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EXHAUST-AIR HEAT-PUMP PERFORMANCE WITH UNSTEADY-STATE OPERATION

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Abstract—The energy performance of two residential exhaust-air heat pumps (EAHP) with different condenser designs was studied experimentally in a laboratory with a focus on transient heat-pump performance associated with time varying requirements for water and space heating. Experimental variables included the total daily volume of hot water required, the schedule of hot-water demand, the temperature of water entering the hot-water tank, hot-water delivery temperature, and the temperature and flow rate of air entering the auxiliary refrigerant-heated fan coil supplied with one of the EAHP units to permit space heating. Based on the data, for a wide range of operating conditions, we derived linear correlations between the heat-pumps' time-average coefficient of performance (COP) and appropriate spatial and temporal average temperatures in the hot-water tanks. With the refrigerant-heated fan coil, the COP varied non-linearly with air flow rate. COPs ranged from 2.0 to 4.2. The control system of the EAHP with two condensers (one is the fan coil) gives priority to water heating. Based on the data, results from our previous hourly modeling of EAHP performance, data from field studies in Sweden, and new calculations, we propose a new control system that usually places priority on space heating and, thus, takes better advantage of the capacity to store heat in the water tank. We estimate that this proposed control system may increase annual energy recovery by approximately 1000 kWh if the EAHP is used in a Portland, Oregon house. Total annual energy savings due to EAHP operation in an all-electric house (compared to the same house with electric resistance space and water heating) is estimated to be approximately 6000–7000 kWh.

INTRODUCTION

Continuous mechanical ventilation of residences has become more common because energy-efficient building practices that conserve energy also reduce natural ventilation through the building envelope. One mechanical ventilation technique that has been widely applied in Scandinavia and is starting to be used in North America is ventilation with an exhaust-air heat pump (EAHP). Such an application is shown schematically in Fig. 1. Exhaust-air is drawn by means of a single fan from several locations in the house such as the bathrooms and the kitchen leading to a slight depressurization of the house. This depressurization leads to inflow of fresh air from the outside either through the unintentional leakage area of the house, or preferably through adjustable registers located in the bedrooms and the living room. In some Scandinavian applications, the fresh air is driven by a second fan (of somewhat smaller capacity than the exhaust fan), and forced through the space-heating unit shown as "fan coil" in Fig. 1 before entering the house.

Also shown schematically in Fig. 1, is the heat pump that extracts energy from the exhaust-airstream by means of a refrigerant evaporator and transfers it by means of a compressor and a condenser either to the tap-water or to the indoor air. Figure 1 shows a configuration with two condensers and a switchable valve that determines which condenser is connected to the refrigerant compressor. This configuration is the main object of investigation in this paper. Another system that is investigated has a single refrigerant condenser located in the hot-water tank and a tap-water recirculation loop to the fan coil for space heating (in some applications, high-pressure radiators or floor heating tubes are used instead of a forced-air coil). A third configuration of the heat sinks that is not studied in this paper is the use of the heat-pump condenser as one heat source of a full hydronic heating system where the additional heat source is, for example, electrical immersion heaters placed in the same radiator water circuit. In most applications of a fan coil for space heating of the type shown in Fig. 1, additional heating capacity is required in the form of electric base-board heaters, for example.

Many kinds of domestic heat pumps for heating duty have received considerable attention in the technical literature. An excellent overview is provided in [1]. However, detailed information on

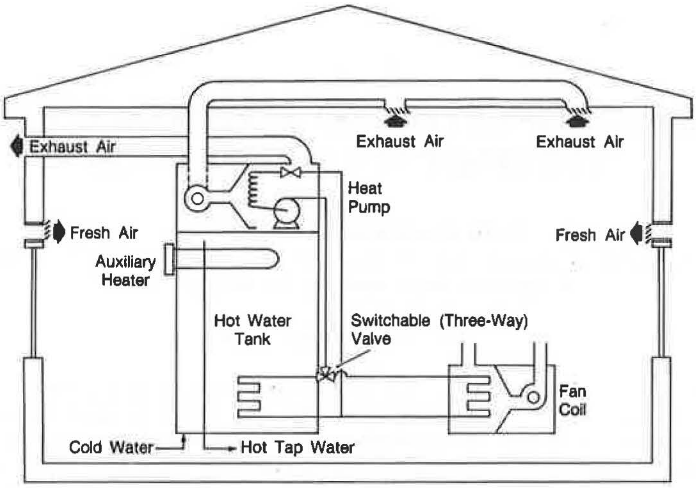

Fig. 1. Application of exhaust-air heat pump for space and water heating.

the performance of EAHPs is quite scarce. This was experienced by the authors in a previous modeling-based investigation of an EAHP system functionally similar to that in Fig. 1 [2]. In that investigation, the authors had to use a manufacturer's bulletin for an EAHP that is no longer available for information on such important parameters as the coefficient of performance (COP).

The investigation [2] indicated that energy savings resulting from EAHP operation in an all-electric house were sensitive to factors that affected both load duration, i.e. compressor on time, and the coefficient of performance of the EAHP. The component of greatest uncertainty with regards to system behavior was the water tank, and specifically the thermal stratification in the tank. It is well known that the time-average temperature in the bottom of the hot-water tank of Fig. 1 is lower than the time-average temperature of the tap-water drawn from the top of the tank, but quantification of this effect by modeling proved difficult. The difficulty of modeling the stratification was a main impetus for the experimental EAHP investigation of this paper.

Although the main objective of the present study was to investigate the COP of EAHPs in typical unsteady-state operation, it became apparent that the entire system consisting of the EAHP as a heat source coupled to the two heat sinks (water and house) warranted further study. A second objective of this paper thus became a critical study and modification of the conventional EAHP control system, so as to improve load duration (compressor operation) of the heat pump. The main control difficulty associated with the EAHP system is one of assigning the correct priority between the two heat sinks, considering that the water heat sink has a significant accumulative capability whereas the house heat sink does not. Both sinks are, of course, unsteady-state in nature because tap-water draw rates are highly variable and so is the weather-induced space-heating demand.

Specifically, the objective of this investigation was (1) to experimentally determine the COP for two different EAHPs under realistic tap-water usage schedules and to correlate the COP with temperatures surrounding the condensers, and (2) to modify the control system for improved load duration for an EAHP with both water- and space-heating load. We discuss our experimental results, propose a new control system, and estimate the energy savings that could result by use of an EAHP with the proposed control system in the all-electric house located in Portland, Oregon.

STUDIED SYSTEMS

Specifications for the heat pumps are listed in Table 1. Unit A has a smaller compressor than Unit B; however, Unit A has a larger COP (according to results discussed later) making the rate of energy extraction from the exhaust-air nearly identical for the two heat pumps. Unit A functions like the EAHP shown in Fig. 1 except that the condenser is wrapped around the entire water

Table 1. Specifications for the exhaust-air heat pumps

	Unit A DEC International Therma-Stor Products Group: Therma-Vent Model HPV-80 with remote space heater	Unit B Elektrostandard of Sweden represented by Fiberglas Canada Inc.: Model Aquaes 270
Tank volume, (l)	260	220
Condenser design	plate wrapped around entire tank	coil in bottom of tank
Average compressor power, (W)	570 (base run)	750 (base run)
Exhaust-fan power, (W)	72 (fixed speed)	120 (max, variable speed)
Resistance-heater power, (kW)	1.7	1.2
Space heater	refrigerant-heated fan coil	option for tap-water heated convector
Space-heater power requirement, (W)	72 (fan)	25 (water pump) + optional fan

tank. Unit B on the other hand, has a condenser in the bottom of the tank like the one shown in Fig. 1. Unit B is equipped with a water pump that can be connected to a hot-water-heated radiator or fan coil, but this option was studied only superficially.

The experimental set up and the procedure for testing the heat pumps have been described in detail elsewhere [3] and will not be repeated here. Only two items pertaining to the experimental procedure need mention here: first, four vertically distributed temperature measurements inside the water tank were important with regard to determination of water-stratification effects. Second, the water demand timing system allowed different hot-water demand schedules to be used. The two water-demand schedules examined are shown in Figs 2 and 3. The intermittent schedule shown in Fig. 3 is considered closer to reality. However, the results were quite similar with these two water-usage schedules. Water-demand volume was varied by scaling all flows shown in Figs 2 and 3.

WATER-HEATING RESULTS

In water-heating mode, the heat pumps were tested in 24-h runs primarily devoted to the study of the following parameters: daily hot-water demand (0–500 l), demand schedule as shown in Figs 2 and 3, water-inlet temperature (10–20°C), and hot-water outlet temperature (EAHP

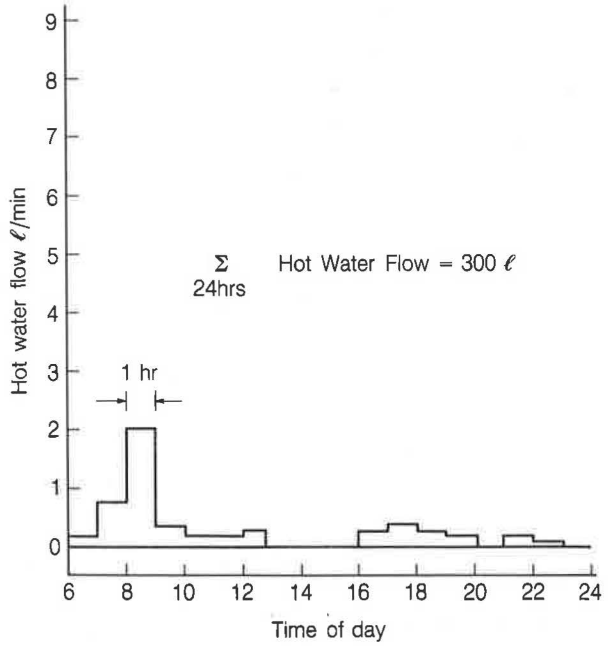

Fig. 2. "Continuous" hot-water usage schedule.

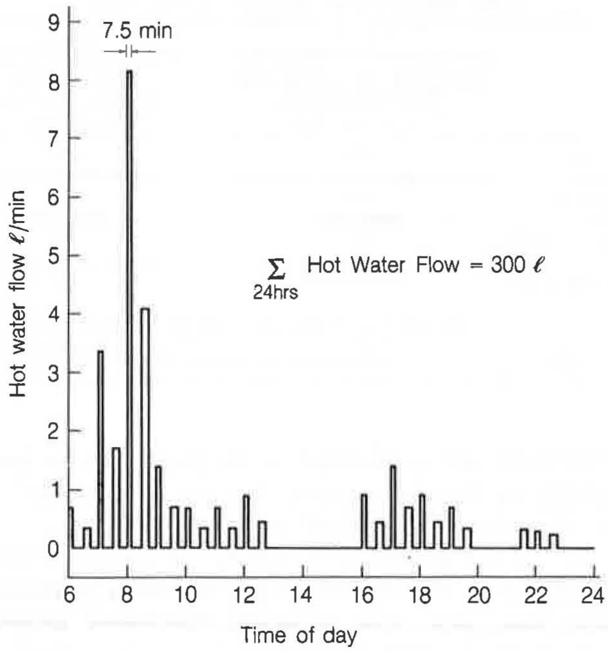

Fig. 3. "Intermittent" hot-water usage schedule.

thermostat setpoint of 40–60°C). Experiments with variable exhaust-air flow rates, temperatures, and humidities were performed but results were as expected [3] and are not reported here. Instead, this paper reports and analyzes all results with variable water-side conditions but with fixed exhaust-air conditions.

Figures 4 and 5 present the experimentally determined COP for Units A and B, respectively, for data where COP is defined as total heat transferred by the heat-pump condenser in one test run

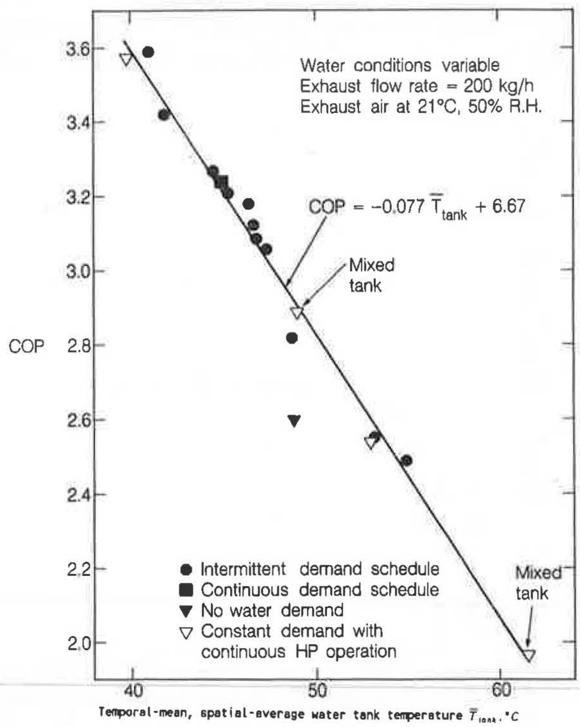

Fig. 4. Unit A heat-pump COP for water heating as function of spatial-average temperature of water in tank (spatial-average refers to space-averaging of the four measurement locations, temporal-mean refers to time-averaging over compressor ON time).

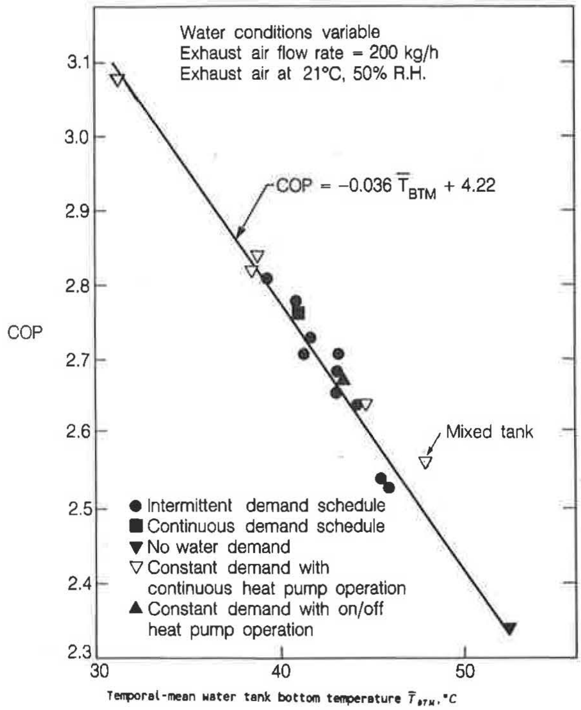

Fig. 5. Unit B heat-pump COP as function of water temperature at the bottom of the tank (temporal-mean refers to time-averaging over compressor ON time).

divided by total energy supplied to the heat-pump compressor in the run. Energy balances for all tests reported in this paper (including space-heating tests) were $97.9 \pm 3.6\%$ for Unit A and $98.6 \pm 1.5\%$ for Unit B (the energy balance is the ratio of heat-pump condenser heat over the sum of evaporator heat and compressor energy). The standard deviations of 1.5 and 3.6% are quite acceptable (the higher number for Unit A is due to the fan-coil runs that had less precision than the water-heating runs), but both units produced a small bias in the energy balance: approximately 2% of condenser heat was unaccounted for on the evaporator side. This bias is possibly explained by the imperfect insulation of the heat-pump air compartment that resulted in a heat gain from the environment unaccounted for in the measurements.

The data in Fig. 5 show that the performance of Unit B, with its condenser in the bottom of the tank, can be correlated to the time-average (average when compressor is on) temperature of the water in the bottom of the tank as expected based on our physical understanding of heat-pump operation. Similarly, Fig. 4 shows that the performance of Unit A with its large wrap-around condenser can be correlated to the time-average temperature of all water within the tank (hereafter referred to as the tank-average temperature). Interestingly, the simple linear correlations between COP and the appropriate water temperature hold despite the large variations in total volume of water demand, demand schedule, and water inlet and outlet temperature. The two heat pumps show significantly different sensitivities of COP to their respective correlating temperature: Unit A has a negative slope (-0.077) approximately double that of Unit B (-0.036). A higher temperature difference between the condensing refrigerant and the water for Unit B is the probable explanation for the different slopes, however, no refrigerant temperatures or pressures were measured in this study.

The second part of EAHP water-heating performance concerns tank stratification, i.e. the relationship between the COP correlating temperature and the hot-water delivery temperature which is the temperature of importance in the overall EAHP system. The correlations obtained in the experiments are shown in Figs 6 and 7. Here, the average delivery temperature is obtained by averaging over delivered hot-water volume. Based on Figs 6 and 7, the stratification effect can be modeled as a single offset between the two temperatures of importance to EAHP performance—COP correlating temperature and the hot-water delivery temperature. This stratification offset is

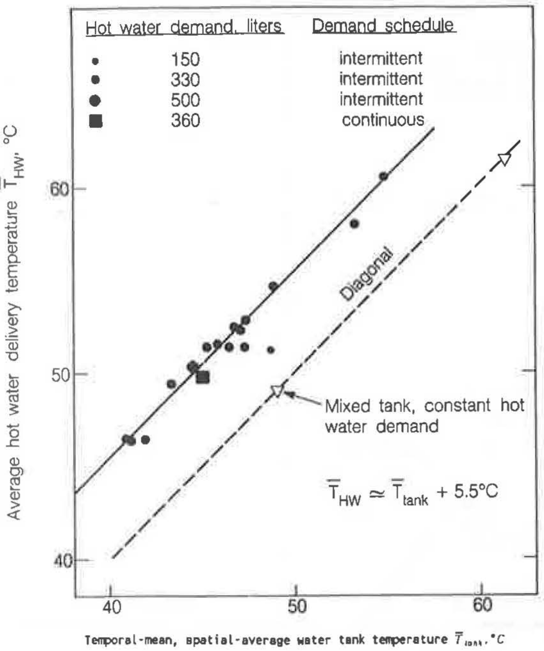

Fig. 6. Stratification in the water tank of Unit A (spatial-average refers to space-averaging of the four measurement locations, temporal-mean refers to time-average over compressor ON time).

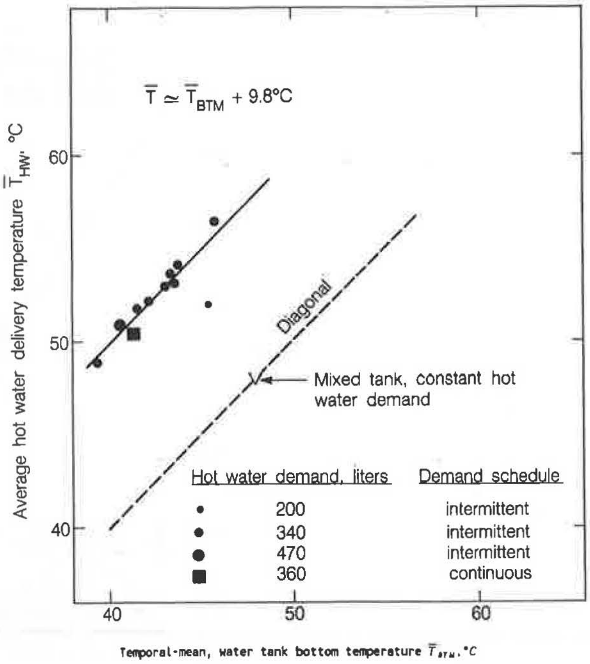

Fig. 7. Stratification in the water tank of Unit B (temporal-mean refers to time-averaging over compressor ON time).

approximately independent of water conditions over the region of practical interest. However, a very low water demand reduces the offset for both heat pumps, but low water demand means little water-heating load, and hence reduced significance in total energy savings computations. As expected, the stratification-induced temperature offset is larger for Unit B (9.8°C in Fig. 7) than for Unit A (5.5°C in Fig. 6). This is, of course, mostly a consequence of the different COP correlating temperatures: i.e. tank bottom temperature for Unit B and the tank average temperature for Unit A.

Combining the results for Unit A in Figs 4 and 6 and for Unit B in Figs 5 and 7, the increase in COP due to the stratification offsets can be determined to be +0.42 for Unit A and +0.35 for Unit B. Thus, stratification causes a 15% increase in COP for both units at a hot-water delivery temperature of 52.5°C. The reason for similarity in the effect of stratification on COP is that Unit B has a stratification temperature offset twice that of Unit A but at the same time its COP has a temperature sensitivity half of Unit A.

For Unit A, the steep drop in COP with tank average temperature (or with hot-water delivery temperature because of the simple empirical offset between the two) has an important consequence—if high hot-water temperatures are required, more energy efficient operation would result by using the heat pump to heat the incoming water from the supply-water temperature $\overline{T}_{win}$ to an intermediate temperature $\overline{T}_{wout}$ and to carry out the final heating with the resistance heater near the top of the tank (according to experimental results, this tank section does not substantially influence the tank average temperature, and therefore does not influence heat-pump COP). The optimal heat-pump outlet temperature $\overline{T}^*_{wout}$ is given by

$$\overline{T}^*_{wout} = -\frac{\beta}{\alpha} - [(\alpha \cdot \overline{T}_{win} + \beta)/\alpha^2]^{1/2} \tag{1}$$

where α and β are the parameters in the COP correlation

$$\mathrm{COP} = \alpha \cdot \overline{T}_{wout} + \beta. \tag{2}$$

For example, if $\overline{T}_{win}$ is 15°C, optimal heat-pump operation is to heat the water to no more than 60.5°C using the heat pump (temporal-mean tank-average temperature being 55°C), and to use electric resistance heating from there on. Although an optimal heat-pump outlet temperature of

60.5°C is somewhat above the region of practical interest, the existence of this optimal point has an important practical consequence—the total energy consumption for water heating is quite insensitive to a small amount of "trim heating" by the electric resistance heater because the total energy consumption curve has a flat minimum at $\bar{T}^*_{wout}$. This in turn means that load shifting from water heating to space heating can be done with the intentional use of electric trim heating for the water load. This holds for Unit A only because Unit B has a much higher $\bar{T}^*_{wout}$ (71°C in the example) and also space heating in Unit B takes place via the hot tap-water.

SPACE HEATING RESULTS

Space heating with Unit B involves recirculation of hot tap-water through a convector. This convector should, if possible, have a capacity at least equal to the heat-pump condenser heating effect so that load duration is not reduced during days with little water usage and a large heating demand. This requirement is difficult to meet without a forced-air convector. The forced-air feature also helps in the distribution of heat throughout the space because any limitation of heat distribution translates to a limitation of the space heat sink implying reduced heat load duration (this also applies to Unit A). A second requirement for the Unit B space heater is that the favorable stratification effect of Unit B should not be distributed by water recirculation. This proved to be a difficult objective. In the water-heating experiments with Unit B, fan-coil operation was simulated by continuously pumping water from the top of the tank to the bottom of the tank by means of the water pump provided. As shown by the "mixed-tank" point on Fig. 7, stratification was totally destroyed by the recirculated water. However, one could possibly improve on this performance by designing a low water flow convector that cools down the recirculated water significantly. This is not an easy task in practice considering the simultaneous requirement of a high heating effect.

The refrigerant-heated fan coil (air-cooled condenser) of Unit A was tested in experiments of a steady-state nature. Figure 8 shows the results obtained with different air flow rates through the fan-coil unit. The 72 W fan provided with the unit was capable of delivering only the lowest air flow rate shown in Fig. 8 (with reasonable air duct dimensions) and limited the COP to approximately 2.8. A more powerful fan, 140 W for example, can substantially increase COP for

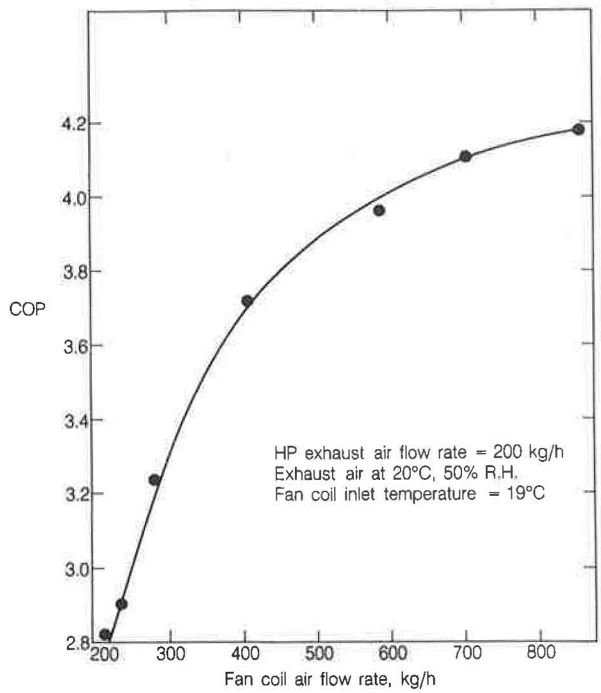

Fig. 8. Unit A heat-pump COP for space heating as function of fan-coil air flow rate.

Unit A in space-heating mode. An air flow rate of 400 kg/h, for example, produces a COP as high as 3.7. (However, fan power is not included in the denominator of the COP calculation because most of the energy supplied to the fan can often be recovered as useful space heat.)

It should be noted that high COPs with the fan coil can also be obtained by lower inlet air temperatures to the fan coil (the COP temperature dependence is -0.073 or about the same as the slope -0.077 of the COP line in Fig. 4 for the case of water heating). However, the benefit of this reduced air temperature can only be realized with outside air blown through the fan coil and not in the more common situation of room air recirculating through the fan-coil unit.

COMBINED WATER AND SPACE HEATING RESULTS

Only heat pump A was tested in combined water and space heating mode. The Unit A control system gave priority to water heating, thus, for combined runs with 24-h compressor operation, slack times (time periods without a water heating demand) were simply used for fan-coil operation with no limit on the space heat sink. The results of a series of such runs are shown in Fig. 9. This series of runs starts out as purely space-heating in the left-most run in Fig. 9, and water load is increased to the right in the figure by increasing the heat-pump's water thermostat setpoint. (The water outlet temperature is maintained at 50°C in all runs due to the electric resistance trim heating.) The experimentally determined COP values (heat transferred by the two condensers divided by compressor energy input) are plotted in the middle diagram while energy savings, defined as heat extracted by the evaporator minus energy required for operation of the 72 W fan-coil fan, are shown in the top part of the diagram. The COP during combined water and space heating can be expressed as simply a superposition of its two parts:

$$\mathrm{COP}_{\mathrm{combi}} = x_{\mathrm{FC}} \cdot \mathrm{COP}_{\mathrm{FC}} + x_{\mathrm{W}} \cdot \mathrm{COP}_{\mathrm{W}} \quad (3)$$

where x_{FC} = Fan coil ON time fraction, x_{W} = Water heating ON time fraction, $\mathrm{COP}_{\mathrm{W}}$ is given by the equation in Fig. 4, and $\mathrm{COP}_{\mathrm{FC}}$ is 2.7 (from results like those in Fig. 8).

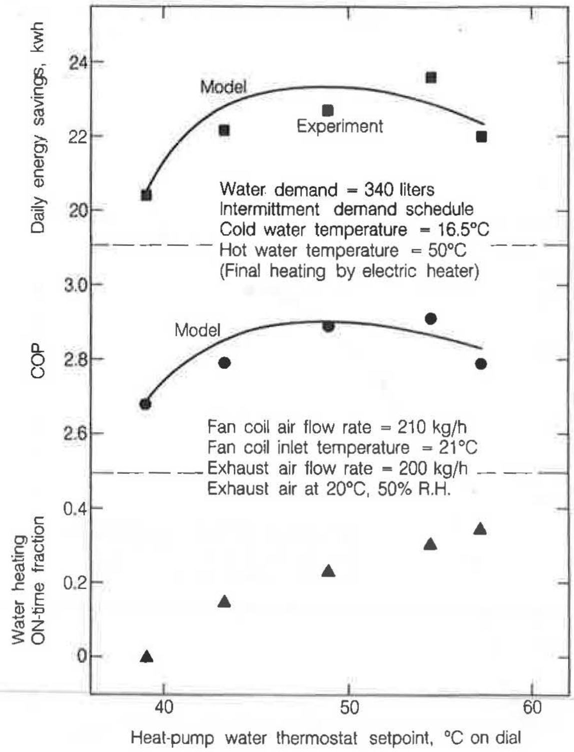

Fig. 9. Unit A heat-pump performance in combined water/space heating runs with 24-h compressor ON time (no space heat sink saturation).

Since the compressor ran continuously in all tests depicted on Fig. 9, $x_{FC} = 1 - x_W$. The daily energy savings with 24-h compressor operation is then:

$$\text{Energy savings} = (\text{COP}_{\text{combi}} - 1)\cdot E_{\text{comp}} - E_{\text{fan}} \tag{4}$$

where E_{comp} is the compressor energy consumption (13.2 kWh for 24-h compressor operation), and E_{fan} is the fan energy consumption (1.7 kWh for 24-h operation).

Based on the equation given above, the energy savings represent the amount of energy recovered from the exhaust-air stream less the amount of energy required to run the heat pump's exhaust fan. In an all-electric house, an EAHP system that operated continuously would reduce total household energy consumption by the amount of the energy savings if the reference-case all-electric house (with electric resistance space and water heating) had the same ventilation rate.

The main conclusion from Fig. 9 is that load shifting between the fan coil and the water tank has little effect on the combined COP value and energy savings. This is because space-heating COP and water-heating COP are of approximately the same magnitude. However, the reason for the flat maximum in COP (and energy savings) is that water heating with a low heat-pump thermostat setpoint produces very high water-heating COPs, but this water-heating COP drops as the setpoint is increased. For a high fan-coil air flow rate such as 400 kg h^{-1}, there would be no maximum in combined COP (nor in energy savings), but rather a monotonic decline as the water heating load is increased. However, even in this case, combined COP and energy savings are relatively insensitive to load shifting between water heating and space heating as long as the combined load provides for nearly continuous operation.

LOAD DURATION ASPECTS

Maximizing heat-pump load duration (maximum compressor operation) is more important than optimal load shifting between water and space heating because energy savings are directly proportional to load duration (with relatively minor adjustments as discussed in the previous section). Hence, the heat-pump control strategy which influences load duration is of utmost importance.

Unit A has a control system that gives priority to water heating, i.e. space heating is only allowed when the water tank is fully heated. This control strategy is satisfactory during cold winter months when there is ample space heating demand 24 h a day. The problem arises when there is temporary saturation of the space heat sink during the day. In these periods the control system should have space heating priority so that load duration is not reduced because of simultaneous saturation of both heat sinks. It is also wrong in principle to assign constant priority to the water heat sink because this heat sink has a larger accumulative capacity than the space heat sink, and therefore allows more flexibility in time-shifting of the load.

Figure 10 shows a proposed new control strategy for a Unit A type heating system; the difference between this proposal and the existing control system is that space heating normally has priority in the control system of Fig. 10 (through the space thermostat TC_3 and the position of the switchable three-way valve which directs heated refrigerant to the fan coil when energized). The existing upper water themostat of Unit A is also modified to be double acting (2 setpoints): The old action of controlling operation of the electric resistance auxiliary heater remains as

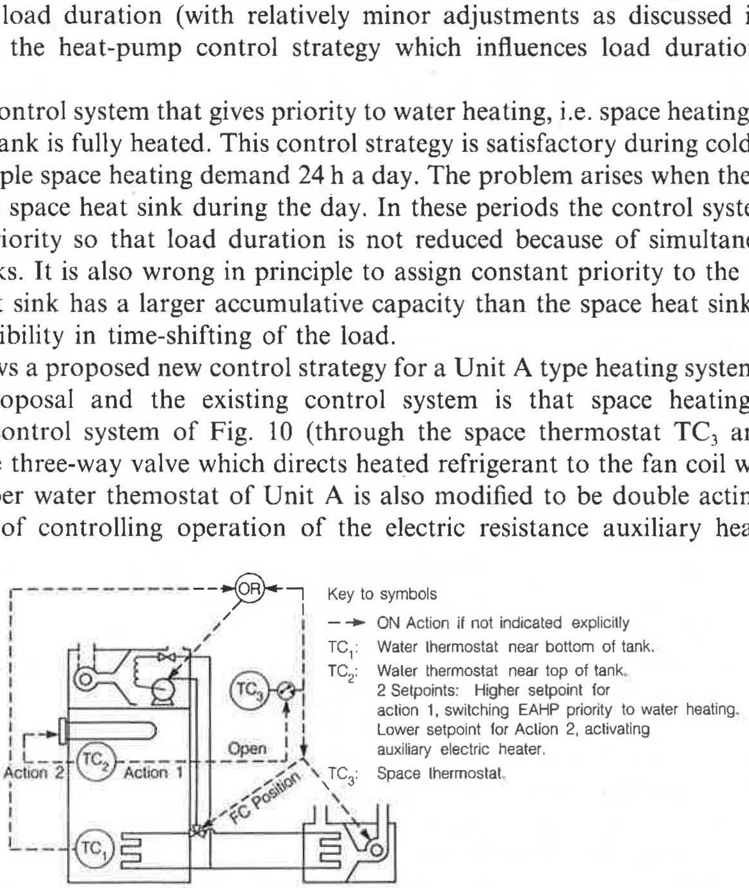

Fig. 10. Proposed exhaust-air heat-pump control strategy for space and water heating.

action 2 activated by a lower setpoint (45°C for example). The new action 1 opens the contact in Fig. 10 and, thus, switches the three-way valve so that condenser heat is used for water heating. Action 1 is normally activated by a higher setpoint than action 2 (50°C for example). In this override situation (action 1), when the water heating (by the heat pump) is given priority over space heating due to low water temperatures near the top of the hot-water tank, compressor operation is governed by the existing thermostat TC_1 near the bottom of the tank which has a setpoint of 50°C, for example. TC_1 also controls compressor operation when TC_3 is satisfied, i.e. when there is no space-heating demand.

It should be noted that the control system of Fig. 10 can directly be applied to an EAHP system of the type studied in [2], namely a fan coil and a water-tank coil heated by a separate water circuit with the heat-pump condenser as the heat source for this circuit. The only difference relative to Fig. 10 is that the three-way valve switches the water flow instead of the refrigerant flow.

For the estimation of the increased load duration achieved with the control system of Fig. 10, reference is made to a Swedish field study [4] of EAHPs similar to Unit B. With space-heating control priority in the relatively cold climate near Stockholm (3900°C-days), the heat pumps were found to have essentially continuous load for 35 weeks of the year, and a total yearly load duration of as much as 80% of the year. An analysis of the data for Spring and Fall season in [4] reveals that the space heat sink for the heat-pump output becomes saturated when the house heating demand, on a weekly basis, falls below 115% of maximum EAHP heating effect (possibly due to temporal variations in heating demand and imperfect distribution of fan-coil heat throughout the house). At this point in Spring, heat-pump load duration starts to decrease from 100% and, as heating demand diminishes it goes down successively, reaching 25% when only water-heating demand remains.

Using the space-heating saturation effect found in [4] on the simulated house in Portland, Oregon, [2], we can estimate the additional energy savings that can be realized using the control strategy of Fig. 10 and continuous operation of the exhaust fan. With this control strategy applied either to Unit A or Unit B, load duration is increased by 23–32% depending on the amount of thermal insulation in the house (lower increase in load duration for a well-insulated house). This increased load duration translates to additional energy savings in Portland, Oregon (2700°C-days) of 800 kWh for a well-insulated 125 m^2 house and 1300 kWh for the same house with more typical insulation.

Based on a combination of original projections of energy savings via an hourly simulation [2] and the estimated additional energy savings given above, the Unit A EAHP system with the controls of Fig. 10 installed in the 125 m^2 all-electric house in Portland, Oregon, is estimated to produce annual energy savings of 5600 kWh in the well-insulated house and 7200 kWh in the standard house. These energy savings are based on a comparison with a house with continuous mechanical exhaust ventilation without heat recovery (0.5 air changes/h of ventilation) with electric resistance space and water heating. These energy savings are considerably higher than those estimated in [2] with a EAHP system that is far from optimal.

CONCLUSIONS

Unsteady-state experiments with one of the tested heat pumps (Unit B) confirm approximately the simulation results of our previous work [2]. The agreement between simulation and experiment does not mean that the model used in the simulations is satisfactory in all aspects. For example, the thermal stratification within the water tank in the model was very small even in the absence of recirculation of water through the space heater. The water-heating experiments on the other hand, showed that average hot-water delivery temperatures were significantly different from the water temperatures surrounding condensers that determine heat-pump COP. For Unit B, this stratification effect (average temperature difference between the supplied hot water and COP-correlating temperature) was as large as 10°C implying a 15% enhancement of COP due to stratification. However, stratification in Unit B disappeared with water circulation for space heating. This brought the experiments with Unit B into agreement with the simulations [2] that had the relatively low average COP of 2.4 for a hot-water supply temperature of 52.5°C.

The other heat pump (Unit A) had a COP approximately 30% greater than the COP used in the simulation [2]. This improved COP results in approximately a 15% increase in energy savings and a 13% decrease in the cost of conserved energy compared to previous predictions [2].

Perhaps most importantly, the experimental data and analysis provided in this paper indicate that a large increase in energy savings (approximately 1000 kWh in Portland, Oregon) is obtained if the EAHP control system is modified to increase load duration by giving priority to space heating most of the time and taking advantage of the energy storage capabilities of the water tank. The total annual energy savings, for a near-optimal application of an exhaust-air heat pump in a typical all-electric, Portland, Oregon house, is approximately 6000–7000 kWh.

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