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GULF POWER
A SOUTHERN COMPANY

February 22, 1999

Ms. Blanca S. Bayo, Director
Division of Records and Reporting
Florida Public Service Commission
2540 Shumard Oak Boulevard
Tallahassee FL 32399-0870

990000

Dear Ms. Bayo:

RE: Docket No. 950520-EI

Enclosed are an original and fifteen copies of the of Gulf Power Company's Natural Gas Research and Demonstration Project Report which was requested in Order No. PSC-95-1146-FOF-EG in the above docket.

Sincerely,

Susan D. Ritenour

Susan D. Ritenour
Assistant Secretary and Assistant Treasurer

lw

Enclosures

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Jeffrey A. Stone, Esquire

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# Field Testing of Gas Engine-Driven Heat Pumps

**Final Report**
**To**
**Electric Power Research Institute**

**AIL Research, Inc.**
**18 Cameron Court**
**Princeton, NJ**

**November 1998**

## ABSTRACT

An engine-driven heat pump (EDHP) was tested for two years at four sites, two in the southeast, one in the mid-Atlantic and one in the mid-west. During both heating and cooling, the EDHP modulated its speed so that its output matched the load on the building. Auxiliary heat was provided by a gas-fired glycol boiler that was located in the outdoor unit along with the engine and compressor. When the building required auxiliary heat, the boiler would turn on and a glycol pump would deliver hot glycol to a heating coil that was located within the indoor air handler. All tested units also used the glycol loop to recover "waste" heat from the engine for space heating when needed.

Three sites were single-family homes and the fourth was a small office building. All sites experienced control and hardware problems that ranged from minor incidents that were detected by the data acquisition system but were not apparent to the owner to the failure of major components (which included the replacement of the engine at one site). The control problems at one site never were resolved despite numerous attempts by the service contractor. At this site, the control problems were related to the operation of the glycol loop. The air handler was located about 15 feet above the engine, which is close to the manufacturer's 18 foot limit on air handler height.

Three of the four EDHPs that were tested were 3-ton units and the fourth was a 3.5-ton unit. The heating and cooling performance of the three 3-ton units during the one-year period starting September 1996 were comparable: cooling COPs ranged from 1.21 to 1.26 and heating COPs ranged from 1.11 to 1.29 (where all COPs are based on gas consumption only). The larger 3.5-ton unit, which was slightly oversized for the site and tended to run at lower engine speeds, had higher cooling and heating COPs: 1.62 and 1.47 respectively. As expected, electric consumption for the gas-fired EDHP was low but not negligible. The EERs for the sites (based on the heating/cooling provided divided by the electricity used to run fans, pumps and controls) ranged from 42.2 to 64.9 kBtu/kWh during the cooling season and 15.7 to 71.5 kBtu/kWh during the heating season. (The 15.7 kBtu/kWh EER during heating occurred at the one southern site that had a very low heating load.)

For the four test sites, the EDHP did provide operating cost savings compared to a HVAC system that used an electric air conditioner and a gas furnace, although the savings were relatively small: between \$53 and \$140. At all sites the savings are less than the higher annual maintenance costs charged by the service contractors in this test: annual maintenance to replace spark plugs, change oil and perform other minor tasks ranged from \$200 to \$350.

## EXECUTIVE SUMMARY

Although gas cooling is now mostly limited to sizes of 100 tons or greater, the gas industry is attempting to move it towards smaller sizes. Twenty to 50 ton engine-driven systems are now made by several manufacturers. It was the objective of this project to test a 3-ton residential engine-driven heat pump (EDHP) that had recently been introduced to the U.S. market.

An EDHP is very similar to a conventional electric heat pump with one major difference: a gas-fired internal combustion engine drives the compressor rather than an electric motor. The speed of the engine is modulated so that the heating or cooling output from the EDHP closely matches the loads on the building. This improves both the efficiency of the unit and the comfort it provides. For the EDHP that was tested, a variable-speed electronically commutated motor drives the indoor fan. It is controlled so that fan power is reduced during part-load heating and cooling.

The EDHPs that were field tested were all "four-pipe" systems in which the indoor air handler has a refrigerant and a glycol heat exchanger. Two of the four pipes transfer refrigerant between the outdoor unit and air handler, and two transfer glycol. The glycol loop is used only in winter to both recover heat from the engine and to provide supplemental heat from the auxiliary burner (during periods when the heat pump's capacity is insufficient or the heat pump is inoperable).

In 1995, EPRI commissioned AIL Research, Inc. to evaluate the performance of an EDHP through two heating and cooling seasons. A major part of this evaluation was the detailed monitoring of four EDHPs--one in the Midwest, one in the mid-Atlantic, and two in the Southeast. Site 1, located in the Columbus, GA, was an office building. The other three sites were single-family homes: Site 2 in Columbus, OH, Site 3 in Roanoke, VA, and Site 4 in Charlotte, NC. Data on each system's performance, comfort conditions within the buildings and the weather at each site were collected at one minute intervals throughout the evaluation with essentially no gaps in the data. This data was then used to meet the project's primary objective: to accurately determine both the seasonal performance and operating costs of the EDHP.

Figure 1 shows a general layout of the EDHP and the instrumentation that was used in the field test. The instrumentation falls into the following categories:

- indoor comfort conditions
- outdoor weather conditions
- air handler operation
- heat pump status
- natural gas flow
- heat pump "internal" operation
- heat pump energy inputs

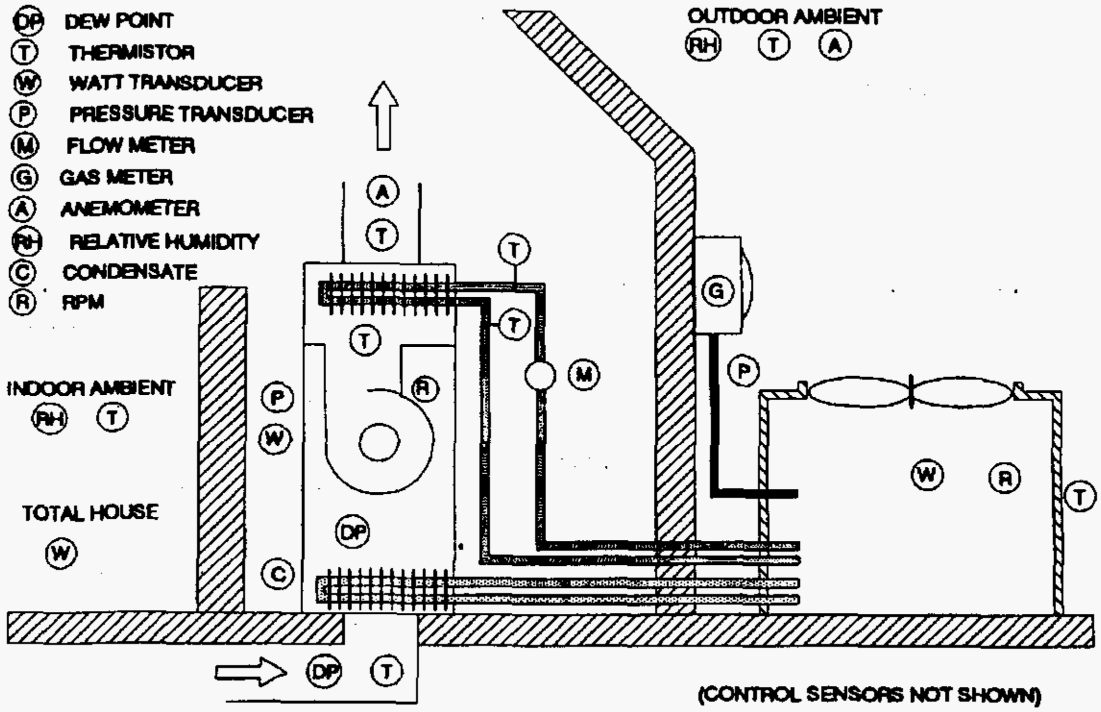


Figure 1 - Placement of Instrumentation

Monitoring of the four test sites began on the following dates:

| | |
|---|---|
| Site 1 | August 25, 1995 |
| Site 2 | October 24, 1995 |
| Site 3 | October 24, 1995 |
| Site 4 | November 23, 1995 |

Monitoring continued for approximately two years and ended on September 23, 1997 at all sites.

The data loggers all operated without problems for the entire field test. The precautions that were taken to secure the databases--in particular, the seven days of on-site data storage and the frequent screening of data--were very effective at minimizing the loss of data. For the approximately two years during which data was collected, data loss averaged about 3 days per site. However, most of this data loss occurred early in the test period when the data loggers were being reprogrammed. For the 20 month period starting February 1, 1996, only 27 one-minute data records were lost out of a total of 3.5 million.

The EDHPs experienced numerous minor and major problems during the test. These are summarized below.

Site 1

- Failure and replacement of engine starter
- Glycol leak

Site 2

- Failure and replacement of engine
- Low-pressure cut-out at -3°F ambient
- Control problems caused by shorted wires (one in thermostat, one in outdoor unit)

Site 3

- System cut-out at 18°F ambient
- Engine starting problems (problem corrected itself without service call)

Site 4

- Chronic control problems apparently caused by glycol loop (warning light came on 34 times in two winters; thermostat replaced twice)
- Engine starting problems (problem corrected itself after homeowner reset the system)

A major objective of this field test was to provide data that could be used to compare the operating costs for the EDHP with alternative heating and cooling technologies. The first step towards meeting this objective is to verify that the field installations were performing close to their catalog specifications. (In this analysis, the operational problems that occurred at the sites were ignored, and the EDHP was studied only during periods when it was running well.)

The steady-state capacity and efficiency of the 3-ton EDHP that was tested as a function of outdoor temperature has been published by the manufacturer. This performance data is based on tests performed in an environmental chamber that maintained indoor conditions at either the ARI summer (80°F, 50% rh) or winter (70°F) test conditions. (Unfortunately, performance data for the 3.5-ton EDHP was not available, and so it was not possible to validate the field performance at Site 4.)

During normal operation, the EDHP continually varies its engine speed to match its output with the load on the building. This complicates a comparison between field-test data and catalog data, since the latter data only applies to steady-state operation. To overcome this problem, we used field data for periods when the EDHP operated at a single speed for at least 20 minutes and discarded the data for the first ten minutes. The only limitation that this procedure imposes is that the EDHP's performance can only be validated at its highest and lowest speeds (i.e., 3000 rpm and 1200 rpm), since these are the only two speeds that will have extended operation at a constant speed.

A second problem, which is described in more detail in Section 7 of the report, is that the measurement of air flow through the air handler had a large uncertainty during low-speed operation.

In general, the in-field performance of the EDHPs at high speed agreed well with the catalog data. Most discrepancies were below 5%. The only exception occurred at Site 2 where the heating capacity and COP of the EDHP at high speed deviated by between 9% and 14%.

The low-speed performance of the EDHPs did not agree as well with catalog data. This is attributed to the previously noted uncertainty in the air flow measurement at low speeds. Most discrepancies were below 15%. Site 2 during heating was again an exception—deviations were between 17% and 23%.

A summary of the comparison between field test data and catalog data appears in Table 1. (In this table, a positive deviation indicates in-field performance was higher than catalog data. Also, there was insufficient heating data to compare the EDHP's performance at Site 1 with catalog data.)

Table 1

High-Speed Performance

| No | Summer | | Winter | |
|---|---|---|---|---|
| | COP | capacity | COP | capacity |
| 1 | +4% | +3% | - | - |
| 2 | -5% | +5% | +9% | +14% |
| 3 | +3% | -0.5% | 4.5% | 3% |

Low-Speed Performance

| No | Summer | | Winter | |
|---|---|---|---|---|
| | COP | capacity | COP | capacity |
| 1 | +15% | +15% | - | - |
| 2 | +9% | +14% | +23% | +17% |
| 3 | -2% | -1% | +12% | +7% |

The performance of the four EDHPs for the one-year period from September 1996 through August 1997 is shown in Table 2.

Table 2

Annual Performance of the EDHPs
(COPs are based on gas use only)

| | Cooling | | | Heating | | | outage | fan only |
|---|---|---|---|---|---|---|---|---|
| Site No. | MBtu | COP | MWh | MBtu | COP | MWh | days | hours |
| Site 1 | 50.5 | 1.26 | 1.20 | 8.2 | 1.11 | 0.52 | 2 | 5,834 |
| Site 2 | 13.0 | 1.21 | 0.30 | 91.7 | 1.27 | 1.71 | 14 | 5,315 |
| Site 3 | 23.3 | 1.22 | 0.40 | 39.9 | 1.29 | 0.58 | 0 | 686 |
| Site 4 | 33.1 | 1.62 | 0.51 | 34.9 | 1.47 | 0.49 | 0 | 1,251 |

The three sites with the 3-ton EDHP (i.e., Sites 1, 2 and 3) had comparable performance: COPs for the cooling season were between 1.21 and 1.26, and for the heating season, between 1.11 and 1.29.

The performance of the 3.5-ton EDHP at Site 4 is significantly better than the performance of the 3-ton units. The larger unit at Site 4 did tend to run at a lower engine speed, which could explain part of the difference. Unfortunately, since catalog performance data is not available for the 3.5-ton unit, it is difficult to determine whether the seasonal performance at Site 4 is reasonable.

The EDHP that was tested will have relatively low operating costs considering the high COPs that were measured during the field test. However, will its operating costs be sufficiently low to justify its selection over conventional gas and electric technologies?

To answer the preceding question, two alternative systems were studied--one an all-electric heat pump with a 11.9 SEER and a 7.85 HSPF, and the other a 11.9 SEER electric air conditioner combined with a 95% AFUE gas furnace. Both these alternatives are high-efficiency premium systems, and so they should appeal to the same customers as the EDHP.

An important aspect of the field test was the direct measurement of the heating and cooling output of the EDHP. With this data and the coincident indoor and outdoor conditions, it was possible to simulate the hour-by-hour performance of the alternative electric systems using manufacturer's steady-state catalog data. To account for cycling effects, the total compressor energy that was calculated from catalog data was increased by 6%.

Table 3 compares the energy consumption and operating costs for the EDHP and the two conventional systems. Operating costs have been calculated using $0.0841 per kWh and $0.605 per therm, which are the 1994 national average rates.

TABLE 3

Comparision of Operation Costs for Alternative Technologies

| Site | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| EDHP | | | | |
| electricity (kWh) | 1219 | 1626 | 928 | 912 |
| gas (therms) | 476 | 829 | 500 | 442 |
| operating cost ($) | 390 | 638 | 380 | 344 |
| Heat Pump | | | | |
| SEER 11.9/ HSPF 7.85 | | | | |
| electricity (kWh) | 5541 | 13146 | 6942 | 7207 |
| gas (therms) | 0 | 0 | 0 | 0 |
| operating cost ($) | 466 | 1106 | 584 | 606 |
| Air Conditioner/Furnace | | | | |
| SEER 11.9/ 94% AFUE | | | | |
| electricity (kWh) | 4639 | 2008 | 2448 | 3085 |
| gas (therms) | 87 | 975 | 424 | 371 |
| operating cost ($) | 443 | 759 | 463 | 484 |

NOTE: field test electricity usage adjusted to eliminate periods of continuous fan operation
$0.605 per therm, $0.084 per kWh

As shown in this table, the EDHP does have the lowest operating costs at all four sites. However, its cost advantage is very small. For the four sites, the differences in operating costs between the EDHP and the combined furnace/air-conditioner are: (1) $53, (2) $121, (3) $83 and (4) $140. At all sites these annual savings do not cover the higher annual maintenance costs for the EDHP: in this field test the annual maintenance to replace spark plugs, change oil, and perform other minor tasks ranged from $200 to $350. (Costs for repairing the major problems that occurred in this field test are obviously not covered by the savings in operating costs.)

The EDHP can achieve a high COP during the heating season by recovering "waste" heat from the engine. Heat is transferred from the engine to the indoor air handler via the same glycol loop that transfers heat from the auxiliary boiler.

In this field test, the glycol loop was a source of problems. At two sites, glycol temperatures would exceed their upper limits and shut down the system.

If the glycol loop were replaced by a conventional gas furnace, how much will the EDHP's heating efficiency be degraded? This can be answered by calculating the percentage of the total heat delivered to the building that was recovered from the engine.

Table 4 presents the percentage of total heat delivered to the building that was recovered from the engine during periods when only the engine was operating (i.e., the auxiliary boiler was off). Ignoring Site 1 which had a very low heating load, the recovered heat averaged 23.6 percent of the total delivered to the building. Thus, if the EDHP has a heating COP of 1.25 when recovering heat from the engine, it will have a 0.955 COP without heat recovery.

Table 4
The Percent Contribution to Total Heating from the Engine Waste Heat

| | |
|---|---|
| Site 1 | 16.5% |
| Site 2 | 24.1% |
| Site 3 | 20.0% |
| Site 4 | 26.8% |

# 1

# INTRODUCTION

In the last ten years, the sales of large-tonnage gas-fired cooling systems has increased significantly, spurred mostly by the introduction of higher efficiency double-effect absorption chillers and engine-driven chillers. Although these gas cooling systems tend to have a higher first cost than alternative electric technologies, the very low summertime price for gas in many parts of the country can produce acceptable payback periods for some customers.

Although gas cooling is now mostly limited to sizes of 100 tons or greater, the gas industry is attempting to move it towards smaller sizes. Twenty to 50 ton engine-driven systems are now made by several manufacturers. It was the objective of this project to test a 3-ton residential engine-driven heat pump (EDHP) that had recently been introduced to the U.S. market.

The EDHP is very similar to an electric heat pump that has its electric motor replaced by an internal combustion engine. Since it is fairly easy to modulate the speed of the engine, the EDHP can operate as a variable-speed heat pump. As with electric heat pumps, variable speed improves both the EDHP's seasonal efficiency and the comfort that it provides.

During the winter, the EDHP that was tested operates as a very high efficiency gas heater by supplementing the heat pump with "free" heat recovered from the engine. When the building needs more heat than the heat pump can provide, a supplemental gas burner is turned on. Glycol (i.e., antifreeze) transfers heat from both this supplemental burner and the engine to the air handler within the building. At the 47°F outdoor air temperature ARI winter rating condition, the EDHP that was tested has a 1.7 COP. The EDHP's cooling COP at the 95°F outdoor air temperature ARI summer rating condition is 0.9. (The preceding COPs are based on the EDHP's gas consumption. Electricity for fans, controls and pumps--which can be significant--are not included in the COPs.)

In 1995, EPRI commissioned an evaluation of the performance of an EDHP through two heating and cooling seasons. A major part of this evaluation was the detailed monitoring of four EDHPs--one in the Midwest, one in the mid-Atlantic, and two in the Southeast. Data on each system's performance, comfort conditions within the buildings and the weather at each site were collected at one minute intervals throughout the evaluation with essentially no gaps in the data. This data was then used to meet the project's primary objective: to accurately determine both the seasonal performance and operating costs of the EDHP.

Also, through surveys and calls to the owners at the test sites, information was gathered on both the maintenance requirements of the EDHPs and the owner's impressions.

## 2

# DESCRIPTION OF THE TEST ENGINE-DRIVEN HEAT PUMP

An EDHP is very similar to a conventional electric heat pump with one major difference: a gas-fired internal combustion engine drives the compressor rather than an electric motor. The speed of the engine is modulated so that the heating or cooling output from the EDHP closely matches the loads on the building. This improves both the efficiency of the unit and the comfort it provides. For the EDHP that was tested, a variable-speed electronically commutated motor drives the indoor fan. It is controlled so that fan power is reduced during part-load heating and cooling.

The EDHPs that were field tested were all "four-pipe" systems in which the indoor air handler has a refrigerant and a glycol heat exchanger. Two of the four pipes transfer refrigerant between the outdoor unit and air handler, and two transfer glycol. The glycol loop is used only in winter to both recover heat from the engine and to provide supplemental heat from the auxiliary burner (during periods when the heat pump's capacity is insufficient or the heat pump is inoperable).

Additional specifications for the EDHP are:

| | |
|---|---|
| Cooling capacity at ARI (95°F) | 36,000 Btu/h |
| Heating capacity at ARI (47°F) | 53,500 Btu/h |
| Cooling COP ARI (95°F) | 0.9 |
| Heating COP ARI (47°F) | 1.7 |
| Engine | single cylinder, four stroke, 5 HP |
| Engine life | 40,000 hours |
| Oil inventory | 3 gallons |
| Compressor | two cylinder, reciprocating |
| Outdoor unit dimensions | 36"x 43"x 38" |

# 3

# FIELD TEST DESIGN

The primary objectives of the field test were to (1) determine the operating and maintenance characteristics of the EDHP, (2) verify its performance in the field, (3) compare its operating costs with those of a conventional electric heat pump, (4) uncover possible generic or site-specific operational problems, and (5) assess users' reactions to it.

Several of the preceding objectives require that the heating and cooling output of the EDHP be measured throughout the field test. In general, it is very difficult to accurately measure in the field the output of a heating and/or cooling system that conditions air. The principle problems encountered are (1) air velocity profiles are very non-uniform due to the turns and intersections in the ducts, and (2) temperature and humidity downstream of the heating/cooling coil are also very non-uniform. The EDHP's glycol loop, however, presents a unique opportunity to measure heating and cooling output. The glycol loop supplies heat from the outdoor burner to the air handler. The heat transferred to the building's supply air by this loop can be accurately determined by measuring the glycol flow rate and its temperature into and out of the air handler. The glycol heat transfer can then be used to calibrate the air-side instrumentation. This calibration procedure is described in Section 7.3.

The heating and cooling output of the EDHP will depend on the outdoor temperature, the temperature, humidity and volumetric flow rate of the return air to the air handler and the engine speed. In addition to measuring these parameters, the electrical power and natural gas consumption of the system must be measured so that the COP, EER and operating costs for the EDHP can be calculated.

The comfort provided by an air conditioner is strongly influenced by its ability to control humidity within the building. The primary diagnostic for measuring the EDHP's latent cooling capacity (i.e., water removal capacity) was a tip bucket that directly measured the condensate that flowed off the cooling coil. Since tip buckets can sometimes be unreliable (e.g., dirt can clog the drain line, the tip arm can stick on its pivot), dew point hygrometers were also installed at the inlet and outlet to the air handler as a redundant measurement.

Indoor temperatures and relative humidities were measured at two locations within each building as a check on comfort conditions. At two sites, temperature sensors were installed in the corner of one room at three different heights to measure possible stratification within the building. (The data from the stratification sensors are not analyzed in this report.)

A complete list of the data collected in this field test appears in Table 3.1.

Table 3.1
List of Data Channels

Engine RPM
Status of Reversing Valve
Status of Coolant Valve
Status of Auxiliary Heat
Fuel Flow (cubic feet)
Absolute Pressure of Fuel (psia)
Outdoor Unit Power (W)
Air Handler Power (W)
Total Building Power (kW)
Status of DAS Power
Battery Voltage
Temperature of Air at Air Handler Inlet (F)
Dewpoint of Air at Air Handler Inlet (F)
Temperature of Air Past Fan in Air Handler (F)
Temperature of Air at Air Handler Outlet (F)
Dewpoint of Air at Air Handler Outlet (F)
Temperature of Glycol into Air Handler(F)
Temperature of Glycol out of Air Handler(F)
Glycol Flow (gpm)
Air Velocity (fpm)
Pressure Differential across Fan (in. w.c.)
Fan RPM
Condensate (lb)
Temperature near Thermostat (F)
rh near Thermostat
Temperature remote from Thermostat (F)
rh remote from Thermostat
Temperature of Air entering Outdoor Unit (F)
Outdoor Air Temperature (F)
Outdoor rh
Wind Speed (mph)
Top Room Stratification Temperature (F)
Mid Room Stratification Temperature (F)
Bottom Room Stratification Temperature (F)

# 4

# DESCRIPTION OF THE DATA ACQUISITION SYSTEM

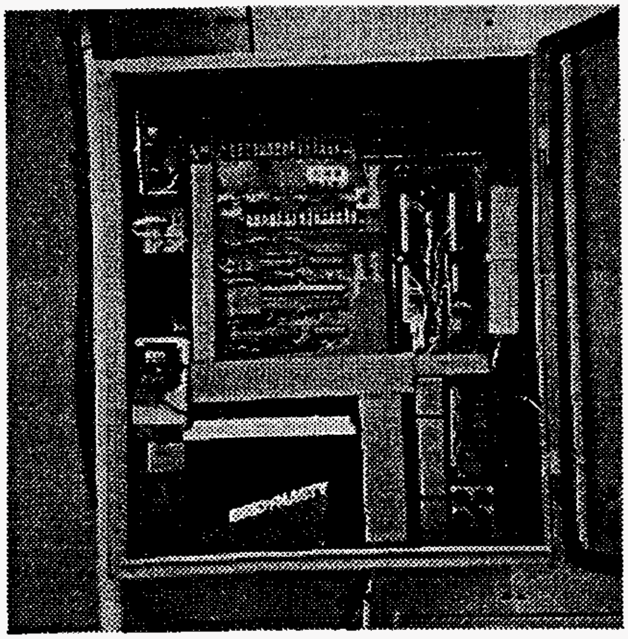

Figure 4.1 – Data Logger for Field Test

The data logger that was used in the data acquisition system (DAS) was the Campbell CR10. This data logger has six differential analog inputs, two pulse counters and eight digital I/O ports. A 13-½ bit A/D converter converts the analog inputs to digital format. Over the temperature range that the data loggers were used in this project, the CR10 introduced an uncertainty of 0.1% of its full-scale voltage measurement into the analog measurements. The CR10 can sample data at rates up to 64 Hz (although in this project, data was sampled at five-second intervals). A Campbell AM416 64-channel multiplexer was used to increase the number of channels of analog data that the DAS could collect. A Campbell SDM-SW8A 8-channel switch closure module was also included in the data logger to increase the number of pulse-counting channels. The CR-10 has 64 kilobytes of random-access memory (RAM) for storing its control program and data. This memory was supplemented with a 750 kilobyte SM-716 storage module. With this extra memory, the DAS could store at least one week of data on-site. Other components included in the data logger were (1) a 14.4 kbps modem, (2) a 33 A-hr battery, (3) a 12V/1.7A DC power supply, and (4) a surge suppressor. All components for the data logger were mounted in a 24" x 20" x 8" steel enclosure. A photograph of the data logger components within the enclosure is shown in Figure 4.1.

Table 3.1 lists the 31 data channels that were logged (34 data channels for the two sites where temperature stratification was studied).

The data logger sampled each channel once every five seconds. Data channels that were not pulse outputs were averaged over one minute and the averages stored. The pulse-output channels were totaled over one minute intervals and stored.

Data was collected nightly from the four sites to a central field-test computer at AIL Research. This "raw" data was regularly processed, typically every one to three days, both to identify data channels that were outside of reasonable bounds and to verify that the test hardware was operating properly. As part of this routine processing, minor adjustments to the data were made. These adjustments included (1) the application of calibration constants that were not included in the data logger program, (2) the correction or deletion of spurious data that was occasionally transmitted from the test site, and (3) the conversion of time from Greenwich Mean Time to local time. These adjustments to the data were made only to a "corrected" database that was stored on a second computer. The "raw" database was always stored as collected from the test sites.

# 5

# Description of Instrumentation

Figure 5.1 shows a general layout of the EDHP and the instrumentation that was used in the field test. The instrumentation falls into the following categories:

indoor comfort conditions
outdoor weather conditions
air handler operation
heat pump status
heat pump energy inputs
heat pump "internal" operation
natural gas flow

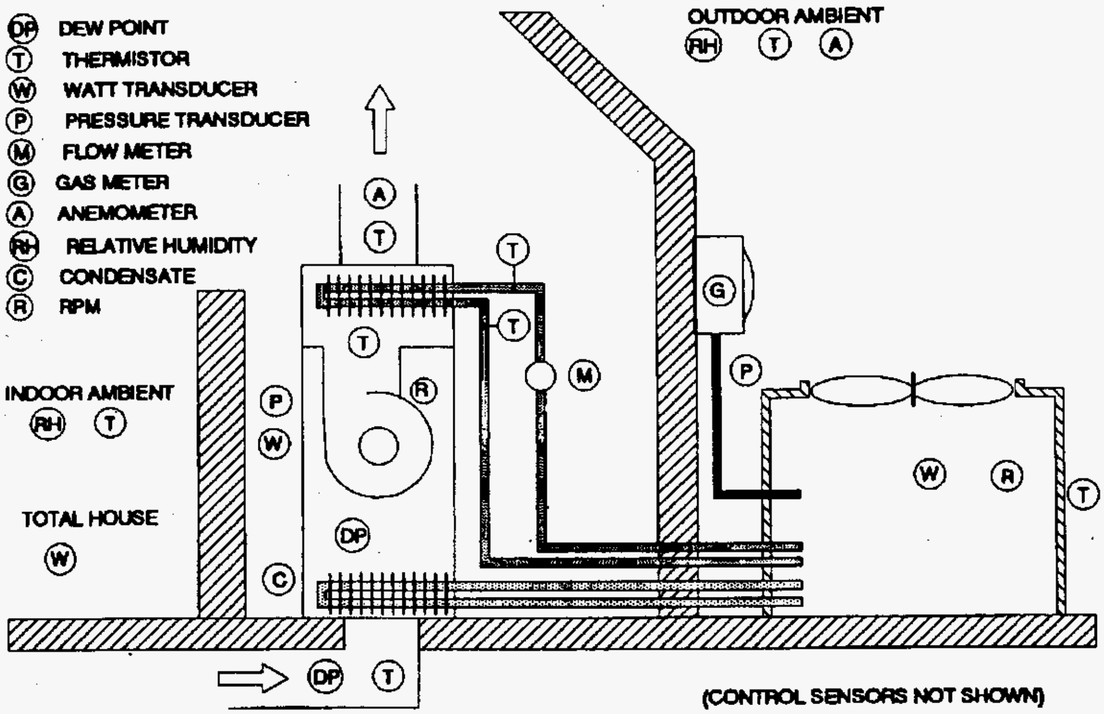


Figure 5.1 - Placement of Instrumentation

## 5.1 Indoor Comfort Conditions

At all sites, indoor temperature and relative humidity were measured at two locations. One T/rh pair was mounted on an internal wall within 12" of the thermostat that controlled the EDHP. For sites that had two levels, the second T/rh pair was located in an upstairs hallway. Otherwise, the second T/rh pair was located remotely from the first.

The indoor temperature sensors were Permalloy RTDs that had an accuracy of ±1°F. The rh sensors were bulk-polymer devices with an accuracy of ±2 points over a range from 3% to 95% rh.

For the two sites where room stratification was studied, thermistors with an accuracy of ±0.3°F were used.

## 5.2 Outdoor Weather Conditions

Outdoor temperature, relative humidity and wind speed were measured at all sites. The temperature sensor for this measurement was a thermistor with an accuracy of ±1.6°F; the rh sensor was a bulk-polymer device with an accuracy of ±3% over a range from 0% to 100% rh; and the wind speed was measured with a cup anemometer with an accuracy of ±2 mph.

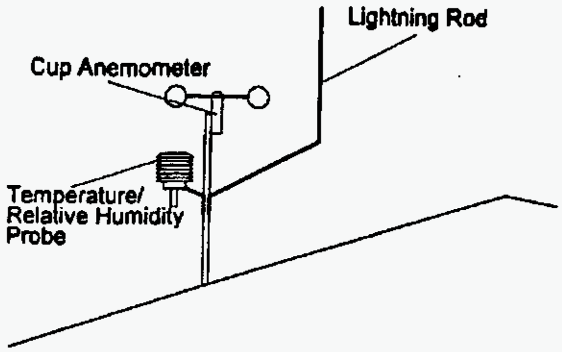


Figure 5.2 – Typical Installation of Weather Instrumentation

The weather instrumentation was mounted at an elevation above the roof of the building on either a stanchion attached to the building or on a high structure, such as a chimney, that was already present. The T and rh sensors were covered by a gill shield to protect them from the weather and direct solar radiation. A typical installation for the weather instrumentation is shown in Figure 5.2.

A platinum RTD was used to measure the temperature of the air entering the outdoor unit. The RTD was a 3' long probe that measures the average temperature over its length. The probe was attached to the grill that covers the outdoor coil at a height of approximately 18" above the ground.

## 5.3 Air Handler Operation

Platinum RTDs and chilled-mirror dew-point hygrometers were used to measure the temperature and humidity of the air entering and leaving the air handler. A third RTD measured the air temperature within the air handler, downstream of the blower but upstream of the glycol coil.

The RTDs were 3' long probes that measure the average temperature over their length. As shown in Figure 5.3, they were bent into a "pretzel" shape that averaged the air temperature over a cross section of the air handler. The locations of the RTDs are also shown in Figure 5.3.

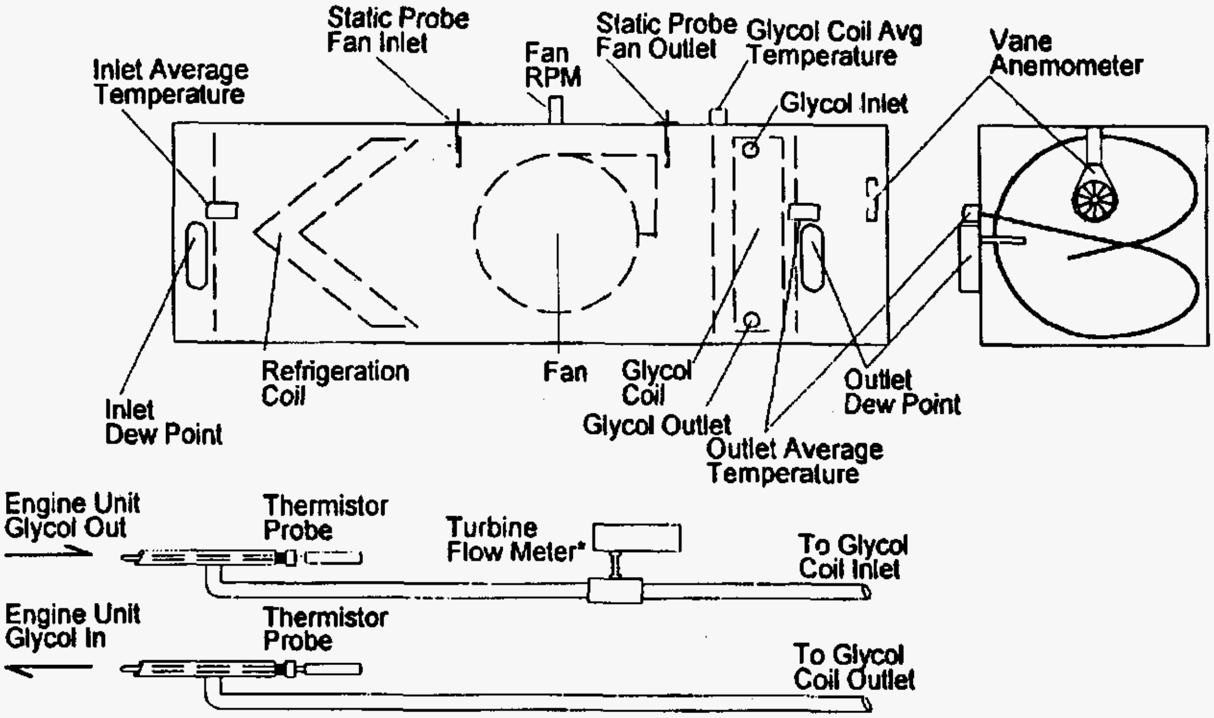


Figure 5.3 - Typical Installation of Air Handler Instrumentation

The RTDs for the supply and return air temperature have an accuracy of ±0.75°F, and the dew-point hygrometers have an accuracy of ±1°F.

The velocity of the supply air was measured with a rotating vane anemometer. Its accuracy was ±10 fpm.

The pressure difference across the air handler's blower was measured with a differential pressure transducer that had an accuracy of ±0.025" w.c.

The blower's speed was measured with an optical retro-reflective sensor. A pulse-counting channel on the data logger accumulated the pulses from this sensor.

The condensate produced by the air handler during the summer is measured with a tip bucket that has an accuracy of ±4% of reading and an output of 0.019 pounds water per pulse.

## 5.4 Heat Pump Status

I/O modules that produce a zero or one output depending on the voltage that is applied to them were used to report on the status of (1) the refrigerant reversing valve (which switches the EDHP between heating and cooling), (2) the glycol valve (which directs the hot glycol towards the indoor air handler during heating and the outdoor radiator during cooling), and (3) the supplemental glycol heater.

## 5.5 Heat Pump Energy Inputs

Separate power transducers were used to measure (1) total building power, (2) air handler power, and (3) outdoor unit power. The transducers that measured the air handler and outdoor unit powers had an accuracy of ±10 W.

## 5.6 Heat Pump "Internal" Operation

The engine speed was measured with an inductive pick-up that detected the spark to the spark plug. A pulse-counting channel on the data logger accumulated the counts.

The temperature of the glycol into and out of the air handler was measured with high-precision thermistors that had an accuracy of ±0.3°F. A turbine flow meter was used to measure the glycol flow rate. This meter had an accuracy of ±2% of reading and an output of 0.00254 gallon per pulse.

## 5.7 Natural Gas Flow

A temperature-compensated gas meter was used to measure the flow of natural gas to the EDHP. This meter had an accuracy of ±1% of reading and an output of 0.05 cubic foot per pulse. Pressure corrections to the gas reading were made by reading the gas delivery pressure with a pressure transducer that had an accuracy of ±0.0375 psi.

For each site, the local gas utility was called to determine the heating value for the natural gas. The heating values that were reported by the utilities were:

| | |
|---|---|
| Site 1 | 1020 to 1035 Btu/cubic foot |
| Site 2 winter | 1020 to 1030 Btu/cubic foot |
| Site 2 summer | 1070 to 1080 Btu/cubic foot |
| Site 3 | 1020 to 1040 Btu/cubic foot |
| Site 4 | 1034 Btu/cubic foot |

## 5.8 Additional Instrumentation

The status of the power to the DAS and its battery voltage were also monitored.

# 6

# Site Descriptions

The EDHP was tested at three residences and one commercial building. The residence in Columbus, OH was heating-dominated, and the commercial building in Columbus, GA was cooling-dominated. Both the residence in Charlotte, NC and the one in Roanoke, VA had heating and cooling loads that were more balanced.

All sites except the residence in Charlotte, NC used a 3-ton EDHP. The Charlotte site used a 3.5-ton unit.

## 6.1 Site 1, Columbus, GA

Figure 6.1 – Front View of Site 1

Site 1, shown in Figure 6.1, was 1,020 square feet of offices that were built within a larger warehouse. The warehouse was a steel-frame/sheet-metal-skin structure built on a concrete slab. It was insulated with 4" plastic-faced fiberglass batts in the walls. The ceiling was insulated with two layers of 6" fiberglass batts.

The EDHP's air handler was located in the crawl space above the office ceiling. The outdoor unit was located on the northwest side of the building outside of the garage area.

Southern Company Services, Inc. was the host utility for this site.

## 6.2 Site 2, Columbus, OH

Site 2 was a 2,300 square foot two-story wood-frame house with an attached garage and basement. The walls were standard 2"x4" stud construction insulated with fiberglass. The garage was built on a concrete slab with a playroom overhead. On the first floor there was a kitchen, living room, formal dining room, half-bathroom, laundry room, entertainment room, foyer and garage. On the second floor there was a playroom, two children's bedrooms, a master bedroom and bath, a spare bedroom and a full bathroom.

The EDHP's air handler was centrally located in the basement. The outdoor unit was directly adjacent to the rear porch. A photograph of the site appears in Figure 6.2.

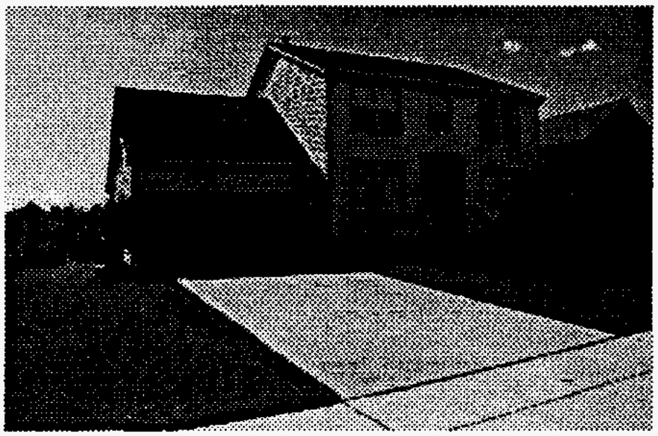

Figure 6.2 – Front View of Site 2

All supply air registers were located near the floor on outside walls. On the first floor, most of the return registers were located near the floor. This included one large return that was located in the floor almost directly above the air handler in the basement. The family room had a second return register that was located near the ceiling. The bedrooms on the second floor each had a high and a low return register. The homeowner adjusted the dampers in these regis: ·s so that the high returns were open in su: ıer and the low returns were open in wir :r.

American Electric Power was the host utility for this site.

### 6.3 Site 3, Roanoke, VA

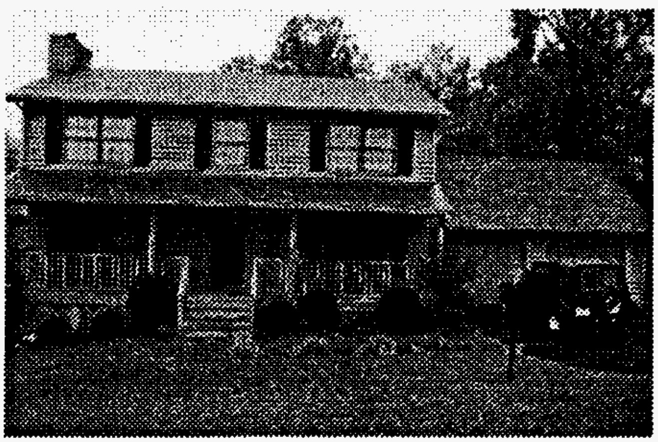

Figure 6.3 – Front View of Site 1

Site 3 was a 1,800 square foot two-story wood-frame house with an attached garage and basement. The walls were standard 2"x4" stud construction insulated with fiberglass. The garage was built on a concrete slab. On the first floor there was a kitchen, living room, formal dining room, half-bathroom and garage. On the second floor there was a master bedroom and bath, two guest bedrooms and a full bathroom.

The EDHP's air handler was centrally located in the basement. The outdoor unit was located on the southwest side of the house by the chimney. A photograph of the site appears in Figure 6.3.

American Electric Power was the host utility for this site.

## 6.4 Site 4, Charlotte, NC

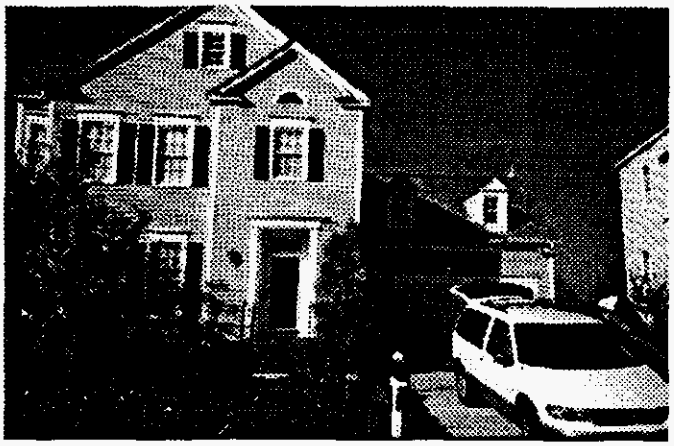

Figure 6.4 – Front View of Site 4

Site 4 was a 2,150 square foot two-story wood-frame house with an attached garage and crawl-space. The walls were standard 2"x4" stud construction insulated with fiberglass. The garage was built on a concrete slab. On the first floor there was a kitchen, living room with 24,000 Btu/h gas fireplace, formal dining room, half-bathroom, foyer and garage. On the second floor there was a playroom over the garage, a master bedroom and bath and two children's bedrooms.

The EDHP's air handler was located in a utility closet next to the second-floor playroom. The outdoor unit was located on the northwest side of the house near the rear entrance to the kitchen. A photograph of the site appears in Figure 6.4.

The Duke Power Company was the host utility for this site.

# 7

# Test Results

Monitoring of the four test sites began on the following dates:

| | |
|---|---|
| Site 1 | August 25, 1995 |
| Site 2 | October 24, 1995 |
| Site 3 | October 24, 1995 |
| Site 4 | November 23, 1995 |

Monitoring continued for approximately two years and ended on September 23, 1997 at all sites.

Numerous minor and major problems occurred at the test sites. These are described in the next section.

## 7.1 Operating Experience

The operational irregularities that were encountered at the four sites are documented in this section. A brief description is presented of each event, when it happened and the site where it occurred.

Site 1 - Starter Malfunction

On January 4, 1996, the engine starter at Site 1 malfunctioned. The starter was replaced on January 15.

Site 1 - Glycol Leak

At 10:30 AM on January 11, 1997 the EDHP at Site 1 failed to start after four attempts in about 30 minutes. This led to a fault condition that disabled the heat pump. A service contractor traced the problem to a glycol leak, which he repaired.

Site 2 - Engine Failure

On December 10, 1995, a warning light on the EDHP at Site 2 came on. A service call to the site diagnosed the problem as low oil (although there was no obvious sign of an oil leak or excessive burning of oil by the engine). The system was reset after adding four liters of oil. Within two hours the warning light again came on and the system shut down. A control board was replaced on January 17, but this failed to correct the problem. A new engine was installed on January 22, 1996.

Site 2 - System Shutdown

At 4:00 AM on February 4, 1996, the warning light on the EDHP came on and the system shut down. The outdoor temperature was -3°F. A service contractor reported that the system shut down due to low system pressure. The system was restarted by the contractor and operated with no problems.

Site 2 - Cycling of Glycol Heater on High Temperature

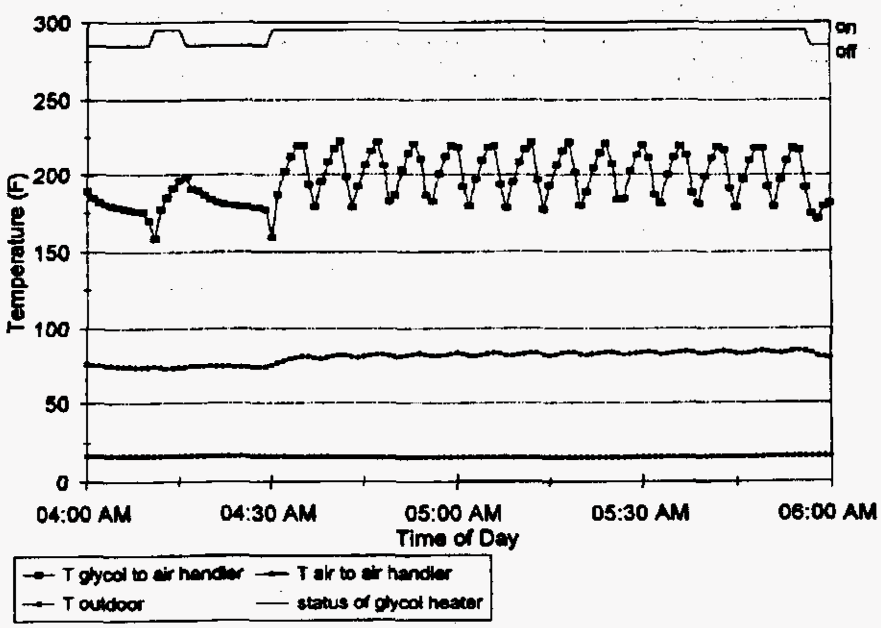


Figure 7.1 – February 13, 1996 Operation at Site 2

On several occasions, the auxiliary glycol heater was observed to cycle on its high-temperature safety cut-out. Figure 7.1 shows one period starting at 4:30 AM and ending at 6:00 AM on February 13, 1996 when heater cycled on and off 14 times. During this period, the EDHP continuously called for heat from the glycol heater as shown by the "high" status channel for the auxiliary heat. Both the engine speed and the glycol flow rate remained constant, the former at 3000 rpm and the latter at 5 gpm.

Site 2 - Control Problem

Starting at 6:00 AM on November 3, 1996, the EDHP at Site 2 cycled for two hours between low-speed engine operation and the glycol heater. This behavior is shown in Figure 7.2. At no time did the engine speed increase above its minimum level of 1200 rpm before the glycol heater turned on. (One would expect the engine to be operating at full speed--3000 rpm--before the glycol heater turns on.) Almost three weeks later, a service technician traced the problem to a shorted thermostat wire within the air handler. However, before this diagnosis was made the thermostat at the site was replaced twice and the EDHP's controller was replaced once.

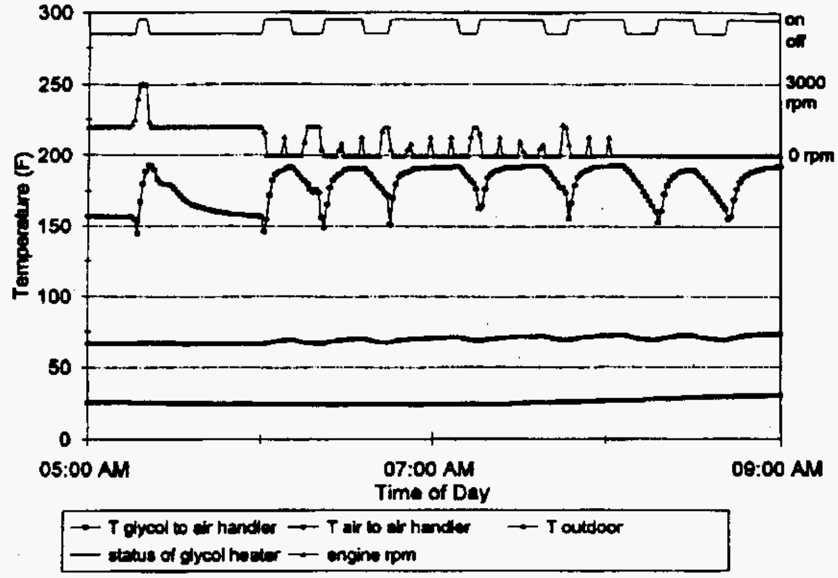


Figure 7.2 – November 3, 1996 Operation at Site 2

## Site 2 - Control Problem

On January 11, 1997 at about 11:15 AM, the EDHP at Site 2 turned off following an unusual defrost cycle in which the system ran in the cooling mode for two minutes without the glycol heater operating. Following this cycle, the system's controller disabled it and all heating for the next 11 days was provided by the glycol heater. On January 23, a service contractor restored normal operation by repairing a control wire that had shorted in the outdoor unit.

## Site 3 - System Shutdown

At 4:30 AM on February 3, 1996, the warning light on the EDHP came on and the system shut down. The outdoor temperature was 18°F, which is too high to shut the engine down on low ambient temperature. A service contractor restarted the heat pump the next day and the system operated with no problems.

## Site 3 - Engine Starting Problems

On April 11, 1996, the EDHP at Site 3 had trouble starting. As shown in Figure 7.3, the system tried to start 14 times from between 2:45 AM and 4:30 AM. For each start there is a large pulse of power to the outdoor unit, which would be the power drawn by the starter, but

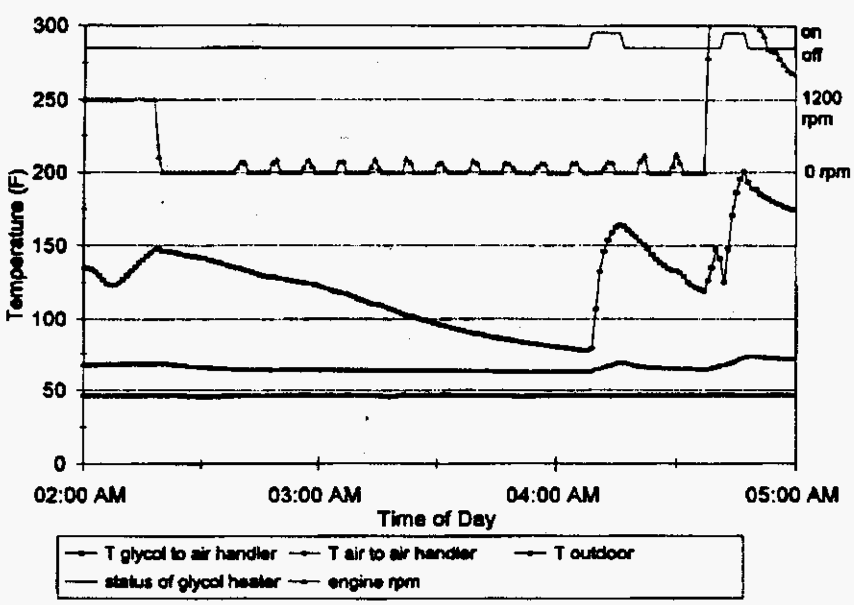


Figure 7.3 – April 11, 1996 Operation at Site 3

very low engine rpm. The occupant at the site reported that the system sounded like it was having trouble starting. No corrective action was taken and the system continued to keep the house at comfortable.

<u>Site 4 - Glycol Loop Problems</u>

During both heating seasons for the field test, the EDHP at Site 4 had chronic problems that showed up as short cycling of the heat pump and glycol heater. For most of the winter, the system would keep the house comfortable, and a homeowner who was not part of this field test might not have been aware of the anomalous operation. However, there were periods when the warning light on the EDHP would frequently come on, requiring either the homeowner or, in a number of instances, a service contractor to reset the system. During the two winters, the warning light came on 34 times and the thermostat was replaced twice.

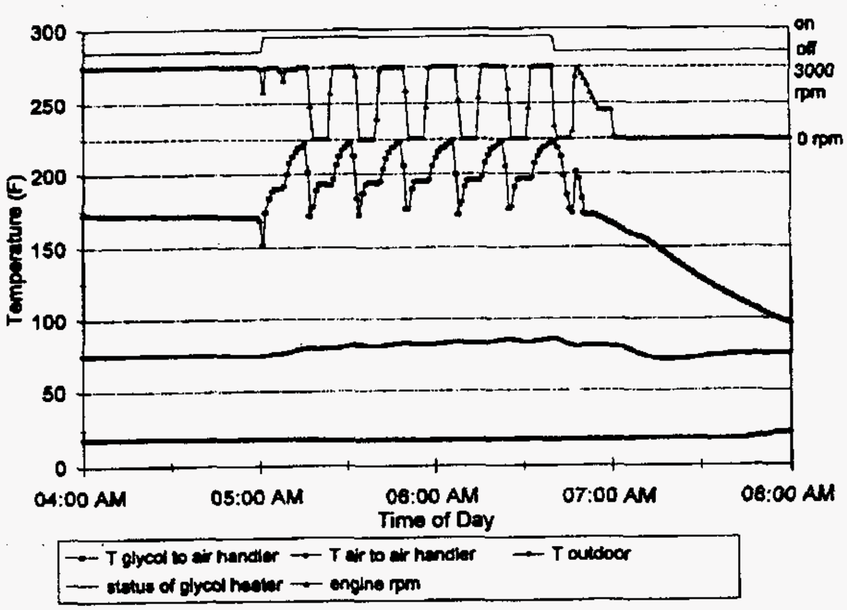


Figure 7.4 – February 6, 1996 Operation at Site 4

Figure 7.4 illustrates the anomalous operation that was observed at Site 4. At 5:00 AM on February 6, 1996, the engine was running at high speed when the system started a defrost cycle. The defrost cycle ended normally after about seven minutes and the glycol heater continued to operate to supply additional heat to the house. As the heater ran, the glycol supply temperature to the indoor air handler steadily increased. When the measured glycol temperature reached 222°F, approximately 15 minutes after the glycol heater started, the heater turned off and the engine continued to operate. After about another minute, the engine turned off and the glycol heater turned back on. The engine restarted after six minutes, but the entire cycle repeated when the glycol temperature reached about 222°F. This cycling repeated six times.

Several factors may have contributed to the chronic anomalous operation of the EDHP at Site 4. This site was the only one to use the 3.5-ton version of the EDHP. It also had the greatest height differential between the outdoor and indoor unit--approximately 15 feet. (The EDHP's manufacturer requires that the air handler be no more than 18 feet above the outdoor unit. Although the height of the air handler may point to insufficient glycol flow as the problem, the measured glycol flow is normal--e.g., about 5 gpm--during the periods of anomalous cycling.)

Although the glycol flow was normal, tests of the glycol heater--which are presented in Section 7.3--showed that Site 3 had the highest heat transfer rates to the glycol and the lowest air flow rates. The combination of these two effects produced air-side temperature rises that approached 50°F. These temperature rises were between 11% and 25% higher than those measured at the other sites.

Site 4 - Engine Starting Problems

At 7:00 AM on January 7, 1997, the EDHP tried to start four times within a half hour, but failed all four times. Following these failed starts, all heat to the house was provided by the glycol heater. At 7:00 PM the homeowner reset the system and it resumed normal operation.

**7.2 The Performance of the Data Acquisition System and Instrumentation**

The data loggers all operated without problems for the entire field test. The precautions that were taken to secure the databases--in particular, the seven days of on-site data storage and the frequent screening of data--were very effective at minimizing the loss of data. For the approximately two years during which data was collected, data loss averaged about 3 days per site. However, most of this data loss occurred early in the test period when the data loggers were being reprogrammed. For the 20 month period starting February 1, 1996, only 27 one-minute data records were lost out of a total of 3.5 million.

Almost all of the 130 sensors that were installed operated reliably throughout the field test. Two exceptions were the gas meter and the sensor measuring the inlet air temperature to the air handler at Site 2. The temperature sensor steadily drifted upward by about 3°F during the first 15 months of the field test. However, since this site almost always operated with the indoor fan running continuously, it was possible to continually recalibrate the faulty sensor. (Once a new sensor was installed in January 1997, the temperature rise across the air handler when just the fan was running was measured to be 0.2°F. Using this temperature rise, the readings from the faulty sensor were then adjusted in the "corrected" database to produce this temperature rise when only the fan was running.)

The gas meter at Site 2 failed on January 11, 1996. The meter could not be replaced until February 26 due to a delay in obtaining a calibrated replacement. This 46 day loss in gas readings produced the only significant gap in the performance data for the four test sites.

The dew-point hygrometers required rebalancing at several of the sites at the start of the second cooling season. At Sites 1 and 3, one of the two dew-point hygrometers continued to read high after the rebalancing, and so there is no data from these sensors for the second cooling season. (This loss of information did not compromise the results for this project since the tip bucket was used to calculate latent cooling.)

The only other sensor to have a significant problem was the one that measured engine rpm at

Site 3. On October 29, 1996 the signal conditioning circuitry for this sensor was replaced. Prior to this date, the sensor would occasionally produce unrealistically low readings when the engine operated at low speed (i.e., values less than 1200 rpm).

### 7.3 Calibration of the Air Flow Measurement

As described in Section 3.0, the air flow measurement was calibrated by performing an energy balance between the air flow and the glycol loop. In this procedure, the heat transfer from the glycol loop to the air stream was first calculated using the temperature of the glycol at the inlet and outlet of the air handler and its flow rate. The air flow rate was then calculated using the temperature change of the air as it flows across the glycol coil. Finally, an effective flow area was calculated at the location within the air handler where the anemometer was located.

It was important that the glycol loop's operation was steady when the calibration was done. This required about 20 minutes of continuous operation for the glycol heater. To insure that this condition was met, the building occupants switched the heat pump to emergency heat and then turned up the thermostat to a high setting for at least 30 minutes.

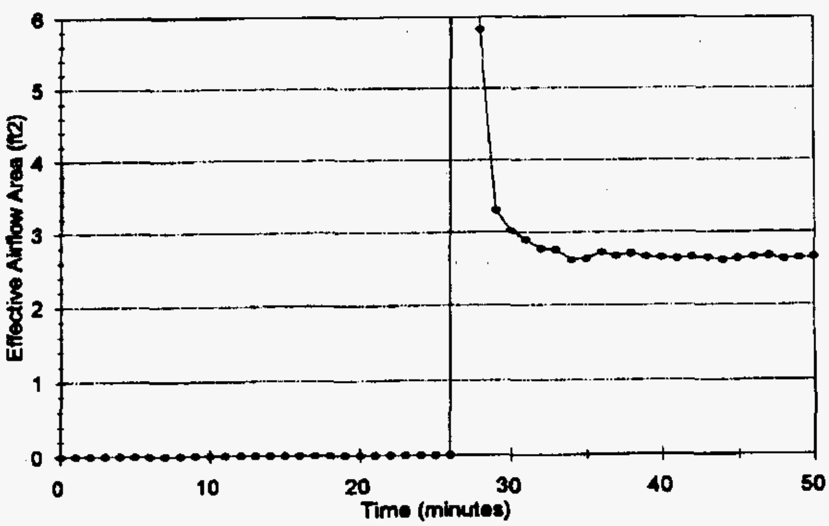


Figure 7.5 – Air Flow Calibration Test

The results of a typical calibration test are shown in Figure 7.5. For each minute of the test, an effective cross-sectional area for air flow at the anemometer was calculated using the preceding energy balance. As shown in this figure, the calculated area reached a steady value after about 10 to 15 minutes of operation.

As part of the air flow calibration, the performance of the glycol heater at each site was calculated and compared to its catalog values. Based on the manufacturer's literature, the glycol heater should have a firing rate, heating rate and efficiency of 75,000 Btu/h, 64,000 Btu/h and 85.3%.

Air-flow calibration tests were performed periodically at the four sites. (For Site 4, where operational problems often "locked out" the heat pump and turned on the glycol heater, there were additional opportunities to check air flow calibration.)

The results of all air-flow calibration tests are shown in Table 7.1. In general, there is good agreement between the measured performance of the glycol heater and its catalog values. The average measured heater efficiencies (i.e., percent of gas energy delivered to air stream) for the four sites were:

| | measured | deviation |
|---|---|---|
| Site #1 | 75.3% | -11.7% |
| Site #2 | 88.0% | +3.2% |
| Site #3 | 86.4% | +1.3% |
| Site #4 | 87.4% | +2.5% |

(The value for heater efficiency from the March 5, 1997 test at Site 4 deviated significantly from the values for the other 13 tests. This data was not included in the average for the site.)

The only site where the measured heater efficiency consistently deviated from the catalog value was Site #1. At this site, the outdoor unit was located very far from the air handler--about 55 feet. This probably caused unusually large line losses that reduced the measured heater efficiency.

### 7.4 Catalog Versus Field Performance

A major objective of this field test was to provide data that could be used to compare the operating costs for the EDHP with alternative heating and cooling technologies. The first step towards meeting this objective is to verify that the field installations were performing close to their catalog specifications. (In this analysis, the operational problems that occurred at the sites were ignored, and the EDHP was studied only during periods when it was running well.)

The steady-state capacity and efficiency of the 3-ton EDHP that was tested as a function of outdoor temperature has been published by the manufacturer. This performance data is based on tests performed in an environmental chamber that maintained indoor conditions at either the ARI summer (80°F, 50% rh) or winter (70°F) test conditions. (Unfortunately, performance data for the 3.5-ton EDHP was not available, and so it was not possible to validate the field performance at Site 4.)

Three complications that arise when comparing field and catalog performance are (1) catalog data is presented for steady-state operation; however, the EDHP is a variable speed heat pump that continually changes its speed during normal operation, (2) although the temperature and air flow sensors respond sufficiently fast to provide accurate minute-by-minute values for sensible performance, it takes several minutes for condensate to travel from the coil to the tip bucket; measured latent performance is therefore always delayed by several minutes, and (3) catalog data applies only to a specified set of operating conditions (i.e., the temperature and humidity of the air at the inlet to the air handler and its flow rate); these conditions cannot be controlled in the field test.

TABLE 7.1

TESTS OF AUXILIARY BOILER

| | | Glycol Coil Temperatures (F) | | | | Glyc Flow | Fan DP | Fuel P | Power (watts) | | Fuel | Fan | Air Flow | Ht Trnsfr (kBtu/h) | | Boiler | Flow A |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Site | Air In | Air Out | Glyc. In | Glyc. Out | gpm | in. WC | psi | air hndlr | out unit | cfm | rpm | scfm | glycol | burner | Eff.(*) | (ft2) |
| 11/22/95 | #1 | 72.0 | 108.9 | 157.1 | 128.7 | 5.04 | 1.02 | 14.9 | 565.4 | 259.7 | 1.40 | 1240 | 1569 | 64.2 | 87.3 | 0.729 | 3.07 |
| 1/8/96 | #1 | 76.4 | 114.4 | 165.0 | 137.0 | 5.18 | 1.01 | 14.9 | 543.1 | 247.9 | 1.38 | 1229 | 1514 | 65.0 | 85.8 | 0.750 | 3.07 |
| 3/27/97 | #1 | 76.9 | 113.1 | 162.7 | 135.1 | 5.17 | 1.12 | 14.9 | 607.1 | 254.1 | 1.30 | 1282 | 1510 | 64.0 | 81.1 | 0.780 | 3.31 |
| Average | #1 | 75.1 | 112.1 | 161.6 | 133.6 | 5.13 | 1.05 | 14.9 | 571.8 | 253.9 | 1.36 | 1250 | 1531 | 64.4 | 84.7 | 0.753 | |
| 1/12/96 | #2 | 75.9 | 114.3 | 192.2 | 163.7 | 4.91 | 1.00 | 14.4 | 477.5 | 205.4 | 1.16 | 1276 | 1564 | 62.7 | 70.0 | 0.887 | 3.15 |
| 4/2/96 | #2 | 80.9 | 117.9 | 193.9 | 165.3 | 4.85 | 0.89 | 14.6 | 442.8 | 212.8 | 1.14 | 1224 | 1578 | 62.2 | 69.7 | 0.883 | 3.14 |
| 3/1/97 | #2 | 82.0 | 119.5 | 193.0 | 164.7 | 4.93 | 1.35 | 14.5 | 570.7 | 224.6 | 1.17 | 1413 | 1505 | 62.6 | 71.3 | 0.868 | 3.37 |
| Average | #2 | 79.6 | 117.2 | 193.0 | 164.6 | 4.90 | 1.08 | 14.5 | 497.0 | 214.3 | 1.16 | 1304 | 1549 | 62.5 | 70.3 | 0.880 | |
| 1/8/96 | #3 | 72.6 | 117.3 | 171.6 | 141.5 | 5.08 | 1.01 | 14.0 | 567.1 | 130.8 | 1.32 | 1277 | 1461 | 68.6 | 77.0 | 0.886 | 1.49 |
| 4/2/96 | #3 | 76.9 | 120.2 | 175.9 | 146.5 | 4.99 | 0.94 | 14.2 | 539.3 | 120.5 | 1.29 | 1246 | 1456 | 66.0 | 76.4 | 0.859 | 1.31 |
| 10/29/96 | #3 | 80.0 | 123.0 | 177.6 | 149.0 | 4.96 | 1.00 | 14.2 | 567.5 | 230.8 | 1.30 | 1285 | 1471 | 63.7 | 77.2 | 0.818 | 1.27 |
| 3/3/97 | #3 | 76.8 | 120.9 | 179.1 | 148.6 | 4.90 | 1.05 | 14.1 | 568.3 | 120.3 | 1.26 | 1303 | 1454 | 67.0 | 74.6 | 0.893 | 1.27 |
| Average | #3 | 76.6 | 120.3 | 176.0 | 146.4 | 4.98 | 1.00 | 14.1 | 560.5 | 150.6 | 1.29 | 1278 | 1460 | 66.3 | 76.3 | 0.864 | |
| 12/25/95 | #4 | 78.0 | 126.4 | 186.2 | 155.8 | 5.12 | 0.95 | 14.6 | 462.5 | 159.4 | 1.34 | 1200 | 1350 | 69.6 | 82.6 | 0.838 | 2.59 |
| 1/2/96 | #4 | 75.6 | 124.1 | 184.3 | 153.8 | 5.17 | 0.96 | 14.6 | 480.8 | 154.7 | 1.32 | 1221 | 1407 | 70.6 | 81.1 | 0.866 | 2.63 |
| 1/16/96 | #4 | 82.4 | 132.0 | 191.0 | 160.7 | 5.15 | 0.97 | 14.9 | 470.0 | 156.1 | 1.29 | 1209 | 1389 | 70.1 | 81.2 | 0.858 | 2.66 |
| 1/23/96 | #4 | 81.9 | 131.2 | 190.6 | 160.3 | 5.14 | 0.97 | 14.8 | 468.7 | 158.6 | 1.31 | 1212 | 1356 | 64.7 | 75.6 | 0.849 | 2.51 |
| 1/23/96 | #4 | 77.6 | 126.7 | 187.3 | 156.2 | 5.16 | 0.98 | 14.8 | 477.0 | 152.5 | 1.30 | 1218 | 1362 | 71.9 | 80.7 | 0.885 | 2.76 |
| 1/25/96 | #4 | 79.5 | 128.9 | 188.5 | 158.3 | 5.14 | 0.97 | 14.8 | 468.5 | 160.3 | 1.31 | 1210 | 1402 | 69.7 | 82.0 | 0.844 | 2.68 |
| 2/2/96 | #4 | 82.8 | 131.1 | 190.0 | 160.9 | 5.15 | 0.99 | 14.7 | 485.5 | 163.1 | 1.17 | 1232 | 1336 | 67.3 | 72.5 | 0.921 | 2.24 |
| 2/3/96 | #4 | 81.1 | 130.0 | 189.6 | 160.2 | 5.15 | 0.99 | 14.7 | 481.3 | 164.2 | 1.17 | 1226 | 1321 | 68.0 | 72.4 | 0.931 | 2.28 |
| 3/19/96 | #4 | 77.9 | 126.6 | 186.0 | 155.3 | 5.09 | 0.95 | 14.4 | 485.9 | 161.7 | 1.30 | 1225 | 1324 | 70.1 | 79.1 | 0.880 | 2.22 |
| 3/19/96 | #4 | 80.2 | 129.6 | 188.8 | 158.8 | 5.10 | 0.95 | 14.4 | 483.8 | 164.1 | 1.32 | 1223 | 1395 | 68.6 | 80.0 | 0.851 | 2.14 |
| 4/2/96 | #4 | 77.5 | 128.5 | 188.4 | 157.0 | 5.11 | 0.94 | 14.7 | 467.4 | 163.8 | 1.29 | 1199 | 1360 | 72.0 | 80.1 | 0.892 | 2.30 |
| 4/7/96 | #4 | 77.5 | 126.3 | 186.3 | 155.5 | 5.08 | 0.94 | 14.6 | 469.2 | 168.2 | 1.30 | 1201 | 1421 | 70.1 | 80.6 | 0.864 | 2.22 |
| 10/30/96 | #4 | 87.4 | 133.9 | 192.4 | 162.4 | 5.07 | 1.03 | 14.6 | 514.2 | 165.4 | 1.25 | 1268 | 1365 | 68.1 | 77.0 | 0.878 | 2.43 |
| 3/5/97 | #4 | 78.3 | 125.9 | 191.2 | 158.2 | 4.96 | 0.83 | 14.7 | 435.0 | 169.2 | 1.21 | 1149 | 1414 | 73.5 | 75.1 | 0.971 | 2.37 |
| Average** | #4 | 80.0 | 128.7 | 188.4 | 158.1 | 5.13 | 0.97 | 14.6 | 478.1 | 160.9 | 1.28 | 1219 | 1368 | 69.3 | 78.8 | 0.874 | |

* Assumes 80% of pump work appears as heat in glycol loop

**data for 3/5/97 omitted from average

**Nominal Operation**

Firing Rate: 75,000 Btu/h

Heating Rate: 64,000 Btu/h

Efficiency: 0.853

The first two complications can be overcome by using field data for periods when the EDHP operated at a single speed for at least 20 minutes and discarding the data for the first ten minutes. The only limitation that this procedure imposes is that the EDHP's performance can only be validated at its highest and lowest speeds (i.e., 3000 rpm and 1200 rpm), since these are the only two speeds that will have extended operation at a constant speed.

The problems introduced by the differences between the inlet air temperature, humidity and flow rate in the field test and the tests that produced the catalog data is addressed by "correcting" the field test data back to catalog test conditions. Unfortunately, the manufacturer's catalog does not give correction factors that can be used to adjust performance for differing inlet air conditions. As an approximation, the following correction factors--which are "borrowed" from another manufacturer's catalog data for a 3-ton electric heat pump--were used for high speed operation:

<u>inlet wet-bulb temperature</u>
total cooling: +2.0% per degree F (wet-bulb)
cooling COP: +1.5% per degree F (wet-bulb)

<u>inlet dry-bulb temperature</u>
total heating: -0.3% per degree F (dry-bulb)
heating COP: -0.8% per degree F (dry-bulb)

<u>air flow rate</u>
total cooling: +1.3% per 100 cfm
cooling COP: +0.7% per 100 cfm
total heating: +0.7% per 100 cfm
heating COP: +1.3% per 100 cfm

(If, for example, test data for the EDHP during heating was collected at 72°F inlet air temperature--two degrees higher than the manufacturer's catalog performance data--the measured heating rate would be decreased by 0.6% (=2*0.3%) and the measured COP would be decreased by 1.6% (=2*0.8%) before they were compared to the catalog data.)

<u>Site 1</u>

The performance of the EDHP at Site 1 during the summer of 1997 is compared with catalog performance in Figures 7.6 and 7.7. For high speed operation, the linear regression curve-fits through the field test data fall very close to the catalog performance curves. However, the average inlet air wet-bulb temperature for the field test data is significantly lower than the catalog test conditions (63°F versus 67°F) and the air flow is significantly higher (1563 cfm versus 1200 cfm). Using the preceding correction factors, the field test data for both the total cooling rate and COP should be between 3% and 4% lower than the catalog curves. This discrepancy is fairly small and can be explained by a combination of measurement error and

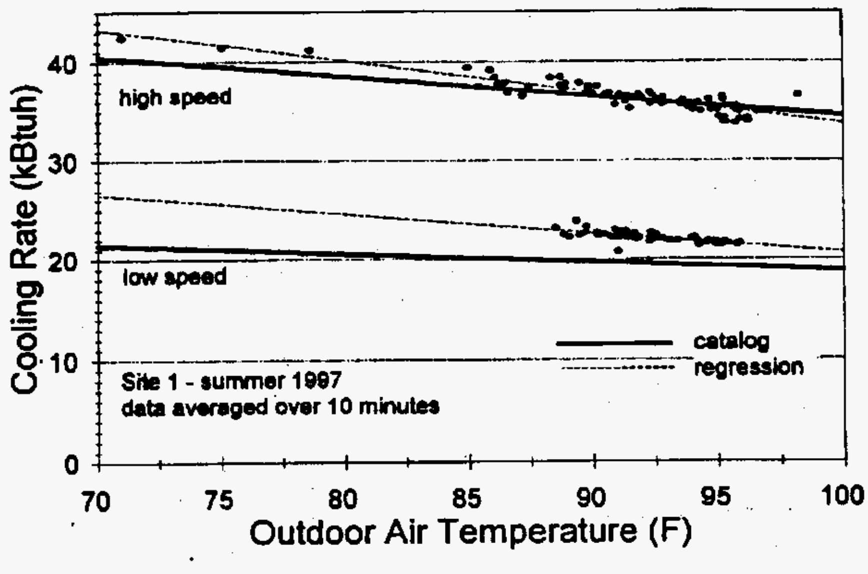


Figure 7.6 – Summer Cooling Capacity at Site 1

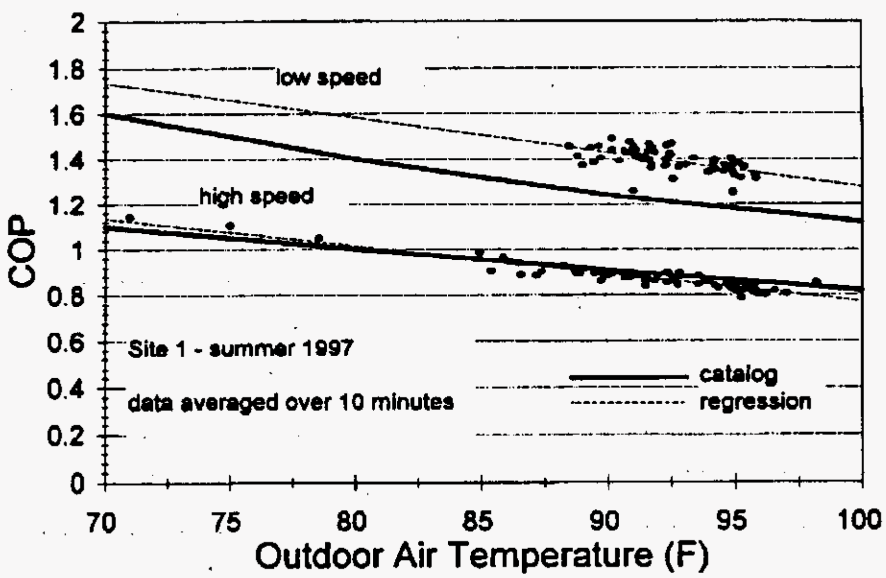


Figure 7.7 – Summer COP at Site 1

normal variations in the heat pump's performance.

Both the cooling rate and COP for the EDHP at Site 1 during low-speed operation are moderately higher than the catalog curves: about 15% deviation for both cooling rate and COP. The average wet-bulb temperature for the inlet air during low-speed operation was only 0.5°F higher than the catalog test conditions. The average inlet air flow was 741 cfm. Since EDHP's manufacturer does not report the air flow at which the low-speed catalog performance was measured, no attempt has been made to correct the field test data for differences in this operating parameter.

It is unlikely that the possible difference in air flow will account for the entire 15% deviation between field test and catalog data at low-speed. Other factors that could be causing the discrepancy are (1) the EDHP at this site is performing better than its catalog performance at low speed, and (2) measurement error--most likely the assumption that the effective flow area at the anemometer is the same for high and low speed operation. (The air flow measurement is calibrated during the operation of the glycol heater. However, the blower in the air handler operates only at high speed when the glycol heater is on. The air flow calibration is extended to lower speeds by assuming that the velocity distribution at the

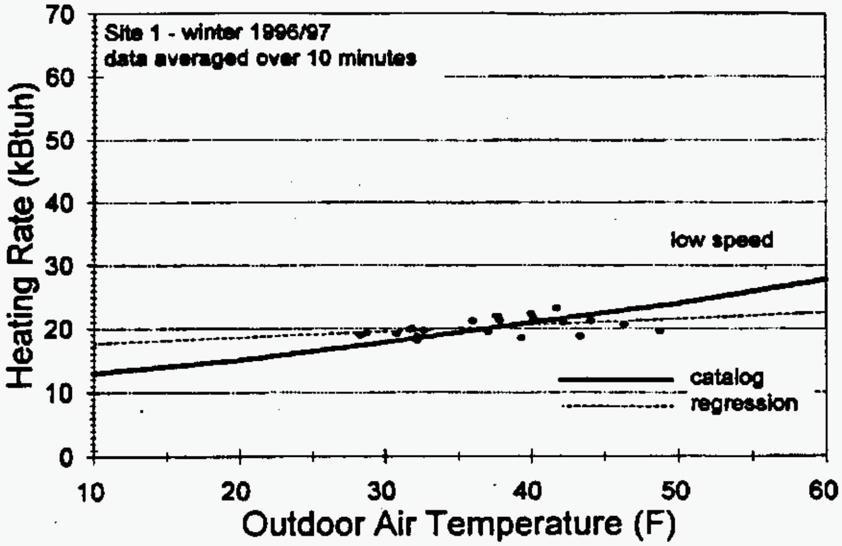


Figure 7.8 – Winter Heating Rate at Site 1

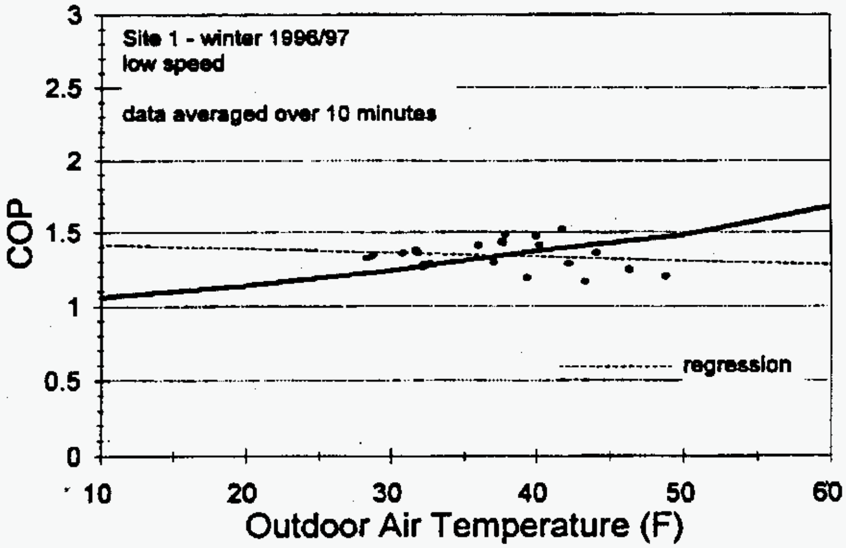


Figure 7.9 – Winter Low-Speed COP at Site 1

location of the anemometer is invariant.)

Because of the relatively mild winters at Site 1, it was not possible to compare the field test performance with catalog data during high-speed heating operation of the EDHP. Low-speed performance could be compared, and it is shown in Figures 7.8 and 7.9 for the winter of 1996/1997. The average inlet air temperature and flow for the low-speed heating operation was 70°F and 734 cfm.

Site 2

The performance of the EDHP at Site 2 during the summer of 1997 is compared with catalog performance in Figures 7.10 and 7.11. For high speed operation, the linear regression curve-fits through the field test data again fall very close to the catalog performance curves: the measured cooling rate is about 2% higher than the catalog curve and the COP is about 8% lower. However, the average inlet air wet-bulb temperature for the field test data is significantly lower than the catalog test conditions (63°F versus 67°F) and the air flow is significantly higher (1655 cfm versus 1200 cfm). Using the preceding correction factors, the field test data for both the total cooling rate and COP should be about 3% lower than the catalog curves. Applying these corrections, the linear regression curve-fits for both COP and cooling rate at high speed are within 5% of the catalog data.

As with Site 1, there is a slightly larger discrepancy between the field test and catalog data at low-speed operation of the EDHP during cooling. As shown in Figures 7.10 and 7.11, the regression curve-fit through the low-speed data for cooling rate is about 14% higher than the catalog curve, and for the COP, it is 9% higher. The average wet-bulb temperature for this data was 61°F and the average air flow was 950 cfm.

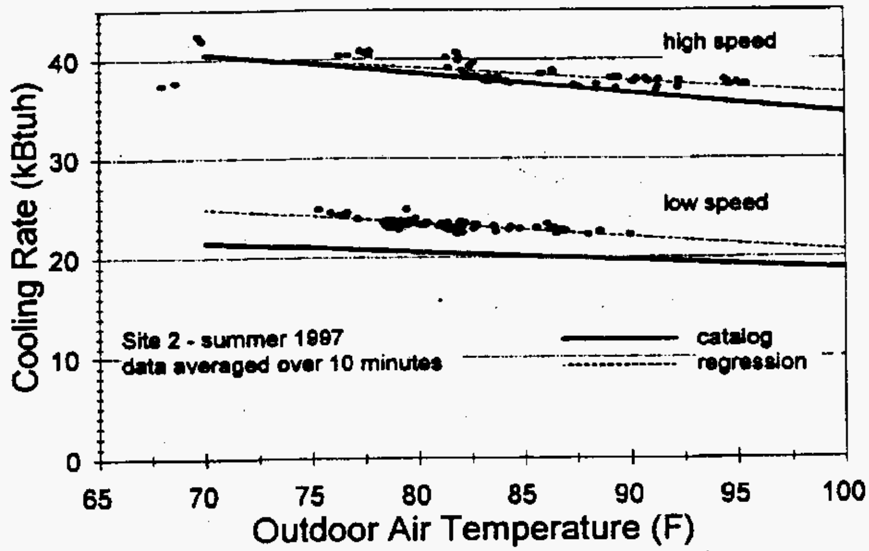


Figure 7.10 – Summer Cooling Rate at Site 2

During heating, the measured performance of the EDHP during high-speed operation did not agree as well with catalog data as did the data for cooling. As shown in Figures 7.12 and 7.13 the regression curve-fits through the high-speed heating data were about 16% higher for the heating rate and 11% higher for the COP. Approximately 2% of these deviations could be explained by a combination of a higher inlet air temperature (74°F for the field test data versus 70°F for the catalog) and higher air flow (1514 cfm versus 1200).

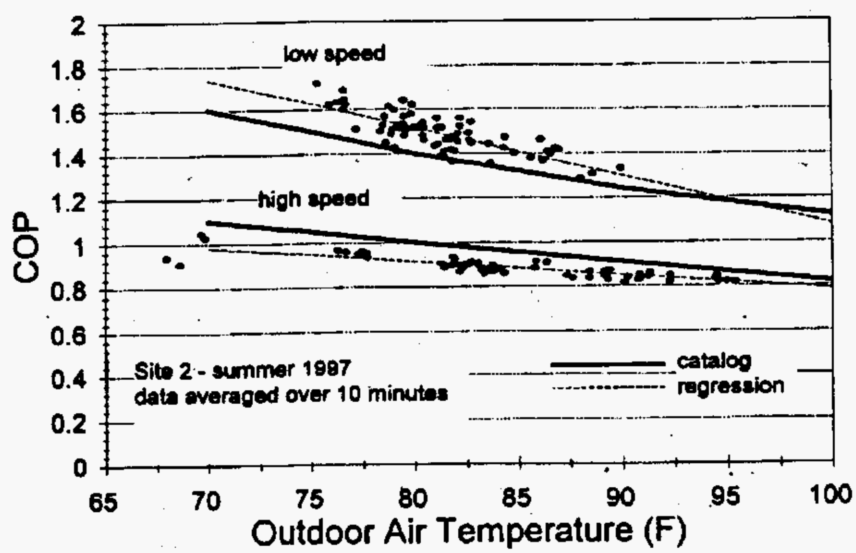


Figure 7.11 – Summer COP at Site 2

The measured performance during low-speed heating also tended to be higher than the catalog curves. As shown in Figures 7.12 and 7.14, the linear regression curve-fit for the heating rate was about 17% higher than the catalog curve, and for the COP it was 23% higher. The average inlet air temperature and air flow for these data were 70°F and 862 cfm.

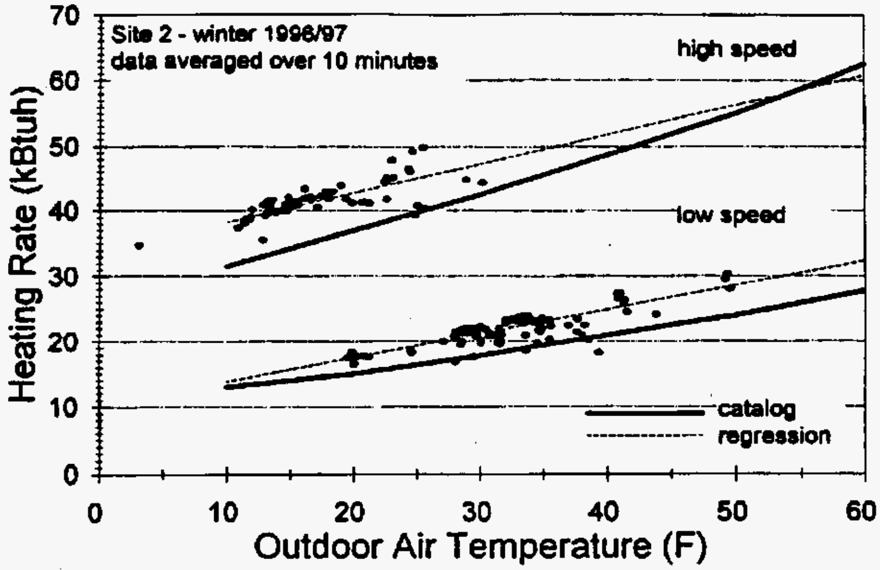


Figure 7.12 – Winter Heating Rate at Site 2

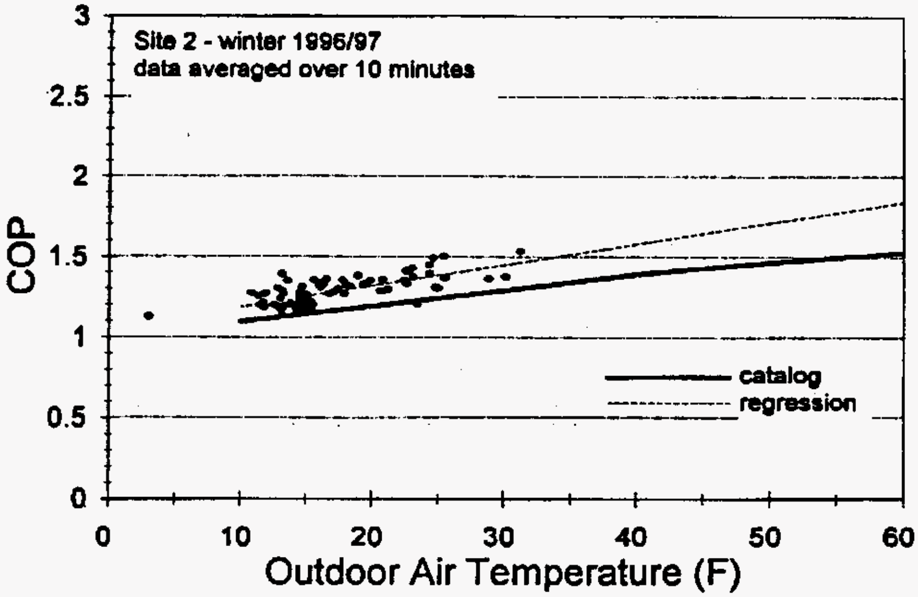


Figure 7.13 – Winter High-Speed COP at Site 2

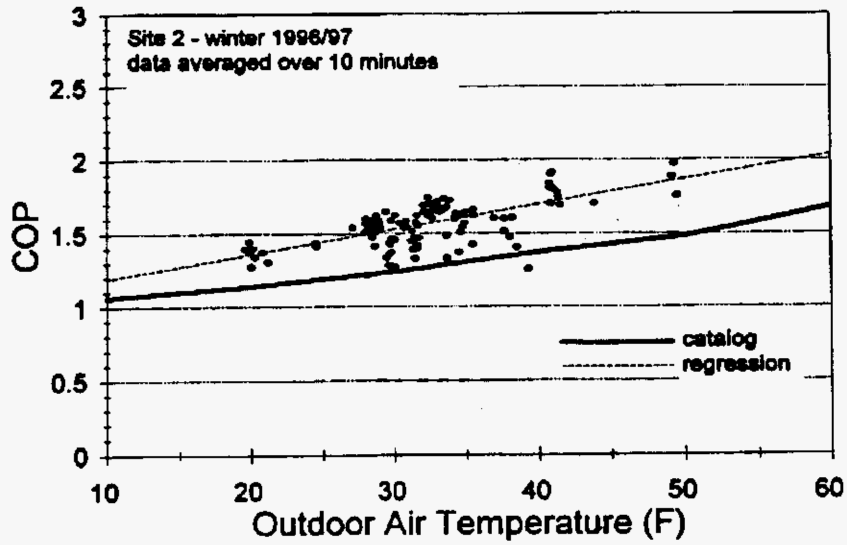


Figure 7.14 – Winter Low-Speed COP at Site 2

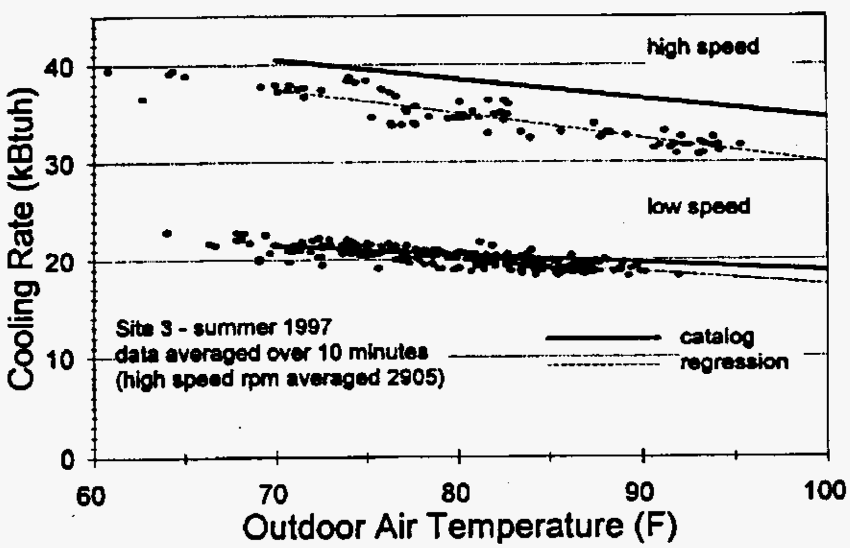


Figure 7.15 – Summer Cooling Rate at Site 3

Site 3

The performance of the EDHP at Site 3 during the summer of 1997 is compared with catalog performance in Figures 7.15 and 7.16. For high speed operation, the linear regression curve-fits through the field test data fall moderately below the catalog performance curves: the measured cooling rate is about 9% lower than the catalog curve and the COP is about 4% lower.

The process for correcting the field test data for air inlet conditions that deviate from the catalog test conditions was complicated by the fact that the dew-point hygrometer at the inlet to the air handler malfunctioned during the summer of 1997 at Site 3. Assuming that the inlet air has relative humidity of 60% (a value frequently seen prior to the failure of the dew-point hygrometer), the inlet wet-bulb temperature averaged 61°F during high-speed operation. The air flow averaged 1473 cfm. Using the correction factors listed above, these inlet air condi-

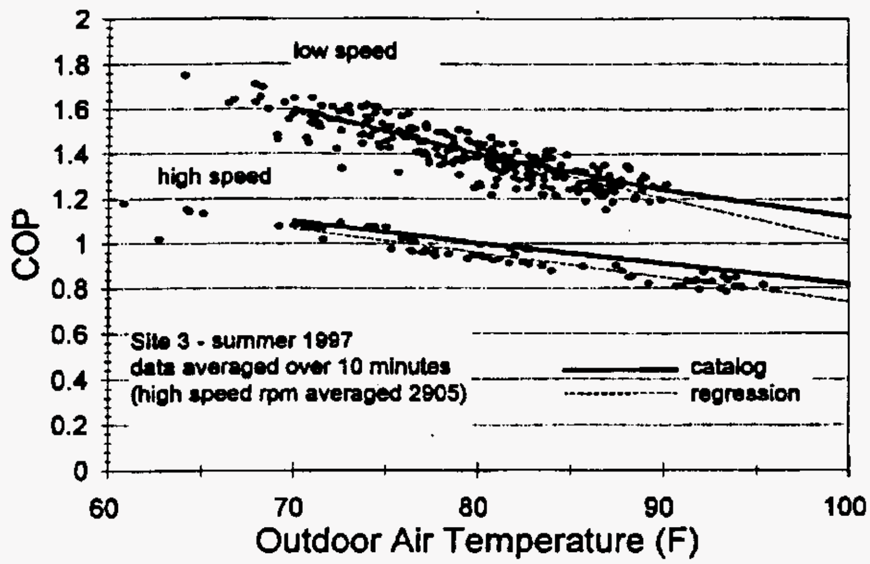


Figure 7.16 – Summer COP at Site 3

tions would produce a cooling rate that was 8.5% below the catalog curve and a COP that was 7% below. These values compare reasonably well with the regression curve-fits through the measured data.

At Site 3 there was good agreement between the meas: red cooling perfo: nance at low speed and c: alog data. As show: in Figures 7.15 and 7.16, the regression curve-fits for both the cooling rate and the COP at low speed were within 2% of the catalog curves. Assuming an inlet air relative humidity of 60%, the inlet air wet-bulb temperature averaged 61°F for the low speed data. The air flow was 781 cfm.

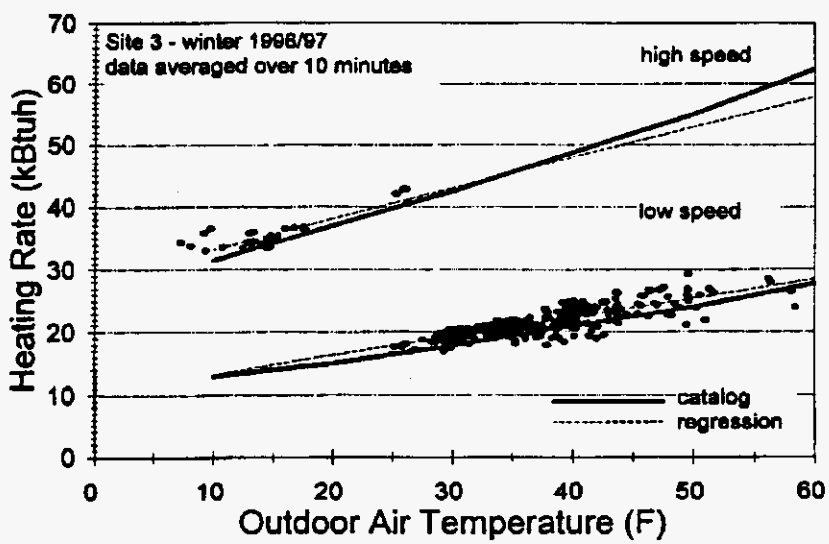


Figure 7.17 – Winter Heating Rate at Site 3

The performance of the EDHP at Site 3 agreed well with catalog data for both low-speed and high-speed heating. As shown in Figures 7.17 and 7.18, the linear regression curve-fits through the data for heating rate and COP at high speed are high by 4% and 7% respectively. The average inlet air temperature for the measured data was 70°F and the air flow was 1389 cfm. The higher air flow would account for a heating rate that was high by 1.3% and a COP that was high by 2.5%.

The measured performance during low-speed heating also tended to be higher than the catalog curves. As shown in Figures 7.17 and 7.19, the linear regression curve-fit for the heating rate was about 7% higher than the catalog curve, and for the COP it was 12% higher. The average inlet air temperature and air flow for these data were 70°F and 740 cfm.

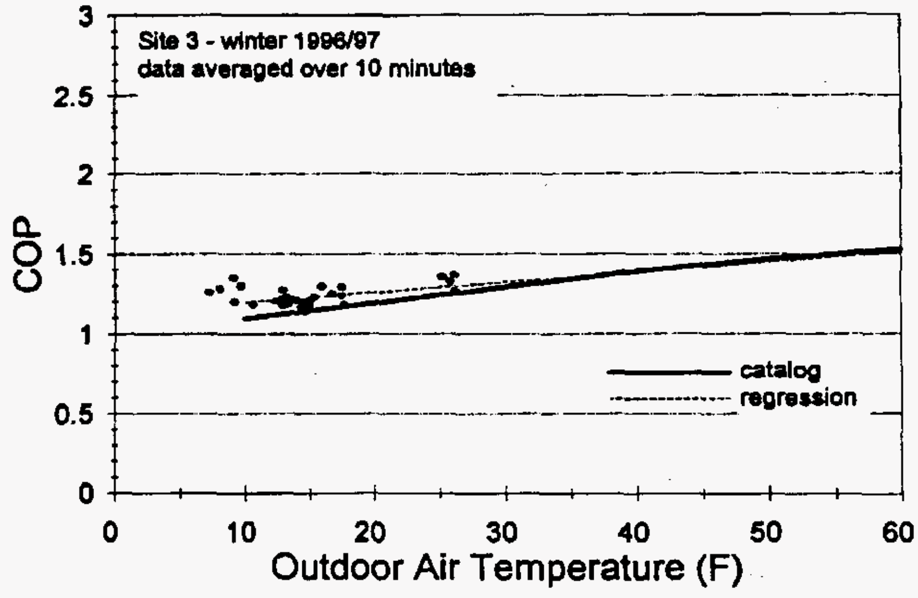


Figure 7.18 – Winter High-Speed COP at Site 3

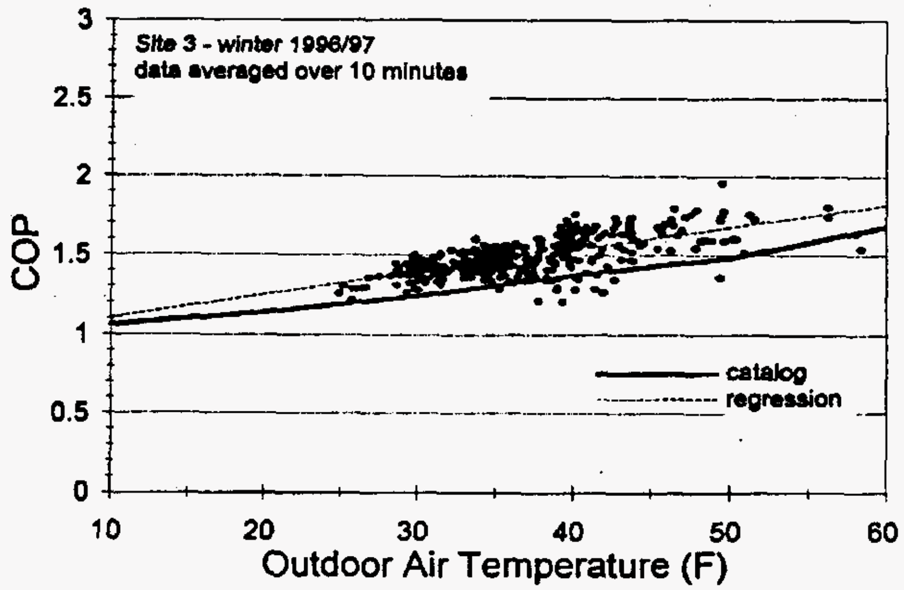


Figure 7.19 – Winter Low-Speed COP at Site 3

## Site 4

Site 4 was unique in that its EDHP was rated at 3.5 tons (as opposed to 3 tons at the other sites). Since performance catalog data for the 3.5-ton model was not available, the field-test data for Site 4 are presented without comparing them to catalog data.

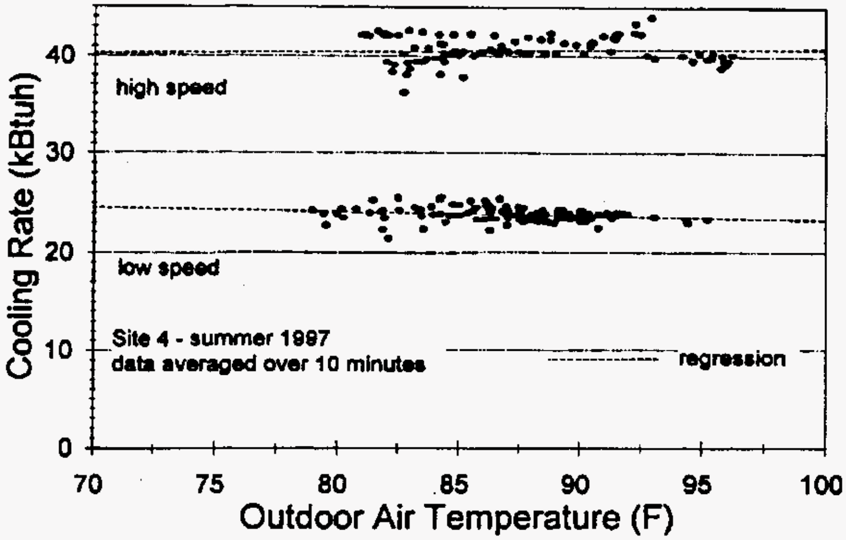


Figure 7.20 – Summer Cooling Rate at Site 4

Figures 7.20 and 7.21 show the cooling rate and COP of the EDHP at Site 4 during high-speed and low-speed operation. For high-speed operation, the average wet-bulb temperature of the inlet air was 66°F and the average air flow was 1545 cfm. For low-speed operation, the average wet-bulb temperature of the inlet air was 67°F and the average air flow was 907 cfm.

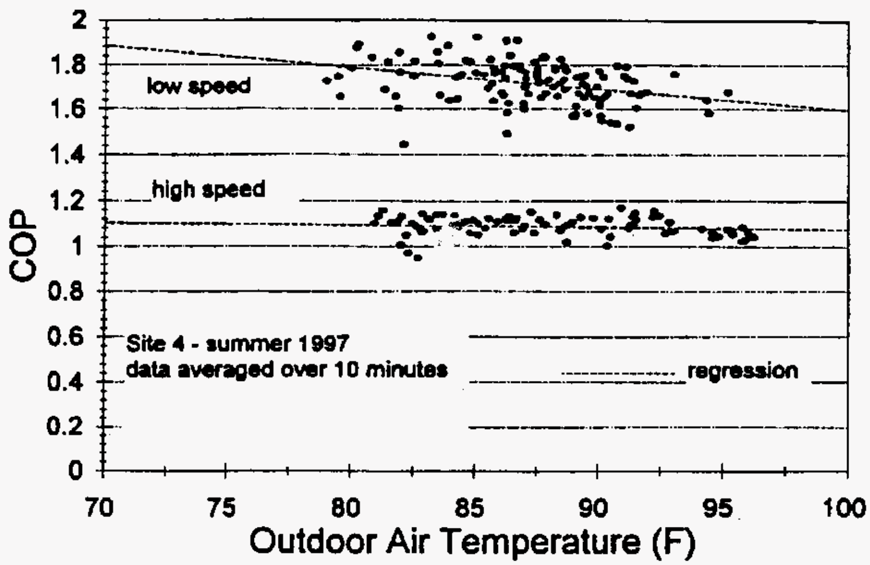


Figure 7.21 – Summer COP at Site 4

Figures 7.22, 7.23 and 7.24 show the heating rate and COP of the EDHP at Site 4 during high-speed and low-speed operation. For high-speed operation, the average inlet air temperature was 77°F and the average air flow was 1440 cfm. For low-speed operation, the average inlet air temperature was 70°F and the average air flow was 802 cfm.

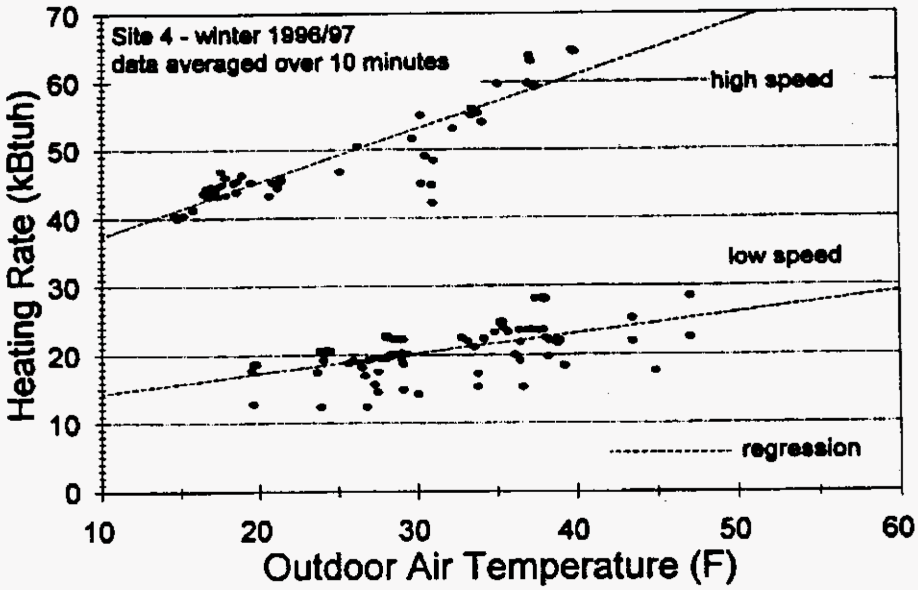


Figure 7.22 – Winter Heating Rate at Site 4

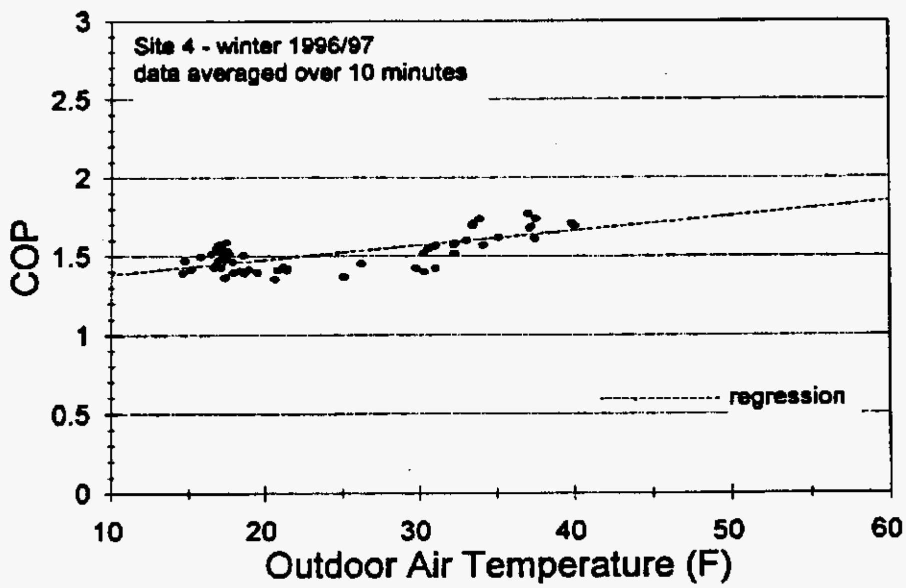


Figure 7.23 – Winter High-Speed COP at Site 4

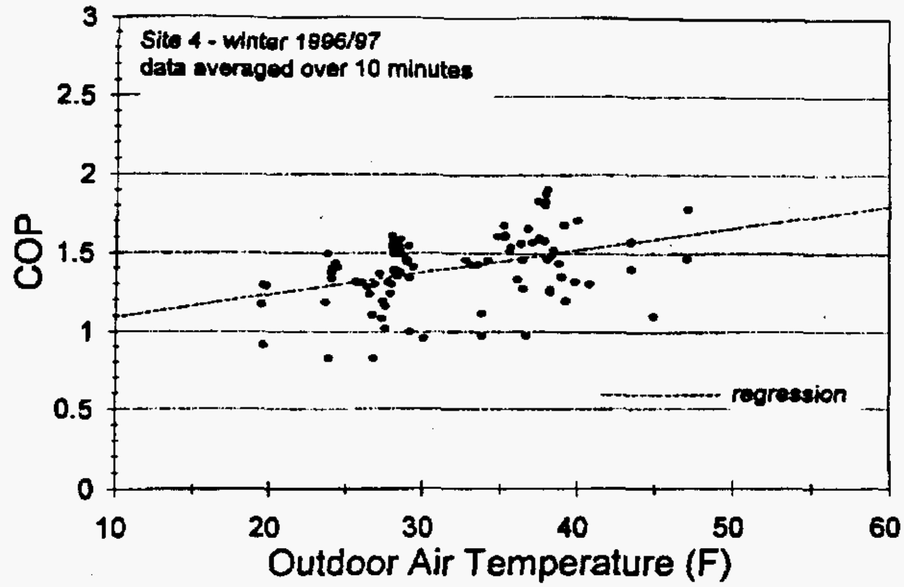


Figure 7.24 – Winter Low-Speed COP at Site 4

## 7.5 Latent Performance

The steady-state data that were used in the preceding section to compare the field performance of the EDHPs with their catalog values were also used to study their latent performance. Table 7.2 presents the results of this analysis.

Table 7.2
The Latent Performance of the EDHP

| | | High Speed | | | | |
|---|---|---|---|---|---|---|
| Site | cfm | DB | WB | DP | tstat | SHR |
| Site 1 | 1563 | 74 | 63 | 56 | 76 | 0.756 |
| Site 2 | 1655 | 69 | 62 | 58 | 83 | 0.776 |
| Site 3 | 1473 | 70 | - | - | 77 | 0.800 |
| Site 4 | 1545 | 78 | 65 | 58 | 80 | 0.790 |

| | | Low Speed | | | | |
|---|---|---|---|---|---|---|
| Site | cfm | DB | WB | DP | tstat | SHR |
| Site 1 | 741 | 80 | 68 | 62 | 82 | 0.713 |
| Site 2 | 950 | 70 | 61 | 56 | 83 | 0.744 |
| Site 3 | 781 | 70 | - | - | 77 | 0.783 |
| Site 4 | 867 | 80 | 67 | 60 | 80 | 0.856 |

The labels in Table 7.2 are defined as follows:

cfm - air flow rate
DB - dry-bulb temperature in Fahrenheit of the air entering the air handler
WB - wet-bulb temperature in Fahrenheit of the air entering the air handler
DP - dew point in Fahrenheit of the air entering the air handler
tstat - air temperature measured at the thermostat
SHR - sensible heat ratio (ratio of sensible cooling to total cooling)

During high-speed operation, the Sensible Heat Ratio (SHR) for the EDHP is between 0.75 and 0.80 at the four sites. For Sites 1, 2 and 4, these SHRs are consistent with the relatively high air flows (i.e., about 520 cfm per ton). At Site 3, the air flow is significantly lower--about 390 cfm per ton--but the SHR is the highest. This could happen if the air entering the air handler was relatively dry. Unfortunately, the lack of dew-point data at this site prevents a more thorough assessment of the SHR.

The SHRs at Sites 1, 2 and 3 decrease to values between 0.71 and 0.78 during low-speed operation. However, at Site 4, the SHR at low-speed increases significantly to 0.856. This behavior was unexpected. Its cause, whether it be problems in the instrumentation or an unusual operating characteristic for the 3.5-ton EDHP, has not been determined.

As shown in Table 7.2, the inlet air to the air handler at Site 2 is 13°F to 14°F cooler than that at the thermostat. This is due to "short-circuiting" of the distribution air from the supply registers to the return registers. As noted in Section 6.2, most of the return registers are located in the floor at this site. This includes a large floor-mounted return register that is located almost directly above the air handler in the basement. Apparently, the cool supply air is sinking to the floor where it is collected by the return registers before it thoroughly mixes with the room air.

## 7.6 Defrost Operation

As with conventional heat pumps, the outdoor coil of the EDHP will accumulate frost when it is removing heat from the outdoor air. Frost accumulation will be most severe when outdoor temperatures are between 30°F and 40°F.

Figure 7.25 shows the number of defrost cycles that occurred during the winter of 1996/97 at Site 2 as a function of the outdoor air temperature. Also shown in this figure is the number of hours of heat pump operation in each outdoor temperature bin. (No attempt is made to normalize the operating time to account for the variable-speed operation.)

Although the greatest number of defrost cycles occur in the 30-35°F temperature bin, this bin also has the most hours of heat pump operation. If the number of defrost cycles per hour of operation is calculated, the bin with the most defrost cycles is 15-20°F. (The 5-10°F bin has the highest defrost cycles per hour of operation. However, only two defrost cycles occurred in this bin so the measured frequency of defrosting is not considered meaningful. For this bin, the average time between defrost cycles was 2.1 hours.)

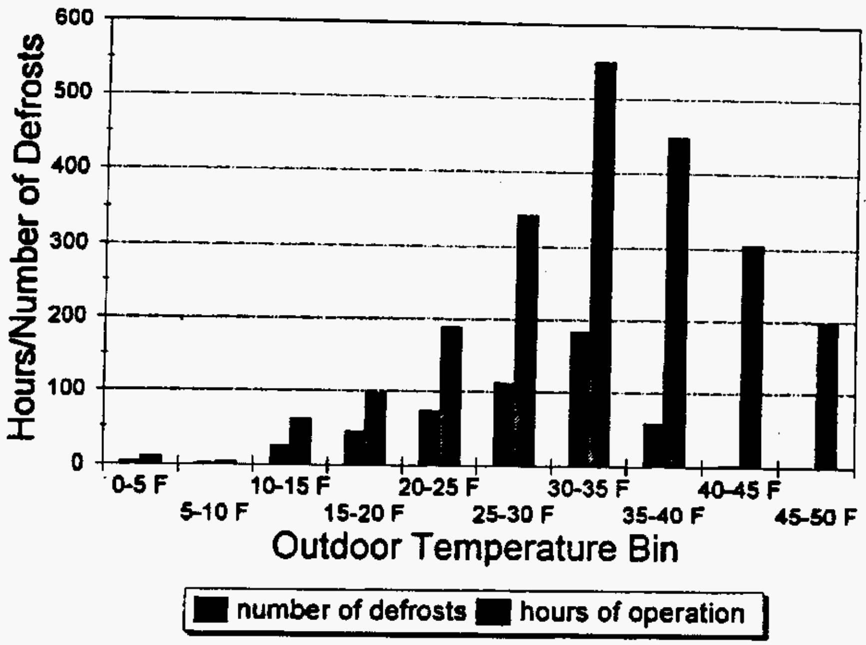


Figure 7.25 – Defrost Cycling versus Outdoor Air Temperature

The most common way to defrost the outside coil of a heat pump is to operate the unit in the cooling mode. This warms the outdoor coil by switching it from an evaporator to a condenser. However, even brief periods of cooling during the winter can inconvenience the homeowner if cold air is supplied to the house.

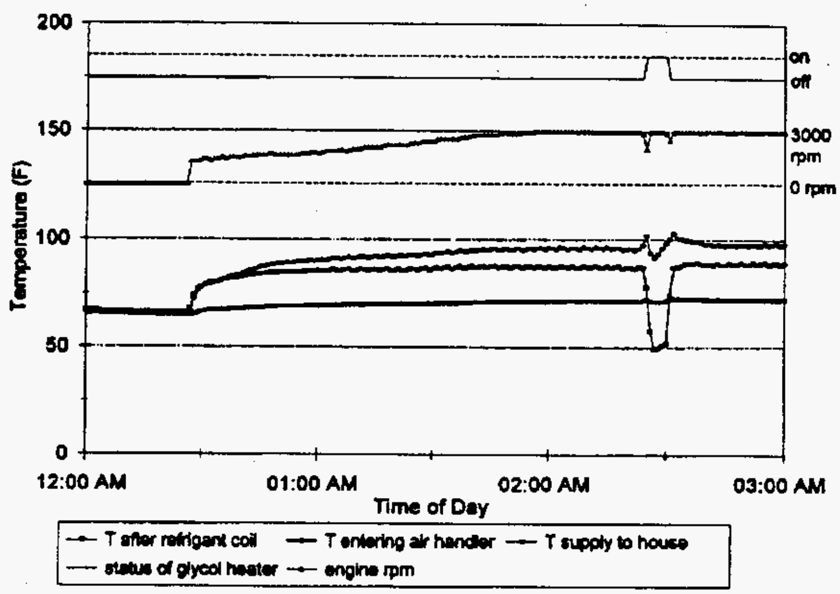


Figure 7.26 – Defrost Performance of the EDHP

The EDHP uses the glycol heater to warm the air supplied to the house during defrosting. As shown in Figure 7.26, this approach is very effective. Although the air temperature after the refrigerant coil drops to 50°F, the glycol heater keeps the air supplied to the house between 92°F and 103°F during the entire defrost cycle.

## 7.7 Supply Air Temperatures

The average supply air temperatures for the EDHPs at the four test sites are shown in Table 7.3. The data in this table were collected after the EDHPs had been operating for at least 10 minutes at either high speed or low speed. Also, the auxiliary glycol heater was off.

Table 7.3
Supply Air Temperatures for the EDHP

| | High Speed | | Low Speed | |
|---|---|---|---|---|
| Site No. | Return | Supply | Return | Supply |
| 1 | - | - | 70.1 | 95.8 |
| 2 | 73.9 | 99.1 | 70.0 | 92.6 |
| 3 | 69.9 | 93.6 | 70.0 | 96.7 |
| 4 | 77.0 | 109.1 | 70.0 | 94.0 |

For the three sites that have 3-ton units (i.e., Sites 1, 2 and 3), the supply air temperatures are comparable to those that would characterize an electric heat pump.

Site 4, which has a 3.5-ton unit, does have a significantly higher supply air temperature than the other sites. Part of the explanation is the higher return temperature. However, this site also had a low air flow rate relative to its heating capacity. Using the measured heating rates during high speed for the four units, the cfm-per-ton for Sites 2, 3 and 4 were: 465, 470 and 352.

When the auxiliary glycol heater supplements the heat pump, supply-air temperatures increase significantly. The glycol heater has a nominal output of 65,000 Btu/h. This will increase supply air temperatures by about an additional 40°F.

## 7.8 Seasonal Performance

A monthly description of the performance of the EDHPs at the four test sites for the one-year period from September 1996 through August 1997 appears in Tables 7.4 through 7.7. The data in these tables include the gas use for the auxiliary boiler during periods when the EDHPs were not in service. Table 7.8 summarizes the seasonal performance at all sites, including the number of days each unit was not in service.

## TABLE 7.4 - ANNUAL SUMMARY FOR SITE 1

**SITE 1** 1996-1997

| | Outdoor Ambient | | | | Indoor Ambient | | | | Air Handler | | | Hours | | | | | Cooling/Heating | | | Total Energy Use | | | System Performance | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Dry Bulb T (F) | | | rh% | Dry Bulb T (F) | | | rh% | Air DB T (F)(1) | | | no aux no eng aux | | | | | (MBtu) | | | Gas | Elec. (kWh) | | Engine | | cool | heat |
| month | min | ave | max | ave | min | ave | max | ave | inlet | fan(2) | outlet | AC | heat | heat | eng | def | S | L | htg | (MBtu) | Indoor | Outdr | rpm | cyc | COP | COP |
| Sep-96 | 55.8 | 74.5 | 96.2 | 78 | 73.5 | 79.3 | 84.5 | 47 | 73.3 | 53.8 | 53.5 | 266 | 0 | 0 | 0 | 0 | 5.74 | 1.31 | 0.00 | 5.21 | 100.3 | 60.5 | 1578 | 129 | 1.35 | - |
| Oct-96 | 37.8 | 64.3 | 90.0 | 78 | 71.1 | 76.8 | 82.6 | 44 | 71.1 | 55.0 | 54.7 | 116 | 3 | 0 | 0 | 0 | 2.46 | 0.36 | 0.03 | 2.03 | 80.5 | 32.7 | 1384 | 130 | 1.44 | - |
| Nov-96 | 31.3 | 56.0 | 83.7 | 73 | 62.7 | 74.5 | 81.5 | 37 | 72.2 | 85.5 | 87.9 | 10 | 39 | 0 | 0 | 0 | 0.25 | 0.03 | 0.78 | 1.00 | 68.6 | 26.6 | 1461 | 88 | 1.31 | 1.00 |
| Dec-96 | 18.2 | 50.9 | 81.3 | 76 | 63.7 | 72.8 | 79.9 | 35 | 71.3 | 91.9 | 95.5 | 0 | 111 | 0 | 1 | 1 | 0.00 | 0.00 | 2.36 | 2.06 | 74.2 | 40.2 | 1388 | 145 | - | 1.15 |
| Jan-97 | 18.5 | 49.4 | 80.4 | 68 | 63.0 | 72.5 | 77.9 | 33 | 72.0 | 92.7 | 96.0 | 0 | 118 | 7 | 1 | 1 | 0.00 | 0.00 | 3.00 | 2.72 | 73.5 | 44.5 | 1357 | 164 | - | 1.10 |
| Feb-97 | 28.4 | 54.2 | 78.6 | 76 | 64.6 | 73.7 | 80.8 | 36 | 72.2 | 92.2 | 94.8 | 4 | 86 | 0 | 1 | 0 | 0.08 | 0.03 | 2.02 | 1.69 | 69.0 | 32.3 | 1379 | 138 | - | 1.25 |
| Mar-97 | 37.8 | 64.9 | 88.0 | 65 | 70.0 | 76.3 | 81.1 | 41 | 71.7 | 56.9 | 57.1 | 76 | 1 | 1 | 0 | 0 | 1.56 | 0.19 | 0.06 | 1.46 | 38.3 | 26.0 | 1449 | 120 | 1.26 | - |
| Apr-97 | 39.5 | 62.5 | 85.6 | 65 | 70.8 | 76.8 | 81.5 | 39 | 72.1 | 62.4 | 63.1 | 52 | 2 | 0 | 0 | 0 | 1.21 | 0.09 | 0.00 | 1.00 | 89.8 | 21.0 | 1520 | 69 | 1.38 | - |
| May-97 | 43.9 | 69.8 | 92.8 | 71 | 72.2 | 77.6 | 86.3 | 48 | 72.8 | 52.8 | 52.7 | 223 | 0 | 0 | 0 | 0 | 4.96 | 0.70 | 0.00 | 4.26 | 110.0 | 52.5 | 1556 | 205 | 1.33 | - |
| Jun-97 | 57.8 | 74.5 | 94.7 | 83 | 74.3 | 78.8 | 84.3 | 52 | 74.2 | 55.1 | 54.9 | 289 | 0 | 0 | 0 | 0 | 5.84 | 1.51 | 0.00 | 5.37 | 108.1 | 64.3 | 1490 | 167 | 1.37 | - |
| Jul-97 | 66.5 | 82.1 | 100.9 | 80 | 70.1 | 78.9 | 85.0 | 51 | 74.8 | 56.8 | 56.5 | 446 | 0 | 0 | 0 | 0 | 9.76 | 3.14 | 0.00 | 11.45 | 172.1 | 97.2 | 1955 | 150 | 1.13 | - |
| Aug-97 | 62.0 | 80.1 | 98.6 | 75 | 69.8 | 78.6 | 85.1 | 50 | 74.3 | 56.0 | 55.6 | 423 | 0 | 0 | 0 | 0 | 8.87 | 2.46 | 0.00 | 9.33 | 146.1 | 91.6 | 1735 | 180 | 1.21 | - |
| total | 18.2 | 65.3 | 100.9 | 74 | 62.7 | 76.4 | 86.3 | 43 | | | | 1904 | 361 | 8 | 3 | 2 | 40.73 | 9.81 | 8.22 | 47.57 | 1130.5 | 588.4 | 1621 | 1705 | 1.26 | 1.11 |

(1) averaged only during engine operation (heating and cooling are combined)
(2) measured after fan
(3) indoor fan ran for 5,834 hours during the year with HP and heater off (includes "extended" fan operation following HP turning off)

## TABLE 7.5 - ANNUAL SUMMARY FOR SITE 2

**SITE 2** 1996-1997

| | Outdoor Ambient | | | | Indoor Ambient | | | | Air Handler | | | Hours | | | | | Cooling/Heating | | | Total Energy Use | | | System Performance | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Dry Bulb T (F) | | | rh % | Dry Bulb T (F) | | | rh | Air DB T (F)(1) | | | no aux no eng aux | | | | | (MBtu) | | | Gas(4) | Elec. (kWh) | | Engine | | cool | heat |
| month | min | ave | max | ave | min | ave | max | ave | inlet | fan(2) | outlet | AC | heat | heat | eng | def | S | L | htg | (MBtu) | Indoor | Outdr | rpm | cyc | COP | COP |
| Sep-96 | 39.8 | 65.0 | 88.1 | 77 | 74.2 | 79.2 | 86.5 | 46 | 71.8 | 78.9 | 80.1 | 66 | 36 | 0 | 0 | 0 | 1.24 | 0.24 | 1.48 | 1.77 | 77.6 | 24.6 | 1387 | 98 | 1.23 | 2.59 |
| Oct-96 | 31.8 | 54.2 | 78.7 | 76 | 68.7 | 77.7 | 84.9 | 41 | 71.1 | 93.1 | 96.5 | 0 | 158 | 0 | 0 | 1 | 0.00 | 0.00 | 4.78 | 2.72 | 86.7 | 37.9 | 1344 | 210 | - | 1.76 |
| Nov-96 | 16.1 | 36.8 | 71.2 | 75 | 70.4 | 75.8 | 79.7 | 33 | 71.7 | 89.7 | 95.2 | 0 | 289 | 74 | 9 | 3 | 0.00 | 0.00 | 12.34 | 10.90 | 91.1 | 88.9 | 1400 | 476 | - | 1.13 |
| Dec-96 | 10.1 | 36.4 | 64.3 | 83 | 70.0 | 76.5 | 80.8 | 34 | 74.0 | 92.9 | 100.1 | 0 | 462 | 0 | 20 | 9 | 0.00 | 0.00 | 16.00 | 11.73 | 115.7 | 115.7 | 1762 | 237 | - | 1.36 |
| Jan-97 | -2.9 | 27.2 | 66.6 | 77 | 69.4 | 76.3 | 80.6 | 32 | 72.9 | 89.7 | 98.3 | 0 | 300 | 153 | 26 | 8 | 0.00 | 0.00 | 20.55 | 19.10 | 186.2 | 121.1 | 2004 | 141 | - | 1.06 |
| Feb-97 | 13.2 | 35.7 | 69.8 | 78 | 70.4 | 76.4 | 80.7 | 30 | 75.1 | 92.7 | 100.5 | 0 | 431 | 1 | 18 | 10 | 0.00 | 0.00 | 13.92 | 10.42 | 133.6 | 110.9 | 1801 | 190 | - | 1.34 |
| Mar-97 | 15.6 | 42.5 | 71.5 | 71 | 70.4 | 76.6 | 80.8 | 29 | 75.8 | 94.9 | 101.1 | 0 | 365 | 4 | 9 | 4 | 0.00 | 0.00 | 10.47 | 8.47 | 125.3 | 88.3 | 1600 | 283 | - | 1.24 |
| Apr-97 | 19.3 | 48.7 | 78.2 | 58 | 70.9 | 77.1 | 83.3 | 30 | 75.0 | 94.7 | 100.2 | 0 | 244 | 0 | 5 | 3 | 0.00 | 0.00 | 7.62 | 5.55 | 86.8 | 59.9 | 1634 | 222 | - | 1.37 |
| May-97 | 34.4 | 57.0 | 83.5 | 66 | 71.8 | 78.0 | 86.4 | 35 | 75.9 | 96.8 | 100.1 | 0 | 138 | 0 | 0 | 0 | 0.00 | 0.00 | 4.16 | 2.71 | 75.6 | 34.3 | 1397 | 177 | - | 1.53 |
| Jun-97 | 47.2 | 70.4 | 93.2 | 73 | 74.3 | 80.9 | 86.3 | 44 | 70.7 | 66.9 | 66.7 | 116 | 5 | 0 | 0 | 0 | 2.46 | 0.55 | 0.24 | 2.57 | 76.8 | 28.5 | 1587 | 56 | 1.21 | 2.74 |
| Jul-97 | 51.1 | 74.1 | 96.1 | 73 | 66.0 | 81.7 | 87.5 | 43 | 69.6 | 53.2 | 53.0 | 226 | 0 | 0 | 0 | 0 | 4.84 | 0.93 | 0.02 | 4.68 | 86.6 | 46.4 | 1556 | 107 | 1.23 | - |
| Aug-97 | 48.4 | 70.1 | 92.8 | 81 | 73.5 | 80.9 | 86.6 | 47 | 70.4 | 54.6 | 54.5 | 110 | 0 | 0 | 0 | 0 | 2.25 | 0.50 | 0.09 | 2.29 | 82.9 | 26.3 | 1569 | 60 | 1.20 | - |
| total | -2.9 | 51.6 | 96.1 | 74 | 66.0 | 78.1 | 87.5 | 37 | | | | 518 | 2428 | 233 | 88 | 38 | 10.79 | 2.22 | 91.67 | 82.92 | 1224.8 | 782.6 | 1648 | 2256 | 1.21 | 1.27 |

(1) averaged only during engine operation (heating and cooling are combined)
(2) measured after fan
(3) indoor fan ran for 5,315 hours during the year with HP and heater off (includes "extended" fan operation following HP turning off)
(4) gas use estimated for January and February

# TABLE 7.6 - ANNUAL SUMMARY FOR SITE 3

SITE 3 1996-1997

| | Outdoor Ambient | | | | Indoor Ambient | | | | Air Handler | | | Hours | | | | | Cooling/Heating | | | Total Energy Use | | | System Performance | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Dry Bulb T (F) | | | rh % | Dry Bulb T (F) | | | rh | Air DB T (F)(1) | | | | no aux | no eng | aux | | (MBtu) | | | Gas | Elec. (kWh) | | Engine | | cool | heat |
| month | min | ave | max | ave | min | ave | max | ave | inlet | fan(2) | outlet | AC | heat | heat | eng | def | S | L | htg | (MBtu) | Indoor | Outdr | rpm | cyc | COP | COP |
| Sep-96 | 45.0 | 65.2 | 92.1 | 79 | 66.5 | 76.6 | 83.6 | 43 | 70.4 | 52.6 | 52.7 | 105 | 0 | 0 | 0 | 0 | 1.93 | 0.52 | 0.00 | 1.96 | 31.4 | 14.9 | 1539 | 126 | 1.24 | - |
| Oct-96 | 32.9 | 57.4 | 82.5 | 71 | 62.6 | 74.6 | 83.6 | 40 | 66.5 | 88.0 | 91.8 | 1 | 28 | 2 | 1 | 0 | 0.01 | 0.04 | 0.89 | 0.71 | 16.0 | 10.4 | 1239 | 69 | - | 1.29 |
| Nov-96 | 18.6 | 40.9 | 73.7 | 66 | 70.5 | 75.0 | 79.9 | 25 | 69.0 | 88.2 | 92.4 | 0 | 337 | 0 | 1 | 4 | 0.00 | 0.00 | 7.25 | 5.85 | 47.8 | 46.7 | 1371 | 397 | - | 1.24 |
| Dec-96 | 10.0 | 41.0 | 68.7 | 71 | 70.2 | 73.6 | 78.4 | 24 | 67.8 | 87.7 | 92.0 | 0 | 364 | 0 | 1 | 5 | 0.00 | 0.00 | 7.95 | 6.33 | 56.6 | 50.8 | 1404 | 456 | - | 1.26 |
| Jan-97 | 5.9 | 35.9 | 73.9 | 58 | 65.1 | 73.5 | 80.7 | 20 | 68.3 | 87.6 | 93.6 | 0 | 397 | 0 | 7 | 6 | 0.00 | 0.00 | 10.77 | 8.56 | 92.9 | 56.9 | 1720 | 265 | - | 1.26 |
| Feb-97 | 23.2 | 42.8 | 67.7 | 65 | 53.5 | 73.1 | 78.2 | 22 | 68.2 | 86.9 | 93.3 | 0 | 253 | 0 | 1 | 4 | 0.00 | 0.00 | 6.23 | 4.69 | 45.0 | 36.6 | 1460 | 263 | - | 1.33 |
| Mar-97 | 25.8 | 49.3 | 77.3 | 57 | 65.9 | 74.9 | 82.3 | 24 | 69.4 | 91.6 | 94.4 | 0 | 154 | 1 | 0 | 1 | 0.00 | 0.00 | 3.65 | 2.57 | 24.3 | 24.2 | 1297 | 219 | - | 1.42 |
| Apr-97 | 28.3 | 52.8 | 82.6 | 50 | 57.9 | 75.2 | 83.4 | 22 | 69.4 | 91.3 | 93.9 | 0 | 106 | 0 | 0 | 0 | 0.00 | 0.00 | 2.45 | 1.75 | 17.9 | 18.0 | 1298 | 154 | - | 1.40 |
| May-97 | 39.4 | 60.7 | 86.7 | 56 | 71.0 | 77.0 | 86.4 | 29 | 70.7 | 69.9 | 71.6 | 63 | 25 | 0 | 1 | 0 | 1.25 | 0.05 | 0.63 | 1.57 | 56.8 | 15.0 | 1410 | 136 | 1.17 | 1.35 |
| Jun-97 | 43.3 | 68.2 | 93.6 | 75 | 71.2 | 78.9 | 89.8 | 38 | 70.7 | 55.3 | 55.6 | 127 | 1 | 0 | 0 | 0 | 2.87 | 0.62 | 0.06 | 2.94 | 64.6 | 17.2 | 1867 | 124 | 1.20 | - |
| Jul-97 | 49.8 | 74.9 | 97.7 | 73 | 72.9 | 78.6 | 93.9 | 40 | 70.8 | 52.8 | 52.8 | 355 | 0 | 0 | 0 | 0 | 6.70 | 1.47 | 0.00 | 6.93 | 86.5 | 37.1 | 1594 | 306 | 1.18 | - |
| Aug-97 | 51.2 | 72.6 | 97.8 | 70 | 74.1 | 76.7 | 80.5 | 41 | 69.9 | 51.6 | 51.7 | 363 | 0 | 0 | 0 | 0 | 6.52 | 1.33 | 0.00 | 6.06 | 71.2 | 37.6 | 1390 | 384 | 1.29 | - |
| total | 5.9 | 55.2 | 97.8 | 66 | 53.5 | 75.7 | 93.9 | 31 | | | | 1015 | 1665 | 3 | 13 | 21 | 19.29 | 4.03 | 39.88 | 49.96 | 613.0 | 365.3 | 1491 | 2899 | 1.22 | 1.29 |

(1) averaged only during engine operation (heating and cooling are combined)
(2) measured after fan
(3) indoor fan ran for 686 hours during the year with HP and heater off (includes "extended" fan operation following HP turning off)

# TABLE 7.7 - ANNUAL SUMMARY FOR SITE 4

SITE 4 1996-1997

| | Outdoor Ambient | | | | Indoor Ambient | | | | Air Handler | | | Hours | | | | | Cooling/Heating | | | Total Energy Use | | | System Performance | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Dry Bulb T (F) | | | rh % | Dry Bulb T (F) | | | rh | Air DB T (F)(1) | | | | no aux | no eng | aux | | (MBtu) | | | Gas | Elec. (kWh) | | Engine | | cool | heat |
| month | min | ave | max | ave | min | ave | max | ave | inlet | fan(2) | outlet | AC | heat | heat | eng | def | S | L | htg | (MBtu) | Indoor | Outdr | rpm | cyc | COP | COP |
| Sep-96 | 51.1 | 71.0 | 93.2 | 76 | 76.3 | 79.4 | 83.0 | 47 | 77.0 | 56.2 | 56.5 | 212 | 0 | 0 | 0 | 0 | 4.47 | 0.88 | 0.00 | 3.13 | 35.6 | 33.3 | 1225 | 260 | 1.71 | - |
| Oct-96 | 36.9 | 60.7 | 83.7 | 73 | 69.9 | 77.6 | 83.3 | 46 | 74.8 | 63.7 | 64.8 | 50 | 8 | 1 | 0 | 0 | 1.05 | 0.14 | 0.31 | 1.07 | 52.7 | 15.4 | 1415 | 98 | 1.40 | 1.47 |
| Nov-96 | 26.3 | 47.4 | 76.1 | 70 | 69.1 | 75.3 | 81.4 | 35 | 73.7 | 88.9 | 96.3 | 5 | 162 | 4 | 10 | 2 | 0.08 | 0.00 | 6.11 | 4.82 | 87.9 | 38.4 | 1722 | 175 | - | 1.29 |
| Dec-96 | 16.3 | 46.4 | 70.0 | 72 | 67.6 | 75.2 | 81.1 | 33 | 73.9 | 95.0 | 103.9 | 0 | 179 | 5 | 11 | 2 | 0.00 | 0.00 | 7.35 | 5.45 | 72.7 | 41.6 | 1792 | 171 | - | 1.35 |
| Jan-97 | 14.6 | 42.6 | 73.3 | 62 | 69.6 | 75.4 | 82.0 | 28 | 74.7 | 95.0 | 102.4 | 0 | 326 | 5 | 2 | 3 | 0.00 | 0.00 | 10.39 | 6.90 | 60.3 | 61.9 | 1631 | 136 | - | 1.51 |
| Feb-97 | 29.0 | 48.3 | 73.7 | 70 | 73.0 | 76.2 | 81.1 | 29 | 74.6 | 95.7 | 102.5 | 0 | 220 | 4 | 1 | 2 | -0.00 | 0.00 | 6.66 | 3.96 | 28.8 | 43.1 | 1397 | 106 | - | 1.68 |
| Mar-97 | 32.1 | 56.0 | 81.9 | 60 | 69.9 | 77.0 | 83.0 | 35 | 74.5 | 84.7 | 88.7 | 16 | 65 | 1 | 1 | 0 | 0.39 | 0.02 | 1.96 | 1.49 | 14.3 | 19.2 | 1376 | 86 | 1.52 | 1.61 |
| Apr-97 | 34.1 | 57.7 | 82.3 | 59 | 69.7 | 77.0 | 84.7 | 32 | 74.3 | 81.4 | 84.4 | 31 | 74 | 0 | 1 | 0 | 0.79 | 0.02 | 2.05 | 1.72 | 16.2 | 22.0 | 1313 | 127 | 1.60 | 1.68 |
| May-97 | 39.8 | 66.1 | 89.6 | 58 | 74.3 | 78.8 | 83.0 | 38 | 76.6 | 56.9 | 57.4 | 112 | 4 | 0 | 0 | 0 | 2.62 | 0.19 | 0.09 | 1.68 | 24.4 | 21.2 | 1231 | 174 | 1.74 | - |
| Jun-97 | 50.7 | 72.1 | 96.6 | 76 | 74.7 | 79.6 | 83.6 | 45 | 77.2 | 58.9 | 59.1 | 235 | 1 | 0 | 0 | 0 | 5.09 | 0.83 | 0.01 | 3.51 | 39.7 | 35.4 | 1275 | 224 | 1.69 | - |
| Jul-97 | 62.4 | 79.8 | 99.2 | 73 | 77.4 | 80.7 | 84.5 | 46 | 78.8 | 59.2 | 59.4 | 365 | 0 | 0 | 0 | 0 | 8.04 | 1.32 | 0.00 | 6.02 | 82.9 | 52.5 | 1373 | 306 | 1.56 | - |
| Aug-97 | 56.4 | 77.0 | 97.6 | 70 | 76.2 | 79.7 | 82.8 | 44 | 78.4 | 57.6 | 57.9 | 318 | 0 | 0 | 0 | 0 | 5.94 | 1.17 | 0.00 | 4.42 | 51.8 | 46.5 | 1217 | 320 | 1.61 | - |
| total | 14.6 | 60.7 | 99.2 | 68 | 67.6 | 77.7 | 84.7 | 38 | | | | 1317 | 1037 | 20 | 26 | 9 | 28.47 | 4.58 | 34.91 | 44.17 | 567.2 | 430.4 | 1421 | 2203 | 1.62 | 1.47 |

(1) averaged only during engine operation (heating and cooling are combined)
(2) measured after fan
(3) Indoor fan ran for 1,251 hours during the year with HP and heater off (includes "extended" fan operation following HP turning off)

Table 7.8
Summary of EDHP Performance
(COPs are based on gas use only)

| | Cooling | | | Heating | | | outage | fan only |
|---|---|---|---|---|---|---|---|---|
| Site No. | MBtu | COP | MWh | MBtu | COP | MWh | days | hours |
| Site 1 | 50.5 | 1.26 | 1.20 | 8.2 | 1.11 | 0.52 | 2 | 5,834 |
| Site 2 | 13.0 | 1.21 | 0.30 | 91.7 | 1.27 | 1.71 | 14 | 5,315 |
| Site 3 | 23.3 | 1.22 | 0.40 | 39.9 | 1.29 | 0.58 | 0 | 686 |
| Site 4 | 33.1 | 1.62 | 0.51 | 34.9 | 1.47 | 0.49 | 0 | 1,251 |

The three sites with the 3-ton EDHP (i.e., Sites 1, 2 and 3) had comparable performance: COPs for the cooling season were between 1.21 and 1.26, and for the heating season, between 1.11 and 1.29.

The performance of the 3.5-ton EDHP at Site 4 is significantly better than the performance of the 3-ton units. The larger unit at Site 4 did tend to run at a lower engine speed, which could explain part of the difference. Unfortunately, since catalog performance data is not available for the 3.5-ton unit, it is difficult to determine whether the seasonal performance at Site 4 is reasonable.

### 7.9 The EDHP versus Conventional Technology

The EDHP that was tested will have relatively low operating costs considering the high COPs that were measured during the field test. However, will its operating costs be sufficiently low to justify its selection over conventional gas and electric technologies?

To answer the preceding question, two alternative systems were studied--one an all-electric heat pump with a 11.9 SEER and a 7.85 HSPF, and the other a 11.9 SEER electric air conditioner combined with a 95% AFUE gas furnace. Both these alternatives are high-efficiency premium systems, and so they should appeal to the same customers as the EDHP.

An important aspect of the field test was the direct measurement of the heating and cooling output of the EDHP. With this data and the coincident indoor and outdoor conditions, it was possible to simulate the hour-by-hour performance of the alternative electric systems using manufacturer's steady-state catalog data. To account for cycling effects, the total compressor energy that was calculated from catalog data was increased by 6%.

Table 7.9 compares the energy consumption and operating costs for the EDHP and the two conventional systems. Operating costs have been calculated using \$0.0841 per kWh and \$0.605 per therm, which are the 1994 national average rates.

As shown in this table, the EDHP does have the lowest operating costs at all four sites. However, its cost advantage is very small. For the four sites, the differences in operating costs between the EDHP and the combined furnace/air-conditioner are: (1) $53, (2) $121, (3) $83 and (4) $140. At all sites these annual savings do not cover the higher annual maintenance costs for the EDHP: in this field test the annual maintenance to replace spark plugs, change oil, and perform other minor tasks ranged from $200 to $350. (Costs for repairing the major problems that occurred in this field test are obviously not covered by the savings in operating costs.)

TABLE 7.9

Comparision of Operation Costs for Alternative Technologies

| Site | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| EDHP | | | | |
| electricity (kWh) | 1219 | 1626 | 928 | 912 |
| gas (therms) | 476 | 829 | 500 | 442 |
| operating cost ($) | 390 | 638 | 380 | 344 |
| Heat Pump SEER 11.9/ HSPF 7.85 | | | | |
| electricity (kWh) | 5541 | 13146 | 6942 | 7207 |
| gas (therms) | 0 | 0 | 0 | 0 |
| operating cost ($) | 466 | 1106 | 584 | 606 |
| Air Conditioner/Furnace SEER 11.9/ 94% AFUE | | | | |
| electricity (kWh) | 4639 | 2008 | 2448 | 3085 |
| gas (therms) | 87 | 975 | 424 | 371 |
| operating cost ($) | 443 | 759 | 463 | 484 |

NOTE: field test electricity usage adjusted to eliminate periods of continuous fan operation

$0.605 per therm, $0.084 per kWh

### 7.10 The Performance of the Glycol Loop during Heating

The EDHP can achieve a high COP during the heating season by recovering "waste" heat from the engine. Heat is transferred from the engine to the indoor air handler via the same glycol loop that transfers heat from the auxiliary boiler.

In this field test, the glycol loop was a source of problems. At two sites, glycol temperatures would exceed their upper limits and shut down the system.

If the glycol loop was replaced by a conventional gas furnace, how much will the EDHP's heating efficiency be degraded? This can be answered by calculating the percentage of the total heat delivered to the building that was recovered from the engine.

Table 7.10 presents the percentage of total heat delivered to the building that was recovered from the engine during periods when only the engine was operating (i.e., the auxiliary boiler was off). Ignoring Site 1 which had a very low heating load, the recovered heat averaged 23.6 percent of the total delivered to the building. Thus, if the EDHP has a heating COP of 1.25 when recovering heat from the engine, it will have a 0.955 COP without heat recovery.

Table 7.10
The Percent Contribution to Total Heating from the Engine Waste Heat

| | |
|---|---|
| Site 1 | 16.5% |
| Site 2 | 24.1% |
| Site 3 | 20.0% |
| Site 4 | 26.8% |

### 7.11 Users' Reactions to the EDHP

Customers' reactions to the four EDHPs that were tested were mixed. Most of the participants in the field test were satisfied with the overall performance of the heat pumps. However, several reported that exhaust emissions were bothersome, particularly during the summer months when they spent more time outdoors. In general, noise from the EDHP was noticeable, but acceptable.

# 8

# CONCLUSION

The two-year field test reported here provided an excellent opportunity to understand the operating characteristics of the engine-driven EDHP heat pump. When the field-test units operated at steady-state they all demonstrated thermal performance that was consistent with catalog data. However, three of the four EDHPs that were tested had significant operational problems that degraded seasonal efficiency. At one site, the engine had to be replaced. At the site where the air handler was mounted on the second floor, numerous service calls were not able to correct a problem that produced repeated trips of the system on high glycol temperature.

Considering its high installed cost and high annual maintenance costs, the EDHP is unlikely to be a viable alternative to a gas furnace combined with an electric air conditioner in most of the country. In warmer climates where all-electric heat pumps compete well against systems that use gas furnaces, the all-electric system will most likely be chosen over the EDHP by most customers.