EXHAUST AIR HEAT PUMP OPERATION AND PERFORMANCE IN FIVE HOMES

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Exhaust air heat pumps (EAHP) are a promising technology that provide ample ventilation while recovering more heat than air-to-air heat exchanger systems. The Bonneville Power Administration funded a Residential Construction Demonstration Project. For this project, five houses located in the Pacific Northwest that are equipped with EAHPs were extensively monitored for one heating season. The EAHPs provided domestic hot water and space heat.

Remote computerized equipment was used to collect and aggregate real time data. Sensors were queried on a 6-second time interval. Data were preprocessed as needed then aggregated to hourly resolution. Data were transferred to a central computer via modems and dedicated phone lines. Over 20 channels were monitored including electric power, temperature, humidity, and flow rates within each home and EAHP. Heat transfer to water and air were computed in real time from flow, temperature, and humidity sensors.

Recovered heat and the coefficient of performance were calculated and control strategies were analyzed for each unit. System efficiency may be increased through changes in control strategies. These changes include coupling the EAHP supplemental space heat condenser thermostat to the main space heat thermostat to prevent superfluous heating and/or excess ventilation during thermostat setback periods and changing the operation strategy when the outdoor temperature drops below the EAHP exhaust air temperature.

INTRODUCTION

Since 1986, Bonneville Power Administration has sponsored the Residential Construction Demonstration Project (RCDP). The Washington State Energy Office operates the RCDP project in the four northwestern states. As subcontractors, energy offices from the three other northwest states--Idaho, Montana, and Oregon--carry out planning and implementation aspects of the project. The project's main purpose is demonstration of energy conserving innovations, but it also contains strong research components. Part of this research involved field monitoring of five houses with exhaust air heat pump ventilation systems (EAHP) located in Montana and Washington. These houses are a subsample within the larger RCDP population of 165 new houses that had both heat recovery ventilators and state-of-the-art envelope construction.

This paper describes each house's physical characteristics, occupant characteristics, and EAHP installation. As configured, the EAHP serves three system functions: water heating, space heating, and ventilation. EAHP system efficiency is effected through system configuration, installation, control design, and occupant operation of controls. EAHP systems are evaluated for system performance and overall efficiency. Ventilation rates achieved by the systems are compared to expected ventilation rates. Energy savings due to heat recovery are evaluated.

Operational parameters examined include the effect of nighttime thermostat setback on energy consumption. Nighttime thermostat setback contributes substantial energy savings. However, system efficiency may be improved through a modified control scheme. Coupling the supplemental space heat condenser thermostat to the main thermostat should prevent over-ventilation during periods of nighttime thermostat setback.

Two types of EAHP systems are monitored. The first type, which is installed in three houses in Washington State, provides space and water heat and extracts energy from the exhaust air stream only. This type of heat pump achieved overall coefficients of performance (COP) between 2.5 and 3.1, and ventilated the homes well above the required 0.25 mechanical air changes per hour during the heating season. COP is a ratio of the energy delivered to water and space heat and the energy required to operate the system. Mean energy recovered from exhaust air by these three heat pumps ranged from 3,183 to 4,731 Btu per hour of operation for the entire monitoring period.

The second type of EAHP, which is installed in two Montana homes, extracted energy from both exhaust and supply air streams. This type of heat pump achieved considerably lower COPs (1.7 and 1.8). The systems are operated less often, recover less heat per hour of operation (1,384 and 1,744 Btu/hr), and do not provide the recommended air change rate. The manufacturer has since discontinued production of this model.

Ventilation operating patterns coincided well with seasonal periods of space heat demand for all types of systems. Wintertime mechanical air change rates fall between 0.09 and 0.56 air changes per hour. The highest readings are from three EAHP units in western Washington, while the lowest readings are from two EAHP units in Montana. The three Washington EAHPs operated almost continuously during the heating season, with less frequent use during the summer months.

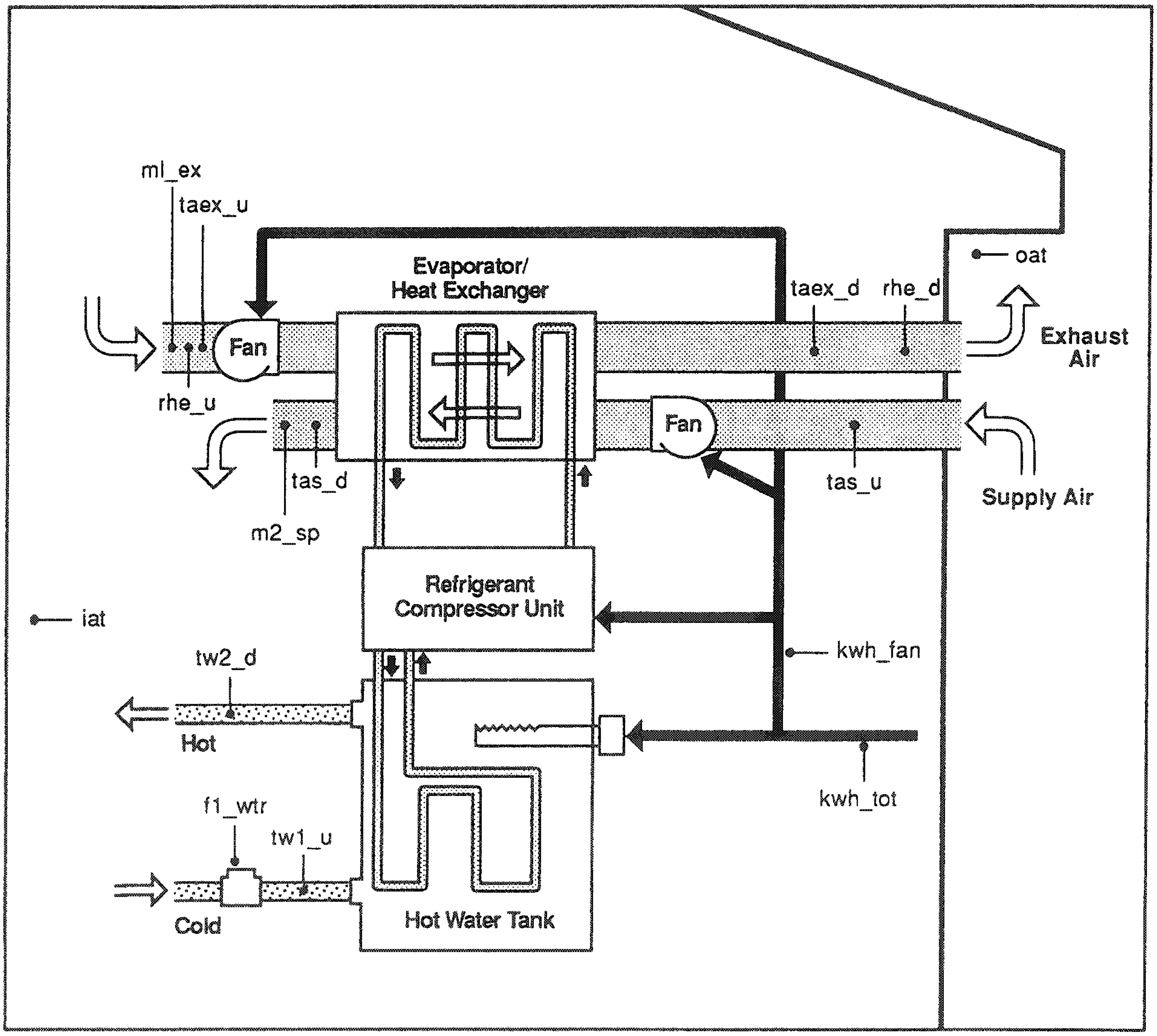
METHODOLOGY

Description of Exhaust Air Heat Pump Systems

Three houses in Washington and two houses in Montana are equipped with EAHP systems. Two different systems are used. All units prioritize water heating above space heating. If neither water nor space heat are required, a timer can be set to operate the exhaust fan periodically for internals as short as 15 minutes for ventilation only. This feature can be overridden by the occupant. Schematics of these systems are contained in Figures 1 and 2.

The Montana units are balanced ventilation systems installed in a utility closet within the heated space. The evaporator is connected to the compressor/tank unit by refrigerant lines. The evaporator has both a supply air and exhaust air passage. Supply air and refrigerant flow parallel to each other and counter to exhaust flow. There is a summer option that changes the flow to precool incoming supply air. The flows of this model are complex. The water tank is heated by a condenser jacket that encircles the tank, contacting the metal surface beneath the insulation. A backup resistance element actuates to heat the upper region of the tank if high water usage drops temperature below a set minimum. Supplemental space heating is provided by the condenser. In summer operation mode, the unit extracts heat from both supply and exhaust air streams for domestic hot water heating and air conditioning. These heat pumps achieved considerably lower efficiencies, were operated less often, and did not provide the recommended air change rate. The manufacturer has since discontinued production of this model.

The Washington units are unbalanced systems installed in a closet within the heated space unless otherwise noted in the house descriptions. Makeup air is provided by two through-the-wall vents on each floor. The unit recovers heat from exhaust air and alternately provides heat to an integral 72-gallon domestic hot water (DHW) tank and a

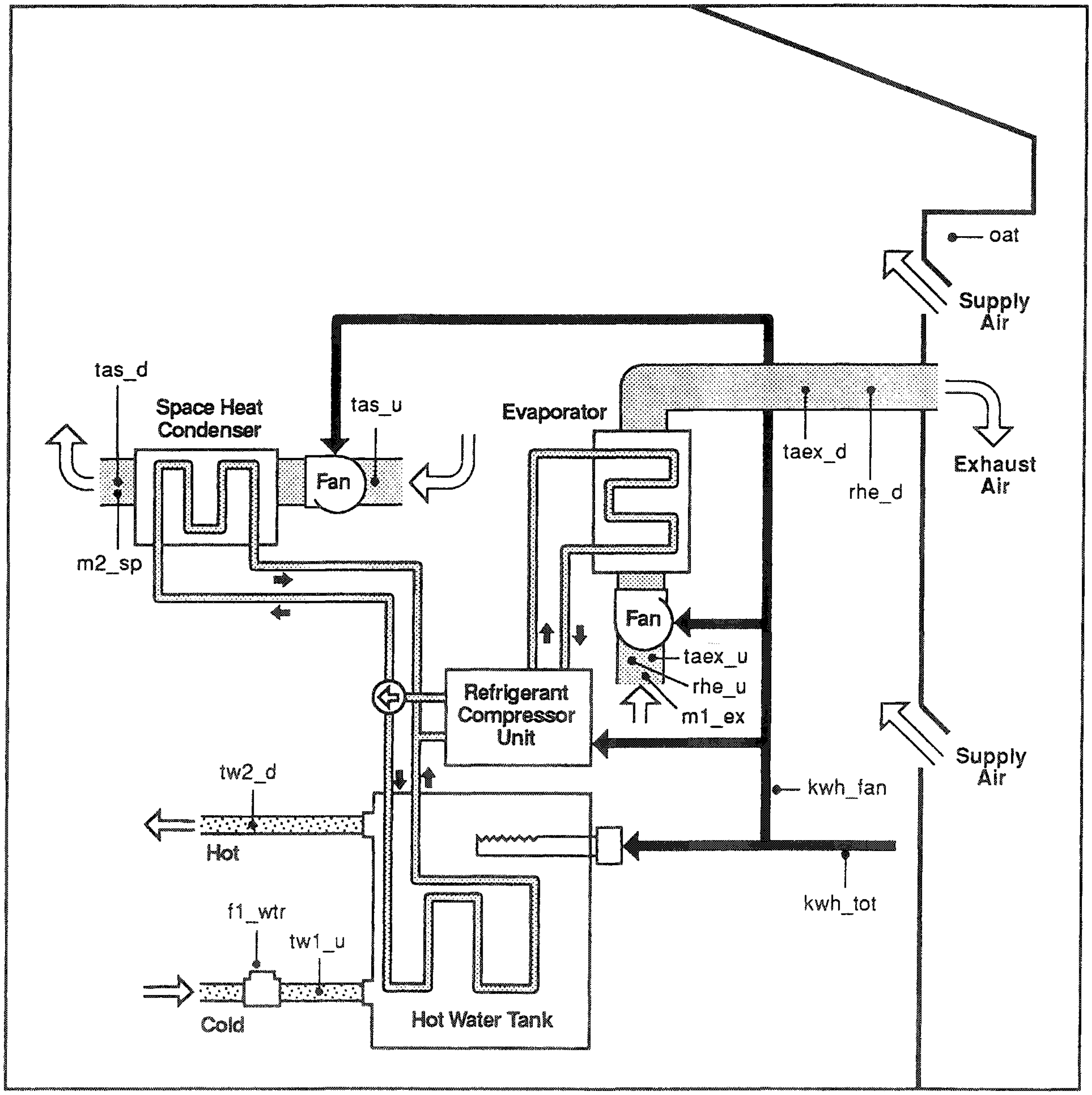


KEY

- Ventilation Ducting
- Refrigerant
- Electric Power
- Water

m1_ex	Exhaust air mass flow in 1bm/hr
m2_sp	Supply air mass flow in 1bm/hr
rhe_u	Relative humidity upstream
rhe_d	Relative humidity downstream
iat	Indoor air temperature in degrees Fahrenheit
oat	Outdoor air temperature in degrees Fahrenheit
taex_u	Exhaust air temperature upstream in degrees Fahrenheit
taex_d	Exhaust air temperature downstream in degrees Fahrenheit
tas_u	Supply air temperature upstream in degrees Fahrenheit
tas_d	Supply air temperature downstream in degrees Fahrenheit
tw1_u	Domestic hot water temperature upstream in degrees Fahrenheit
tw2_d	Domestic hot water temperature downstream in degrees Fahrenheit
f1_wtr	Domestic hot water mass flow in 1 bm/hr
kwh_fan	Compressor and fan energy consumption in kW/hr
kwh_tot	Total unit energy consumption in kW/hr

Figure 1. Montana EAHP Systems



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kwh_fan	Compressor and fan energy consumption in kW/hr
kwh_tot	Total unit energy consumption in kW/hr

Figure 2. Washington EAHP Systems

remote supplemental space heat condenser. The supplemental space heat condenser is connected to the compressor/tank unit by refrigerant lines. The water tank is heated by a condenser jacket that encircles the tank, contacting the metal surface beneath the insulation. A backup resistance element actuates to heat the upper region of the tank if high water usage drops the temperature below a set minimum.

Descriptions of Houses and Occupants

Site 228. Site 228 is a 2,356 ft^2, split-entry house with two stories, including the heated daylight basement. It is located in Helena, Montana at an altitude of 3,500 feet. Over the 12 months ending April 30, 1988, 7,422 degree-days (65°F base) were recorded at the site. Primary space heating is by wall mounted electric resistance baseboard heaters. Zone thermostats are distributed throughout the house. There is no automatic night setback system, but occupants reported a daily routine for manually setting the various thermostats. This house was monitored for 14 months from February 1987 to April 1988. The house was initially occupied by a family of two adults, a mother and daughter. The daughter moved out in May of 1987. The occupant (mother) works a normal shift week. The occupant considers the home comfortably ventilated except when guests smoke. The occupant indicated that there are instances of inadequate hot water when both she and her daughter are living at home. There is a strong passive solar contribution in the winter months. The occupant reported that this sometimes makes the living/dining area too warm. Smoke testing revealed a large amount of crossover leakage from the exhaust side into the supply side of the evaporator/heat exchanger component. There is also considerable leakage out of the component cabinet and at the duct connections.

Site 235. Site 235 is a 1,821 ft^2, two-story house with a vented crawl space. It is located in Helena, Montana at an altitude of 3,850 feet. Primary space heating is by wall-mounted electric resistance baseboard heaters. Zone thermostats are distributed throughout the house and there is no automatic night setback system. This house was monitored for 9 months from December 1986 to September 1987.

Site 439. Site 439 is a 2,798 ft^2, two-story house with a vented crawl space. It is located in Yacolt, Washington at an altitude of 720 feet. Over the 12 months ending April 30, 1988, 4,957 degree-days (65°F base) were recorded at the site. Primary space heating is by a conventional electric forced air furnace. Blower door technicians ranked the furnace ducting as the second largest leak (to outdoors) in the house. Additional space heating is provided by the supplemental space heat condenser coil of the exhaust air heat pump. This house was monitored for 14 months from February 1987 to April 1988. The home is occupied by a family of two adults and three school age children. Usually, all family members are at home during the evenings. One adult works days, five days per week, and the other works outside the home intermittently (substitute teaching). The occupants understood their heating and ventilation systems and generally operated and maintained them according to manufacturers' recommendations. Nighttime setback of the main thermostat was regularly practiced. Mass flow data for the first several months of operation are not recoverable from this site. During that time, the sensor had come loose and true flow data are not estimable from monitored data. Nine hundred thirty four hours of space heat data or 8.8 percent of all space heat data on Site 439 are excluded due to 60 Hz power line interference.

Site 449. Site 449 is a 1,967 ft^2, two-story house with a vented crawl space. It is located in Bothell, Washington at an altitude of 500 feet. Over the 12 months ending April 30, 1988, 4,639 degree-days (65°F base) were recorded at the site. Primary space heating is by a ceiling electric resistance radiant system. Zone thermostats are distributed throughout the house and there is no automatic night setback system. This house was monitored for 14 months from February 1987 to April 1988.

The house is occupied by two adults with no children living at home. The occupants travel frequently and the house is often vacant for weeks at a time. They did not appear to have a good understanding of their heating and ventilating systems. At the time of monitoring equipment installation, the EAHP space heating fan/coil thermostat was found to be set (incorrectly) lower than the space heating thermostat in the same zone. The occupants reported that their EAHP operated quietly and that they never ran short of hot water. Initially, there was a perception of inadequate ventilation in the family room. This is reasonable because there are no EAHP exhaust or space heating registers in the room and there were no slot vents installed. Slot vents were placed in the family room when they were installed throughout the house in April 1988. The water tank thermostat is set for 130°F. This is within the unit's operating capability, but the manufacturer recommends 120°F because the vapor cycle circuit achieves a better coefficient of performance (COP) at lower water temperature.

There are aspects of the EAHP installation that degrade system efficiency including: long runs of supply ducting from the supplemental space fan/coil routed through the vented crawl space; exhaust ducting routed through unheated attic space or outside walls; large area uninsulated exhaust inlet plenum in the unheated garage; and noticeable leaks in the EAHP casing with air escaping from the space between the fan and the evaporator. The latter may cause a significant energy loss because the unit is outside the conditioned space.

Site 450. Site 450 is a 2,183 ft^2, two-story house with a vented crawl space. It is located in Olympia, Washington at an altitude of 100 feet. Construction is timber framing with 6 3/8" stress-skin wall panels and 8 3/8" stress-skin vaulted roof panels. The monitoring period did not span an entire heating season. Over the 8 months from January 17, 1987 through September 17, 1987, 2,517 degree-days (65°F base) were recorded at the site. Primary space heating is by wall mounted electric resistance fan coil heaters. Zone thermostats are distributed throughout the house and there is no automatic night setback system. The house is occupied by a family of two adults with two young children (one of which was an infant during the monitoring period). One adult worked full time on a normal shift; the other did not work outside the home. The occupants understood their heating and ventilation systems and generally operated and maintained them according to manufacturers' recommendations.

Data Acquisition System Overview

There are two levels of field thermal performance data acquisition in the RCDP project. All the RCDP houses are instrumented to record thermal performance with weekly or more resolution. This five-house subsample represents a higher time resolution experiment with more data points per house. The monitoring system for these homes aggregates data from 21 channels every hour. Data acquisition system design, installation, and data retrieval and archiving are performed by a subcontractor.

All equipment was purchased "off the shelf." Data loggers were originally developed for use in remote unmanned weather stations. They are packaged in a tamper-proof box along with a modem connected to a dedicated phone line. Programming is contained in volatile memory. The program may be reloaded or modified remotely. An internal battery retains the program and data for several hours in the event of a power interruption. Recorded channels include indoor and outdoor ambient temperatures as well as all parameters necessary to perform an energy balance for the heat recovery ventilators. All points are queried every 6 seconds and aggregated to hourly resolution and running averages. At least once per week, the units are interrogated by a central computer and reset. Hourly data are then archived. The monitoring protocol is described in detail by Douglass (1989).

Air flow rates are determined with hot tip air velocity transducers. These sensors give an output proportional to the velocity of air at standard temperature and pressure. The transducer output is converted directly to mass flow rate by an algorithm in the data logger. Flow monitoring transducers are installed toward the downstream end of the longest straight ducting run available. To ensure accuracy, a duct traverse was done on installation to find a position for the transducer probe where the measured velocity accurately represents mass flow.

Humidity is monitored both upstream and downstream of the heat recovery units in the exhaust stream. These measurements allow latent heat to be calculated. Bulk resistance type relative humidity sensors are used. Accuracy is within 4 percent between 30 percent and 90 percent relative humidity. From 90 percent to 99.9 percent relative humidity, 5 percent accuracy applies. Air and water temperatures are measured with RTD-type sensors to within 0.5°F (0.3°C). Electrical energy consumption is measured with power transducers. These sense voltage and current and give true power, not volt amperes to the data logger. Accuracy is within the greater of 15 W or 3 percent.

Hot water flow rate is sensed with a pulse initiating nutating (for example, wobble plate) meter. Accuracy is within 3 percent at flows above 0.2 gpm (13 mL/s), but problems occur when there is no flow as discussed below. Heat transfer parameters are derived by the data logger in real time (6-second intervals) from other parameters listed above. BTU sensors are not used.

Data Quality and Monitoring Problems

Data quality for this set of homes is very high. However, three sensors--hot water flow, space heat energy consumption, and exhaust ventilation mass flow--exhibit systematic quality problems. In the case of exhaust ventilation mass flow, recorded data are lower than actual flow due to fouling of the mass flow sensor. Systematic errors in hot water flow and space heat energy consumption data are more sporadic in nature.

Exhaust Ventilation Mass Flow Errors

Airborne aerosols, dust, and cooking grease tended to accumulate on the hot tip mass flow sensors and reduce readings. The rate of degradation varied considerably depending upon factors including system on time and amount of airborne contaminants. Contamination reduced the measured air flow by as much as one half and was seen exclusively in the exhaust flow sensors as they are exposed to many more contaminants than fresh supply air flow sensors. Since much of the proposed analysis is dependent on mass flow data and measured flow rates correspond to actual flow rates, a procedure was created to correct for contamination of exhaust mass flow sensors.

All sensors were cleaned and verified as accurate by duct traverses at the time of installation, at least once during the monitoring period, and again for calibration at the time of removal. Accordingly, the periods immediately following cleanings are treated as benchmark readings. The algorithm contained in Equation 1 was created to adjust monitored data to estimated actual data. This algorithm depends upon beginning and ending benchmark readings, daily maximum flow, and system on time. On those days when the daily maximum flow is less than 85 percent of the previous day's maximum reading (for example, the system did not operate for a continuous hour), daily maximum flow is assumed to be equal to the previous day's reading. The first part of the algorithm adjusts each measurement as a ratio of the initial maximum flow to the daily maximum flow. The second part of the algorithm adjusts the new flow data as a ratio of the final maximum flow to initial maximum flow multiplied by a percent of total system on-time. For the purpose of this analysis, exhaust fan energy consumption is considered analogous to system on time.

$$M_a = (M_n*(M_I/M_{max}))*(1-((1-(M_f/M_I))*((T_n-T_I)/(T_f-T_I))) \quad (1)$$

where M_a = Adjusted mass flow lbm/hour

M_n = Monitored Mass flow for Observation n in lbm/hour

M_{max} = Daily maximum monitored flow in lbm/hour

M_f = Final Mass flow in lbm/hour

M_I = Initial Mass flow in lbm/hour

T_n = System On Time for Observation n in hours

T_f = Final On Time in hours

Heat transfer data are generated in real time from the product of air mass flow and enthalpy difference between upstream and downstream points in the EAHP evaporator ducting. These heat transfer data are corrected as a ratio of the adjusted and measured mass flow.

While the algorithm corrects for sensor contamination, potential errors in exhaust mass flow readings are much greater than random errors generated by the specified inaccuracy of sensing equipment. These errors are propagated with the calculation of heat transfer rates. Moreover, these errors are difficult to quantify.

Space Heat Energy Consumption Errors. Space heat data contain outlying high values for Site 439 that occur during discrete blocks of time. These values are typically two to four times the capacity of the heating system and corresponded to exactly 60 pulses per second. Thus the sensor or data logger had periodically picked up interference from 60 Hz power line hum. When the power transducer sensors were replaced, and the problem was corrected.

Unfortunately, there is no way to determine if all readings below the furnace capacity ceiling are good. They could have been influenced by the power line interference for part of the hour, but not enough to exceed the furnace capacity ceiling. The extreme readings occur in clusters within single days and groups of days. Other groups of days seem to be free of the high readings. Upon this basis, entire blocks of data are excluded from the analysis if some data within the block are suspected.

Domestic Hot Water Flow Errors. Hot water flow data are of the lowest quality. Water meters utilize a nutating disk that turns a rotor with a Hall effect (magnetic) switch. This generates electrical pulses that are counted and scaled into gallon units by the data logger. When the rotor randomly stops near the position that triggers a pulse, the sensor can emit electronic noise. Flow rates as high as 6,000 gallons per hour are recorded. Spurious data are dispersed randomly and frequently throughout the data set. While extremely high readings are obviously incorrect, there is no direct way to validate moderate level readings. Effort was made to remove extreme values, but confidence in the remaining domestic hot water flow data is very low. Consequently, hot water flow data are not used in this analysis.

RESULTS

Examination of System Ventilation Performance

Program design standards require 0.25 mechanical ACH. Table 1 summarizes ventilation system capacity, annual mean air change rate, and heating season air change rate in air changes per hour (ACH). These air change figures are for mechanical ventilation only. Air change rate figures are averaged over all hours regardless of system operation. All of the systems had enough capacity to meet programmatic ventilation requirements. However, the two Montana systems (Sites 228 and 235) would have had to operate almost continuously to meet programmatic requirements and were not operated frequently enough to satisfy those requirements. The Washington homes were over-ventilated during the heating season. This over-ventilation is partially due to insufficient control strategies and is examined in the evaluation system heating performance below.

Examination of System Heating Performance

All heat recovered is used for space or domestic water heat. The total amount of heat recovered by the EAHP units is calculated based on the temperature and relative humidity differences and mass flow rate across the EAHP. (Both sensible and latent heat recovery are accounted.) Methodologies used to calculate total heat recovered are taken from ASHRAE (1989).

Table 2 summarizes heat recovered, ventilation heat loss, net heat recovered, and the ratio of the mean heat recovered heat per mean exhaust air flow. Note that Sites 235 and 450 were not monitored for an entire heating season. Most of the hours of observation for these two houses occurred during non-heating months. Only those hours when the systems were operated (exhaust flow greater than zero) are compiled in Table 2. Inclusion of those hours with no exhaust air flow will reduce mean heat recovered statistics.

Table 1. Mechanical Ventilation Air Changes Per Hour

Site	Exhaust Flow Capacity ACH	Annual Mean Air Changes ACH	Annual Standard Deviation ACH	Heating Season Mean Air Changes ACH	Heating Season Standard Deviation ACH
228	0.26	0.12	0.10	0.14	0.10
235	0.27	0.09	0.09	0.09	0.07
439	0.57	0.35	0.22	0.46	0.12
449	0.73	0.42	0.29	0.56	0.18
450	0.60	0.23	0.26	0.55	0.10

Table 2. EAHP Heat Recovered Heat Entire Monitoring Period

Site	# of Hours	Mean Heat Recovered Btu/hr	Ventilation Heat Loss Btu/hr	Net Heat Recovered Btu/hr	Heat Recovered per lb_m Flow Btu/lbm
228	6,442	1,744	2,897	-1,153	5.3
235	4,022	1,348	770	578	8.1
439	7,381	4,731	3,677	1,054	6.1
449	8,329	3,183	3,351	-168	5.5
450	2,728	3,710	3,390	320	5.7

Recovered heat only explains part of EAHP system performance. Other factors that impact system performance are ventilation heat loss and operating energy consumption. Ventilation heat loss is a function of indoor air temperature, outdoor air temperature, heat capacity of air, and air flow rate. Mechanical ventilation heat loss is calculated with Equation 2. Net heat recovered is the difference between heat recovered and ventilation heat loss. Where net recovered heat is negative, more heat is needed to warm makeup air than is extracted from exhaust air. Ventilation heat loss in Table 2 is equivalent to the heat loss that would be experienced by a non-heat recovery ventilation system operating at the same exhaust rate.

$$Q_m = MCp(T_i - T_o) \qquad (2)$$

where Q_m = Heat needed for make-up air due to mechanical ventilation in Btu/hr

M = Mass flow rate due to mechanical ventilation in lbm/h

Cp = Mass heat capacity of air in Btu/lbm°F

T_i = Interior temperature in °F

T_o = Incoming air temperature in °F

COP Calculation. The heating coefficient of performance (COP) is calculated with Equation 3. This figure is the ratio of the recovered energy to the energy consumption by the compressor and fan. Note that the numerator contains both the energy recovered from the ventilation air stream and the energy utilized by the EAHP because all waste heat from the EAHP will also be recovered.

$$COP = (Q_r + Q_f)/Q_f \qquad (3)$$

where COP = the heating system coefficient of performance

Q_r = the energy recovered by the EAHP

Q_f = the energy consumed by the EAHP for compressor and fan operation

The COP does not include energy required to reheat makeup air. The ventilation standard to which these houses are designed requires mechanical ventilation of 0.25 air changes per hour. Any ventilation above this rate is considered excess and the associated heat loss reduces the COP. The mean adjusted heating coefficient of performance (ACOP) is calculated by substituting net recovered energy into Equation 3. See Table 3.

No attempt is made to compare water heating COP to space heating COP. This would have required invasive tank instrumentation beyond the scope of this study. Moreover, this question has already been explored in lab testing conducted by a national laboratory (Wallman, Fisk, and Grimsrud 1988). This research noted significantly higher COP for space heating than for water heating. This is due to the lower temperature required for space heating. Revision of controls to give priority to space heating was suggested. The manufacturer has since made such modifications. Units with redesigned controls are currently being monitored in Cycle 2, RCDP. Data acquisition, similar to that reported herein, was completed at the end of April 1990.

The magnitude of net heat recovered is a function of exhaust air and outdoor temperature difference. If the outdoor temperature is less than the EAHP exhaust temperature, more heat is required to heat makeup air than is extracted from exhaust air. Figure 3 plots the difference between outdoor and exhaust air temperatures versus the total amount of heat recovered for each of the EAHP homes. As the outdoor temperature decreases, net recovered heat also decreases. Energy savings could be increased if the EAHP is operated only when the exhaust air temperature is less than the exterior air temperature.

Effect of Nighttime Thermostat Setback. Nighttime thermostat setback was practiced by occupants in two of the houses (Sites 228 and 439). This is the most significant occupant behavior in these houses. Space heat thermostat is usually turned down between 10:00 p.m. and 11:00 p.m. and turned up between 5:00 a.m. and 8:00 a.m. Reheat energy consumption is particularly difficult to infer because the time duration of reheat is confounded by varying temperature differentials. Worse yet, space heat energy consumption tapers off gradually, before the apparent set point is reached, as if the thermostat is significantly affected by self heating from it's anticipator. High shell insulation levels also create gradual temperature decay.

An envelope heat loss approach was used to evaluate the effects of nighttime thermostat setback. This required information on overall heat loss coefficient and internal gains and solar heat gains. A set of whole days with good space heat data and no temperature differentials less than 10° F was selected. A regression of total space heating (supplemental space heat condenser plus furnace energy consumption) against temperature differential was performed for these days. Calculated heat loss coefficients and internal gains are included in Table 4.

$$Q = UA\,(T_I - T_O) - IG \qquad (4)$$

where Q = Total space heat required in kW
UA = Heat transfer coefficient in kW/° F
IG = Internal and solar heat gains in kW
T_I = indoor temperature in ° F
T_O = outdoor temperature in ° F

Regression values for the heat loss coefficient and internal gains, and actual air temperature differential are substituted into Equation 4 to derive space heat energy consumption. Table 4 contains the resulting hourly space heat values that are summed over the calendar year ending on April 30,

Table 3. EAHP Coefficient of Performance and Adjusted Coefficient of Performance

Site	# of Hours	Mean COP	Standard Deviation	Mean ACOP	Standard Deviation
228	6,435	1.83	0.37	0.44	0.58
235	4,022	1.70	0.32	1.29	0.45
439	7,381	3.09	0.58	2.39	1.27
449	8,329	2.46	0.48	1.56	1.02
450	2,728	2.71	0.29	1.93	2.01

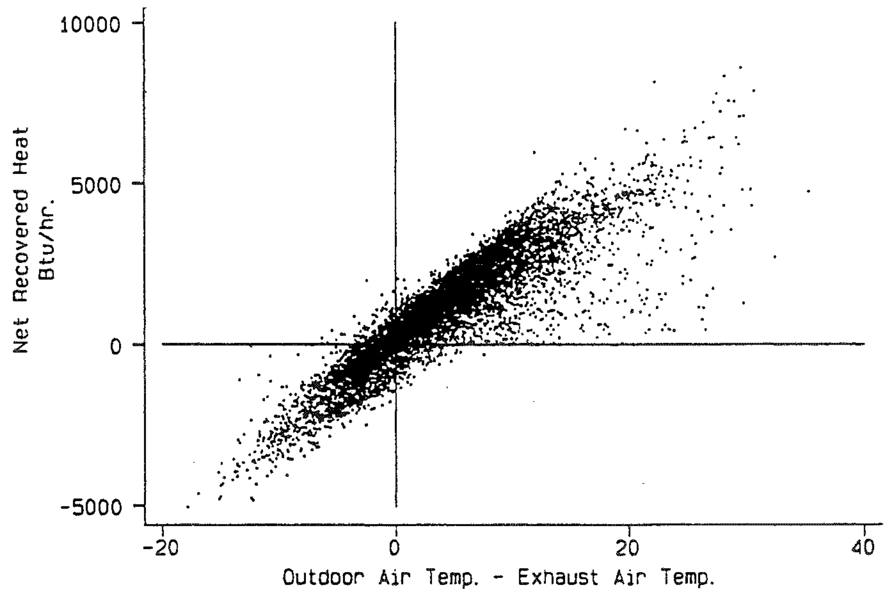

Figure 3. Outdoor-Exhaust Air Temperature Differential vs. Net Recovered Heat - Site #439

Table 4. Evaluation of Energy Savings from Nighttime Setback

Site	UA Btu/F-hr	Gains Btu/hr	52-Week Actual Space Heat with Setback kBtu	52-Week Estimated Space Heat without Setback kBtu
228	269	5,022	21,458	25,320
439	889	8,319	51,903	61,464

1988. The above computation is rerun with indoor air temperature data modified so that it never drops below 68°F, the inferred temperature if setback is never done.

The savings from night setback are 18.0 percent and 18.4 percent for Sites 228 and 439, respectively. Energy consumption was found to be significantly lower due to nighttime setback. However, these savings are limited by the EAHP control strategy.

The EAHP is set to give priority to domestic hot water (DHW) if there is demand for such heat. If water heat is not required, priority is given to space heat provided that there is demand. The thermostat for the supplemental space heat condenser is set at a higher temperature than the main space heat thermostat. This is to ensure that the EAHP provides space heat before electric resistance heat is required.

Since the EAHP supplemental space heat condensers operated at peak capacity during the setback hours of winter months, it is assumed that the supplemental space heat condenser thermostat was not set back. Consequently, the main thermostat was set back to a level well below the EAHP. Greater energy savings could have been achieved by setting back both the main space heat and the supplemental space heat condenser thermostats. The supplemental space heat condenser thermostat should be set on a temperature differential to the main thermostat. As the main thermostat is set back, the supplemental space heat condenser thermostat would also be set back an equal increment. Further analysis is required to determine morning reheat energy requirements compared to supplemental space heat condenser requirements under set back and non-set back scenarios. If reheat energy is served by resistance heating, it may be more economical to run the supplemental space heat condenser when ventilation is needed and indoor temperature is falling below the morning setting of the electric furnace thermostat. The energy economics are more complicated when considering over-ventilating in order to operate the supplemental space heat condenser.

CONCLUSION

Five homes located in Montana and Washington were equipped with exhaust air heat pump systems and monitored with hourly data acquisition systems. These systems were evaluated for ventilation and heating performance.

Two different types of systems were installed. The Montana systems had a complex design and lower performance than the Washington systems. The manufacturer has since discontinued production of the Montana systems. All systems had enough ventilation capacity to meet programmatic ventilation requirements. However, the Montana systems were not operated with sufficient frequency to meet program requirements. The Washington homes had excess ventilation.

These systems recovered between 5,422 and 34,922 kBtu over the monitoring period and operated with COPs between 1.70 and 3.09. The Montana systems operated with lower COPs and are no longer being manufactured. The three Washington EAHP systems over-ventilated the structure due to insufficient supplemental space heat condenser control strategies. Real EAHP efficiency could be increased by limiting the operation for space and water heating to those times when outdoor temperature exceeds the expected exhaust temperature or when ventilation is required. Effectiveness of nighttime set back could be enhanced by coupling the main space heat and supplemental space heat condenser thermostats.

ACKNOWLEDGMENTS

The research reported herein is sponsored by the United States Department of Energy, Bonneville Power Administration. The Project was operated by the Washington State Energy Office. Under contract to Washington State, The Fleming Group implemented the actual monitoring, including specifying verification and quality control procedures and selecting all hardware. Energy agency personnel in the states of Idaho and Montana served as site liaison, identifying candidate houses, performing prewiring inspection, and performing some equipment maintenance.

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