FIELD MONITORING OF A VARIABLE-SPEED INTEGRATED HEAT PUMP/ WATER-HEATING APPLIANCE

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ABSTRACT

A variable-speed integrated heat pump/water-heating appliance was monitored for two years while meeting the space-conditioning and water-heating needs of an occupied residence. Experimental results are presented that show the total energy consumed by the residence was significantly reduced compared to previous years in which electric baseboard heat, a wood stove, and window air conditioners were used. During the two space-heating seasons, the variable-speed integrated heat pump/water-heating appliance used 60% less energy than would have been consumed by an electric furnace with the same air distribution system and a storage-type electric water heater.

The monthly space-cooling-only coefficients of performance (COP) ranged from 2.50 to 4.03, whereas the monthly space-heating-only coefficients of performance ranged from a low of 0.91 to a high of 3.33. A proposed index to quantify the overall system performance of integrated water-heating/ space-conditioning appliances, referred to as the combined performance factor, ranged from 1.55 to 3.50. The majority of larger values occurred during months in which space cooling dominated. The combined performance factor for the entire two-year study was 2.45.

A conventional watt-hour meter supplied by the local electrical utility and an electronic digital power analyzer were used to measure the energy consumption of the variable-speed heat pump to discern if variable-speed equipment introduces errors in conventional utility metering equipment. Measurements made using the two instruments were in excellent agreement.

The monthly energy consumption and peak electrical demands of the residence, integrated heat pump/water-heating appliance, supplemental space heater, and water heater are discussed. The influence of outdoor temperature on electrical power demand is presented.

INTRODUCTION

Nearly one out of three new single-family houses built in the United States uses an electric heat pump for space conditioning. Since the early 1950s, when heat pumps were introduced into the residential market, manufacturers have continuously strived to improve reliability, increase efficiency, improve thermal comfort, and meet increasingly stringent environmental concerns while remaining competitive with oil- and gas-fired residential space-conditioning equipment.

During the past decade, heat pumps have evolved from units utilizing a single-speed reciprocating compressor to units that incorporate variable-speed components, rotary and scroll compressors, microprocessor controls, demand-limiting features, and integrated water heating. These technological advances have resulted in heat pumps that are significantly more efficient than traditionally designed units, although they have yet to gain a significant share of the heat pump market. Additional technological advances, including thermal storage, multiple condensers, and adaptive controls, are currently being introduced into the market.

This paper summarizes a research project (Fanney 1993) in which a commercially available, technologically advanced integrated heat pump/water-heating appliance (hereafter referred to as the heat pump) was extensively instrumented and monitored under field conditions. The heat pump selected for this study incorporates a variable-speed compressor and indoor fan, microprocessor controls, and a refrigerant-to-water heat exchanger. The unit attempts to improve thermal comfort by controlling the ratio of indoor fan speed to compressor speed in a manner that improves humidity control during the cooling season and avoids the occasional blowing of cool air during the heating season. Rather than extracting heat from the conditioned space to defrost the outdoor coil, the heat pump uses energy stored within the water heater.

The research was undertaken to determine the thermal performance of an integrated heat pump system and the electrical load characteristics of an occupied residence equipped with such a unit. Secondary objectives were to investigate whether the variable-speed components within the heat pump system introduce errors in conventional residential watt-hour meters and to provide data for future computer simulation validation studies.

This paper describes the residence, heat pump system, and monitoring equipment. Results are presented that include a comparison of the total electrical energy consumption of the residence prior to and after installation of the heat pump system, the portion of energy used by each end use

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within the residence, a comparison of the heat pump's energy consumption using a conventional watt-hour meter and an electronic digital power analyzer, and the hourly electrical demands imposed on the utility. The thermal performance of the heat pump system is reported on monthly, seasonal, and annual bases using conventional performance indicators in addition to using an index that quantifies the overall system performance of integrated appliances.

EXPERIMENTAL APPARATUS

Residence

The residence in which the integrated heat pump was tested is located approximately 19 km (12 mi) south of Hagerstown, Maryland. Space heating and cooling is provided to 153.6 m^2 (1,653 ft^2) of living area within the residence. The integrated heat pump system and accompanying water heater are located in the 53.5-m^2 (576-ft^2) unconditioned below-grade basement. The first level of the conditioned space includes a family room, bathroom, and laundry room. A living room, dining room, and kitchen are located on the second level. Three bedrooms and two full baths are contained on the third level. During this study the residence was occupied by a working adult, a homemaker, and two preschool children.

The first level's exterior walls consist of nominal 50.8-mm by 101.6-mm (2-in. by 4-in.) studs on 0.406-m (16-in.) centers, with R-11 fiberglass batts between the studs, positioned against a 0.203-m (8-in.) concrete block wall. The exterior walls of the second and third levels consist of nominal 50.8-mm by 101.6-mm (2-in. by 4-in.) studs spaced 0.406 m (16 in.) apart. The walls are insulated with nominal R-11 fiberglass batts between the studs and 25.4 mm (1 in.) of extruded polystyrene sheathing fastened to the exterior beneath vinyl siding. The uninsulated exterior walls of the basement are constructed of 0.305-m (12-in.) concrete blocks.

The windows throughout the residence are vinyl-clad, double-glazed, double-hung wood units. Exterior insulated steel doors are located in the kitchen and living room. The remaining exterior doors—one in the walkout family room and one leading onto the deck from the dining room—are double-glazed sliding units. Attic insulation consists of R-30 fiberglass batts placed between the 0.610-m (24-in.) on-center roof trusses.

Prior to the installation of the variable-speed integrated heat pump, the residence was heated by electric baseboard heaters in each room and a wood stove in the family room. Approximately five cords of wood were burned each heating season prior to the installation of the heat pump. Space cooling was provided to approximately one-third of the residence through the use of two 1.46-kW (5,000-Btu/h) window air conditioners, one located in the living room and the second one in the master bedroom. The domestic water-heating needs of the residence were met through the use of a nominal 0.197-m^3 (52-gallon) electric storage-type water heater.

Heat Pump Description

The variable-speed integrated heat pump system was installed in the residence in September 1989 and was fully operational prior to October 1, 1989. Detailed monitoring of the heat pump system began in June 1990. The system consists of three distinct components: (1) an indoor fan coil, (2) an outdoor fan coil, and (3) a compressor section that accommodates the compressor, a refrigerant-to-water heat exchanger, a water pump, a bidirectional electronic expansion device, and the majority of the heat pump controls. The heat pump system incorporates several unique technological innovations. The compressor and indoor fan both have electronically commutated, permanent-magnet motor drives. These variable-speed components are used to enhance both comfort and efficiency. The control logic maximizes the efficiency of the heat pump by operating the blower and compressor at the lowest possible speeds at which the conditioned space temperature can be maintained within 0.6°C (1°F) of the thermostat's setpoint. During defrost cycles, energy is removed from the water heater as opposed to extracting energy from the conditioned space.

The heat pump is connected to a 0.197-m^3 (52-gallon), dual 4,500-watt element, electric storage-type water heater. The water heater's thermostats were maintained at 51.7°C (125°F) throughout the study. Water is removed from the water heater by a tee connection on the cold-water supply pipe, circulated through the refrigerant-to-water heat exchanger, and subsequently returned to the storage tank through the port normally used to drain the water heater.

The heat pump can operate in five distinct operating modes: space cooling only, combined space cooling and water heating, space heating only, combined space and water heating, and water heating only. A microprocessor controls the operating mode of the system and the heating elements within the storage-type water heater. A brief description of the logic employed by the heat pump system for each mode follows.

Space Cooling Only The heat pump system provides space cooling in proportion to the space load by varying the compressor and indoor fan speeds. The humidity level within the residence is controlled by varying the indoor fan speed relative to the compressor speed. When the humidity is above the setpoint, the compressor and fan speeds are varied to satisfy the sensible cooling load with the lowest practical indoor coil temperature.

Space Cooling Plus Water Heating In this mode, heat removed from the house is transferred to the water heater and outside air. If the outdoor fan is off, the heat pump operates in a full condensing mode, in which the majority of the heat is delivered to the water heater. Whenever the outdoor fan is operational, the heat pump operates in a desuperheater mode, in which the majority of the heat is dissipated by the outdoor coil and a smaller portion to the water heater. Outdoor fan operation is based on the cooling and water-heating loads. If a substantial cooling load exists, the unit will operate in the desuperheater mode since the space-cooling capacity is not diminished. The heat pump will operate in a fully condensing mode if the space-cooling load is small and a large water-heating load exists. Although the space-cooling capacity is decreased in this mode, the maximum contribution to the water-heating load is obtained. If both the space-cooling and water-heating loads are small, the heat pump will operate in the desuperheating mode.

Space Heating Only The control logic varies the compressor speed to meet the space-heating load while the indoor speed is varied to maintain comfortable supply air temperatures. The maximum compressor speed is 5,300 rpm; however, at temperatures between −8.3°C (17°F) and −17.8°C (0°F) the compressor speed is limited to 3,600 rpm. If the space-heating load exceeds the heat pump's capacity, supplemental resistive heat is added. At temperatures below −17.8°C (0°F), the compressor is turned off and the total space-heating load is met using auxiliary resistive heating.

Space Plus Water Heating The condenser heat is shared by the space- and water-heating loads. The compressor is typically ramped to and maintained at maximum speed until operation in the space- plus water-heating mode is discontinued. The distribution of heat between these two loads depends on the indoor fan speed, which is controlled to meet the space-heating load. At a high fan speed, the majority of heat is supplied to the conditioned space. As the fan speed is lowered, heat delivered to the water heater increases. If the space-heating requirement exceeds that of the compressor, water heating by the heat pump ceases, and the electric resistance heaters provide all needed water heating.

Water Heating Only The integrated heat pump used in this study can heat water even if a space-conditioning load does not exist. The system monitors the water temperature in the bottom of the tank by means of a sensor located in the water heater drain port. If the controller senses that the water temperature is low, it will energize the pump, which circulates water from the water heater through the refrigerant-to-water heat exchanger for three minutes so that a representative water temperature can be obtained. At the end of this sampling period, the compressor and outdoor fan will start if the water temperature is low. The unit will operate in the water-heating mode when the outdoor temperature is between 8.3°C (17°F) and 35°C (95°F). The heating elements within the storage tank will be enabled by control logic within the heat pump system if the water-heating capacity of the heat pump is insufficient to meet the water-heating load. The heat pump will heat water to 54.4°C (130°F) under most conditions, and then the electric elements are energized if the water heater thermostats are set higher.

The Air-Conditioning and Refrigeration Institute (ARI), an independent rating organization, lists the cooling capacity of this heat pump as 10.78 kW (36,800 Btu/h) and the seasonal energy efficiency ratio (SEER) as 14.05. This SEER corresponds to a cooling season COP of 4.12. The heating capacity and heating seasonal performance factor (HSPF) are listed as 10.37 kW (35,400 Btu/h) and 9.05, respectively. This HSPF corresponds to a heating season COP of 2.65. These values are obtained from tests where the heat pump operated in the space-heating-only and space-cooling-only modes. Existing test methods do not capture the additional benefit of integrated water heating. A test method is currently being developed to quantify the benefits of water heating for all integrated space-conditioning/water-heating appliances.

Instrumentation

The integrated variable-speed heat pump in the Hagerstown house was extensively instrumented as shown in Figure 1. All instrumentation was calibrated prior to the beginning of the monitoring period and upon completion of the two-year data-collection period.

Watt/watt-hour transducers were used to measure the energy consumption of the house, water heater, and resistance strip heaters. Each watt/watt-hour transducer was individually calibrated using a watt-hour standard. An electronic digital power analyzer was used to measure the electrical power consumed by the heat pump. The electronic digital power analyzer was used in lieu of a watt/watt-hour transducer due to concerns regarding the accuracy of the watt/watt-hour transducers when subjected to harmonics produced by the variable-speed components (Baldwin 1982; Grady 1991). The electric utility to which the residence is connected installed a watt-hour meter upstream of the electronic digital power analyzer. This permitted an assessment of the error that would be encountered if conventional residential metering equipment was used.

The indoor temperature of the residence was monitored in three locations—the family room, the living room, and the master bedroom—using radiation-shielded thermocouples. The outdoor temperature sensor consisted of a shielded thermocouple located on the north side of the residence. An eight-junction thermopile was used to determine the air temperature difference across the heat pump. Sheathed thermocouples were used to measure the cold-water inlet temperature, the hot-water supply temperature, and temperatures within the storage tank at the center of six equal vertical volumes.

The inlet and outlet water temperatures associated with the refrigerant-to-water heat exchanger were measured using sheathed platinum resistance temperature detectors immersed in the water at the entrance and exit of the heat exchanger.

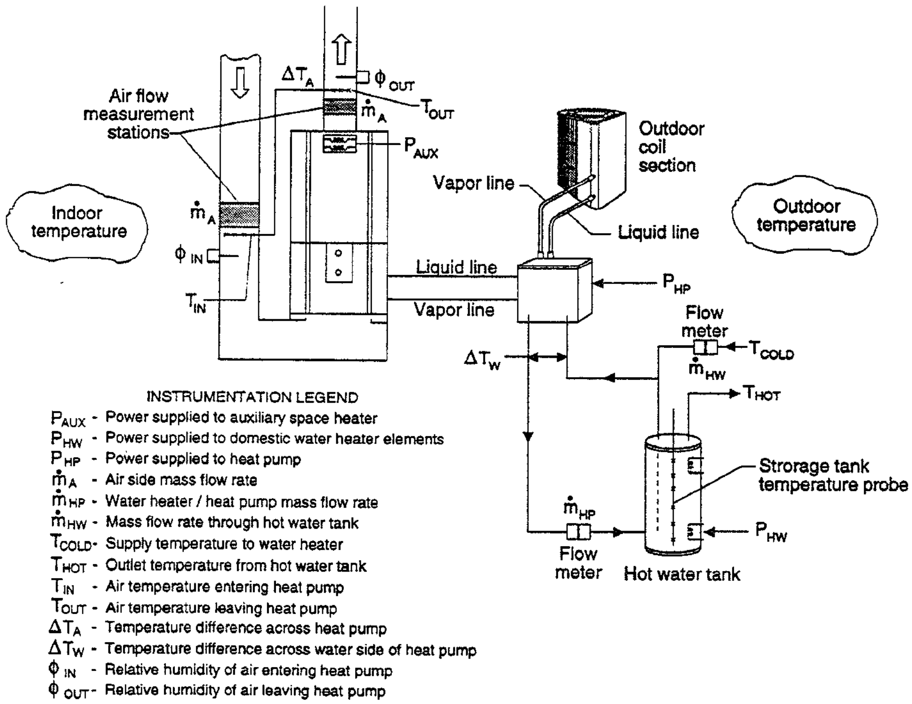

Figure 1 *Instrumentation schematic.*

The thermocouples, thermopile, and platinum resistance temperature detectors were calibrated at the field site by immersing each sensor in a constant-temperature bath containing a calibrated thermometer. The temperature sensors were calibrated over the temperature range to which they were subjected during the monitoring period. The indicated temperatures were compared to the calibrated thermometers, and correction algorithms were developed for the data reduction software.

A turbine flowmeter was used to measure the flow rate through the water side of the refrigerant-to-water heat exchanger. An integrating-type flowmeter measured the quantity of water circulated through the water heater. This flowmeter provides a pulse output with a resolution of 3.8×10^{-5} m^3 (0.01 gallons). An airflow measurement station was used to measure the velocity of the air in the 0.457-m (18-in.) by 0.609-m (24-in.) return duct. This device contains parallel-cell honeycomb panels that straighten the airflow prior to its arrival at the averaging pitot tube station. The averaged total and static pressures are supplied to an electronic differential pressure manometer that incorporates an integral square root extractor to convert the differential pressure into an analog output voltage proportional to the flow rate. This unit was factory calibrated and checked periodically during the two-year period by comparing it to a second reference unit.

The relative humidity entering and leaving the indoor coil of the heat pump was measured using thin-film capacitive humidity sensors. The relative humidity transmitters were calibrated three times during the two-year study: at the beginning, after one year, and at the conclusion of the study. During calibration, the sensors were placed in a small environmental chamber with temperature and humidity control. A calibrated thermocouple and dew-point hygrometer were placed near the relative humidity transmitters within the chamber. Using the measured dew-point and chamber temperatures, the relative humidity within the chamber was determined and compared to that indicated by the relative humidity transducers.

The status of the indoor fan, water circulation pump, defrost valve, and reversing valve was determined by measuring their respective control signals. Logic within the data reduction software was able to discern the operational mode of the heat pump system based on these control signals.

All of the instrumentation used in the study was connected to a microprocessor-based data-acquisition system controlled by a personal computer. The computer converted the sensor signals into engineering quantities, corrected the measured data using calibration data, integrated or averaged the data over appropriate time intervals, displayed the current measurements on a video monitor, and recorded the data. All sensors were scanned every 10 seconds.

EXPERIMENTAL RESULTS

Electrical Energy Consumption

Prior to the installation of the advanced integrated heat pump system, the residence was not monitored. Thus, a direct comparison between the energy required for space conditioning, water heating, and other uses before and after installation of the integrated heat pump system is not possible. It is possible, however, to compare the bimonthly billing data for the entire residence. For the two-year period from October 1986 through September 1988, the total electrical energy consumed by the residence was 116,143 MJ (32,262 kWh). In addition, approximately 10 cords of oak wood were burned during the two heating seasons. Assuming a fuel value of 28,784 MJ (27.3 million Btu) per cord (Gray 1977) and a 40% wood stove efficiency (ASHRAE 1992), an additional 115,116 MJ (31,977 kWh) of electricity would have been consumed if electric resistance heat had been used in lieu of the wood stove, resulting in a total estimated energy consumption of 231,259 MJ (64,239 kWh) for the residence. Space cooling was provided to approximately 30% of the home through the use of two 1.46-kW (5,000-Btu/h) window air conditioners during the cooling seasons.

The energy consumed by the residence from October 1989 through September 1991 was 110,405 MJ (30,668 kWh), representing a 4.9% reduction in electrical energy consumption and an estimated 52% reduction in total energy consumption. The total electrical energy consumed by the residence was reduced by 6.1% for the period in which space heating was required and 1.0% for the months in which space cooling occurred. During this two-year period, the integrated heat pump system was operational, the wood stove was not used, and the entire residence was space conditioned during both the heating and cooling seasons. It should be noted that differences in weather conditions, temperatures within the conditioned spaces, and hot water consumption before and after installation of the integrated heat pump system are not taken into account in this comparison.

A comparison can be made between the total energy consumption of the advanced integrated heat pump and associated water heater during the monitoring period and the energy that would have been consumed if an electric furnace and an identical water heater had been used during the months in which space heating occurred. Months during which both space heating and cooling occurred—September 1990 and May 1991—were not included in this comparison. The measured space-heating load, energy supplied to the water heater, and energy supplied from the heat pump to the water heater were combined to represent the equivalent amount of electrical energy required to space condition the residence with an electric resistance furnace and provide hot water (Table 1). The use of an electric furnace with the same air distribution system and an identical electric water heater

TABLE 1
Space-Heating Load, Energy Transferred from Heat Pump to Water Heater, and Energy Consumption During Space-Heating Months

Year	Month	Space Heating Load		Energy Transferred from Heat Pump to Water Heater		Energy Consumption: Heat Pump		Energy Consumption: Supplemental Heater		Energy Consumption: Water Heater	
		MJ	kWh	MJ	kWh	MJ	kWh	MJ	kWh	MJ	kWh
1990	October	2109.0	585.8	864.0	240.0	1172.5	325.7	10.8	3.0	0.0	0.0
1990	November	5618.9	1560.8	1100.2	305.6	2405.2	668.1	10.1	2.8	4.7	1.3
1990	December	9244.4	2567.9	1128.6	313.5	3573.4	992.6	492.1	136.7	56.2	15.6
1991	January	12254.4	3404.0	1263.6	351.0	4672.1	1297.8	1312.2	364.5	112.0	31.1
1991	February	8767.8	2435.5	1028.2	285.6	3404.5	945.7	403.9	112.2	24.8	6.9
1991	March	6692.0	1858.9	1289.9	358.3	2927.5	812.7	85.3	23.7	7.2	2.0
1991	April	2678.4	744.0	1102.0	306.1	1396.1	387.8	4.0	1.1	9.0	2.5
1991	October	2143.1	595.3	923.4	256.5	1261.8	350.5	0.0	0.0	5.0	1.4
1991	November	5958.0	1655.0	963.4	267.6	2541.2	705.9	74.9	20.8	20.9	5.8
1991	December	8948.2	2485.6	1229.0	341.4	3740.4	1039.0	298.8	83.0	79.2	22.0
1991	January	10747.4	2985.4	1109.5	308.2	4063.7	1128.8	931.7	258.8	146.9	40.8
1991	February	8852.8	2459.1	1099.4	305.4	3548.5	985.7	397.8	110.5	115.2	32.0
1991	March	7456.3	2071.2	1274.0	353.9	3297.6	916.0	79.2	22.0	54.7	15.2
1991	April	2638.1	732.8	1110.6	308.5	1553.4	431.5	3.6	1.0	9.0	2.5
1992	May	515.9	143.3	996.5	276.8	779.4	216.5	0.0	0.0	5.4	1.5
TOTAL		94624.7	26284.6	16482.3	4578.4	40337.3	11204.8	4104.4	1140.1	650.2	180.6

would have resulted in 112,094 MJ (31,137 kWh) of energy being consumed to meet the space- and water-heating loads over the two seasons, assuming that the heating elements within the water heater have an efficiency of 98%. This compares to an actual energy consumption of 45,092 MJ (12,526 kWh) to perform these two functions, or a savings of 67,002 MJ (18,612 kWh), a 60% reduction.

A comparison can also be made to the total energy that would have been consumed if an identical water heater and electric baseboard heat had been used. The entire duct system is located within interior wall partitions and between floor joists. Approximately 70% of the air distribution system is located in the conditioned space, with the remaining portion within the unheated basement. During installation every effort was made to minimize air leakage by sealing each individual seam. The portion of the air distribution system within the basement was insulated to an R-5 level. Thus, for this particular duct system, a thermal efficiency value of 85% is assumed. With this assumption, the space-heating loads using electric baseboard heat would be 15% less than the values given in Table 1. Thus, the use of an identical water heater and electric baseboard heat during months in which space heating was required would have resulted in 97,900 MJ (27,194 kWh) of energy being consumed to meet the space- and water-heating requirements compared to the measured energy consumption of 45,092 MJ (12,526 kWh), a 54% reduction.

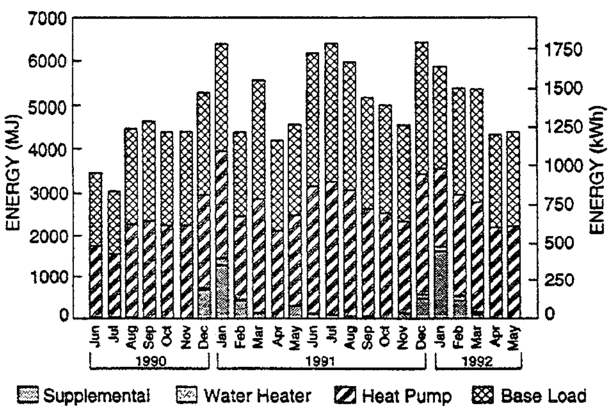

Figure 2 *Monthly energy consumption for base load, heat pump, supplemental space heater, and water heater.*

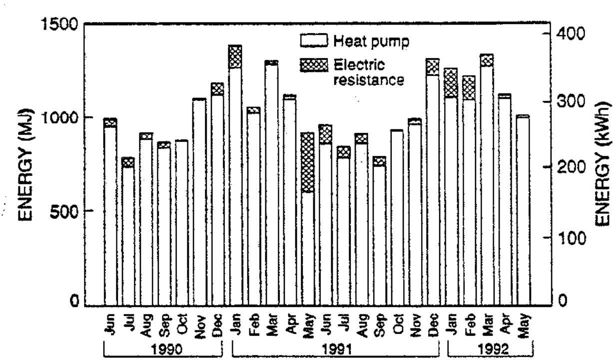

Figure 3 *Energy contributions of heat pump and electrical resistance elements to water heater June 1990-May 1992.*

The energy consumed by the residence, water heater, supplemental heaters, and heat pump is given in columns D, E, F, and G of Tables 2 and 2a on monthly, seasonal, and annual bases. Table 2 uses the SI system of units, whereas Table 2A uses the conventional I-P system. The base load, which includes all miscellaneous uses of electricity within the residence such as lights, range, clothes washer, dryer, refrigerator, and small appliances, may be determined by subtracting the electrical energy consumption of the heat pump, water heater, and supplemental heater from the total energy consumption of the residence. Figure 2 shows the monthly energy consumption for each end use. The heat pump system is the primary end user for the months of November through March. The primary end user of electricity during the remaining months is the base load.

The energy supplied to the resistive heating elements within the water heater represents a small fraction of the total energy consumption, ranging from 0 MJ (0 kWh) in October 1990 to 306 MJ (85 kWh) during the month of May 1991. The value for May 1991 is unrepresentative due to a one-week test during this month in which the heat pump was prevented from heating water. During this interval, an energy balance was performed on the water heater to ensure that the hot water load was being accurately measured. The energy supplied by the heat pump to the water heater and energy consumed by resistive elements within the water heater are shown for each month in Figure 3. During the monitoring period, the water heater represented only 1.2% of the total energy consumed, followed by the supplemental duct heaters (3.6%), the heat pump (44.7%), and the base load (50.5%) (Figure 4).

An objective of this project was to compare the measured energy consumption of the heat pump system (using an electronic digital power analyzer) to measurements made using a conventional residential watt-hour meter. The utility industry is concerned whether the harmonics produced by variable-speed equipment will introduce errors in conven-

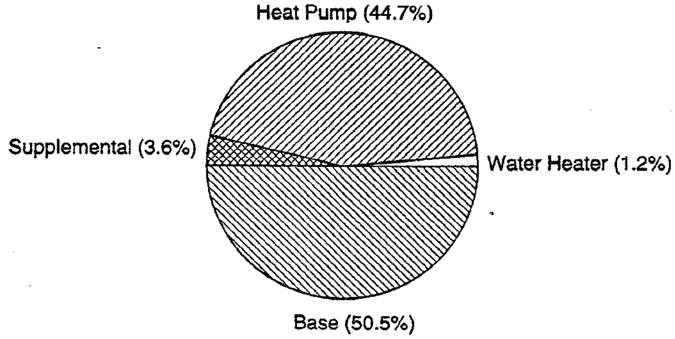

Figure 4 *Distribution of energy consumption.*

TABLE 2
Monthly, Seasonal, and Annual Performance Summary (SI Units)

A	B	C		D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R
							HEAT PUMP ENERGY CONSUMPTION											
MONTH	YEAR	HOUSE THERMAL LOAD (MJ)		TOTAL RESIDENCE ENERGY CONSUMPTION (MJ)	WATER HEATER ELECTRICAL CONSUMPTION (MJ)	SUPPLEMENTAL HEATER ELECTRICAL CONSUMPTION (MJ)	TOTAL (MJ)	WATER HEATING ONLY (MJ)	SPACE HEATING ONLY (MJ)	SPACE AND WATER HEATING (MJ)	SPACE COOLING ONLY (MJ)	SPACE COOLING AND WATER HEATING (MJ)	AVERAGE DAILY HOT WATER CONSUMPTION (m^3*10^3)	OPERATIONAL STATUS OF HEAT PUMP	MONTHLY COOLING COEFFICIENT OF PERFORMANCE (-)	MONTHLY HEATING COEFFICIENT OF PERFORMANCE (-)	WATER HEATING COEFFICIENT OF PERFORMANCE (-)	COMBINED PERFORMANCE FACTOR (-)
		HEATING	COOLING															
JUNE	1990	0.00	2432.88	2815.6	34.9	0.0	1071.0	n/a	n/a	n/a	n/a	n/a	167.7	COOL	3.60	n/a	n/a	2.88
JULY	1990	0.00	3926.88	2770.9	38.9	0.0	1242.0	n/a	n/a	n/a	n/a	n/a	140.8	COOL	3.62	n/a	n/a	3.50
AUGUST	1990	0.00	3599.28	3482.3	25.2	0.0	1228.3	247.0	n/a	0.0	360.7	510.8	150.3	COOL	4.03	n/a	1.91	3.36
SEPTEMBER	1990	281.38	1362.96	3215.9	20.2	0.0	891.0	383.0	40.7	28.1	149.4	168.1	146.1	COOL/HEAT	3.85	1.81	1.84	2.42
OCTOBER	1990	2108.95	0.00	3381.8	0.0	10.8	1172.5	442.4	335.9	254.5	0.0	0.0	127.2	HEAT	n/a	2.75	1.70	2.28
NOVEMBER	1990	5618.88	0.00	4617.0	4.7	10.1	2405.2	562.7	972.4	780.1	0.0	0.0	156.3	HEAT	n/a	3.30	1.39	2.62
DECEMBER	1990	9244.44	0.00	6489.0	56.2	492.1	3573.4	626.8	1711.8	1153.1	0.0	0.0	143.1	HEAT	n/a	2.69	1.24	2.43
JANUARY	1991	12254.4	0.00	8574.1	112.0	1312.2	4672.1	586.1	2423.9	1555.6	0.0	0.0	157.9	HEAT	n/a	2.19	1.11	2.13
FEBRUARY	1991	8767.8	0.00	5814.0	24.8	403.9	3404.5	498.6	1820.9	1013.4	0.0	0.0	142.0	HEAT	n/a	2.69	1.23	2.45
MARCH	1991	6692.04	0.00	5747.0	7.2	85.3	2927.5	679.3	1316.2	837.7	0.0	0.0	171.1	HEAT	n/a	3.01	1.36	2.50
APRIL	1991	2678.4	0.00	3494.2	9.0	4.0	1396.1	559.1	455.4	295.2	0.0	0.0	186.6	HEAT	n/a	3.33	1.71	2.50
MAY	1991	185.76	762.84	3141.7	306.0	0.0	706.3	255.6	27.0	38.5	87.1	114.1	162.4	COOL/HEAT	4.03	0.91	1.85	1.65
JUNE	1991	0.00	3851.64	4529.5	95.4	0.0	1392.1	219.6	0.0	0.0	465.1	616	168.8	COOL	3.58	n/a	2.01	3.07
JULY	1991	0.00	4849.56	4816.1	54.7	0.0	1590.8	145.4	0.0	0.0	633.2	720.4	138.5	COOL	3.68	n/a	2.05	3.31
AUGUST	1991	0.00	4326.84	4508.6	47.9	0.0	1498.0	178.9	0.0	0.0	557.3	673.2	154.4	COOL	3.61	n/a	2.02	3.22
SEPTEMBER	1991	0.00	916.56	3312.0	39.2	0.0	707.4	350.3	0.0	0.0	139.3	112.0	135.9	COOL	2.50	n/a	1.86	1.99
OCTOBER	1991	2143.08	0.00	3760.2	5.0	0.0	1261.8	499.7	425.2	231.8	0.0	0.0	138.9	HEAT	n/a	2.92	1.62	2.21
NOVEMBER	1991	5958	0.00	4858.9	20.9	74.9	2541.2	474.5	1168.6	810.4	0.0	0.0	129.8	HEAT	n/a	2.94	1.33	2.49
DECEMBER	1991	8948.16	0.00	7135.9	79.2	298.8	3740.4	587.5	1827.0	1233.7	0.0	0.0	166.2	HEAT	n/a	2.65	1.26	2.37
JANUARY	1992	10747.44	0.00	7542.7	146.9	931.7	4063.7	472.3	2147.0	1359.0	0.0	0.0	152.6	HEAT	n/a	2.33	1.15	2.24
FEBRUARY	1992	8852.76	0.00	6489.4	115.2	397.8	3548.5	550.4	1719.0	1208.2	0.0	0.0	163.5	HEAT	n/a	2.63	1.21	2.37
MARCH	1992	7456.32	0.00	6037.9	54.7	79.2	3297.6	630.7	1552.3	1028.9	0.0	0.0	165.4	HEAT	n/a	2.87	1.31	2.42
APRIL	1992	2638.08	0.00	3719.9	9.0	3.6	1553.4	616.3	460.1	386.3	0.0	0.0	146.5	HEAT	n/a	2.88	1.55	2.16
MAY	1992	515.88	0.00	2972.5	5.4	0.0	779.4	525.2	65.5	92.5	0.0	0.0	156.3	HEAT	n/a	1.61	1.80	1.55
1990 COOLING SEASON		n/a	11322	10897.6	119.2	0.0	4098.2	424.8	0.0	0.0	510.1	679.0	147.6		3.72	n/a	1.92	3.18
1990-91 HEATING SEASON		47832.05	n/a	41936.8	510.8	2318.4	20244.6	4398.5	9103.7	5955.5	0.0	0.0	153.7		n/a	2.63	1.42	2.34
1991 COOLING SEASON		n/a	14707.44	17875.4	246.2	0.0	5414.4	910.4	0.0	0.0	1881.7	2235.6	147.6		3.52	n/a	1.96	3.06
1991-92 HEATING SEASON		47259.72	n/a	42517.4	436.3	1786.0	20785.7	4356.7	9364.7	6350.8	0.0	0.0	151.4		n/a	2.63	1.40	2.31
ANNUAL 6/90 - 5/91		47832.05	12084.84	53542.8	639.0	2318.4	24568.6	4822.9	9103.7	5955.5	510.1	792.7	151.8		3.73	2.63	1.47	2.48
ANNUAL 6/91 - 5/92		47259.72	13944.6	59684.4	673.6	1786.0	25974.4	5267.2	9365.0	6350.8	1794.6	2121.5	150.3		3.50	2.63	1.50	2.45

(1) Operation in Heating Mode began 9/18/90 at 00:00; Operation in Cooling Mode began 5/27/91 at 13:00.

(2) Cooling Coefficient of Performance is computed for space cooling only operation. Parasitic energy consumption is included.

(3) Heating Coefficient of Performance is computed for space heating only operation. Supplemental and parasitic energy consumption is included.

(4) Water Heating Coefficient of Performance is the ratio of the energy delivered to the water heater divided by the energy consumed while operating in the water heating only mode.

(5) The Combined Performance Factor is the sum of the space conditioning and hot water loads divided by the sum of the energy consumed by the heat pump, supplemental heaters, and hot water heater.

TABLE 2A
Monthly, Seasonal, and Annual Performance Summary (I-P Units)

A MONTH	B YEAR	C HOUSE THERMAL LOAD (KWH)		D TOTAL RESIDENCE ENERGY CONSUMPTION (KWH)	E WATER HEATER ELECTRICAL CONSUMPTION (KWH)	F SUPPLEMENTAL HEATER ELECTRICAL CONSUMPTION (KWH)	HEAT PUMP ENERGY CONSUMPTION						M AVERAGE DAILY HOT WATER CONSUMPTION (GALLONS)	N OPERATIONAL STATUS OF HEAT PUMP	O MONTHLY COOLING COEFFICIENT OF PERFORMANCE (-)	P MONTHLY HEATING COEFFICIENT OF PERFORMANCE (-)	Q WATER HEATING COEFFICIENT OF PERFORMANCE (-)	R COMBINED PERFORMANCE FACTOR (-)
		HEATING	COOLING				G TOTAL (KWH)	H WATER HEATING ONLY (KWH)	I SPACE HEATING ONLY (KWH)	J SPACE AND WATER HEATING (KWH)	K SPACE COOLING ONLY (KWH)	L SPACE COOLING AND WATER HEATING (KWH)						
JUNE	1990	0.00	675.80	782.1	9.7	0.0	297.5	n/a	n/a	n/a	n/a	n/a	44.3	COOL	3.60	n/a	n/a	2.88
JULY	1990	0.00	1090.80	769.7	10.8	0.0	345.0	n/a	n/a	n/a	n/a	n/a	37.2	COOL	3.62	n/a	n/a	3.50
AUGUST	1990	0.00	999.80	967.3	7.0	0.0	341.2	68.6	n/a	0.0	100.2	141.9	39.7	COOL	4.03	n/a	1.91	3.36
SEPTEMBER	1990	78.16	378.60	893.3	5.6	0.0	247.5	106.4	11.3	7.8	41.5	46.7	38.6	COOL/HEAT	3.85	1.81	1.84	2.42
OCTOBER	1990	585.82	0.00	939.4	0.0	3.0	325.7	122.9	93.3	70.7	0.0	0.0	33.6	HEAT	n/a	2.75	1.70	2.28
NOVEMBER	1990	1560.80	0.00	1282.5	1.3	2.8	668.1	156.3	270.1	216.7	0.0	0.0	41.3	HEAT	n/a	3.30	1.39	2.62
DECEMBER	1990	2567.90	0.00	1802.5	15.6	136.7	992.6	174.1	475.5	320.3	0.0	0.0	37.8	HEAT	n/a	2.69	1.24	2.43
JANUARY	1991	3404.00	0.00	2381.7	31.1	364.5	1297.8	162.8	673.3	432.1	0.0	0.0	41.7	HEAT	n/a	2.19	1.11	2.13
FEBRUARY	1991	2435.50	0.00	1615.0	6.9	112.2	945.7	138.5	505.8	281.5	0.0	0.0	37.5	HEAT	n/a	2.69	1.23	2.45
MARCH	1991	1858.90	0.00	1596.4	2.0	23.7	813.2	188.7	365.6	232.7	0.0	0.0	45.2	HEAT	n/a	3.01	1.36	2.50
APRIL	1991	744.00	0.00	970.6	2.5	1.1	387.8	155.3	126.5	82.0	0.0	0.0	49.3	HEAT	n/a	3.33	1.71	2.50
MAY	1991	51.60	211.90	872.7	85.0	0.0	196.2	71.0	7.5	10.7	24.2	31.7	42.9	COOL/HEAT	4.03	0.91	1.85	1.65
JUNE	1991	0.00	1069.90	1258.2	26.5	0.0	386.7	61.0	0.0	0.0	129.2	171.1	44.6	COOL	3.58	n/a	2.01	3.07
JULY	1991	0.00	1347.10	1337.8	15.2	0.0	441.9	40.4	0.0	0.0	175.9	200.1	36.6	COOL	3.68	n/a	2.05	3.31
AUGUST	1991	0.00	1201.90	1252.4	13.3	0.0	416.1	49.7	0.0	0.0	154.8	187.0	40.8	COOL	3.61	n/a	2.02	3.22
SEPTEMBER	1991	0.00	254.60	920.0	10.9	0.0	196.5	97.3	0.0	0.0	38.7	31.1	35.9	COOL	2.50	n/a	1.86	1.99
OCTOBER	1991	595.30	0.00	1044.5	1.4	0.0	350.5	138.8	118.1	64.4	0.0	0.0	36.7	HEAT	n/a	2.92	1.62	2.21
NOVEMBER	1991	1655.00	0.00	1349.7	5.8	20.8	705.9	131.8	324.6	225.1	0.0	0.0	34.3	HEAT	n/a	2.94	1.33	2.49
DECEMBER	1991	2485.60	0.00	1982.2	22.0	83.0	1039.0	163.2	507.5	342.7	0.0	0.0	43.9	HEAT	n/a	2.65	1.26	2.37
JANUARY	1992	2985.40	n/a	2095.2	40.8	258.8	1128.8	131.2	596.4	377.5	0.0	0.0	40.3	HEAT	n/a	2.33	1.15	2.24
FEBRUARY	1992	2459.10	n/a	1802.6	32.0	110.5	985.7	152.9	477.5	335.6	0.0	0.0	43.2	HEAT	n/a	2.63	1.21	2.37
MARCH	1992	2071.20	n/a	1677.2	15.2	22.0	916.0	175.2	431.2	285.8	0.0	0.0	43.7	HEAT	n/a	2.87	1.31	2.42
APRIL	1992	732.80	n/a	1033.3	2.5	1.0	431.5	171.2	127.8	107.3	0.0	0.0	38.7	HEAT	n/a	2.88	1.55	2.16
MAY	1992	143.30	n/a	825.7	1.5	0.0	216.5	145.9	18.2	25.7	0.0	0.0	41.3	HEAT	n/a	1.61	1.80	1.55
1990 COOLING SEASON		n/a	3145.00	3027.1	33.1	0.0	1138.4	118.0	0.0	0.0	141.69	188.6	39.0		3.72	n/a	1.92	3.18
1990-91 HEATING SEASON		13286.68	n/a	11649.1	141.9	644.0	5623.5	1221.8	2528.8	1654.3	0.0	0.0	40.6		n/a	2.63	1.42	2.34
1991 COOLING SEASON		n/a	4085.40	4965.4	68.4	0.0	1504.0	252.9	0.0	0.0	522.7	621.0	39.0		3.52	n/a	1.96	3.06
1991-92 HEATING SEASON		13127.70	n/a	11810.4	121.2	496.1	5773.8	1210.2	2601.3	1764.1	0.0	0.0	40.0		n/a	2.63	1.40	2.31
ANNUAL 6/90 - 5/91		13286.68	3356.90	14873	177.5	644.0	6824.6	1339.7	2528.8	1654.3	141.69	220.2	40.1		3.73	2.63	1.47	2.48
ANNUAL 6/91 - 5/92		13127.70	3873.50	16579	187.1	496.1	7215.1	1463.1	2601.4	1764.1	498.5	589.31	39.7		3.50	2.63	1.50	2.45

(1) Operation in Heating Mode began 9/18/90 at 00:00; Operation in Cooling Mode began 5/27/91 at 13:00.

(2) Cooling Coefficient of Performance is computed for space cooling only operation. Parasitic energy consumption is included.

(3) Heating Coefficient of Performance is computed for space heating only operation. Supplemental and parasitic energy consumption is included.

(4) Water Heating Coefficient of Performance is the ratio of the energy delivered to the water heater divided by the energy consumed while operating in the water heating only mode.

(5) The Combined Performance Factor is the sum of the space conditioning and hot water loads divided by the sum of the energy consumed by the heat pump, supplemental heaters and water heater.

tional watt-hour meters. For the current and voltage ranges used in this experimental investigation, the accuracy of the electronic digital power analyzer is specified as 0.8% of the actual power over a band width of 20 to 100 Hz. The electric utility company calibrated a conventional watt-hour meter in accordance with ANSI Standard C12-1975 and found the meter accurate to within 0.2%.

Table 3 gives the energy consumed by the integrated variable-speed heat pump as measured by the electronic digital power analyzer, the conventional watt-hour meter, and the percent difference. With the exception of four months, there is less than a 0.63% difference, which is within the error bands of the two instruments. The cause of the greater disagreements observed for the months of June 1990, October 1990, May 1991, and May 1992 is not known. Over the entire two-year monitoring period, the energy consumption measured by both instruments was essentially identical—50,662 MJ (14,073 kWh) vs. 50,669 (14,075 kWh). This finding is consistent with studies performed by Baldwin (1982), who found that the average meter error due to distorted current waveforms caused by a single-phase variable-speed AC motor drive was less than 0.8%. Grady (1991), however, has voiced concern over possible metering errors that may result if a significant number of variable-speed motors are connected on the same utility feeder.

Hourly Electrical Demands

The maximum hourly electrical energy demand for the total residence, heat pump, water heater, and supplemental heaters is shown on a monthly basis in Figure 5. Excluding the peak for the residence (which is the sum of the base load, water heater, heat pump, and supplemental heaters), the greatest peak demand was due to the supplemental space heaters—5.30 kJ/s (5.30 kW)—which occurred during January 1991. The peak demand imposed by the heat pump ranged from 1.89 kJ/s (1.89 kW), measured in August 1991, to 3.70 kJ/s (3.70 kW), measured in February 1991. The highest electrical demand recorded for the water heater was 2.97 kJ/s (2.97 kW) during February 1992. The peak demand imposed by the total residence ranged from 5.80 kJ/s (5.80 kW) in June 1990 to a high of 17.00 kJ/s (17.00 kW) in November 1990.

The average hourly energy consumed by the heat pump, supplemental heaters, and water heater is plotted versus outdoor temperature for the 1990-1991 heating season (Figures 6 [SI] and 6a [I-P]) and the 1991 cooling season (Figures 7 [SI] and 7a [I-P]). The graphs were produced by averaging the hourly energy consumption measurements within each two-degree outdoor temperature increment. The heating season includes all hours of data between September 1, 1990, and May 27, 1991, the period during which the homeowner operated the heat pump in the space-heating mode. The homeowner operated the system in the space-cooling mode between May 28, 1991, and September 31, 1991. Each tem-

TABLE 3
Comparison of Measured Energy Consumption Using Electronic Digital Power Analyzer and Electric Utility Watt-Hour Meter

Year	Month	Electronic Power Digital Analyzer		Electric Utility Watt Hour Meter		Percent[1] Difference (%)
		MJ	kWh	MJ	kWh	
1990	June	1071.0	297.5	1057.3	293.7	+1.3
1990	July	1242.0	345.0	1235.5	343.2	+0.5
1990	August	1228.3	341.2	1226.9	340.8	+0.1
1990	September	891.0	247.5	894.2	248.4	-0.4
1990	October	1172.5	325.7	1199.9	333.3	-2.4
1990	November	2405.2	668.1	2397.6	666.0	+0.3
1990	December	3573.4	992.6	3556.8	988.0	+0.5
1991	January	4672.1	1297.8	4673.9	1298.3	-0.0
1991	February	3404.5	945.7	3399.1	944.2	+0.2
1991	March	2925.7	812.7	2927.5	813.2	-0.1
1991	April	1396.1	387.8	1390.7	386.3	+0.4
1991	May	706.3	196.2	712.8	198.0	-0.9
1991	June	1392.1	386.7	1394.6	387.4	-0.2
1991	July	1590.8	441.9	1539.7	442.7	-0.2
1991	August	1498.0	416.1	1497.2	415.9	0.0
1991	September	707.4	196.5	703.1	195.3	+0.6
1991	October	1261.8	350.5	1266.8	351.9	-0.4
1991	November	2541.2	705.9	2542.7	706.3	-0.1
1991	December	3740.4	1039.0	3739.0	1038.6	+0.0
1992	January	4063.7	1128.8	4063.3	1128.7	0.0
1992	February	3548.5	985.7	3545.3	984.8	0.1
1992	March	3297.6	916.0	3297.2	915.9	0.0
1992	April	1553.4	431.5	1563.1	434.2	-0.6
1992	May	779.4	216.5	790.6	219.6	-1.4
Total		50662.4	14072.9	50668.9	14074.7	0.0

[1]Percent Difference = $\frac{\text{Electronic Digital Analyzer - Watthour Meter Measurements}}{\text{Electronic Digital Analyzer Measurement}}$

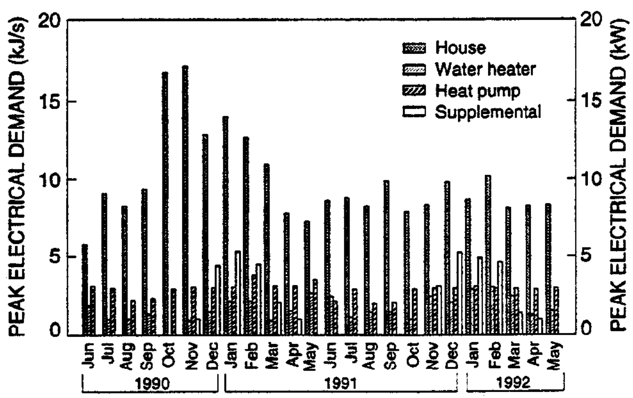

Figure 5 *Monthly peak electrical demands.*

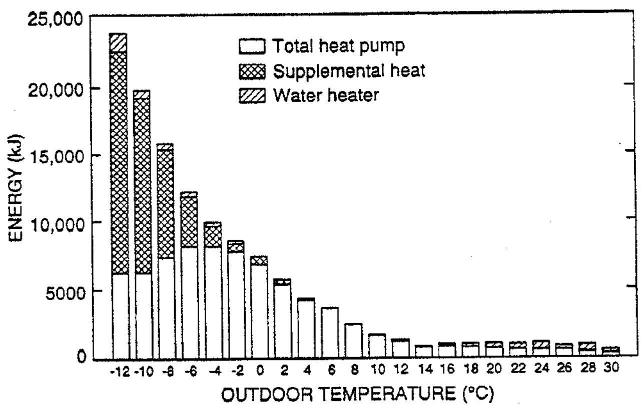

Figure 6 *Average energy consumption of heat pump, supplemental space heater, and water heater vs. outdoor temperature during 1990-91 heating season (SI units).*

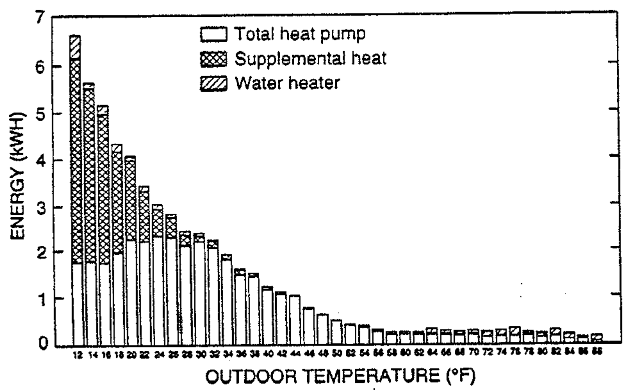

Figure 6a *Average energy consumption of heat pump, supplemental space heater, and water heater vs. outdoor temperature during 1990-91 heating season (I-P units).*

perature entry includes all hours during which the temperature was greater than or equal to the value and less than the temperature entry to its right. For example, the temperature increment labeled 0°C (32°F) in Figure 6 includes all hours of data during which the outdoor temperature was greater than or equal to 0°C (32°F) but less than 2°C (35.6°F).

During the heating season (see Figure 6), at temperatures above 4.0°C (39.2°F) supplemental energy was not required to meet the space-heating load. As the outdoor temperature decreased below 4.0°C (39.2°F), the use of supplemental heat increased, reaching an average hourly value of 15.94 MJ (4.43 kWh) when the outdoor temperature was equal to or above −12.0°C (10.4°F) and less than −10.0°C (14°F). It is interesting to note (Figure 6a) that as the outdoor temperature decreased below −6.7°C (20°F), the average hourly energy consumed by the heat pump decreased and became essentially constant. This is attributed to the heat pump's control logic, which limits the compressor speed to approximately two-thirds its maximum value for outdoor temperatures below −8.3°C (17°F). The accompanying reduction in space-heating capacity resulted in the heat pump operating continuously during each hour in an attempt to meet the space-heating load, with the supplemental heat providing the difference between the space-heating load and the heating capacity of the heat pump.

The difference between the actual temperature at which the hourly energy consumed by the heat pump decreased, −6.7°C (20°F), and the control logic value of −8.3°C (17°F) is attributed to a temperature difference between the outdoor temperature sensor and the sensor utilized by the heat pump's

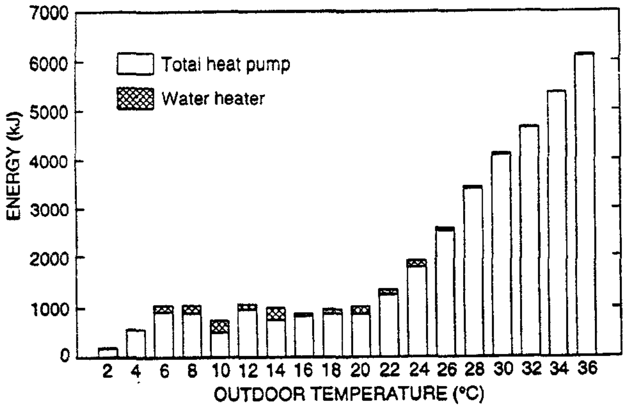

Figure 7 *Average energy consumption of heat pump and water heater vs. outdoor temperature during 1991 cooling season (SI units).*

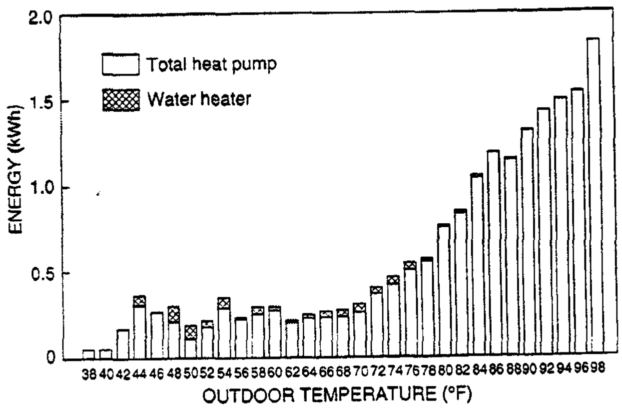

Figure 7a *Average energy consumption of heat pump and water heater vs. outdoor temperature during 1991 cooling season (I-P units).*

control logic. The outdoor sensor is located on the north side of the residence at an elevation of eight feet. The heat pump's sensor is located on the exterior of the heat pump at an elevation of six inches. A short-term experiment was conducted in which two calibrated thermocouples, one positioned at the outdoor temperature sensor and a second one adjacent to the heat pump's outdoor temperature sensor, were used to measure the temperature at these two locations for 10 days. During the experiment the outdoor temperature ranged from −7.8°C (18°F) to 0.6°C (33°F). The average temperature measured adjacent to the heat pump's sensor was found to be 1.3°C (2.4°F) less than that measured by the ambient sensor, which explains the discrepancy in outdoor temperature at which the compressor's speed reduction occurred.

The energy consumed by the resistance elements within the water heater tends to be negligible for outdoor temperatures between 0°C (32°F) and 18°C (64.4°F) during the heating season. At temperatures below this range, the energy consumed by the water heater increased with decreasing outdoor temperature. As the space-heating load increases, the indoor fan speed increases, supplying more heat to the space-heating load and less to the water-heating load. Additionally, whenever supplemental heat is required, the control logic prevents water heating by the heat pump and only the resistance heaters within the water heater are used to meet the water-heating load. Finally, when the outdoor temperature is less than −8.3°C (17°F), the control logic does not permit water heating by the heat pump.

As expected, the average hourly energy consumption of the heat pump increased with outdoor temperature during the cooling season (Figures 7 and 7a). The average energy consumed by the water heater is less than 0.36 MJ (0.1 kWh) for any two-degree temperature increment and in general decreased as the outdoor temperature increased. The greater

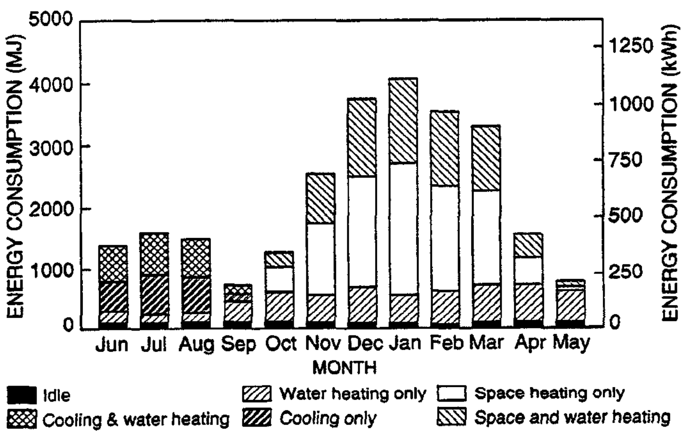

Figure 8 *Monthly energy consumption of heat pump in various operational modes.*

use of energy for heating water at the lower ambient temperatures during the cooling season may be attributed to the fact that the lower outdoor temperatures tend to occur during the early morning hours, which coincide with the greatest demand for hot water within this residence.

Advanced Integrated Heat Pump System Performance

The advanced integrated heat pump system became operational in February 1990, three months before the monitoring effort commenced. After three years of operation, two problems have been associated with the heat pump system. A noisy electrical contactor was replaced during the monitoring period with no loss of data. The water pump used to circulate water from the water heater through the refrigerant-to-water heat exchanger failed after the monitoring was complete.

The performance of the heat pump is summarized in Tables 2 and 2a on monthly, seasonal, and annual bases. The thermal load of the residence is separated into heating and cooling loads (column C). The space-conditioning loads for the two heating seasons were essentially equal, whereas the 1991 space-cooling load was substantially greater than the space-cooling load measured in 1990—14,707 MJ (4085.4 kWh) vs. 11,322 MJ (3145.0 kWh). During September 1990 and May 1991 the heat pump operated in both the space-heating and space-cooling modes. The total energy consumed by the heat pump system (neglecting the supplemental heater) (column G) is divided into energy consumed in the various operational modes in columns H through L. Figure 8 shows the energy consumed by the heat pump, ignoring the supplemental heater, in its various modes for each month from June 1991 to May 1992. The data collection software did not initially permit the breakdown of total heat pump energy consumption into the energy consumed during various modes of operation, and was altered at the end of July 1990 to obtain this additional information. During the 1991 cooling season (May 1 through September 31), 2,235.6 MJ (621.0 kWh) of energy was consumed in the combined space-cooling/water-heating mode compared to an energy consumption level of 1,881.7 MJ (522.7 kWh) while operating in the space-cooling-only mode. When a substantial space-heating load exists (for example, November 1991 through March 1992), the majority of energy is consumed during the times in which the heat pump is operated in the space-heating-only mode (8414 MJ [2337.2 kWh]), followed by space and water heating (5640 MJ [1566.7 kWh]), and finally water heating only (2715.5 MJ [754.3 kWh]). The heat pump consumes the largest amount of energy while operating in the water-heating-only mode during temperate months such as September, October, April, and May. Column N lists the space-conditioning mode in which the heat pump operated during each month.

The monthly cooling COPs (column O) and monthly heating COPs (column P) are computed using data collected when the heat pump was operating in the space-conditioning-only mode. Thus, the energy consumed by the heat pump and space-conditioning loads during time intervals in which the heat pump was operating in the combined or water-heating-only modes is not included in these columns. The total parasitic energy—the energy consumed when the unit is not space conditioning and/or water heating—is included in the calculation of the monthly cooling and heating COPs. Parasitic energy is computed by taking the total heat pump energy (column G) and subtracting the energy consumed for each of the operational modes (columns H through L). During months in which both space heating and cooling took place, the parasitic energy is allocated in proportion to the length of time the heat pump operated in each mode.

The monthly cooling COP is defined as

$$\mathrm{MCCOP} = \frac{Q_{LSCO}}{E_{SCO} + E_{PAR}} \tag{1}$$

where

Q_{LSCO} = monthly space-cooling load when the heat pump provides space cooling only (MJ [MBtu]),

E_{SCO} = monthly energy consumed by the heat pump when the unit operates in the space-cooling-only mode (MJ [kWh]), and

E_{PAR} = monthly energy consumed by the heat pump when the unit is not providing any space conditioning and/or water heating (MJ [kWh]).

The monthly cooling COP ranged from a low value of 2.50 in September 1991 to 4.03 recorded in June 1990. The inclusion of parasitic or "standby" energy has a significant impact on both heating and cooling COPs. This impact is greatest for months in which a low space-conditioning and/

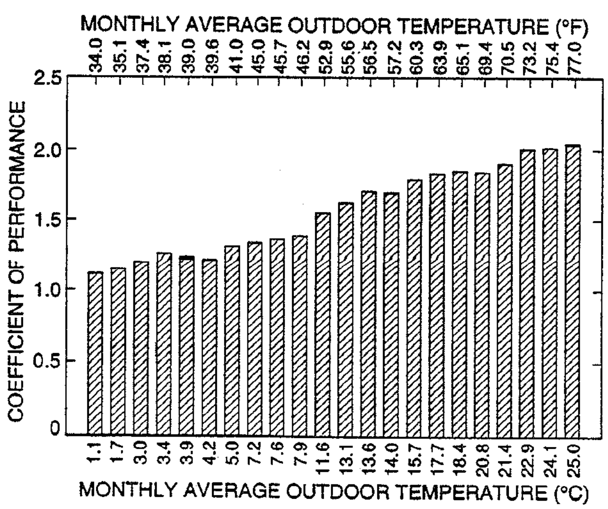

Figure 9 *Monthly water-heating COP vs. average monthly outdoor temperature.*

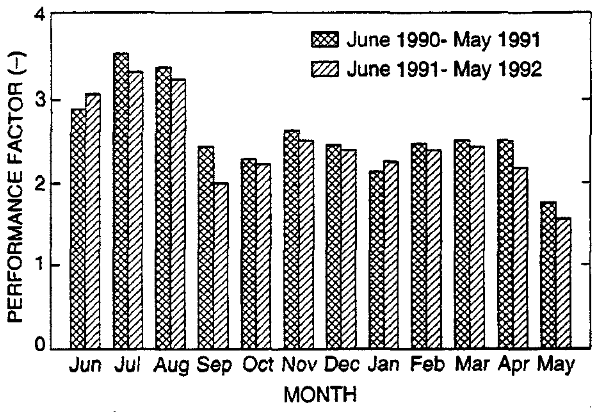

Figure 10 Monthly combined performance factor.

or water-heating load exists. For example, in September 1991 the COP was 2.5 but would have been 4.40 if no parasitic energy was consumed during standby intervals.

The cooling seasonal COP is defined as

$$\mathrm{CSCOP} = \frac{\sum_{i=1}^{n} Q_{LSCO}}{\sum_{i=1}^{n} E_{SCO} + \sum_{i=1}^{n} E_{PAR}} \tag{2}$$

where the monthly quantities Q_{LSCO}, E_{SCO}, and E_{PAR} are summed over the number of months, n, in which space cooling only took place. The cooling seasonal COP for the 1990 season was 3.72 (corresponding to a SEER of 12.69 Btu/Wh), whereas for the second cooling season, the heat pump achieved a cooling seasonal COP of 3.52 (corresponding to a SEER of 12.01 Btu/Wh). Without parasitic energy consumption, the SEER values would have risen to 14.83 Btu/Wh and 14.49 Btu/Wh, respectively. The annual cooling COPs—3.73 for the first year of the study and 3.50 for the second year—are calculated using Equation 2 and summing the variables over each year.

In a similar manner, the monthly heating COP is defined as

$$\mathrm{MHCOP} = \frac{Q_{LSHO}}{E_{SHO} + E_{SUPP} + E_{PAR}} \tag{3}$$

where

Q_{LSHO} = monthly space-heating load when the heat pump provides space heating only (MJ [MBtu]),

E_{SUPP} = monthly energy consumed by the supplemental electric resistance heaters (MJ [kWh]), and

E_{SHO} = monthly energy consumed by the heat pump when the unit operates in the space-heating-only mode (MJ [kWh]).

The monthly heating COP ranged from a low of 0.91 in May 1991 to 3.33 for April 1991. The lower values tended to occur primarily during months in which the energy consumed by the heat pump in the "standby" mode was significant in comparison to the energy consumed by the heat pump while operating in the space-heating-only mode. The higher heating COPs occurred for months in which a significant space-heating load existed, which was met without the assistance of the supplemental resistance heaters.

The heating seasonal COP is computed in an identical manner to that used to compute the monthly heating COP (Equation 3), except the space-heating load, energy consumed by the heat pump while operating in the space-heating-only mode, supplemental energy consumption, and parasitic energy are the seasonal totals, rather than monthly values. The heating seasonal COPs and annual heating COPs were all identical, with a value of 2.63 corresponding to a heating seasonal performance factor of 8.97. Without parasitic energy consumption, the heating seasonal COP would have risen to 2.82 for the 1990-91 heating season and 2.80 for the 1991-92 heating season.

Column Q, the water heater COP, is a performance index based on data collected during times in which the heat pump operates in the water-heating-only mode, i.e.,

$$\mathrm{COP}_{WH} = \frac{Q_{HPWH}}{E_{WHO}} \tag{4}$$

where

Q_{HPWH} = energy delivered by the heat pump to the water heater when the unit operates in the water-heating-only mode (MJ [Btu]) and

E_{WHO} = energy consumed by the heat pump when the unit operates in the water-heating-only mode (MJ [kWh]).

The monthly water-heating COPs are plotted versus average monthly outdoor temperature in Figure 9. As expected, the higher the average outdoor temperature, the greater the water-heating COP, ranging from a value of 1.11 in January 1991 to an upper value of 2.05 for July 1991. The seasonal and annual values were calculated by summing the variables in Equation 4 over the appropriate time intervals. The seasonal water-heating COPs for the 1991 and 1992 cooling seasons are 1.92 and 1.96, compared to 1.42 and 1.40 for the heating seasons. The annual water-heating COPs were 1.47 for June 1990 through May 1991 and 1.50 for June 1991 through May 1992.

The combined performance factor (column R) is an index proposed by Dougherty (n.d.; DOE n.d.) to quantify the performance of combined appliances. This factor represents the ratio of the energy delivered by the heat pump and water heater in the form of space-heating, space-cooling, and/or water-heating loads to the energy consumed by the heat pump, supplemental heaters, and water heater, i.e.,

$$\mathrm{CPF} = \frac{Q_{CL} + Q_{HL} + Q_{HWL}}{E_{HP} + E_{SUPP} + E_{WH}} \qquad (5)$$

where

Q_{CL} = total monthly space-cooling load (MJ [MBtu]),
Q_{HL} = total monthly space-heating load (MJ [MBtu]),
Q_{HWL} = total monthly domestic water-heating load (MJ [MBtu]),
E_{HP} = total monthly energy consumed by the heat pump excluding supplemental energy consumption (MJ [MBtu]),
E_{SUPP} = total monthly energy consumed by the supplemental heaters (MJ [MBtu]), and
E_{WH} = total monthly energy consumed by the water heater (MJ [MBtu]).

Figure 10 shows the combined performance factor for each month, which ranged from a low of 1.55 to a high of 3.5. The combined performance factors are substantially greater during the cooling months as opposed to months when space heating is required (Tables 2 and 2a). This can be partially attributed to the fact that during the space-cooling season, energy removed from the house is transferred to the water heater, the supplemental resistive heaters are not needed, and energy is not removed from the water heater to defrost the outdoor coil.

The cooling season, heating season, and annual combined performance factors (Tables 2 and 2a) are computed in an identical manner to that used to compute the monthly combined performance factors, except the values in Equation 5 are summed over appropriate time intervals. The combined performance factor for the entire 24-month monitoring period is 2.47. Thus, for every unit of energy consumed by the heat pump and water heater, 2.47 units were supplied in the form of domestic water heating, space heating, and/or space cooling. The effect of parasitic energy consumption has a relatively small effect on the combined performance factor. For example, if the heat pump had not consumed any parasitic energy during the second year, the combined performance factor would have increased from 2.45 to 2.51, a 2.4% increase.

SUMMARY

An integrated heat pump/water-heating appliance was extensively monitored for two years. The heat pump system incorporated variable-speed components, microprocessor-based control logic, and a refrigerant-to-water heat exchanger. The heat pump used in this study operates in five distinct modes: space cooling only, space cooling and water heating, space heating only, space and water heating, and water heating only.

The heat pump was instrumented with temperature, relative humidity, watt/watt-hour, airflow rate, and water flow rate transducers. An electronic digital power analyzer was used to monitor the energy consumption of the heat pump (in addition to a conventional residential watt-hour meter) to address concerns regarding the accuracy of conventional utility metering equipment when variable-speed equipment is present within a residence.

The total energy consumed by the residence after installation of the heat pump system was an estimated 52% less than the amount consumed prior to the study by electric baseboard heaters, a wood stove, and window air conditioners. The window air conditioner was used to cool approximately one-third of the residence. A comparison between the electrical energy consumed by the heat pump and the water heater during the monitoring period to the energy that would have been consumed if an electric furnace with the same air distribution systems and an identical water heater had been used during the space-heating months suggests a 60% reduction in total electrical energy consumption.

The electric resistance heaters within the water heater accounted for 1.2% of the total energy consumed by the residence after the integrated heat pump/water-heating appliance was installed, followed by the supplemental resistance heaters (3.6%), heat pump (44.7%), and base load (50.5%). Agreement between the measured energy consumption of the heat pump system using two different instruments—a residential watt-hour meter and an electronic digital power analyzer—over the two-year monitoring period was excellent. Month-to-month variations ranged from −1.4% to 2.4%, with variations for the vast majority of months being within the error bands of the two instruments, 0.6%.

The maximum hourly demand for the residence, heat pump, water heater, and supplemental heaters was recorded for each month. The peak demand of the residence ranged from 5.80 kJ/s (5.80 kW) in June 1990 to a high of 17.00 kJ/s (17.00 kW) in November 1990. The highest peak demands recorded during the study were 3.70 kJ/s (3.70 kW) for the heat pump, 5.30 kJ/s (5.30 kW) for the supplemental resistance heaters, and 2.97 kJ/s (2.97 kW) for the water heater.

The average energy consumed by the heat pump, supplemental heaters, and water heater was computed for 2°C and 2°F increments over the range of outdoor temperatures encountered during the study. During the heating seasons, the use of supplemental heaters was not required to meet the space-heating load when outdoor temperatures exceeded 4.4°C (40°F). As expected, the average hourly energy consumption of the supplemental heaters increased with decreasing outdoor temperature, reaching a value of 15.94 MJ (4.43 kWh) as the outdoor temperature approached −11.1°C (12°F). As the outdoor temperature fell below −6.7°C (20°F), the hourly energy consumed by the heat pump became constant due to control logic, which limits the compressor speed to two-thirds of its maximum value for outdoor temperatures below −8.3°C (17°F), while supplemental energy consumption increased to meet the space-heating load. During the cooling season, the average hourly energy consumption of the heat pump increased, with increasing outdoor temperature reaching a value of 6.59 MJ

(1.83 kWh) for ambient temperatures exceeding 36.7°C (98°F). The average hourly energy consumption values for the water heater during the cooling season were always less than 0.05 MJ (0.05 kWh), independent of the outdoor temperature.

The majority of energy used by the heat pump during the heating season is consumed while providing space heating only, closely followed by energy consumed when operating in the combined space- and water-heating mode. During the cooling seasons, the energy consumed by the heat pump while operating in the space-cooling mode is roughly equivalent to that consumed by the heat pump while operating in the combined space-cooling and water-heating mode. The quantity of energy consumed while operating in the water-heating-only mode tends to be substantially less than that used during operation in the space-conditioning-only or combined modes. The exception is for temperate months such as September, October, April, and May. The monthly parasitic energy was relatively constant, having an average value of 100.8 MJ (28 kWh).

The monthly cooling COP ranged from a low value of 2.50 to 4.03. The inclusion of parasitic or "standby" energy had a significant impact on the monthly cooling efficiency ratio for months in which the space-cooling load and/or water-heating load is small. For example, the lowest value of the monthly cooling COP was 2.5, but it would have been 4.40 if parasitic energy was not consumed during standby intervals. The seasonal cooling COPs were 3.72 (SEER of 12.69 Btu/W) and 3.52 (SEER of 12.01 Btu/W) for the 1990 and 1991 cooling seasons, respectively.

The monthly heating COPs ranged from 0.91 to 3.33. The higher values were recorded during months in which a significant space-heating load existed, which was met without the assistance of supplemental resistance heaters. The lower monthly values occurred during periods in which parasitic energy was a significant portion of the total energy consumed by the heat pump. The heating seasonal COPs were identical for the two heating seasons—2.63, equivalent to an HSPF of 8.97 Btu/Wh.

The monthly water-heating COPs ranged from 1.11 to a high value of 2.05. The values for the cooling seasons were 1.92 for 1990 and 1.96 for 1991, as compared to the values of 1.42 and 1.40 recorded during the 1990-91 and 1991-92 heating seasons. The water-heating COP was found to be proportional to the outdoor temperature.

A performance indicator proposed by Dougherty (n.d.; DOE n.d.) to quantify the overall performance of combined appliances was utilized in this study. This indicator, called a combined performance factor, is the ratio of the space-conditioning and water-heating loads divided by the energy consumed by the heat pump for space heating and cooling, supplemental heaters, and the water heater. A combined performance factor of 2.0 would indicate that for every unit of energy consumed by the heat pump, supplemental heaters, and water heater, two units of energy were delivered by the heat pump and water heater to meet the space-cooling, space-heating, and/or water-heating loads. The monthly combined performance factor ranged from a low of 1.5 to a high of 3.5. The seasonal combined performance factors for the cooling seasons were substantially higher than those obtained during the heating seasons. For example, the 1990 cooling season's combined performance factor was 3.18 compared to a value of 2.34 for the 1990-91 heating season. The combined performance factor for the entire 24-month monitoring period was 2.45. Thus, 2.45 units of energy were delivered for space conditioning (heating and cooling) and water heating for every unit of electricity purchased.

REFERENCES

ASHRAE. 1992. *1992 ASHRAE handbook—HVAC systems and equipment*, p. 30.5. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Baldwin, A.J. 1982. Evaluation of electrical interference to the induction watthour meter. Report No. EL-2315. Palo Alto, CA: Electric Power Research Institute.

DOE. n.d. Department of Energy proposed rule on a uniform method for measuring the energy consumption of combined heat pump-water heating appliances. To be published in the *Federal Register*.

Dougherty, B.P. n.d. Laboratory performance of a combined heat pump-water heating appliance. Report in preparation. Palo Alto, CA: Electric Power Research Institute.

Fanney, A.H. 1993. *Field monitoring of a variable-speed integrated heat pump/water heating appliance*. National Institute of Standards and Technology Building Science Series 171. Gaithersburg, MD: National Institute of Standards and Technology.

Grady, W.M. 1991. Power system harmonics and the potential impact of residential ASD heat pumps. Presentation to the EPRI Advanced Heat Pump Interest Group.

Gray, L. 1977. *The complete book of heating with wood*. Charlotte, VT: Garden Way Publishing.