LBL-32451
UC-350
Preprint

Lawrence Berkeley Laboratory

UNIVERSITY OF CALIFORNIA

ENERGY & ENVIRONMENT DIVISION

Submitted to ASHRAE Transactions

Energy Performance of Heat Pumps in New Commercial Buildings in the Pacific Northwest

M.A. Piette, O. de Buen, and B. Nordman

April 1992

ENERGY & ENVIRONMENT DIVISION

Prepared for the U.S. Department of Energy under Contract Number DE-AC03-76SF00098

LBL-32451
UC-350

ENERGY PERFORMANCE OF HEAT PUMPS IN NEW COMMERCIAL BUILDINGS IN THE PACIFIC NORTHWEST*

M. A. Piette, O. de Buen, and B. Nordman

Energy & Environment Division
Lawrence Berkeley Laboratory
University of California
Berkeley, CA 94720

April 1992

*This work was jointly supported by the Bonneville Power Administration and by the Assistant Secretary for Conservation and Renewable Energy, Office of Building Technology, U.S. Department of Energy, under Contract No. DE-AC03-76SF00098. Appeared in ASHRAE Transactions 1992, Vol. 98, Pt. 2 (paper # BA-92-3-2).

ENERGY PERFORMANCE OF HEAT PUMPS IN NEW COMMERCIAL BUILDINGS IN THE PACIFIC NORTHWEST*

LBL-32451
UC-350

M. A. Piette, O. de Buen, and B. Nordman

Energy & Environment Division
Lawrence Berkeley Laboratory
One Cyclotron Road
Berkeley, CA 94720

ABSTRACT

This paper reviews the performance of efficient heat pumps in new commercial buildings located in the Pacific Northwest. The objective of the analysis is to explore differences between modeled and monitored heat pump performance, and examine the performance of high-coefficient of performance (COP) heat pumps. A detailed case study compares hourly monitored heat pump data to results from a DOE-2.1C computer simulation model. Design predictions and heat pump characteristics are presented for five small buildings.

The simulation overestimated fan energy use since, unlike in the actual building, the fan is modeled as "on" for full hour increments. In general, the measured COP values show that the heat pumps are operating well, averaging slightly higher than the modeled average. The measured COP is within 10% of the average measured COP for the majority of the hours. We found that the need to control and minimize the use of supplemental electric resistance heat can be as important as the use of efficient heat pumps. In the case-study building, ramp-up thermostats could have reduced electricity used for morning warmup and lowered the peak electric demand.

The average COP investigations suggest that as an alternative to detailed modeling, analysts interested in evaluating the actual energy impacts of efficiency improvements in occupied buildings should consider basing results on short-term monitoring. This paper reviews a simplified evaluation of the energy savings from high-COP heat pumps based on annual compressor energy use and average COP values. At a minimum, simple instrumentation to disaggregate compressor and resistance heat energy use can provide useful data to evaluate actual heat pump energy performance.

---

*This work was jointly supported by the Bonneville Power Administration and by the Assistant Secretary for Conservation and Renewable Energy, Office of Building Technology, U.S. Department of Energy, under Contract No. DE-AC03-76SF00098.

INTRODUCTION

In 1986 an electric utility initiated a program to evaluate the potential for energy savings in new commercial buildings. One major goal of the program is to design, construct, and assess new commercial buildings that reduce energy use by at least 30% below a hypothetical base-case building. The 28 buildings in the program range from fast-food restaurants to large offices. Efficient heat pumps were selected in 11 buildings as one of the technologies to achieve the targeted energy savings. The systems include air-to-air, ground-source, water-source, and air-to-water (hydronic) heat pumps.

This paper focuses on the energy performance and the heating characteristics of three air-to-air heat pumps in a small ophthalmic clinic. The clinic is a 3,030-ft^2 (281-m^2) building in Ashland, Oregon. The primary technical objectives of the evaluation are to determine if each building met the 30% energy savings target, and to establish the savings from each individual measure.[1] The analysis described below is part of a U.S. laboratory's review of the building energy modeling activities within the program, which are based on the DOE-2.1C simulation program. Part of our review includes comparing the results from detailed monitoring with modeling. We have also sought to identify additional energy savings opportunities that may be present in the buildings. For example, in the case-study building, the use of ramp-up thermostats could have reduced electricity used for morning warmup, and lowered peak electric demands.

The analysis described below is being applied to several other buildings, although at this midpoint in the evaluation, the data available for each building are limited. Also included is a review of the characteristics and predicted energy savings of the heat pump improvements in five commercial buildings.

PERFORMANCE DATA

We briefly review general characteristics of the evaluation methodology since it determines the type of monitored data and model simulations available for this analysis of the heat pumps.

Simulation Models

The evaluation of the buildings and the individual energy-saving measures relies on a combination of hourly end-use metering and DOE-2.1C simulation modeling.[2,3] The development of the calibrated model begins with an "as-built" DOE-2.1C model that is based on information from both the standardized "documentation package" that contains a description of the efficiency improvements and the as-built site plans, plus the on-site "operations and maintenance (O&M) audits." Next, the building schedules and site weather data derived from monitoring are added to the as-built model. The calibration of each model, or "tuning," is done by modifying the input assumptions until the output meets pre-defined tolerances. Input assumptions changed in the model calibration include infiltration, fan mode ("on-always" vs. "on-demand"), and outside airflow rates. The criteria for tuning the models are that the simulated monthly end-use must be within 30% of the monitored energy end uses and within 10% of the whole-building seasonal energy use. The five end uses in the monthly tuning include total heating, ventilating, and air conditioning (HVAC), indoor and outdoor lighting, water heating, and plug loads.

A "tuned baseline" model is derived by subtracting all of the energy conservation measures from the tuned model. Each measure is individually modeled against the tuned baseline, and the levelized cost is calculated. The baseline is designed to represent the Northwest regional model conservation standards (MCS), which are based on ASHRAE Standard 90-80A.[4] Model documentation is generated for each building. The DOE-2.1C iterations used in tuning the case-study building and the analysis of each efficiency improvement are described in a report by the modeling team.[5]

Unfortunately, the final tuned model for the case-study building did not meet the predefined tuning tolerances during two of the twelve months of 1989 data. In the tuned model simulation run used in this analysis, the spring season HVAC energy use was 30% less than the monitored spring season HVAC energy use, and the autumn HVAC energy use was 55% greater than monitored.[†] Similar problems in simulation "tuning" have been noted within the program for several of the small buildings, which tend to have more irregular control settings, operation, and occupancy than the larger buildings.

Multi-Year Hourly Monitoring

The monitoring plans were designed to produce equivalent levels of hourly end-use data among the 28 buildings, which were to be used to calibrate the DOE-2.1C models. Additional data include indoor and outdoor dry-bulb temperatures, outdoor wet-bulb, and limited solar and wind measurements. End-use disaggregation included, at a minimum, submetering lighting, total HVAC water heating, and miscellaneous plug loads. At least one HVAC zone is instrumented with the following channels:

- *Airflow* - hourly averages are derived from a differential pressure measurement.
- *Mixed and supply air temperatures* - an average of seven discrete sensors spread across a hexagonal pattern with one sensor in the center.
- *HVAC energy use* - measured with a watt transducer at the electrical panel.

In some cases additional measurements include

- *Fractional on-time* - fraction of the hour when the fan, the compressor (both during heating and cooling), or the resistance heat operated.

Indoor ambient dry-bulb and wet-bulb measurements are available for selected zones within each building. We do not have the capability to measure latent loads, restricting our ability to analyze the heat pumps in the cooling mode.

[†]Following our analysis, two additional parametric simulation runs were conducted to improve the fit to the monitored energy use data, but the final model was still not within the predefined tolerances described above.

The monitoring contractor estimates that the flow measurements are correct within 10%.[6] For example, in many of the buildings the flow measurement was made in duct runs that were shorter than optimal, causing interference from duct bends. Also, the heat pump's thermal output measured with the simple instrumentation, will vary slightly from the actual output of the heat pump due to several effects. Cycling causes thermal transients in the data on the hourly average heat supplied by the HVAC since the sensors take about one minute to reach equilibrium. A related issue is that duct losses can reduce the heat supplied by the HVAC system that is effectively delivered to the respective zone.

CASE STUDY OF THE OPHTHALMIC CLINIC

To review the performance of the HVAC systems at the clinic, we compare the calibrated DOE-2.1C simulation results with the monitored results for the largest of the building's three heat pumps (referred to as heat pump 1 - HP1). HP1 has a rated heating capacity of 14.3 kW_t (kW thermal), and a COP_r of 3.5. Heat pumps 2 and 3 are smaller with lower efficiencies (rated at 7.0 and 3.1 kW_t, and COP_rs of 3.2 and 2.8, respectively). We use the subscript r to designate that the COP is the rated, instantaneous, steady-state COP based on Air-Conditioning and Refrigeration Institute (ARI) Standard 240,[7] including the ventilation supply fan power. Since the data from DOE-2.1C and from the monitoring are based on hourly measurements that include cycling and defrost losses, we also examine average COPs. We use COP_{af} to designate an average COP including fan power and COP_{an} to designate an average COP excluding fan power.

Simulated Performance of Heat Pump 1

Results from the simulation are presented in Figure 1. The plot shows the average hourly thermal power (kW_t) supplied by the heat pump and the hourly average electric power (kWh/h_e) for a full year of operation. This plot shows three distinct modes:

A1. *compressor heating alone,*

A2. *compressor heating with defrost cycle, and*

B. *compressor and resistance heat combined.*

We discuss the total energy use and heat pump efficiency of the three modes below.

Some features of the simulation input are as follows. The heating capacity of the heat pump is modeled as a function of the return air wet-bulb temperature and the outside dry-bulb temperature. The compressor provides heat until the efficiency is reduced to near that of resistance heating. The compressor cutout was modeled at an outdoor dry-bulb temperature of -30°F (-34.4°C). Not surprisingly, this low cutout temperature was never reached. The compressor heat with defrost cycle occurs only when the outdoor temperature is below 35°F (1.7°C). The simulation models defrost cycles equivalent to 3 minutes of defrosting every 90 minutes during normal heat pump operation.

We are particularly interested in identifying opportunities to minimize the use of resistance heat. For most heat pumps, the three conditions that result in the use of resistance heat are

- *compressor cutout at low outdoor temperatures,*
- *thermostat senses a large thermal load* - one that cannot be met by the compressor alone, and
- *emergency heat* - manual thermostat lockout of compressor.

All of the resistance heat in this simulation falls into category 2, when the load cannot be met within the hourly time step by the compressor alone. The resistance heat is never on alone. Within the simulation, the resistance heat is locked-out for outdoor temperatures above 60°F (15.6°C).

Most of the use of resistance heat occurs during morning warmup, which can be seen in Figure 2. These plots show the median hourly weekday electric load profiles for the total HVAC system in January, February, and March. Maximum, minimum, upper and lower quartiles, and the mean (dashed line) are indicated on the figures. Weather and hourly building schedules are from 1989 data. The plots show the morning peak in the first two months, which was greatly reduced by March. Also, the nighttime HVAC operation is significant during the first two months. The erratic HVAC load profile in the early morning hours of March is due to the DOE-2.1C hourly time step. The heat pump first comes on at 6 a.m., satisfying the interior temperature requirements. The load at 7 a.m. does not require that the heat pump remain on, and the interior temperature floats within the deadband. During the third morning hour, at 8 a.m., heat pump heat is again required.

Monitored Performance of Heat Pump 1

The modes of heat pump operation for the actual building, shown in Figure 3, differ from the regimes identified in the computer simulation in Figure 1. We find the following four modes:

- *compressor heating with fan always on* - the compressor is on part of the hour, and fan is on during the whole hour,
- *compressor and resistance heating combined,*
- *resistance heat only* - resistance heat is on part of the time while the fan is on the whole hour, and
- *fan on demand* - fan and the compressor are on together only for a fraction of the hour.

During normal operating hours the fan is always on to supply ventilation (regimes A and B), and in the on demand mode with the night setback control (regime D). However, during parts of February and March 1988 the fan was operated on demand during weekday daytime hours (regime D). Unfortunately, we are unable to accurately determine the thermal load provided by the HVAC during the fan on-demand hours because we do not know the fraction of the hour that the fan operated. The fractional on-time data, available for some of the other program buildings, were not available for the monitoring of the clinic during the periods covered in this analysis. The "resistance heat only" operation (regime C) may be due to manual lockout since it does not occur at severe outdoor temperatures. However, these occur randomly through two winter months in 1989, during day and late nighttime hours, suggesting they are not due to occupant control.

Figures 4 and 5 show the monitored hourly weekday electric load profiles for HP1 in January, February, and March 1988 and 1989. The 1988 data are more similar to the modeled load profile than the 1989 data. It appears that during 1989 there is no night setback and a less severe morning warmup spike.

COMPARISON OF MODELED AND MONITORED PERFORMANCE

As mentioned above, the simulation model was tuned to the monitored data using monthly and seasonal end-use totals. Simulated total annual HVAC energy use is 20% greater than the monitored total (19.1 MWh vs. 16.0 MWh). The monitored data presented below are for September 1988 through August 1989. The 65°F (18.3°C) heating degree-days are 5% greater for the modeled year than the monitored year. The modeling is based on calendar year 1989.

Method to Disaggregate HVAC Energy Use

DOE-2.1C simulation output includes disaggregated HVAC energy use, reporting fans, resistance heat, compressor heat, and cooling energy use. The monitored heat pump energy use data were not disaggregated. To evaluate how the simulation output compared with the monitoring we developed a method to disaggregate the monitored total HVAC energy use and tested it on the DOE2.1C hourly HVAC energy use to ensure reliable results.

Audit and simulation input data were used to determine average power ratings of the supply fan, compressor, and resistance heat. The disaggregation is based on determining which components of the heat pump are on and assigning hourly HVAC energy use to each component. We assume that the resistance heat energy use is the energy that exceeds the maximum hourly fan plus compressor energy use. We also assume the fan is always on during work hours (8 a.m. to 6 p.m., Monday through Friday) and on demand during other hours, which was the case for the majority of the year.

An important difference between the simulation and the actual HVAC system is that in DOE-2.1C the fan is on for the full hour if the compressor heat is needed for any fraction of the hour. For the actual building, when the fan was operating in the always-on mode, the energy use assigned to fans ranges from nearly zero to the total fan power (further described in the example). During fan on-demand operation, energy is divided between the fan and the compressor, since both operate simultaneously.

Our method to disaggregate HVAC energy slightly differed among the three heat pumps. To separate heating energy from cooling we used the thermal HVAC load measurements for

HP1 (Figure 3) and HP2. There are no thermal HVAC measurements for HP3, only total energy use, so the separation of heating and cooling for HP3 is based on outdoor temperature. Figure 6 shows that the regimes of heating and cooling are distinct, with the division near 60°F (15.6°C). Further differences in the disaggregation for each heat pump are discussed below.

Disaggregation of Heat Pump 1

To illustrate the methodology, we present results for HP1. According to the model documentation, the rated power of the compressor is 4.78 kW_e; with the exterior fan (0.3 kW_e), the total is 5.08 kW_e. The ventilation supply fan is rated at 0.6 kW_e. The actual consumption of the heat pump is significantly less than the 5.68 kW_e the ratings suggest. The average power of the fan, identified during full hours of fan only operation, is 0.49 kW_e. We assume the ventilation fan uses 0.49 kW_e and that the compressor consumes less than the rated power. (The DOE-2.1C simulation modeled the fan at the full rated power). Compressor power varies with load. For example, at the ARI coefficient of performance rating point (outdoor air of 47 °F [8.3°C] dry-bulb and 43 °F [6.1°C] wet bulb, with air entering equipment at 70 °F [21.1°C] dry bulb) heat pump power is 15% less than the rated capacity (heating capacity of 14.4 kW_t divided by the COP [3.52] is 4.1 kW_e).

The results of this disaggregation and the test of our method on the model output are shown in Table 1. The first two rows show the DOE-2.1C simulation output and our estimate of disaggregated total DOE-2.1C HVAC energy use. The HVAC subcategories compare well, although we assigned more of the energy to compressor heat than modeled, and underestimated fan energy use. The resistance heat energy differed by only 1%.

Table 1 also shows our estimate of the disaggregated HVAC energy use; modeled HVAC energy use for HP1 is 20% greater than the monitored total. The energy use for resistance heat from the DOE-2.1C simulation totaled 1993 kWh and our estimate, based on the monitoring, is 1.34 MWh. This total, though less than the simulated total, is 21% of the total heating energy use. The relative fraction of the energy use of the resistance heating to the total heating energy by the DOE-2.1C simulation is similar (23%).

The fan energy use is higher in the simulation than our estimate from the monitored data due to the hourly time-step for DOE-2.1C and the higher full-load power use. The largest difference in HVAC disaggregation was the cooling end-use, where modeled energy use was less than one-third of monitored energy use.

Comparison of Total HVAC Energy Use

Table 2 shows our estimate of the disaggregated HVAC energy use for the three heat pumps compared to the simulation output. Again we see the modeled fan energy use is higher than monitored--nearly twice as high for the three heat pumps combined. Monitored cooling energy use is again greater than simulated. However, the comparison between modeled and monitored heating and cooling greatly differs among the three heat pumps, showing differences in the zones. HP3 consumes the majority of the cooling energy in the model output (1047 kWh_e), while the monitored data show the majority of the cooling energy is from HP1 (2482 kWh_e).‡

Table 2 also shows the thermal heat load delivered by each heat pump and the load delivered by compressor heating (Comp Ht). All of the load data for HP1 are direct measurements, except for the fan on-demand hours, when the load was measured for an unknown fraction of the hour (regime D in Figure 3). Using the data from the fan always-on mode (regime A in Figure 3) we estimate a COP_{an} of 2.92. This is an average COP, including cycling and defrost losses, without the fan power. Using this average COP, and an average compressor kW_e, we estimate the HVAC load during the fan on-demand hours. The COP_{an} (2.92) for HP1 is 83% of the ARI COP_r (3.5).

During the whole year HP2 operated in a fan on-demand mode (similar to regime D in Figure 3), rendering the load measurements useless in the absence of fractional on-time data. For HP2 we used an estimate of the COP based on 83% of the ARI COP_r of 3.2 to derive the delivered load. We followed this approach for HP3 as well. This fraction of the average to the rated COP may be high since HP1 operated more fully loaded than the other two heat pumps.** However, the use of a slightly high COP for these small heat pumps will not have a major impact on the analysis since they together represent less than a third of the HVAC energy use.

Factors Influencing COPs

There are subtle differences between Figures 1 and 3 that influence our COP comparisons. The load supplied by the HVAC in DOE-2.1C does not include the heat of the supply fan motor, although in the actual building the motor is in the duct. The monitored "load" includes the fan heat, which has the effect of shifting all the points 0.5 kW_t to the right since the fan consumes about 0.5 kW_e. The simulated fan uses 0.68 kW_e, or about 35% more. These differences affect our comparisons of fan energy, HVAC loads, and COPs. The simulation begins every cycle with 0.35 kW_e, which does not deliver load but apparently accounts for cycling losses. The monitored cycles cross the hour boundary making it difficult to identify short cycles.

The average heating COP derived from the monitoring is slightly higher than the modeled COPs. The COP_{af} includes the fan energy only while the compressor operates, based on the ratio of the average fan power to the average compressor power. The key reason for measured COP exceeding the modeled COP is that the simulation includes more short cycle periods than the monitored operating data used to calculate the average COP. Dynamic losses cause the average measured performance to differ from rated performance. (Dynamic losses from cycling increase with cycle frequency, which will be greater in oversized systems.[8] Figures 7 and 8

‡Subsequent iterations of the DOE-2 model increased the total cooling to 2.39 MWh_e, still well below our estimate of 3.10 MWh_e.

**In subsequent analysis of monitored heat pump data at another program building, we found an average COP of 74% of the ARI rating.

show the average hourly COPs versus outdoor temperature for the modeled and monitored performance of HP1. The modeled heat pump shows numerous hours when the average COP_{af} is below two and the outdoor temperature is above 60°F (15.6°C) (Figure 8). One might expect the average COP to increase with outdoor temperature. However, we see a large distribution in hourly average COPs during warmer weather. This is due to the heat pump quickly meeting heat demands with shorter cycles and, less steady-state operation, and increased dynamic losses. Two symbol types are shown in Figure 8 that represent hours when the heat pump operated for greater and less then 20 minutes. The hours of less than 20 minutes have lower average COPs.

Figure 7 shows that the monitored COP_{af} is not strongly temperature dependent. Also shown in Figures 7 and 8 is the manufacturer's rated COP as a function of outdoor temperature. (Note the ARI rating point: COP_r is 3.5 at 47°F [8.3°C].) The step in the COP below 35 °F [1.7°C] for the modeled output is due to defrost power (Figure 8), which cannot be seen in the monitored data because of hourly averaging (Figure 7).

All-Resistance vs. All-Compressor Energy Use

Table 3 shows the range in total heating energy use that might occur under various combinations of resistance and compressor heating. The first three columns repeat our estimate of the total energy for resistance and compressor heating energy use for each of the three heat pumps. The fifth column shows the energy use required to heat the building if only electric resistance heating was used (25.1 MWh), derived with the average COP for each heat pump (shown in column 4). The final three columns show scenarios of heating energy use with all-compressor heating. The first of these shows the total energy use (8952 kWh) assuming the resistance heat would have been compressor heat based on the derived COPs listed in the same table. The other two scenarios show the energy use for slightly lower and higher COPs (COP_{an} of 2.5 and 3.5).

Not only is electric resistance heat less efficient than compressor heating, it can lead to spikes in electric load profiles, as shown in Figures 2 and 4. While electric load management issues have not been a priority in the program evaluation, many electric utilities will benefit from reduced morning peak loads.

Savings from MCS to Actual COPs

Also shown in Table 3 is a scenario representing an MCS base case of a $COP_r = 2.7$. To derive an average COP we used a constant fraction of the measured to the rated COP (83%) derived from HP1, leading to an estimated COP_{an} of 2.24. The MCS base-case column shows the total heating energy use with the same energy use of the resistance heat as monitored, and estimated compressor energy use had the COP been the MCS base case. We estimate that the MCS base-case building would have consumed 12057 kWh to heat the building. That is, the improvements in the COPs beyond MCS saved 2291 kWh in heating energy use.

We can compare this with the results from the program modeling. After the DOE-2.1C model was calibrated with the monitoring, it was modified to model an MCS base case building.

The base-case MCS heat pumps were modeled with COP_f of 2.7. The total modeled savings from the increased COP is 1.18 MWh (0.39 kWh/ft^2·yr [4.2 kWh/m^2·yr].) This estimate is about half our simplified estimate. The modeled savings include reduced heating energy use plus reduced cooling energy use since the heat pump also had a higher cooling COP. On the other hand, cooling energy use is only a small part of the HVAC energy use.

Note also that the savings from the increased COP are similar to the total energy use of the electric resistance heat, which, with better control, could also be reduced.

HEAT PUMP PERFORMANCE DATA FOR FIVE SMALL BUILDINGS

In the following section we review the energy savings predictions, characteristics, and comfort issues of the heat pumps in the five program buildings with air-to-air heat pumps. This is presented to give some greater context for the case study results described above.

Design Predictions

The program began with a design competition that required applicants to submit energy savings estimates for each efficiency improvement. Table 4 summarizes the early design data for the air-to-air heat pumps in five all-electric buildings. The buildings are small "office-type" buildings. Compared to a standard heat pump that meets the MCS code, the incremental costs to install the high-COP systems range from \$0.10/ft^2 (\$1.08/m^2) to \$1.79/ft^2 (\$19.27/m^2). These costs include design, equipment, and installation costs. The incremental costs (area normalized) increase with decreasing building size. However, the cost data must be evaluated with caution since there are significant differences in the assumptions and components considered in the cost estimates.

Most of the early energy-saving estimates were generated using DOE-2 simulations. Improvements in heat pump COPs were estimated to save from 0.14 kWh/ft^2·yr (1.5 kWh/m^2·yr) to 1.03 kWh/ft^2·yr (11.09 kWh/m^2·yr) (a range of nearly an order of magnitude). The savings shown in Table 4 include both heating and cooling savings. At the clinic the predicted savings include economizer savings, while the "tuned" savings described above are for the heat pump alone (both heating and cooling savings). The range in estimates of energy savings is due to several factors such as differences in climate, further discussed below. Operating schedules and control strategies differed among the buildings. A smart, rampup thermostat was used in one building to reduce the use of resistance heat during morning warmup, while the clinic was controlled with a conventional seven-day setback thermostat. There is significant variation in the COP targets. Not all of the early design estimates used the same baseline COP, which ranged from 2.3 to 2.8. The improved COPs range from 3.0 to 3.4. Future comparisons within the program's evaluation activities will use more comparable results from the "tuned" models.

As-Built Heat Pump Characteristics

The equipment installed differed from the early design plans. Installed COPs tended to be slightly higher than the early design assumptions, probably a result of improved availability of better equipment. The installed COPs ranged from 2.9, a small increase over the MCS standard, to 3.5, a substantial improvement beyond even the 1992 standards. Within individual buildings the heat pump COPs varied. At the clinic this variation was significant: the smallest heat pump (3.1 kW_t) has a COP of 2.9, and the largest (14.3 kW_t) has a COP of 3.5.

Correct sizing of heat pumps influences energy efficiency. We suggest that energy analysts interested in evaluating the field performance of heat pumps examine cycle frequency. While the system must be large enough to provide adequate thermal comfort, if it is too large it may have a frequent duty cycle, and performance may suffer from increased cycling losses. If it is undersized, there may be frequent use of resistance heat. The rated thermal heating capacity of the heat pumps listed in Table 4 ranged from 6.5 W/ft^2 (70.0 W/m^2) to 11.9 W/ft^2 (128.1 W/m^2) (the coldest climate). Electric resistance backup heating added another 7.1 W/ft^2 (76.4 W/m^2) to 12.5 W/ft^2 (134.6 W/m^2).

Comfort Complaints

In general, the heat pumps are performing well. However, there were reports of comfort problems in most of the buildings. In one building some of the offices and the production area alternated between being too cold and too hot. Two offices adjacent to the computer room at another building were consistently warm. At the clinic, the occupants reported comfort problems, with cold temperatures during the early mornings, especially on Mondays. Similarly, in another building the occupants reported hot and cold spots in the building. Due to the lack of results on occupant surveys in a large sample of small commercial buildings, it is difficult to say whether these complaints are unique to heat pump systems, or whether the complaints are common to most HVAC systems.

Duct Losses and Zoning

Although we found that the COP of HP1 was relatively high compared to the rated COP, we have not yet explored how effectively the fan and duct system is in delivering the heat to individual zones. Intensive short-term monitoring at one building found that the "effective COP" was poor when distribution losses were considered, although the building's ducting is located in unconditioned space (whereas the clinic's ductwork is in conditioned space).

PERFORMANCE ISSUES: CONTROLS AND EVALUATION PROCEDURES

Within the program, heat pump conservation measures were evaluated as incremental improvements in the COP. We suggest that demand-side management programs and energy-efficiency design evaluations give equal consideration to heat pump control equipment and operating strategies.

The Need for Smart Controls

We have found that the majority of resistance heating is used during morning warmup. In many buildings additional energy savings appear to be available with the use of smart, rampup controls to reduce resistance heating. These systems calculate an optimal start time for the heat pump based on the temperature in the zone and the target temperature for morning warmup. Compressor heating is brought on earlier in the morning warmup schedule to minimize the use of resistance heating. Savings in monthly peak demand costs can be significant when morning warmup peaks are reduced.

The energy savings from improved controls are difficult to model. DOE-2.1C cannot directly simulate optimal start controls that are designed to minimize the use of supplemental resistance heat. Rather, they trade shorter runtimes for higher heating efficiency by estimating the shortest amount of time needed to warm the building by applying the full capacity of the heating system, including resistance heat. To simulate an optimal start, one can modify the heating schedule to include one or more rampup hours. At one building this was done by setting the rampup hour setpoint equal to half the difference between the occupied setpoint temperature and the minimum zone temperature reported by the simulation.[9]

Beyond the difficulty of modeling the energy savings from control strategies is the challenge of insuring that the system is installed, calibrated, and commissioned properly within the building. Establishing the use of optimal control strategies is particularly difficult in small buildings that lack full-time, sophisticated building operators.

Lessons on Performance Evaluation

Short-Term Monitoring of Heat Pump Performance. The program's monitoring plans were designed prior to the development of a detailed work plan. Ideally we would have conducted additional short-term measurements under steady-state conditions for resistance-heat-only and compressor-heat-only modes. Flow measurements would not have been necessary because flow could be calculated from the heat rate and the temperatures obtained from resistance-heat-only mode measurements (with constant-volume units).[10] The heat rate can be obtained and compared with compressor power to get a more robust COP measurement. If the test conditions (especially outside temperature) differ from the rated COP conditions, a comparison with the values of capacity and power from the manufacturer's technical sheet should be done (for the test conditions).

Alternative Approach to Modeling Energy Savings. Our analysis suggests that there may be alternatives to detailed simulation modeling to evaluate the savings from improvements in heat pump efficiency. Simulation modeling is often extremely time consuming and may not adequately represent the operating characteristics of an HVAC system. We found that the COP remained relatively constant (within 10 % of the average) over the majority of compressor heating hours. Given this result, we can evaluate a change in the average efficiency using a simple estimate of the energy impact of a change in COP on annual compressor heating energy use. The energy savings are:

$$\text{Savings} = (\text{compressor energy use}) \times (\text{COP1}^{-1} - \text{COP2}^{-1}).$$

This estimation requires disaggregated resistance and compressor heat pump energy use data. A second analytical task would be to evaluate the use of resistance heat in the building to identify potential energy-saving opportunities with improved controls.

Clearly there are several simplifications in this technique, which may be considerable drawbacks under certain conditions; they warrant further investigation. For example, the variation of COP with outdoor temperature may be greater in colder climates or if a rampup thermostat were used to allow greater use of cold morning compressor heating. This simplified estimation method is offered in consideration of the need to develop simplified methods to analyze the field performance of energy-efficiency measures based on short-term metering. Further comparisons of the merits of modeling versus monitoring are needed, including the costs to collect field data compared with the costs to develop and calibrate sophisticated building models.

CONCLUSIONS

The analysis of the heat pumps has shown that the data collection and analysis methods used in the program illustrate several important lessons about the heat pumps in small commercial buildings. First, the COP measurements in the case-study building's largest heat pump show that it has been operating well and slightly better than the simulation model results. While the heat pump is performing well, the use of electric resistance heat could be further reduced with a rampup thermostat. New commercial building design studies should carefully consider optimal HVAC control strategies while evaluating efficient HVAC components. To ensure that these controls are properly applied, the system should be properly commissioned and the building operator trained to understand the optimal use of the control system.

The case study results showed that the DOE-2.1C model, calibrated to the monitored data using total HVAC energy use, showed significant differences in fan, compressor heating, resistance heating, and cooling energy use. The simulation overestimated fan energy use since it is modeled as always on during full-hour increments, which is not the case in the actual building. As an alternative to detailed modeling, it may be desirable to evaluate the energy savings from changes in the COP of a heat pump using annual compressor energy use and an average COP. At a minimum, simple instrumentation to disaggregate compressor and resistance heat energy use can provide useful data to evaluate actual heat pump energy performance.

ACKNOWLEDGMENTS

The authors are indebted to many other participants in the program, including BPA staff, program sponsors, contractors, and building owners. We also thank Helmut Feustel, Rene Meldem, Jeff Harris, Alan Meier, and Rick Diamond at Lawrence Berkeley Laboratory for their assistance.

REFERENCES

1. Diamond, R.C., J.P. Harris, O. de Buen, B. Nordman, 1990. "Evaluating Actual Performance of New Commercial Buildings: The Energy Edge Demonstration Program." *Proceedings* of the ACEEE 1990 Summer Study on Energy Efficiency in Buildings, American Council for an Energy Efficient Economy, Pacific Grove, CA, 3: 3.77-3.89.
2. Kaplan, M., B. Jones, and J. Jansen, 1990. "DOE-2.1C Model Calibration with Monitored End-Use Data." *Proceedings* of the ACEEE 1990 Summer Study on Energy Efficiency in Buildings, American Council for an Energy Efficient Economy, Pacific Grove, CA, 10: 10.115-125.
3. Gardner, C.M., and L.A. Lambert, 1987. "Monitoring Methodology for Energy Edge." *ASHRAE Transactions*, Volume 93, Part 1.
4. NWPPC, 1985. "Model conservation standards equivalent code" February. Northwest Power Planning Council, Portland, OR.
5. KE and PECI, 1991. "Energy Edge Siskiyou Ophthalmic Clinic Simulation Tuning" Prepared for the Bonneville Power Administration, July 16. Kaplan Engineering and Portland Energy Conservation, Inc, Portland, OR.
6 Lambert, L, 1991. Personal communication, March.
7. ARI, 1981. Standard 240. *"Standard for Air-Source Unitary Heat Pump Equipment."* Air-Conditioning and Refrigeration Institute, Arlington, VA.
8. Miller, W.A, 1988. "Laboratory Examination and Seasonal Analyses of the Dynamic Losses for Continuously Variable-Speed Heat Pumps." *ASHRAE Transactions*, Volume 94, Part 2.
9. Kaplan, M., and J. McFerran, 1989. "West Yakima Medical Facility As-Built Model Documentation Report." Prepared for Pacific Power and Light Co.
10. Proctor J., et al., 1990. Pacific Gas & Electric heat pump efficiency and super weatherization pilot project. Final Report to the Pacific Gas and Electric Company.

Table 1. HVAC disaggregation for HP1 for simulated and monitored energy use.

	HVAC Disaggregation (MWh_e/year) Resist	Fan	Comp.Ht	Comp. Cl	Total
DOE-2.1C Results:					
Actual	1.99	4.64	6.79	0.73	14.16
Derived estimate	1.97	4.01	7.33	0.74	14.05
Monitored:					
Derived estimate	1.34	2.32	5.18	2.48	11.32

Table 2. HVAC energy use and COPs for simulated and monitored energy use.

	DOE-2.1C HP1	HP2	HP3	Total	Derived from Monitoring HP1	HP2	HP3	Total
MWh_e								
Fan	4.64	0.83	0.37	5.85	2.32	0.63	0.15	3.09
Resist	1.99	0.21	0.007	2.21	1.34	0.12	0.02	1.48
Comp Ht	6.79	2.44	0.36	9.58	5.18	2.85	0.26	8.29
Comp Cl	0.73	0.15	1.05	1.93	2.48	0.46	0.16	3.10
HVAC	14.16	3.02	1.94	19.12	11.32	4.05	0.59	15.96
MWh_t								
Tot Heat	20.66	6.27	0.78	27.71	16.86	7.66	0.67	25.18
Comp Heat	18.66	6.07	0.77	25.50	15.12	7.54	0.65	23.31
COP_{an}	2.75	2.97	1.48	2.66	2.92	2.65	2.31	2.81
COP_{af}	2.52	2.83	1.46	2.50	2.67	2.54	2.26	2.64
COP_r	3.52	3.2	2.9		3.52	3.2	2.9	

MWh_e - Electrical MWh

MWh_t - Thermal MWh

COP_{an} - Average COP with no fan

COP_{af} - Average COP with fan

COP_r - Rated ARI COP

Table 3. Actual, all-resistance, MCS base case and all-compressor energy use.

	Heating Energy *(MWh/year)*							
	Derived from Monitoring			All	MCS	All Compressor		
	Resist. Heat	Comp Heat	Total Heat	Resist. Heat	Base Estim.	COP_{an} *Derived*	*2.5*	*3.5*
Heat Pump 1	1.34	5.18	6.52	16.86	8.27	5.77	6.74	4.82
Heat Pump 2	0.12	2.85	2.97	7.66	3.48	2.89	3.06	2.19
Heat Pump 3	0.02	0.26	0.28	0.67	0.31	0.27	0.27	0.19
TOTAL	1.48	8.29	9.77	25.18	12.06	8.95	10.07	7.20

MCS base $COP_r = 2.7$ is equivalent to $COP_{an} = 2.24$.

All-compressor heating cases are shown for three different COP assumptions. The COP_{an} values for the derived case are 2.92 for HP1, 2.65 for HP2, and 2.31 for HP3.

Table 4. Design data for the air-to-air heat pumps in five buildings.

Building Name (City)	Area	Design Predictions				Actual Characteristics			
		COP_r(1)		Costs(2)	Savings	Installed	Capacity	Compressor	Strip Heat
	(ft^2)	Baseline	Improved	($/$ft^2$)	(KWh/yr-ft^2)	COP_r(3)	(W_t/ft^2)	(W_e/ft^2)	(W_e/ft^2)
West Yakima (Yakima, WA)	16221	2.3	3.0	0.10	0.14	3.1	7.6	2.68	12.37
O'Ryan (Vancouver, WA)	6020	(6.7)*	(8.5)*	0.39	N.A.	3.0	7.2	2.7	8.0
East Idaho (Ihaho Falls, ID)	5300	N.A.	N.A.	0.79	0.19	3.4	11.9	3.6	12.5
Siskiyou (Ashland, OR)	3030	2.8	3.3	1.03	1.03	3.4	8.1	3.0	8.2
Caddis & McFaddin (Spokane ,WA)	2100	2.6	3.4	1.8	0.87	3.3	6.5	1.9	7.1

Notes:

* Value is Heating Seasonal Performance Factor (HSPF). HSPF is the total heating output of a heat pump during its normal annual usage period for heating in Btu, divided by the total electric power input, including resistance heating, during the same period, in kWh.

(1) Design stage values (ARI Standard rating).

(2) Includes design and construction costs.

(3) Weighted average based on capacity (ΣCapacity*COP/ΣCapacity)(ARI Standard rating).

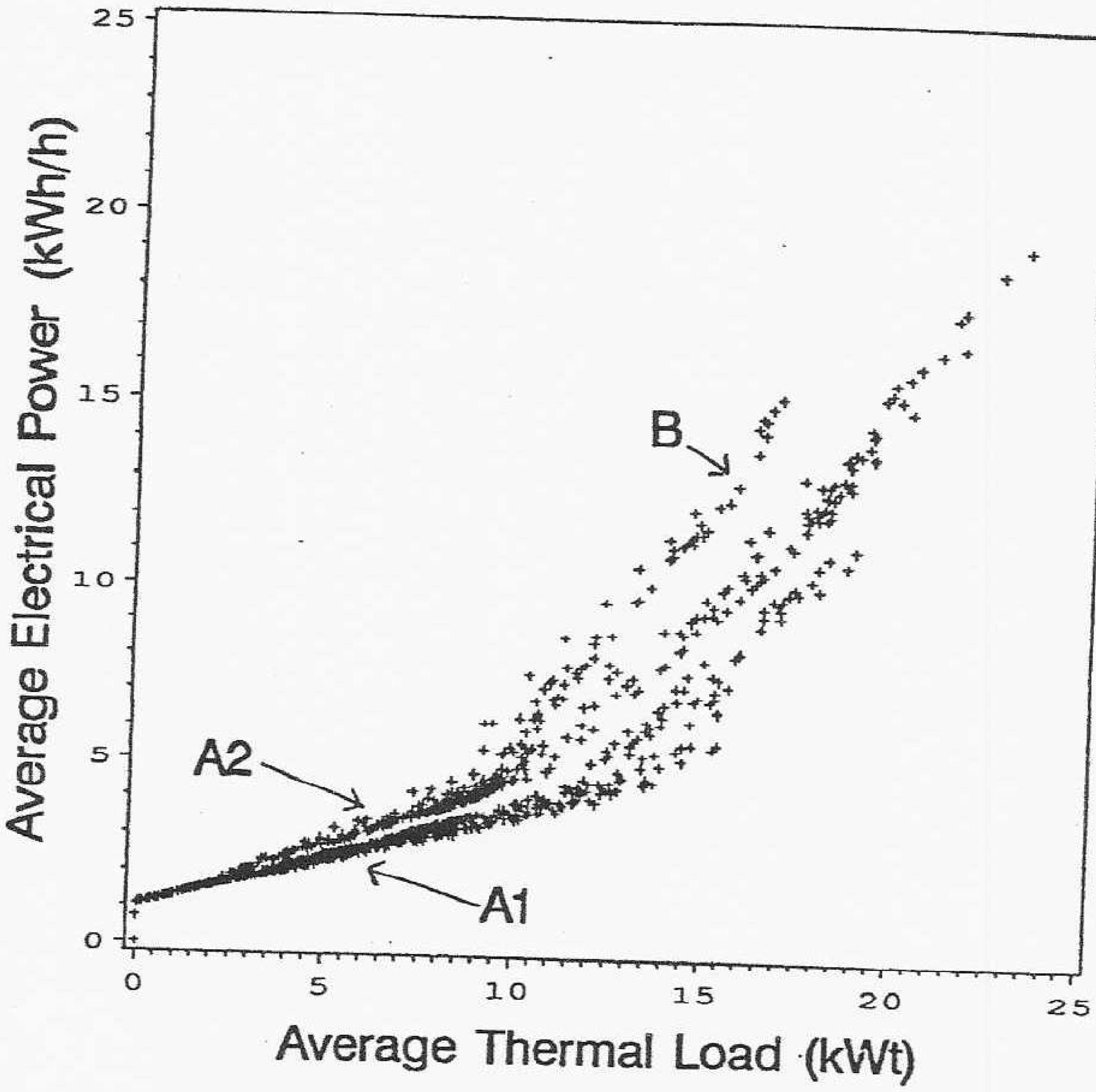

Figure 1. Simulated average hourly thermal load (kW_t) delivered by the heat pump, and hourly average electric power (kWh/h_e) for a full year of operation. Three operation modes are labeled: A1 - compressor heating alone, A2 - compressor heating with defrost cycle, and B - compressor and resistance heating combined.

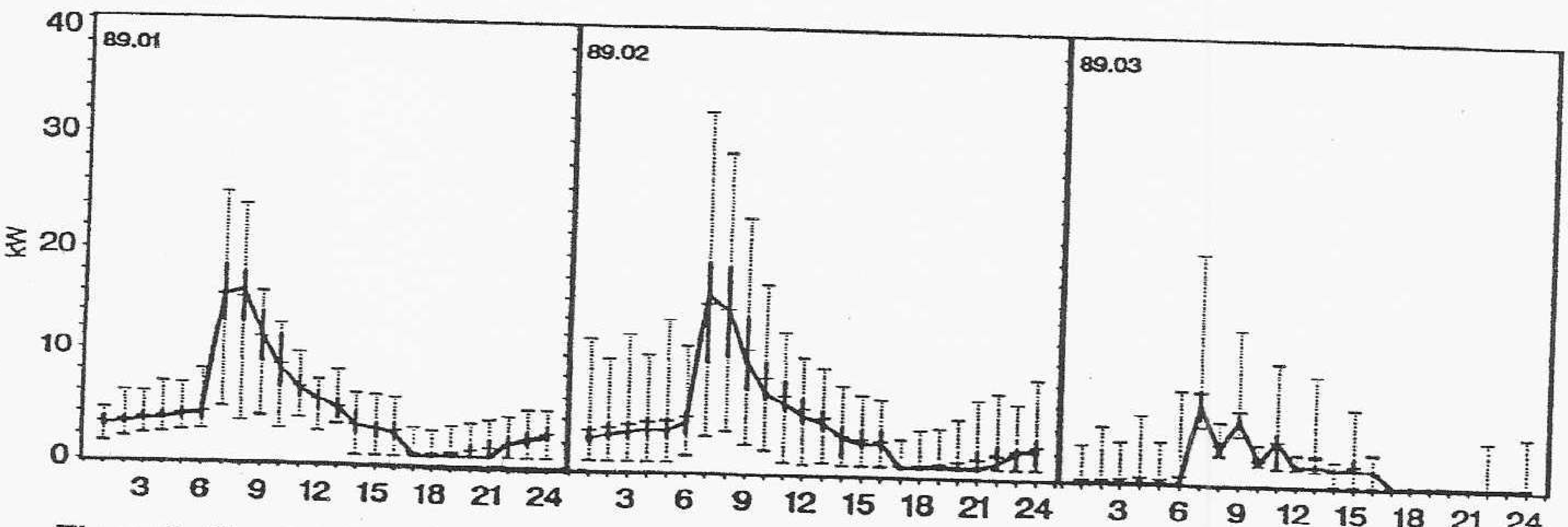

Figure 2. Simulated median hourly weekday electric load profiles for HP1 in January, February, and March. Maximum, minimum, upper and lower quartiles, and the mean (dashed line) are indicated on the figures.

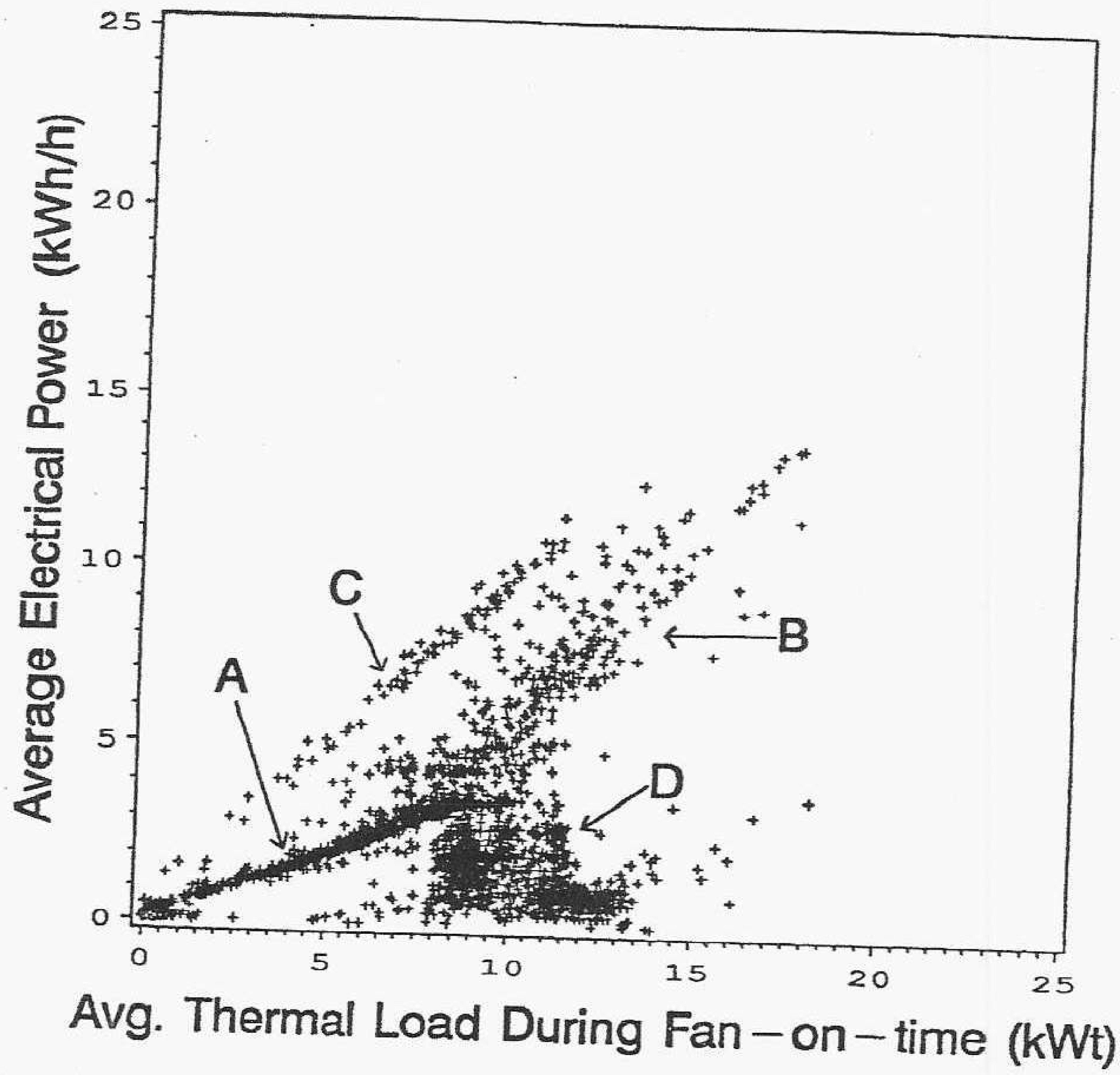

Figure 3. Monitored average thermal load during fan on-time (kW_t) delivered by the heat pump versus hourly average electric power (kWh/h_e) for a full year of operation. Four operating modes are labeled: A - compressor heating with fan on during the whole hour, B- compressor and resistance heating combined, C - resistance heat only, and D - fan on-demand, fan and compressor are on together for a fraction of the hour.

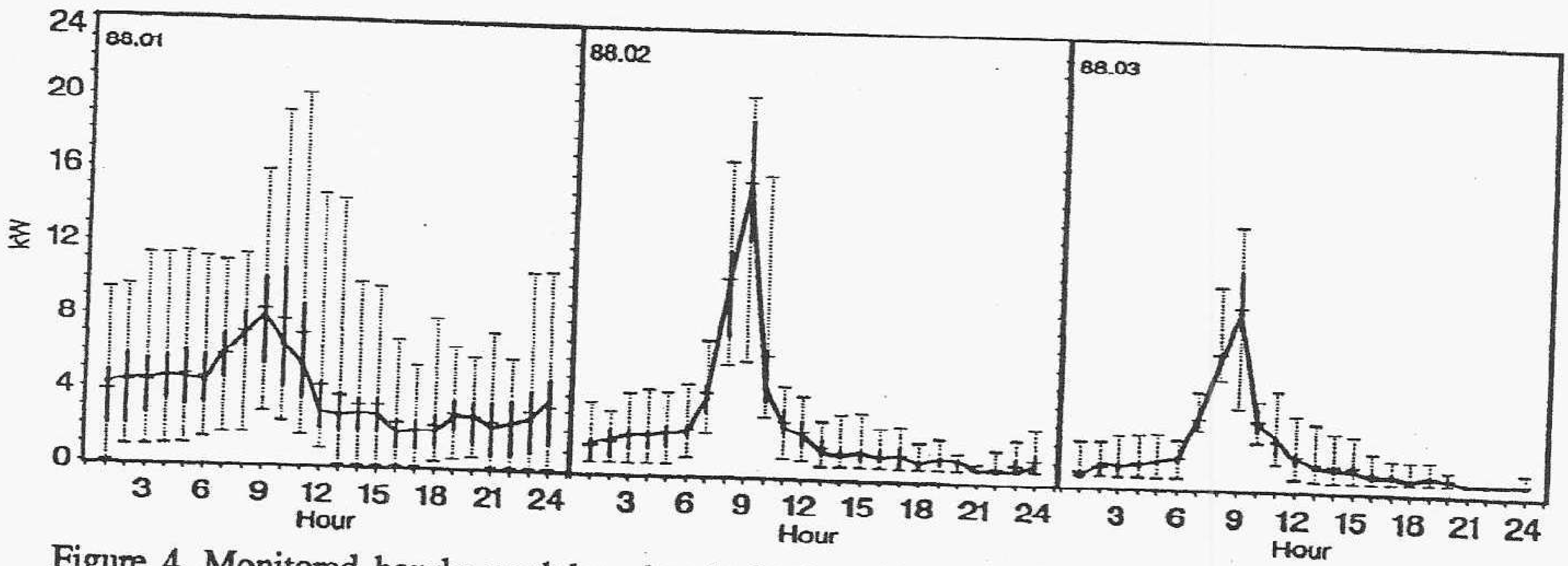

Figure 4. Monitored hourly weekday electric load profiles for HP1 in January, February, and March 1988. Maximum, minimum, upper and lower quartiles, and the mean (dashed line) are indicated on the figures.

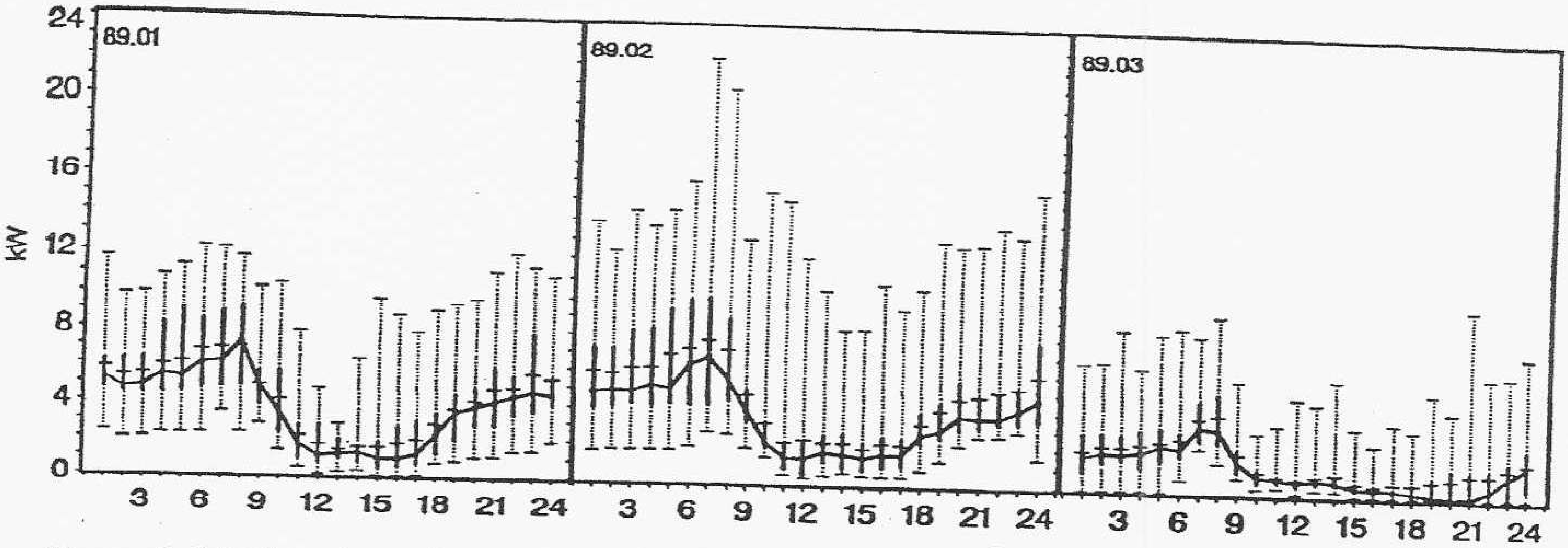

Figure 5. Monitored hourly weekday electric load profiles for HP1 in January, February, and March 1989. Maximum, minimum, upper and lower quartiles, and the mean (dashed line) are indicated on the figures.

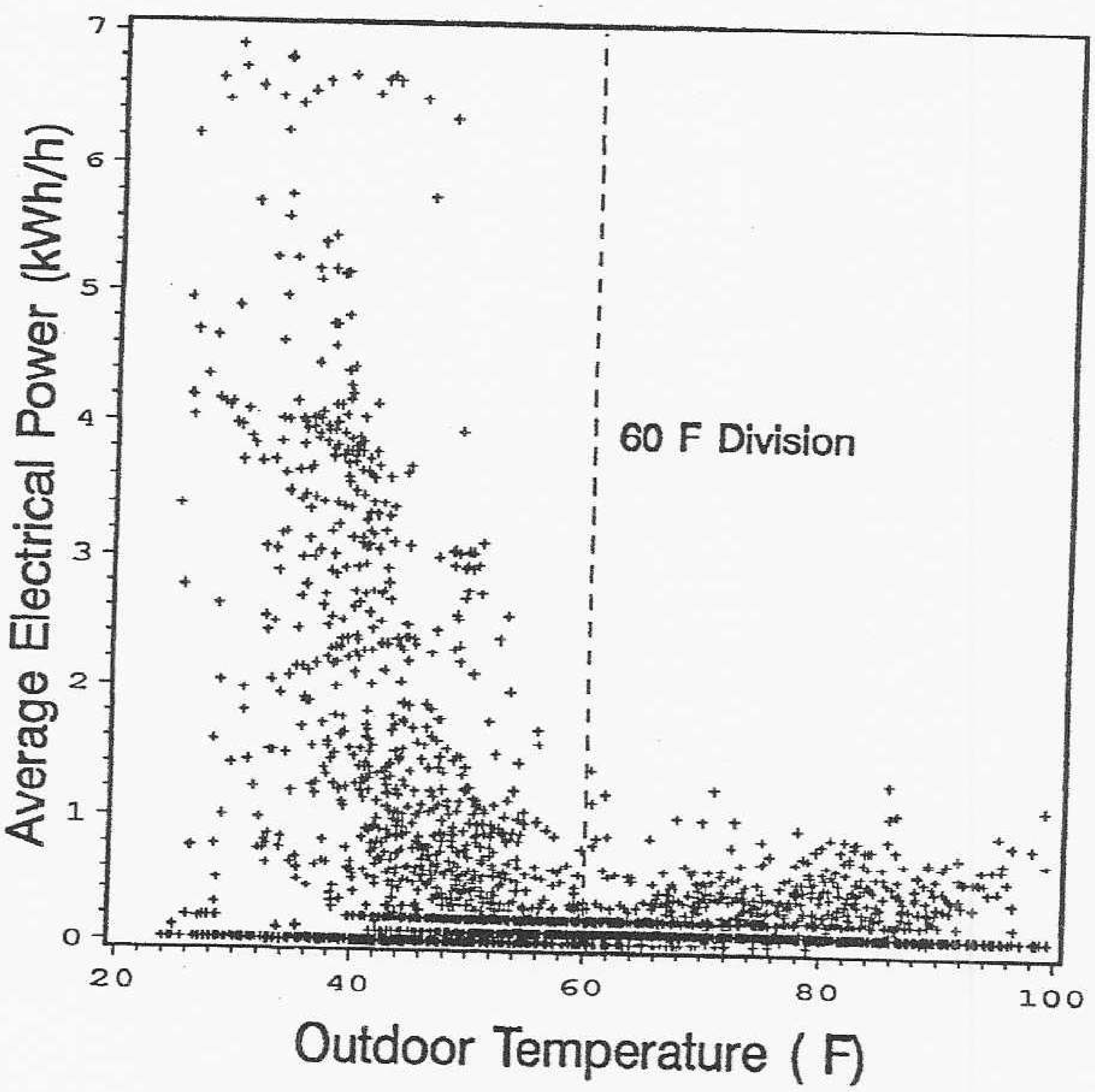

Figure 6. Hourly energy use versus outdoor temperature for HP3. The 60°F (15.6°C) division was used to separate heat pump heating from heat pump cooling energy use.

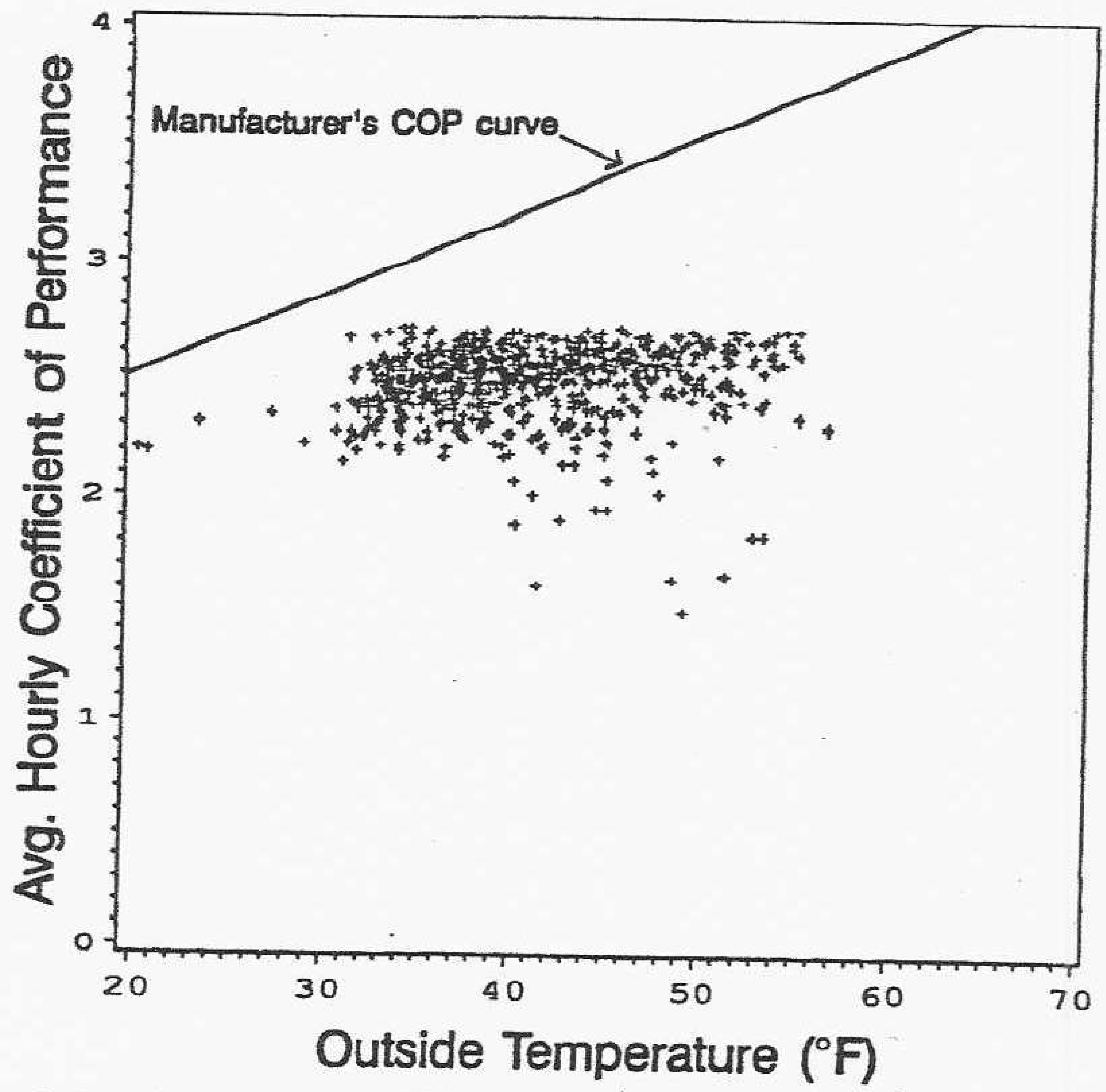

Figure 7. Measured hourly average COP_{af} versus outdoor temperature. Also shown is the manufacturer's rated COP as a function of outdoor temperature. (Note the ARI rating point: COP_r is 3.5 at 47°F [8.3°C].)

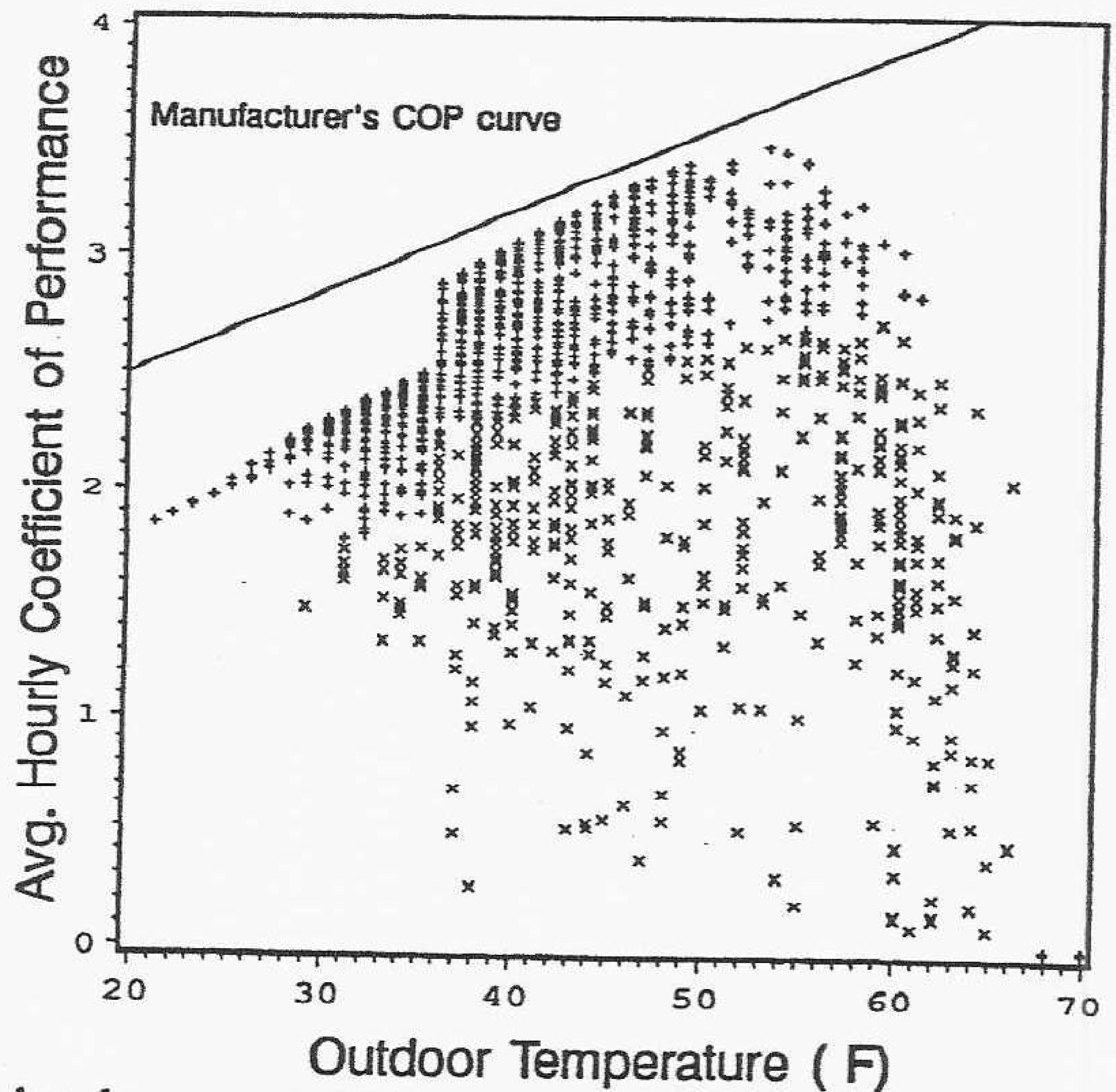

Figure 8. Modeled hourly average COP_{af} versus outdoor temperature. Also shown is the manufacturer's rated COP as a function of outdoor temperature. Two symbols are shown. A "+" indicates the heat pump operated for more than 20 minutes during the hour. An "x" indicates the heat pump operated for less than 20 minutes, representing a short cycle length, and therefore lower average COP.