circulation pump) has undergone both durability and field performance testing. Results of two rounds of durability testing and evaluation of as-installed field efficiency vs ambient conditions at HPWH location are summarized in this paper. Ten of the HPWHs were run through an initial durability test designed to represent seven to ten years of normal compressor cycling to meet hot water needs of a residence. The first generation control system proved to be the least reliable component of the units with a 40% failure rate for its temperature input sensors and one complete system failure. After modification of the controls a second durability test was run with five units. There were essentially no operational failures during the second test. In parallel with the durability tests, HPWH units were installed in occupied residences at 18 US locations and field performance data were collected for 1-2 years. Results for two units with differing degrees of exposure to seasonal ambient air temperature and supply water temperature variations are presented and interpreted in this paper. For the first unit, installed in a conditioned space with well water supply, such seasonal temperature variations were small. The average field-measured coefficient of performance (COP) over a 75-week test period for this unit was 2.44 ±5% and weekly averages ranged from 9% below to 10% above the overall average. For the second unit, installed in an unconditioned space with city water supply, the seasonal temperature variations were substantial. Its average COP was 1.81 ±5% over a 104-week test period and weekly averages ranged from 38% below to 28% above that value.

NOMENCLATURE

COP – coefficient of performance
DOE – United States Department of Energy
EF – energy factor
EWH – electric resistance water heater
HPWH – heat pump water heater

INTRODUCTION

The HPWH examined in this study was intended to be a "drop-in" replacement for residential electric water heaters (EWH), and is shown in a schematic cut away view in Figure 1. A small air-to-water vapor compression heat pump unit (about 3400 Btu/h or 1 kW heating capacity), which uses R-134a as the refrigerant, is located on top of a conventional EWH water tank (50 gallon or 190 L nominal capacity). Heat to the evaporator is provided by ambient air. The condenser coil is wrapped around the bottom half of the water tank to provide heat to the water. Conventional EWH electric resistance heating elements (one at top and one at bottom of tank) are included to provide backup to the heat pump unit (or emergency heating in event of heat pump failure).

The design is based on a patented concept originally developed in 1999 [1,2]. Development of this HPWH design is described fully by Baxter and Linkous [3] in a detailed project report. Ten prototype units were built in late summer of 2000 for the first of two durability test programs discussed in this paper. Another eighteen units were built for a field test program [4]. The HPWHs have a solid-state, microprocessor based control system. The system includes a programmable microprocessor chip, which contains the control program, a thermostat (variable resistance potentiometer) and four temperature inputs – compressor discharge temperature, evaporating refrigerant temperature, and upper and lower tank water temperature. The control program operates the HPWH based on values of the inputs and the control logic.

DURABILITY TESTING

Round 1 reliability results

The durability testing consisted of running the HPWHs through about 7300 water heat cycles over a 9-10 month period to represent about 7-10 years of normal operation in a residence. A test facility was designed and installed to operate the test units under the accelerated cycling schedule. It included an environmental chamber to house the test units and subject them to a set of representative ambient conditions that grew progressively harsher with time and number of cycles (Table 1). The goal was to identify design and component weaknesses that could impact the reliability and performance of the HPWH over 10 years of simulated residential use. The testing protocol and test facility along with detailed round 1 results are described in detail by Baxter and Linkous (2002 and 2003) and briefly summarized herein.

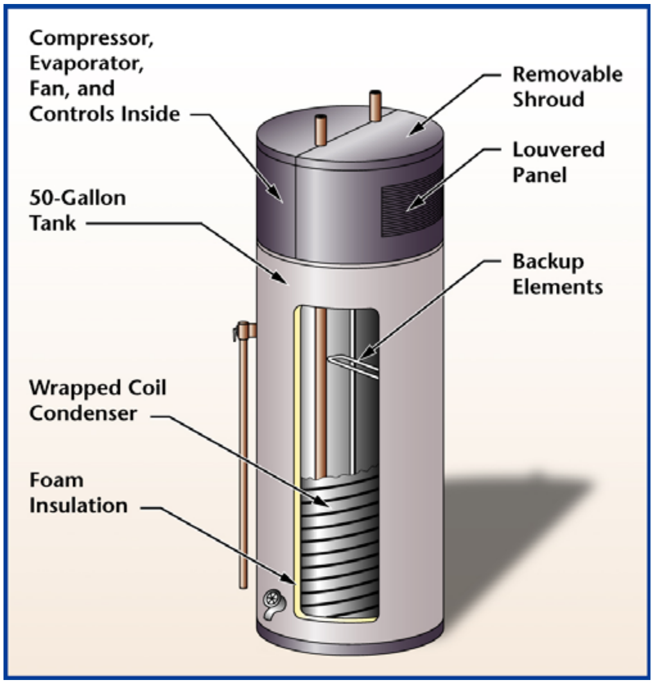

Figure 1 - Cutaway schematic view of the HPWH

Table 1: Operating conditions for each stage of durability test protocol

Stage	Cycles	Ambient air conditions	HPWH power supply voltage
1	2000	75-80 °F (23.9-26.7 °C) dry bulb temperature 50% relative humidity	230 volts AC
2	2000	75-80 °F (23.9-26.7 °C) dry bulb temperature 80% relative humidity	230 volts AC
3	2000	100 °F (37.8 °C) dry bulb temperature 50% relative humidity	230 volts AC
4	1200	100 °F (37.8 °C) dry bulb temperature 50% relative humidity	15% reduced voltage
5	100	67.5 °F (19.7 °C) dry bulb temperature 50% relative humidity	230 volts AC

No durability testing was conducted at lower ambient temperatures (<41 ºF or 5 ºC). However, see the discussion in the last paragraph under the field test section of this paper for some commentary.

The initial durability test program commenced on December 21, 2000, and ended October 7, 2001. There were no major mechanical failures on any of the HPWHs over the course of the first test period. The principal source of problems was the control system temperature input sensors. Sixteen of the sensors out of 40 (total for all ten HPWHs) failed giving a 40% failure rate. These sensors were thermistors that had very fine 28 gauge lead wires. They came with about 6 in (152 mm) lead lengths necessitating a spliced connection to provide connecting leads from the thermistor location to the control board terminal points (see Figure 2). All of the thermistor failures were due to failures of these splices either as shorts or open circuits. Five failures occurred on one unit alone, and this same unit also experienced failures of two control boards. Several units had problems with thermostats and control boards coming loose from their mountings and one thermostat failure occurred. In addition all ten control units experienced occasional erratic behavior caused by random electronic noise spikes in the low voltage control circuitry. The source of the noise problem is not known with complete certainty but is felt to be due in large part to the fact that the low voltage sensor wires were bundled with the high voltage power wiring for the heat pump and backup electric heating elements.

Both the tanks and compressors were subjected to tear down and examination after the durability test run. No excessive or unusual wear was noted on any of the tanks. Nor were any problems evident with the condenser coil wrap (condenser remained firmly adhered to tank wall). The compressors were also in very good shape but there was a consistent wear pattern noted on the crankpin bushing on all of the units. Discussions with the compressor manufacturer indicated that this wear pattern was probably due to heavy loading associated with excessive compressor discharge temperature (>240 ºF or 115.5 ºC) experienced during the testing by all ten units.

Based on the results a number of recommendations were made for improvement of the control system reliability. These include the following.

● Upgrade thermistor sensors to heavier gauge lead wires and eliminate splices.

● Separate low voltage sensor cables from high voltage power wiring to reduce susceptibility to random noise.

● Modify control program to sample thermistor and thermostat inputs multiple times and make control decisions based on the average value. This would help offset the impact of a single aberrant reading.

● Institute a vigorous factory quality assurance (QA) program to minimize occurrences such as unsecured thermostats, control boards, and similar items.

Round 2 reliability results

Following the initial durability test program, our manufacturing partner made a number of changes to the control system design to reduce production costs and to implement the recommendations above. Five new HPWH units featuring the new control design were shipped to our laboratory for a second round of durability tests. Figure 2 illustrates the major changes made to the control system.

The second durability test commenced on February 11, 2003, and continued until November 14, 2003 using the same range of ambient and voltage conditions as for the first round (Table 1). As in Round 1, there were no mechanical system or component failures of any kind. In addition there were no control system failures at any time while chamber ambient conditions were maintained at the planned Stage 1-4 levels. There was, however, one inadvertent departure from the planned testing conditions that resulted in control failures on two units. Just after Stage 4 started, the test facility suffered two shutdowns on successive nights. As a result the dry bulb temperature in the test chamber rose to about 135 ºF (57 ºC) and the dew point temperature to about 120 ºF (49 ºC). After exposure to this extreme condition for a total of about 12 hours the control boards and evaporator temperature sensors failed on two of the test units. After replacement of these components, both units operated without incident for the remainder of the durability test period. In addition, unlike the first test sequence none of the HPWHs in Round 2 experienced any erratic behavior of the controls. The design modifications seem to have effectively eliminated electrical-noise-related problems with the control systems.

Post-test tank and compressor examination revealed no excessive or unusual wear on any components, including the compressor bushings. The updated HPWH controls did not permit compressor discharge temperatures to exceed about 225 ºF (107 ºC) during this test period, which would tend to reduce the loading on the bushing.

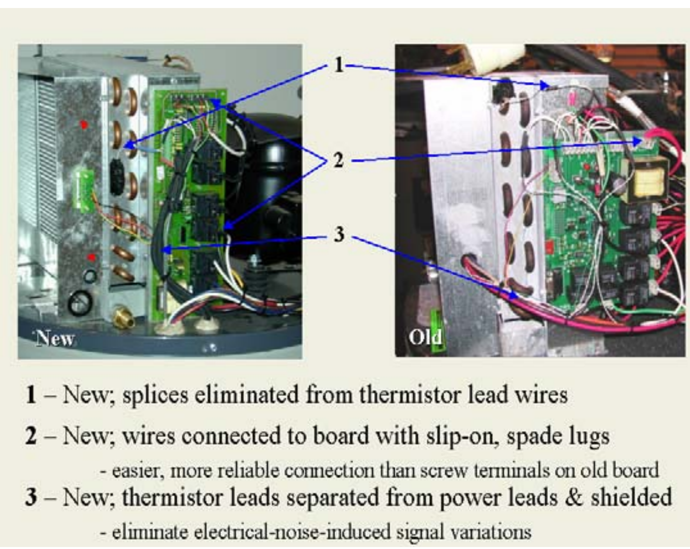

Figure 2 – HPWH control board with changes made after Round 1 durability test

Unit efficiencies

The standard approach to reporting efficiencies for rating residential storage-type water heaters is to measure the energy factor (EF). EF is measured using a standard test of 24-hr duration as outlined in the DOE Simulated Energy Use Test Procedure [5]. Energy factors were measured for each of the test HPWHs during both durability test runs. Results showed an average EF of 2.1 with values ranging from 1.8 to 2.4. The uncertainty for the measured EFs is estimated at ±5% (see Appendix A for discussion of uncertainty limit). In comparison, EF values for conventional electric water heaters range from 0.86 (minimum allowed) to about 0.95. Several of the compressors from Round 1 were subjected to calorimeter tests before tear down. Results of these tests are given in Table 2 along with manufacturer's reference values (for new compressors). None of the compressors tested showed any significant capacity or COP loss compared to the "as new" performance levels.

Table 2: HPWH compressor calorimeter test results [130 ºF (54.4 °C) saturated condensing temperature; 45 ºF (7.2 °C) saturated evaporating temperature; 220 Volts, 60 Hz power supply]

Compressor	Test date	Capacity (Watts)	Power (Watts)	COP
1	2/15/02	1043.5	443.8	2.35
2	12/4/01	1028.6	445.5	2.31
4	2/15/02	1081.8	444.6	2.43
7	12/4/01	1047.3	461.2	2.27
8	2/15/02	1029.1	451.5	2.28
9	2/15/02	1061.4	450.4	2.36
10	2/15/02	1051.4	450.0	2.34
Reference[a]	-	1034.6	452.6	2.29

[a] Manufacturer's reference performance in "as new" condition.

FIELD TESTING

Eighteen HPWHs were instrumented, pre-tested in a laboratory environment, installed in a wide variety of occupied host homes across the United States, and monitored to determine performance over 1-2 years [4]. Results indicated that performance was sensitive to hot water usage (amount and pattern), ambient temperature, supply water temperature, and thermostat setting. Daily, seasonal, and weekday/weekend variations were examined. Measured energy usage averaged about 55% less for the HPWH than for a conventional EWH.

Seasonal variations in performance were expected to be relatively small in field situations where both the ambient temperature and the supply water temperature were generally constant throughout the year. Unit 13, located in a Georgia (GA) residence, was installed in a conditioned utility room (characterized by relatively small fluctuations in ambient temperature) and was supplied by well water (characterized by relatively small fluctuations in supply water temperature). Weekly values of minimum evaporator temperature, average ambient air temperature, and normalized (to average) COP acquired over a time span of 75 weeks are shown in Figure 3. Periods with incomplete or atypical (vacations, etc.) data have not been included. The data for the ambient air temperature measured near the HPWH vary between 64 and 72°F (17.8 – 22.2 ºC) during this period. The measured evaporator temperature data generally follow the same pattern, varying between 44 and 54°F (6.7 – 12.2 ºC) during the same period. Of course, the evaporator temperature is displaced considerably below (approximately 19°F or 10.6 ºC) the ambient air temperature to accommodate the required heat transfer from the ambient air to the evaporator. The corresponding normalized

COP data in the same figure show relatively small fluctuations (from 9% below to 10% above the average) during the same period. The average measured COP for this unit was 2.44 ±5% over the entire period.

Unit 14, located in a North Carolina (NC) residence, was installed in an unconditioned garage and was supplied by a city water system. This situation subjected the HPWH to substantial seasonal fluctuations in both the ambient temperature and the supply water temperature. The corresponding weekly averages acquired over a period of 104 weeks are shown in Figure 4. In contrast to the previous figure, significant seasonal variations are evident. The data for weekly average ambient air temperature measured near the heat pump water vary between 37 and 82°F (2.8 – 27.8 ºC) during this period. Minimum evaporator temperature data follow a similar pattern, varying between 27 and 58°F (-2.8 – 14.4 ºC) during the same period. The normalized COPs also show relatively large fluctuations (from 38% below to 28% above the average). Each of the three parameters are in phase (that is, low COP coincides with low ambient temperatures and low evaporator temperatures) and show cyclic variations closely correlated with the change in seasons and the associated change in outdoor average air temperatures. The average measured COP for this unit was 1.81 ±5% over the entire period. This is lower than that of Unit 13 for several reasons – 1) the overall average ambient temperature and minimum ambient temperature for Unit 14 was lower (see discussions below), 2) the water use pattern of the NC family included many more heavy hot water use incidences than for the other family causing more usage of the backup electric elements, and 3) the hot water thermostat was set somewhat higher by the NC family than by the GA family resulting in higher condenser temperature operation for Unit 14.

It should be noted that, for Unit 14, the displacement of the evaporator temperature below the ambient air temperature varied from 27°F (15 ºC) at higher ambient air temperatures to 9°F (5 ºC) at lower ambient air temperatures. This reduction in heat transfer temperature difference reflects the reduction in air heat removal capacity of the heat pump system at low ambient air temperatures. Of course, the power requirement of the heat pump system also drops at low ambient air temperatures because of reduced compressor load. However, as the normalized COP data show, the heat removal capacity reduction is greater than the power reduction. Decreased ambient temperatures were expected to cause (1) decreased ideal (that is, Carnot potential) cycle performance because the system was required, in this situation, to lift heat over a larger temperature range and (2) increased standby losses (more of the heat provided by the heat pump system was being used to make up heat lost through conduction of heat from the tank to the surrounding air).

Another characteristic of the normalized COP data for Unit 14 in Figure 4 is that the depth of the "valleys" at low ambient temperatures is greater than the height of the "peaks" at high ambient temperatures as compared to the average. In fact, the extremes of the "valleys" coincide with the lowest ambient air and evaporator temperatures seen. In these situations, in order to prevent potential frost buildup on the evaporator coil, the HPWH controls automatically terminated operation of the heat pump system (compressor and fans) and activated the tank's lower resistance element to provide the required heat. The resulting electric resistance heat operation substantially reduced the weekly average COP below that achievable with the heat pump system alone (if frost were not a problem).

CONCLUSIONS

From the durability testing

1. The basic heat pump system hardware seems to be very robust. During both durability runs none of the HPWHs experienced any mechanical system component (compressor, fan, or power-switching relays) failures.

2. During Round 1, the control system experienced erratic behavior and suffered failures of sixteen temperature input sensors (thermistors) failed – a 40% failure rate. Two control boards also failed. As a result of this test program, fixes were identified and implemented and no failures occurred during the planned testing conditions for Round 2. {NOTE: The only problems occurred after a test loop outage resulted in inadvertent exposure of the units to extremely hot and humid conditions [>135 °F (57 °C) dry bulb and >120 °F (49 °C) dew point]. Two control boards and two temperature sensors failed after 12 hrs of exposure to these conditions. However, it is not likely that a residential water heater will be installed where such ambient conditions are expected to be prevalent. So this is not considered to be a long-term reliability problem.}

3. The units' efficiency looks very promising. Energy factors (EF) measured during the durability tests were at least twice that of conventional electric water heaters. Compressor calorimeter test results indicated no efficiency degradation after undergoing over 7000 repetitive cycles – representing about 10 years of normal service for a residential application.

From the field tests

1. Seasonal variations in weekly average COP were found to be relatively small in a residential field test situation where the unit was installed in a conditioned space with well water supply where both the ambient temperature and the supply water temperature were relatively constant throughout the year. In this case, the weekly average COP varied from 9% below to 10% above the overall average for the 75-week test period.

2. In a second situation, where the unit was installed in an unconditioned space with city water supply, substantial seasonal variations in ambient air and supply water temperature occurred. In this case, the weekly average COP varied from 38% below to 28% above the overall average for the 104-week test period.

3. Average field efficiency (COP) for this HPWH design varies significantly for different locations depending upon ambient air temperature levels, hot water usage patterns, thermostat setting, and supply water temperatures. These variations will also affect energy savings vs a conventional EWH.

4. For the two specific field test HPWHs discussed in this paper, the overall average measured COPs were 1.81 and 2.44. In comparison, the rated efficiency for conventional EWHs ranges from 0.86 (minimum allowed by DOE) to about 0. 95. Thus this HPWH is more than

twice as efficient as an EWH. Average energy savings for all eighteen field test units compared to the EWH alternative was 55% [4].

APPENDIX A - EFFICIENCY MEASUREMENT UNCERTAINTY

The average hot water delivery efficiencies, or COPs, from the field tests reported in this paper are calculated by

$$COP = Q/W \qquad \text{(A-1).}$$

Q is the total hot water energy delivered by the HPWH to the house delivery system over a given time period and is calculated by

$$Q = m \cdot c_p \cdot \Delta T_{avg} \qquad \text{(A-2),}$$

where

m – total mass of hot water delivered to delivery system over the given time period, lb (kg),

ΔT_{avg} – average difference between hot water leaving HPWH (T_{out}) and cold water entering HPWH (T_{in}) over the given time period, °F (°C), and

c_p - specific heat of water at the average of the entering and leaving water temperatures.

W is the total energy consumed by the HPWH over the given time period.

Water used by the HPWH was measured using a positive displacement flow meter having a stated accuracy of ±1.5% for flow rates of interest in this study - from 0.5 to 15 gpm (0.11 to 3.40 m^3/h). Energy consumption was measured using a power transducer with a stated accuracy of ±0.5%. Water temperatures (T_{in} and T_{out}) were measured using Type T thermocouples immersed in the water streams. These thermocouples have a standard commercial specification of about ±1.5 °F (± 0.8 °C) for the temperature range of interest in the field and durability tests. During operation the ΔT is very small at the beginning of a hot water use but reaches a value of 70 °F or more after about 5 seconds of water flow. Most hot water used during the field tests occurred during usages lasting for several minutes. Accepting 70 °F (38.9 °C) as the nominal value for ΔT_{avg} and using the standard thermocouple accuracy above yields a nominal accuracy for the temperature difference measurement of approximately ±4.3%. Given the accuracies for the individual quantities in equations A-1 and A-2, the overall uncertainty in COP is estimated to be about ±5%.

The energy factor (EF) values reported in the durability test portion of the paper are measured in essentially the same way as described above. Therefore the overall uncertainty in EFs reported herein is also about ±5%. The basic difference between the EF as determined according to the standard test procedure [5] and the field average COPs is that EF is normalized for standard water temperatures and ambient air conditions so it can be used as a rating value to compare efficiencies of different water heaters. The test procedure requires normalization to a ΔT_{avg} of 77 °F (42.8 °C) [T_{in} = 58 °F (14.4 °C) and T_{out} = 135 °F (57.2 °C)] and for an average ambient air temperature of 67.5 °F (19.7 °C).

ACKNOWLEDGMENTS

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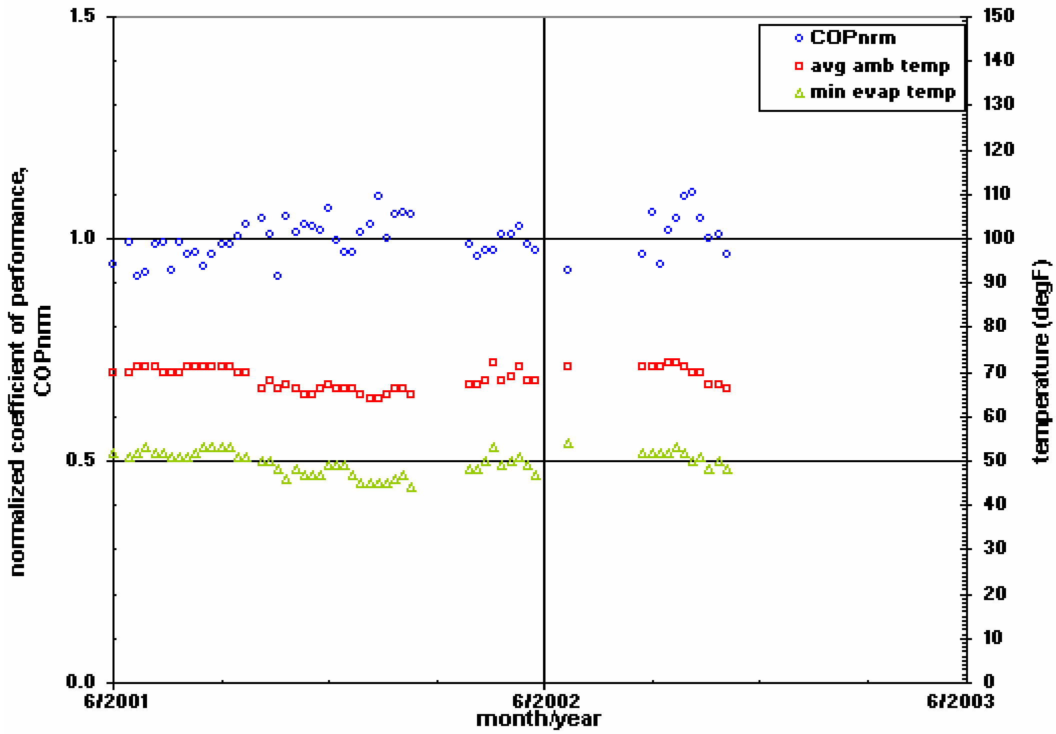

Figure 3 – Weekly average ambient temperature, minimum evaporator temperature, and normalized COP for field test Unit 13

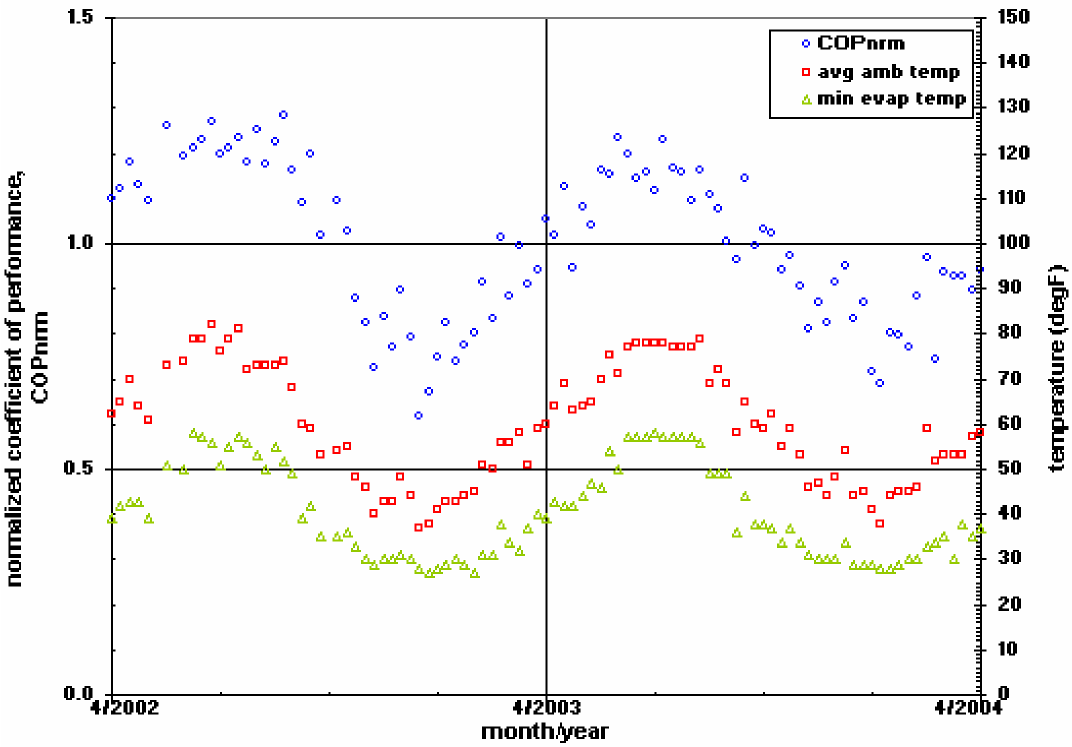

Figure 4 – Weekly average ambient temperature, minimum evaporator temperature, and normalized COP for field test Unit 14