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EXHAUST-AIR HEAT PUMP STUDY:

EXPERIMENTAL RESULTS AND UPDATE OF

REGIONAL ASSESSMENT FOR THE PACIFIC NORTHWEST

By

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EXECUTIVE SUMMARY

The desire for improved energy efficiency and thermal comfort in residences has resulted in construction practices that minimize air leakage through the building envelope. This implies that ventilation air required for acceptable indoor air quality must be supplied mechanically. Several techniques are available for this application. This report describes a laboratory evaluation of a mechanical ventilation technique that has recently been introduced into the United States, exhaust ventilation employing a heat pump to recover energy from the exhaust air stream.

Heat pumps designed to recover energy from an exhaust airstream are referred to as exhaust-air heat pumps (EAHP). These heat pumps are ideally suited for supplying heat at high temperatures to domestic hot water and can be used exclusively for this purpose. However, the amount of heat that can be extracted from the exhaust airstream by a heat pump is typically more than is required to meet hot water needs. Consequently, larger heat pumps can be employed to supply heat to the indoor space as well as the water.

This report describes one portion of a larger project supported primarily by the Bonneville Power Administration to determine the suitability of EAHPs as mechanical ventilation systems in residences in the Pacific Northwest. Phase one of the project consists of two components: a preliminary assessment of the performance of EAHPs in the Pacific Northwest based on computer simulations (completed earlier), and laboratory evaluations of EAHP performance plus a brief update to the preliminary assessment. The second component is the subject of this report. Phase two of this project, which is being conducted by another organization, is a field study of EAHP performance in actual residential settings. Originally, a third phase of the project was planned -- this would have been a comprehensive final modeling effort, similar to the preliminary assessment, but taking the results of Phases 1 and 2 into account.

The laboratory results reported here represent the first complete evaluation of exhaust-air heat pumps available in the open literature. Prior information had to be gleaned from the limited manufacturer's data that were available. The results reported here now allow the performance of two EAHP systems available in this country to be modeled with a much higher degree of confidence.

The performance of two EAHP systems was monitored, and one system was evaluated with and without operation of an optional fan-coil condenser which can be used to deliver the recovered energy to the indoor air. The influence of numerous parameters (e.g., inlet air temperature, humidity, and flow rate; water demand volume and schedule; inlet water temperature; and thermostat set points) on EAHP performance was determined. (The primary measures of performance are the coefficient of performance (COP) and the energy savings relative to electric-resistance heating.) It was possible to correlate the measured COPs to temperatures of water within the hot water tanks and also to average hot-water delivery temperatures. Therefore, simple empirical models of EAHP performance were developed and are presented in the report.

The measured COPs varied between approximately 2.3 and 4.2 depending on the EAHP and the operating conditions. One of the EAHP systems studied experimentally has an average COP that is approximately 30 percent greater than assumed in preliminary modeling. This implies a 15 percent increase in energy performance and a 13 percent decrease in the cost of conserved energy compared to previous predictions. Revised estimates of total yearly energy savings and costs of conserved energy are provided for three Pacific Northwest cities.

The experimental data and analysis provided in this report also indicate that large increases in energy savings (1000-1500 kWh per year) might be obtained if EAHP systems were modified so that more heat is supplied to the indoor air and greater advantage is taken of the energy storage capabilities of the water tank. Considering the magnitude of these potential increases in energy savings, we recommend that experimental evaluations of EAHPs with modified control systems be performed in the future.

INTRODUCTION

This paper describes an evaluation of a mechanical ventilation technique that has recently been introduced in the United States, exhaust ventilation employing a heat pump to recover energy from the exhaust air stream.

Figure 1 shows the application of a typical exhaust-air heat pump (EAHP) to a residential building. Here, exhaust air is drawn by means of an exhaust fan from several locations such as the bathrooms and the kitchen, leading to a slight depressurization of the house. This depressurization in turn leads to inflow of fresh air from the outside, either through the natural leakage pathways of the house, or, preferably, through adjustable fresh air inlets as shown in Fig. 1. The adjustable inlets allow the occupants to determine where fresh air enters the house, and thus allow drafts to be avoided in locations selected by the occupant.

The heat pump extracts energy from the exhaust air stream by means of an evaporator and transfers it to the tap water and/or to the interior of the house itself. Figure 1 shows a configuration with two condensers and a three-way solenoid valve that determines which condenser is connected to the heat pump's compressor.

The ventilation aspects of EAHPs and a preliminary evaluation of their impact on residential energy consumption and cost effectiveness have been discussed elsewhere [1,2]. EAHPs provide a nearly constant ventilation rate throughout the year (with the exhaust fan ON all the time), while natural ventilation by infiltration drops almost to zero during certain times of the year.

The objective of this study was to experimentally evaluate the energy performance of three EAHP systems. Two of the systems supplied heat only to the domestic hot water. The third system (which is a modified version of the first system) contained a second condenser so that heat could be supplied to the air within the building. The experimental data obtained in this study was required to confirm and allow updating of a computer model used for the previously mentioned preliminary evaluations of this ventilation technology.

EXPERIMENTAL SYSTEM AND CALIBRATION

Specifications for the heat pumps are listed in Table 1. Two units were studied and the performance of one of these units was determined with and without usage of the optional fan-coil condenser which helps heat the indoor air. The first, manufactured by DEC, is called Unit A throughout the remainder of this report. The second, manufactured by Elektrostandard, is referred to as Unit B hereafter. There are several differences between the two heat pump units. Unit A has a smaller compressor than Unit B; however, Unit A has a larger coefficient of performance (according to results discussed later), making the rate of energy extraction from the exhaust air almost identical for the two heat pumps. Unit A has a refrigerant-heated fan coil condenser and the refrigerant can be directed either to this fan coil or to the condenser which heats the domestic hot water. Unit B can be connected to a hot water-heated fan coil; however, its performance was not evaluated when used with a fan coil. There is a major difference in the water heating condenser design: Unit A has large condenser that wraps around most of the water tank, while Unit B has a conventional coil-type condenser inside the lower quarter of the water tank. This latter coil has double walls so that any accidental refrigerant leakage would occur to the house air rather than to the domestic hot water. The Unit A condenser also has this safety feature.

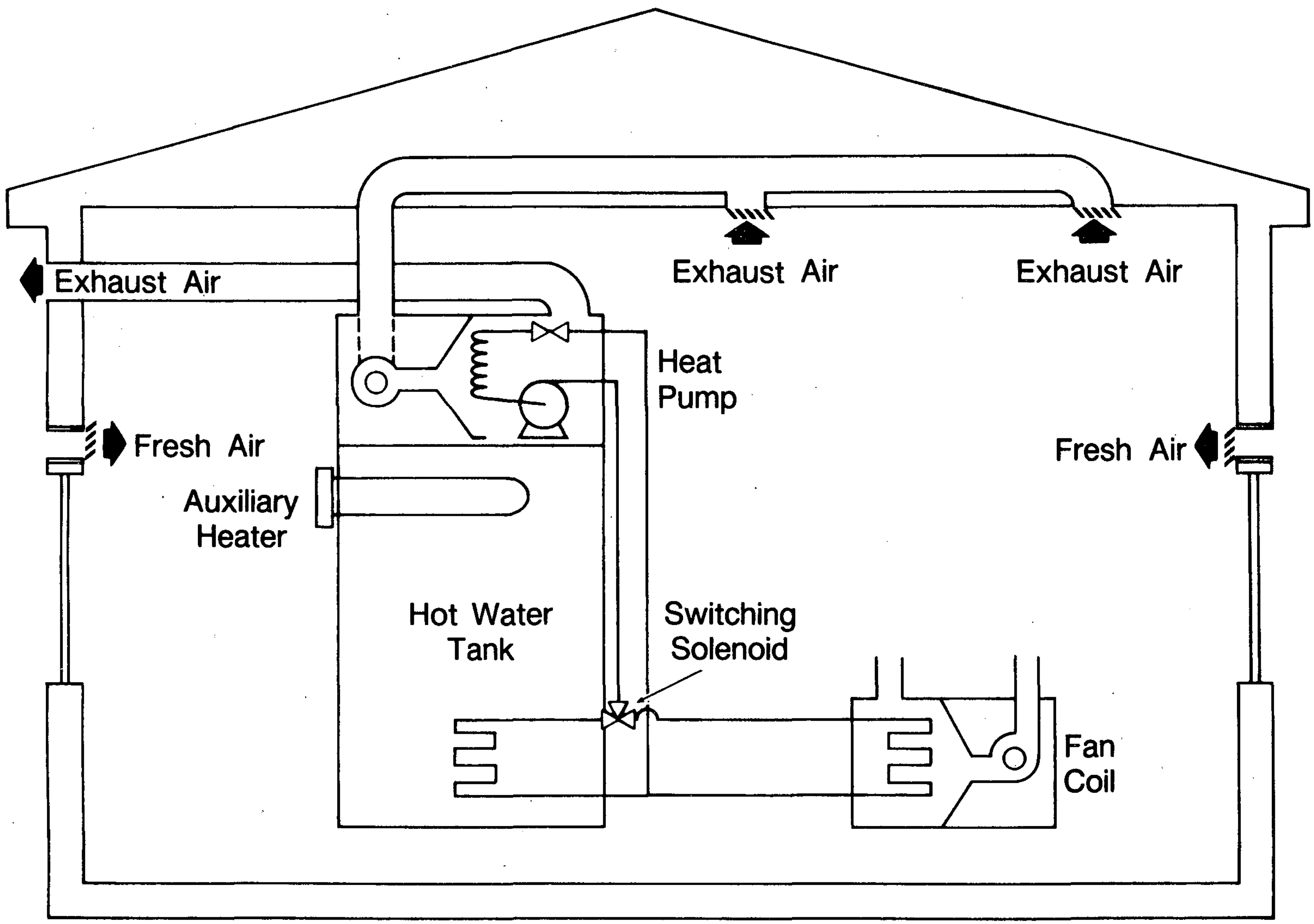

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Figure 1. Application of exhaust-air heat pump for water and space heating.

These two units were selected because they were the only EAHPs scheduled for marketing in North America. The designs of these two units have subsequently been modified and significant changes in performance (probably improvements) may have resulted from these design changes.

Figure 2 shows schematically how the heat pumps were set up for experimental testing; Table 2 lists the instrumentation used; and Table 3 specifies the data logging system. As shown in Fig. 2, the experimental setup included needle valves, ON/OFF solenoid valves, and a programmable timer for setting a desired hot water demand schedule and volume. The two schedules used in this study are shown in Figs. 3 and 4. Demand volume and schedule were set by adjusting the pressure in the supply water tanks and by adjusting the needle valves and opening and closing solenoid valves, as required. The equipment used to condition the air supplied to the heat pump and the water supplied to the heat pump's water tank is described briefly in Appendix A.

All resistance temperature sensors (RTDs) were calibrated against a high-precision NBS-traceable thermometer in a constant-temperature bath. High-precision, highly accurate (estimated accuracy of $\pm 0.1°C$) RTDs were to measure the fan-coil air temperatures and, when no fan coil was operated, for redundant measurement of the temperatures of water entering and exiting the water tank. The thermocouples and the "standard" RTDs, coupled with the data acquisition system showed poorer precision and accuracy. By averaging three consecutive readings for every measurement, the achieved standard deviation was better than 0.1°C over the whole temperature range and the estimated accuracy of thermocouples and the standard RTDs is $\pm 0.3°C$.

The displacement water flow meters were calibrated by weighing the volume of water collected in the calibration tests. For accurate measurement of water volume, two flow meters had to be used with different sensitivities, because the flow rates varied by a factor of 20. For every test, total water volume from the meters was compared to the volume indicated by the change in water level in the supply tank. The error of total water volume in these checks was typically less than 1%.

The orifice plate used for the exhaust-air flow rate determination was checked against a pitot-tube traverse in a branch duct of smaller diameter (to increase the air velocity, and thus the accuracy of the pitot-tube measurement). ASME [3] computational methods were used for orifice plates; the orifice plate system yielded flow rates within 2% of the flow rates determined with the pitot-tube traverse. The differential-pressure transducer used with the orifice plate system was calibrated using a manometer which contained a micrometer for measuring changes in water level.

Before the actual experimental runs, the measurement system was cross-checked in a few special tests. One such test was to start with a full tank of hot water, draw out the entire inventory and replace it with cold water (all water heating OFF), and compare the change in tank internal energy as determined by the tank thermocouples, to the total enthalpy change of the flowing water, as determined by the RTDs and the flow meters. With a measurement interval of 1 minute, the change in tank internal energy agreed with the calculated enthalpy change within 2%. A measurement interval of 45 seconds did not improve accuracy; consequently, most runs were carried out with a measurement interval of 1 minute. Another test compared the measured water enthalpy change to the electric heating energy, when using only the auxiliary electric heater. The result showed an error of only 1.5%. Hence, water enthalpy changes were quite accurately determined by the experimental system used for this investigation.

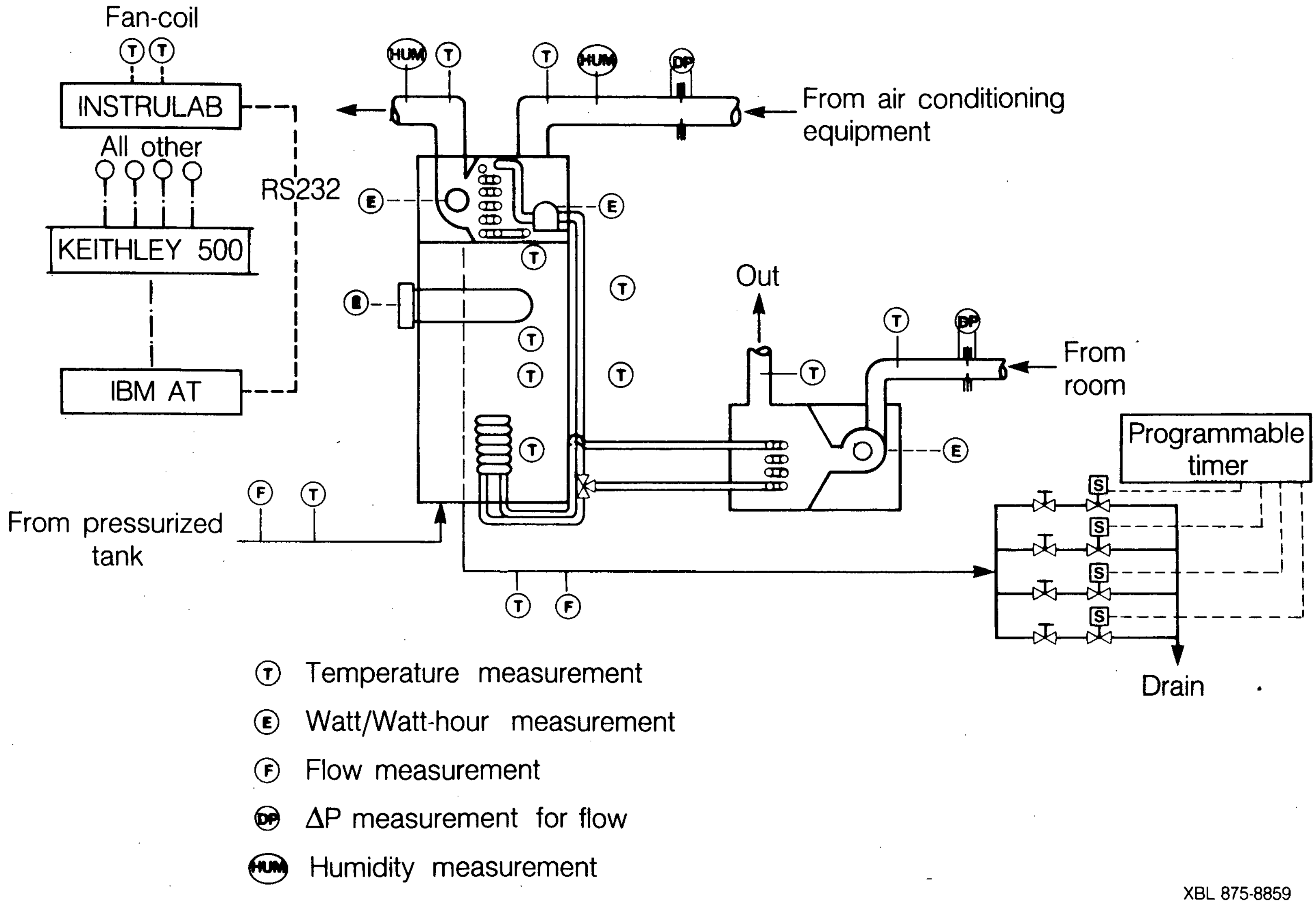

Figure 2. Heat pump experimental system.

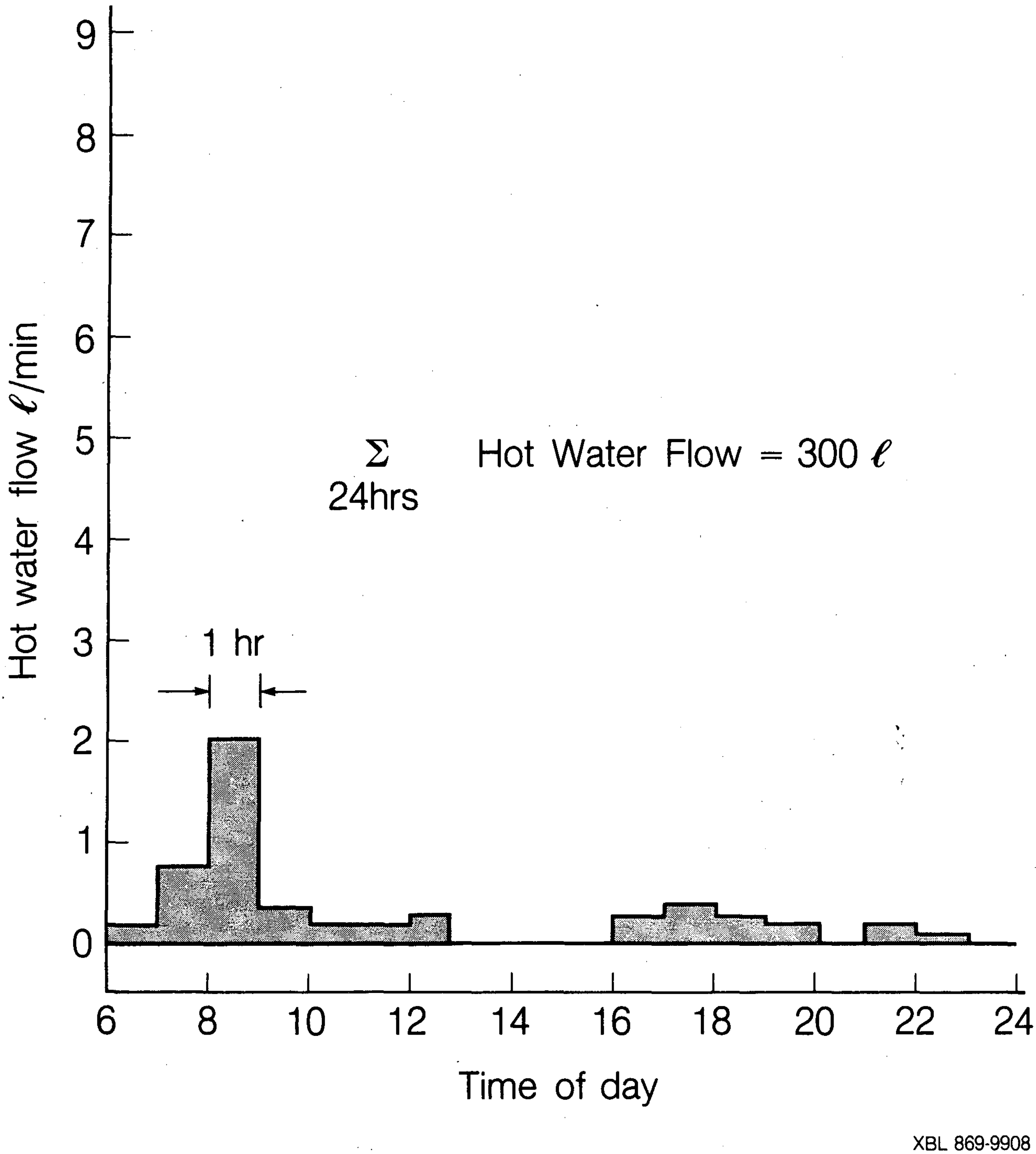

Figure 3. "Continuous" hot water usage schedule.

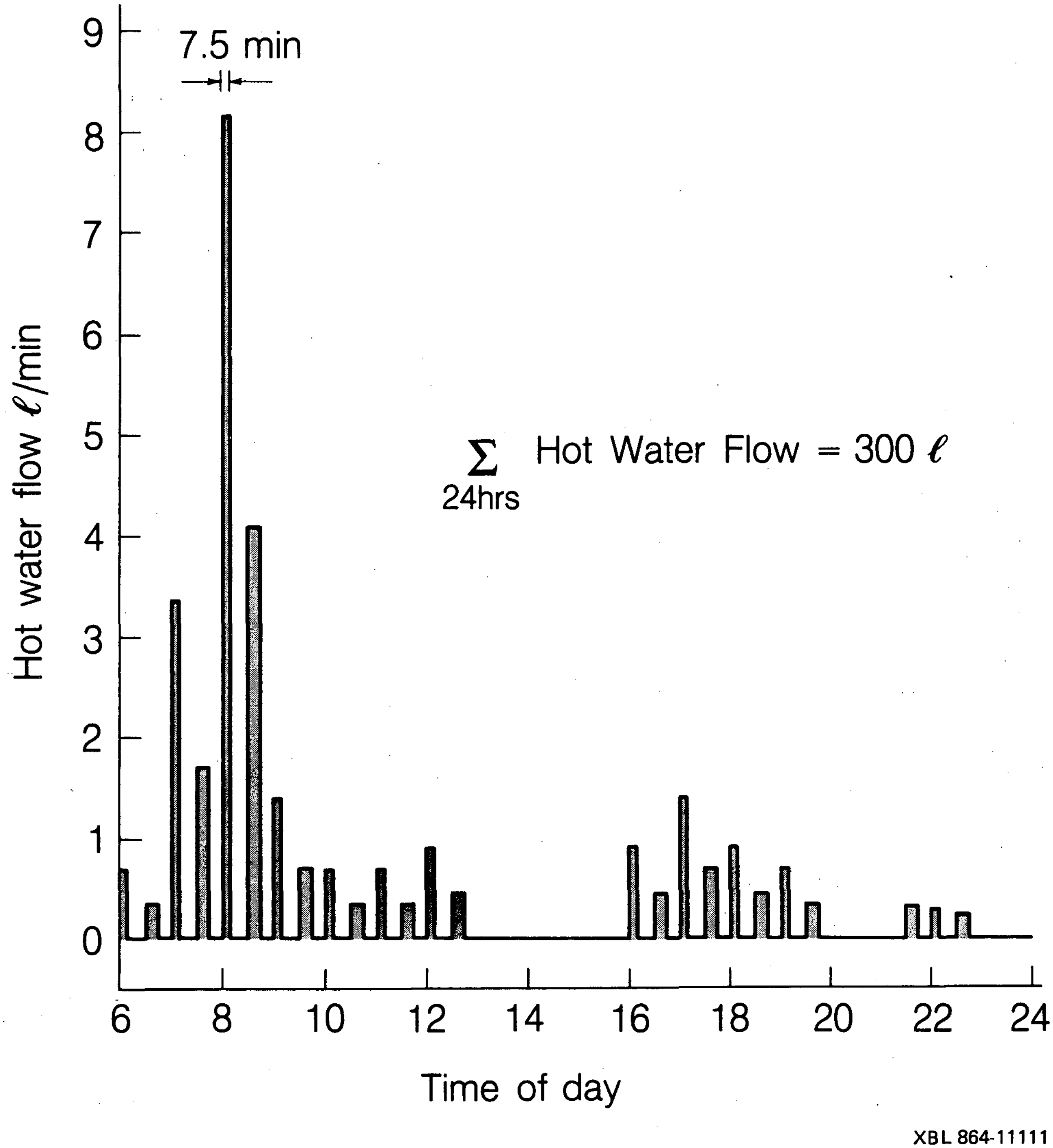

Figure 4. "Intermittent" hot water usage schedule.

EXPERIMENTAL PROCEDURE

The experimental runs with Unit A were divided into three categories: water heating only, space heating only, and combined water and space heating. Unit B was run in water-heating mode only.

In the water-heating mode, the heat pumps were tested in 24-hour runs, each run devoted to the study of one of the following: daily total hot water demand, demand schedule, water inlet and outlet temperature, auxiliary-resistance-heater set point, exhaust-air flow rate, exhaust-air temperature, and exhaust-air humidity. The experimental procedure included a short tank mixing period (by an external pump) at the beginning and end of each run to allow accurate determination of average temperature of water within the tank and, thus, any change in internal energy. Run starting times were selected to be in the middle of the night (simulated time) when there was no water demand, so that the water within the tank had time to stratify thermally before the water draw started at 6:00 a.m. The intermittent demand schedule of Fig. 4 was used in most runs because we believe that it is closer to reality. However, the results were essentially the same when the demand schedule depicted in Figure 3 was utilized.

In the space-heating mode (i.e., all heat delivered to the fan coil), experimental runs were of shorter duration - - typically only six hours because the fan-coil tests were steady state in nature and the fan-coil operation reached steady state quickly. The impact of fan-coil air flow rate and inlet air temperature were studied.

In the combined mode, heat-pump compressor operation was continuous and water heating was given priority (a characteristic of the Unit A control system). The effects of varying water thermostat set point and hot water demand parameters were studied in this mode.

DATA ANALYSIS

A computer-based data logging system recorded the variables listed in Table 2 at any desired multiple of the measurement interval. The software also kept a log of running sums and running sums of squares for the variables listed in Table 4. The running sum of squares allowed later off-line calculation of the variance around the mean value for any sub-period of the total run. The variance provided, for example, a check on the constancy of such input parameters as inlet water temperature or inlet air temperature.

One of the parameters calculated is the heat pump energy balance which is made up of several terms: including the heat transferred from the condenser to the water, the heat transferred from the fan-coil condenser to the air, and the heat extracted by the evaporator from the exhaust air. For a time interval Δt, the heat transferred from the condenser to the water is itself expressed as the sum of several terms. The first term in this sum indicates the total enthalpy of water that exits the hot water tank minus that of water which enters the tank. Subsequent terms account for any change in internal energy of the water within the tank during the run, the conductive heat loss through the tank shell, and the energy supplied to the tank's electric-resistance heater. Symbolically, for a twenty-four hour run, the energy transferred from the condenser to the water is represented by:

$$Q_{Wcond} = \int^{24h} W \cdot C_W \cdot (T_{Wout} - T_{Win})\, dt + \Delta E_{tank} + \int^{24h} U \cdot A \cdot (T_{tank} - T_{env})\, dt - Q_{Res.Heat} \qquad (1)$$

where:

Q_{Wcond}	=	Heat transferred by tank condenser to water
W	=	Water flow rate
C_W	=	Average water heat capacitance
T_{Wout}	=	Water outlet temperature
T_{Win}	=	Water inlet temperature
ΔE_{tank}	=	Internal energy of tank water at the end of run, minus that at the beginning of the run
$U \cdot A$	=	Heat-loss coefficient times area of tank surface
T_{tank}	=	Average tank water temperature
T_{env}	=	Average environment temperature
$Q_{Res.Heat}$	=	Electric energy input to resistance heater.

The computation of the heat transferred by the fan-coil condenser to the air is somewhat simpler. The first term represents the total amount of energy transferred to the air which flows through the fan coil. The second term accounts for the energy supplied to the fan-coil fan which also ends up heating the air stream. Symbolically we have:

$$Q_{FCcond} = \int_{\substack{\text{when comp ON}\\ \text{in FC mode}}}^{24h} F_{FC} \cdot C_{air} \cdot (T_{FCout} - T_{FCin})\, dt - \int_{\substack{\text{when comp ON}\\ \text{in FC mode}}}^{24h} P_{FCfan}\, dt \qquad (2)$$

where:

Q_{FCcond}	=	Heat transferred by fan-coil condenser to air
F_{FC}	=	Air flow rate through fan coil
C_{air}	=	Average air heat capacitance
T_{FCin}	=	Air inlet temperature to fan coil
T_{FCout}	=	Air outlet temperature from fan coil
P_{FCfan}	=	Electric power of fan coil fan (actually integrated by transducer).

The final expression required for a heat pump energy balance describes the heat extracted by the evaporator from the exhaust air stream. The dominant term is the sensible energy removed from this air stream which results in a reduction in air temperature. Added to this are terms describing the heat of condensation of the condensate collected during the time interval and the power supplied to the air stream by the exhaust fan. Symbolically:

$$Q_{Evap} = \int_{(\text{When } T_{Eout} < T_{Ein})}^{24h} F_E \cdot C_{air} \cdot (T_{Ein} - T_{Eout})\, dt + Q_c + \int_{\text{When Comp On}}^{24h} P_{Efan} \cdot dt \qquad (3)$$

where:

Q_{Evap}	=	Heat extracted by evaporator from exhaust air
F_E	=	Exhaust air flow rate
C_{air}	=	Average air heat capacitance
T_{Eout}	=	Exhaust air outlet temperature
T_{Ein}	=	Exhaust air inlet temperature
Q_c	=	Heat of condensation of condensate collected in 24 hours
P_{Efan}	=	Electric exhaust fan power (integrated by transducer).

The integrals of Eqs (1), (2), and (3) were calculated as discretized sums as discussed with reference to Table 4. The first integral of Eq (3) is integrated over all times with $T_{Eout} < T_{Ein}$ (instead of just times when the compressor is ON) since the evaporator mass introduced significant thermal transients: after the compressor shut off, T_{Eout} remained below T_{Ein} for about 15 minutes, implying warmup of the evaporator mass. An equivalent amount of energy was assumed to be extracted from the evaporator mass when compressor operation started.

Equations (1), (2), and (3) were used for the energy balance, EB:

$$EB = \frac{Q_{Wcond} + Q_{FCcond}}{Q_{Evap} + E_{Comp}} \tag{4}$$

where E_{Comp} = Electrical energy input to compressor (integrated by transducer).

The parameter "EB" should have a value close to unity. Heat losses from the compressor shell and the refrigerant tubing are not accounted for in the energy balance but were minimal. The Unit A compressor did not run very hot, and most of the Unit B compressor heat loss occurred to the exhaust air stream, and was thus recovered.

The calculated condenser and evaporator heat transfer also allowed the heat pump coefficient of performance (COP) to be computed according to

$$COP = \frac{Q_{Wcond} + Q_{FCcond}}{E_{Comp}} \tag{5}$$

where, in the water-heating mode $Q_{FCcond} = 0$, and in the space-heating mode $Q_{Wcond} = 0$.

The energy savings in any one run could be equated to the evaporator heat transfer Q_{Evap}; however, the following definition was used:

$$\text{Energy Savings} = Q_{Evap} - \int^{24h} P_{Efan} \cdot dt \tag{6}$$

i.e., the energy savings were debited with the energy required for 24-hour operation of the exhaust fan. This debit amounted to 1.7 kWh per day for Unit A and 1.2 - 2.7 kWh per day for Unit B (function of exhaust-air flow rate).

EXPERIMENTAL RESULTS

The major results are discussed under the subheadings; additional details are given in Appendix B, which contains the primary experimentally-derived performance curves for the heat pumps, together with a brief description of each figure.

1. General Results

Tank heat losses were determined by measuring the decline of the tank-average water temperature with time. The calculated average product of the heat transfer coefficient and the tank surface area (U·A) for use in Eq (1) was 2.7 W/°C for Unit B and 2.8 W/°C for Unit A. In practice, both of these heat transfer coefficients yielded a heat loss of about 80 Watts.

The energy balance, Eq(4), was 98.6 $\pm$ 1.5% and 97.9 $\pm$ 3.6% for all Unit B runs and all Unit A runs, respectively. The standard deviations of 1.5 and 3.6% are quite acceptable (the higher standard deviation for the tests with Unit A is due to the fan-coil runs that had less precision than the water heating runs), but both series of runs produced a clear bias in the heat balance: approximately 2% of the condenser heat transfer was unaccounted for on the evaporator side. This bias is possibly explained by the fact that the air compartment after the evaporator was poorly insulated, and therefore had a significant heat gain from the environment that was unaccounted for. Consequently, condenser-side heat transfer results are considered to be slightly more accurate.

A consequence of the relatively good energy balance is that EB in Eq (4) can be set equal to one, resulting in the following expression for Q_{Evap} that can be used in Eq (6) when predicting EAHP performance:

$$Q_{Evap} = (Q_{Wcond} + Q_{FCcond})\,(1 - \frac{1}{COP}) \qquad (7)$$

A practical result of a general nature is that both heat pumps had problems with air tightness around the evaporator: Unit B had a leaky air seal in the evaporator access door while Unit A leaked at the slide valve for exhaust-air flow rate adjustment. Elimination of these leaks was necessary for this study, but would not be a routine practice in actual field installations. Hence, this aspect offers the manufacturers an opportunity to improve their products.

2. Water-Heating Results

Figures 5 and 6 summarize the primary experimental data from tests when the EAHPs are used only for water heating. The experimental data indicate that the performance of Unit B can be correlated to the time-average (average when compressor is ON) temperature of water in the bottom of the tank and that the performance of Unit A can be correlated to the time-average temperature of all water within its tank (hereafter referred to as the tank-average temperature). These correlations hold despite variations in water demand volume, demand schedule, and water inlet and outlet temperature. The correlating temperatures are the ones expected from a thermodynamic point of view when one considers the condenser design of the two units: a bottom coil design for Unit B and a tank wrap-around design for Unit A. What was not expected is the linearity of the correlations between COP and water temperature over the range of temperatures studied. This linearity allows time-averaged values to be used without significant error.

Figures 5 and 6 also illustrate the influence of the second most important variable affecting water heating COP: exhaust-air flow rate. An increase in the exhaust-air flow rate is seen to increase the COP, a result expected on the basis of the increase in average evaporator temperature with air flow rate. Correlation equations which fit the experimental data are included on Figs. 5 and 6. Interestingly, the two heat pumps show significantly different sensitivities to their respective correlating temperature: Unit A has a slope approximately double that of Unit B. A higher temperature difference between the condenser and the water for Unit B is one possible explanation for the different slopes. However, no refrigerant pressure or temperature measurements were made to confirm this potential explanation.

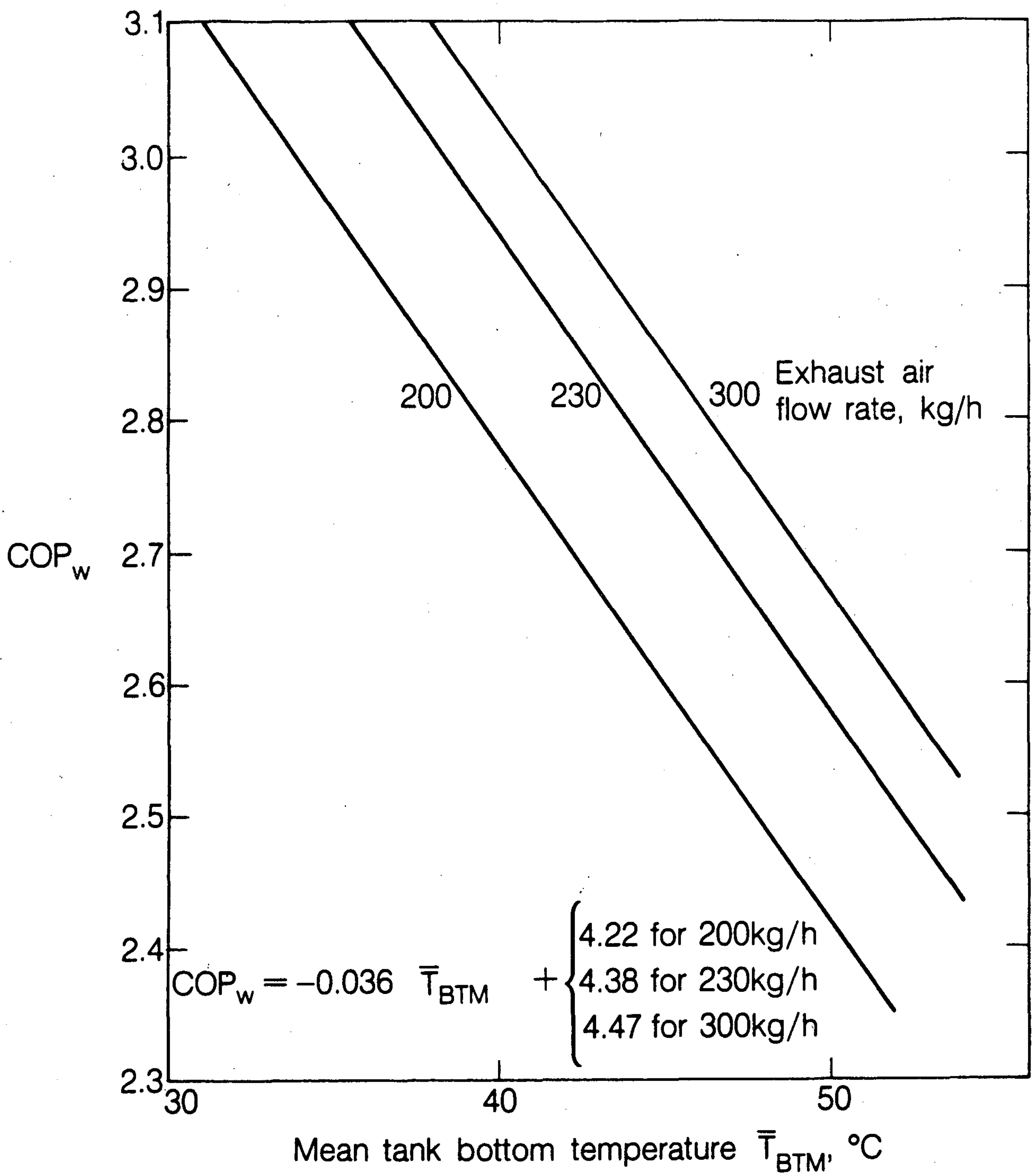

Figure 5. Unit B heat pump coefficient of performance for water heating (COP_w). The exhaust air temperature is 21°C and the exhaust air relative humidity is 50%.

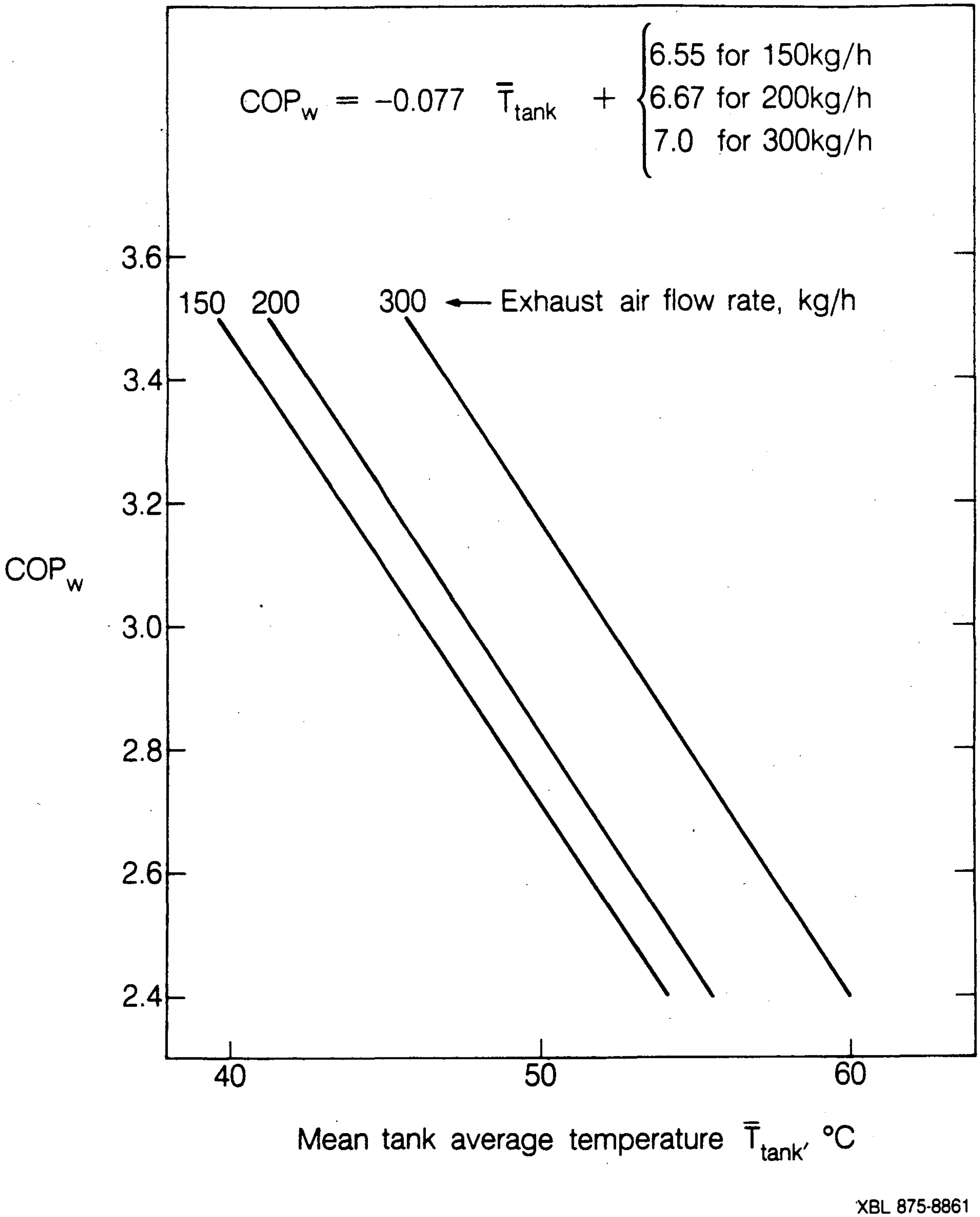

Figure 6. Unit A heat-pump coefficient of performance for water heating (COP_w). The exhaust air has a temperature of 21^{o}C and relative humidity of 50%.

Before comparing Figs. 5 and 6 quantitatively, the correlating temperatures (tank-bottom temperature and tank-average temperature) must be related to some common variable. The relationship between the COP-correlating temperature and the average hot water delivery temperature is shown in Fig. 7 for Unit B and in Fig. 8 for Unit A. These relationships are indicators of the degree of thermal stratification in the water tanks. Discounting the out-lying data points from tests with a low water demand, the degree of stratification can be expressed as approximately a constant ΔT for each heat pump: 9.8°C for Unit B and 5.5°C for Unit A. These empirical values are important. For example, if an average hot water temperature of 52.5°C is desired, then Unit B should be operated with a mean tank-bottom temperature of only 42.7°C and Unit A with a mean tank-average temperature of 47°C. Now the two heat pump COPs can be compared, yielding 2.68 for Unit B and 3.05 for Unit A (using an exhaust-air flow rate of 200kg/h). Higher water temperatures reduce both COPs, but Unit A's COP falls much faster. At a hot water delivery temperature of 61.5°C, either unit would have a COP of only 2.36. However, this temperature exceeds the range of practical interest.

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 desired, it is more energy efficient to use the heat pump to heat the incoming water only to an intermediate temperature and to carry out the final heating with the resistance heater near the top of the tank. (This tank section does not influence the tank-average temperature for condenser operation appreciably.) For example, if the supply water temperature is 15°C, optimal heat-pump operation is to heat the water to no more than 60.5°C using the heat pump (tank-average temperature = 55.0°C) and to use electric resistance heating from there on. In general, the optimal heat-pump outlet temperature T^*_{Wout} is given by

$$T^*_{Wout} = -\frac{\beta}{\alpha} - \sqrt{(\alpha \cdot T_{Win} + \beta) / \alpha^2} \qquad (8)$$

where α and β are the parameters in

$$COP_W = \alpha \cdot \overline{T}_{Wout} + \beta. \qquad (9)$$

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 electrical resistance heater in Unit A, i.e., the total energy consumption curve is flat near the minimum. However, this is not true for Unit B because of the much higher T^*_{Wout} of 71°C (due mainly to the smaller α).

Figures 9 and 10 show that the COP-correlating temperatures can be estimated if the settings of the heat pump thermostats are known. However, as shown by Figs. 9 and 10, the difference between these two temperatures is not independent of water demand, especially for Unit B.

The COP results discussed above are directly applicable to calculation of energy savings in water heating mode according to Eqs (6) and (7). Here, energy savings scale with Q_{Wcond} as a first order effect, i.e., increasing water heating demand (for example,) increases energy savings proportionally. A smaller effect is the increase in energy savings with increased COP, given by Eq(7). A third, and negligible, effect on energy savings results from the variation in compressor power with varying operating conditions. If, for example, the daily water demand is 300 liters, supply water temperature 15°C, and average hot water delivery temperature 52.5°C, Unit B saves 2,810 kWh per year and Unit A saves 3,060 kWh per year. Here, the heat pump condenser load, $Q_{W_{cond}}$ in Eq (7), includes not only the direct water heating load but also a constant heat loss from the tank of 80W. (Both heat-pump exhaust fans are assumed to require 72W for this calculation.)

3. Space-Heating Results

Figure 11 shows the main results obtained with the Unit A space-heating fan-coil condenser. The fan coil air inlet temperature has an effect on COP similar to the effect of tank-average temperature in the water-heating mode. Interestingly, the slopes of the COP curves are very similar: -0.077 in the water-heating mode and -0.073 in the space heating mode. However, the sensitivity of the COP to the air inlet temperature is less important from a practical point of view because the fan coil is expected to experience an inlet air temperature of around 20°C most of the time. What is more important is the strong effect of fan-coil air flow rate, shown clearly in Fig. 11. At an air flow rate of 400 kg/h and with an inlet air temperature of 20°C, a COP of 3.6 is obtained. Higher air flow rates than about 400 kg/h are not necessary because, although the COP continues to climb for a while, the compressor power starts to drop, implying a limitation in the capacity of the heat pump. For details on this issue, reference is made to Appendix B, Figs. B25 and B26.

Energy savings can be calculated according to Eqs (6) and (7). However, contrary to the water-heating case, the space heat sink is more difficult to define because the building itself is part of a dynamic system influenced by the climate. In addition, the fan-coil system has no significant capacitance for energy; space-heating load cannot, therefore, be shifted in time. As an interesting upper limit on possible energy savings, the compressor and fan-coil can be assumed to be ON continuously for 24 hours. Then, the daily maximum energy savings is given by

$$\text{Max. Energy Saving} = \int^{24h} (COP - 1)\, P_{comp} \cdot dt - \int^{24h} P_{Efan} \cdot dt \tag{10}$$

where P_{comp} = Compressor power.

Using the average compressor power of Unit A of 570 W and a COP_{FC} of 3.6, the maximum energy savings for 24-hour operation of the fan coil is 34 kWh/day. If the Unit B water-heated fan coil is assumed to operate at the same COP as obtained in water-heating mode (COP = 2.68), the corresponding savings with this unit is 29 kWh/day. These maximum achievable energy savings can be compared to the average daily heating requirements for a well-insulated house (built to the Model Conservation Standard used in the Bonneville Power Administration area) during the heating season (December - February in Portland and November - March in Spokane and Missoula): 32 kWh in Portland, 43 kWh in Spokane, and 46 kWh in Missoula [2]. Consequently, the fan-coil units can supply a substantial portion of the heating requirement in the Pacific Northwest. These maximum achievable energy savings for space heating should also be compared to the energy savings for the water-heating example discussed above: on the order 8 kWh/day for both units.

4. Water- and Space-Heating Results

In the combined mode, Unit A performed as expected, based on the sum of its two parts. Figure 12 shows the comparison between experiment and model for the case of a relatively low fan-coil air flow rate producing a COP_{FC} of only 2.7. The model is simply a superposition of its two parts:

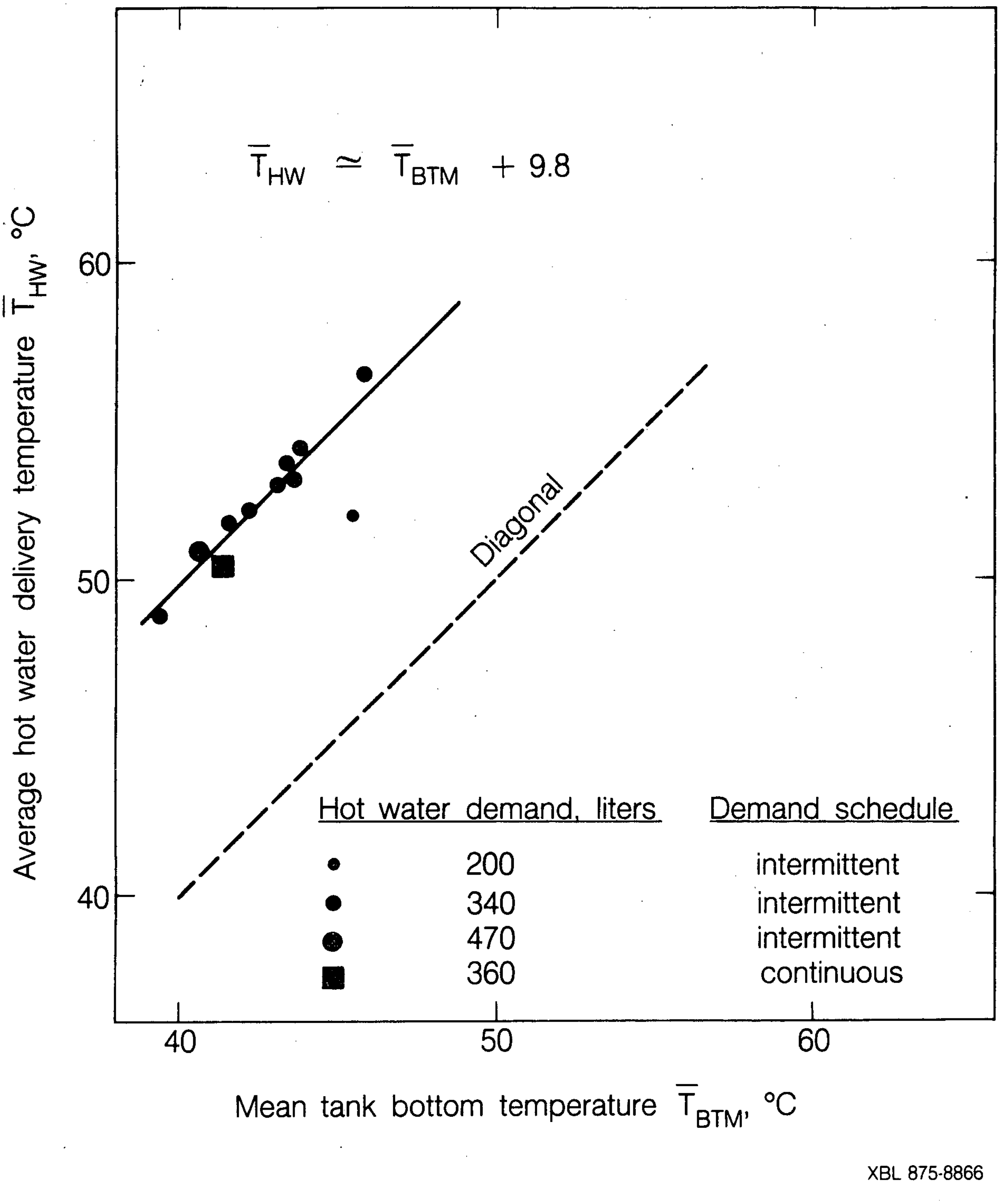

Figure 7. Stratification in the water tank of Unit B.

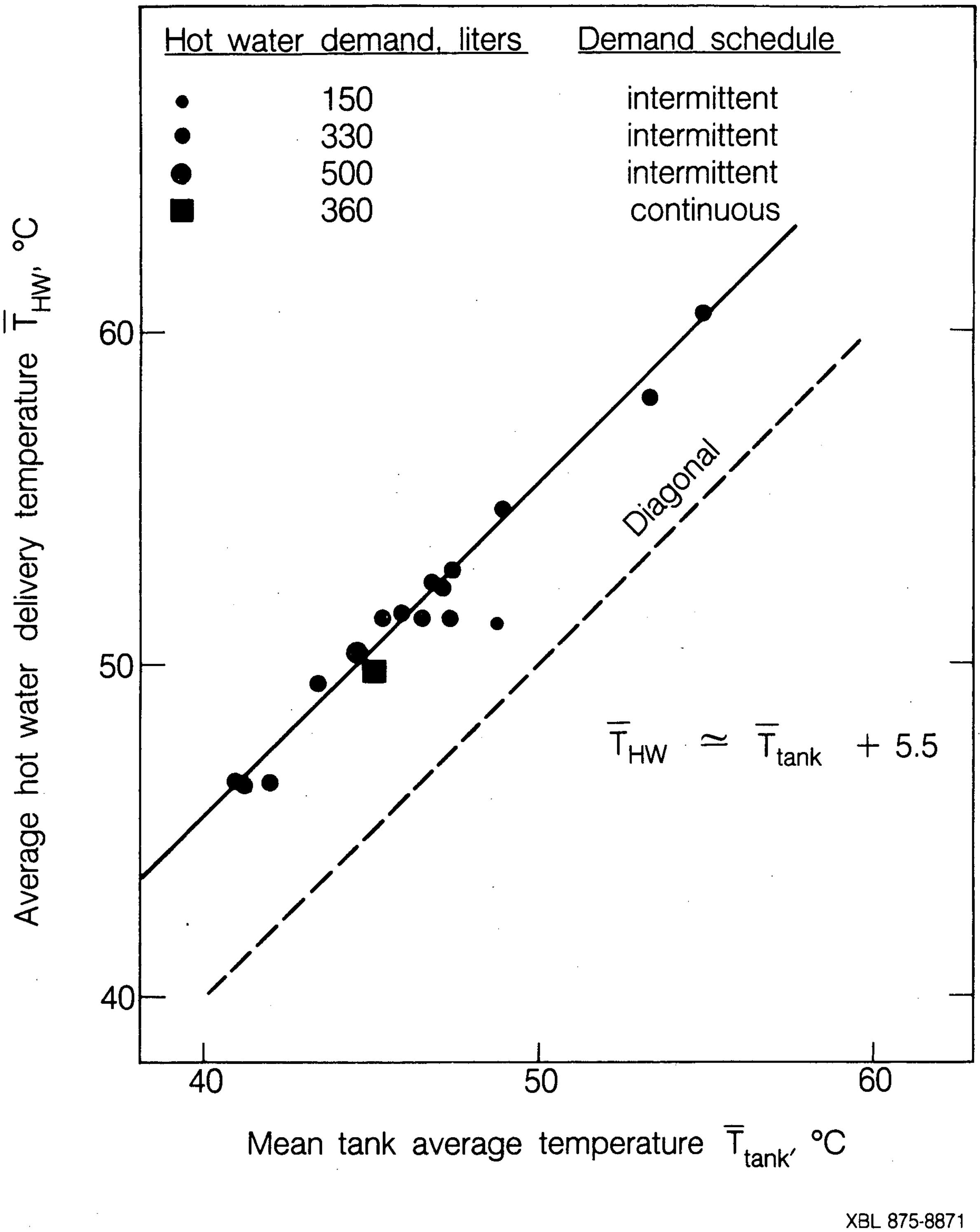

Figure 8. Stratification in the water tank of Unit A.

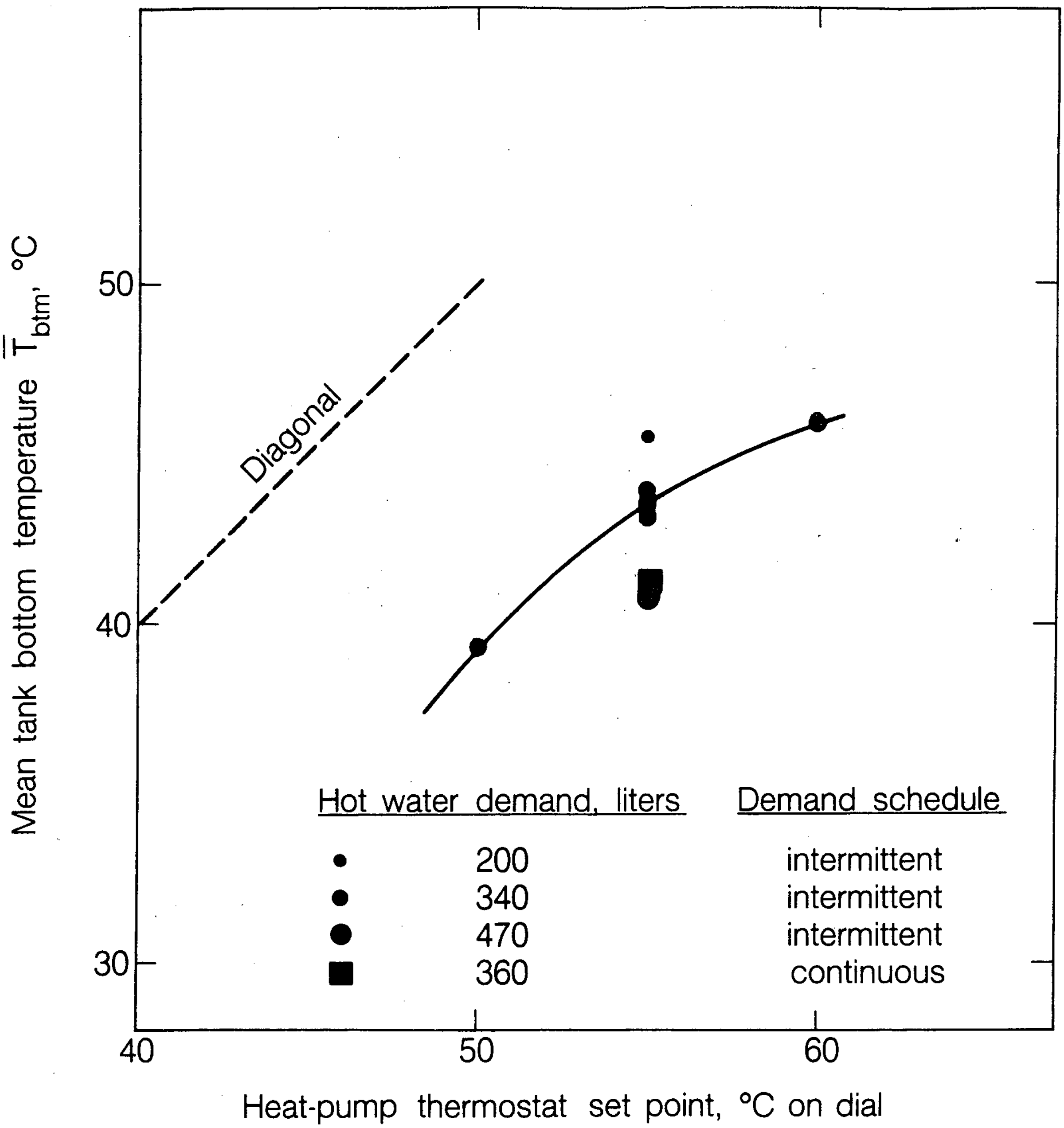

Figure 9. Temperature at the bottom of the water tank of Unit B as a function of thermostat set point.

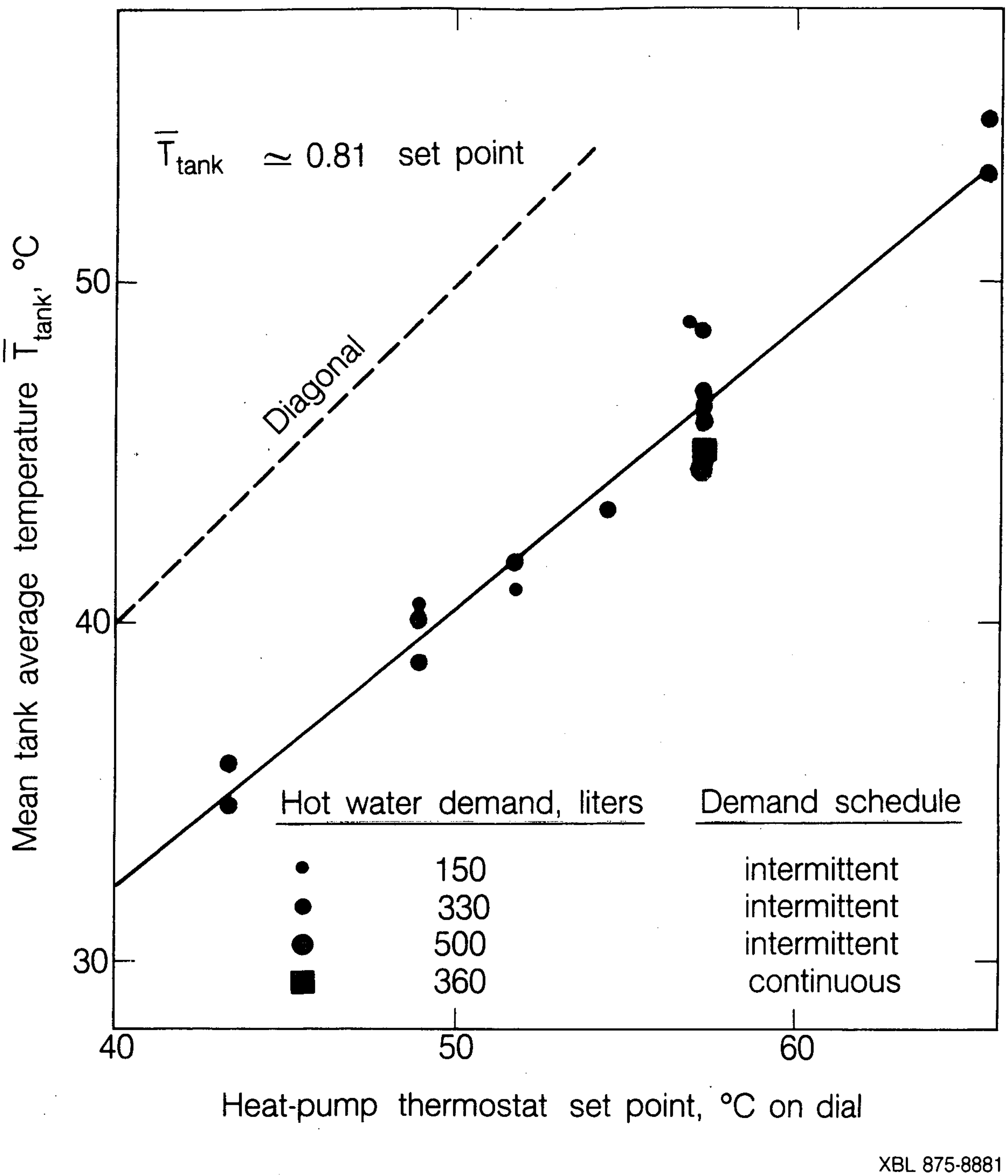

Figure 10. Mean temperature of water in tank of Unit A as a function of thermostat set point.

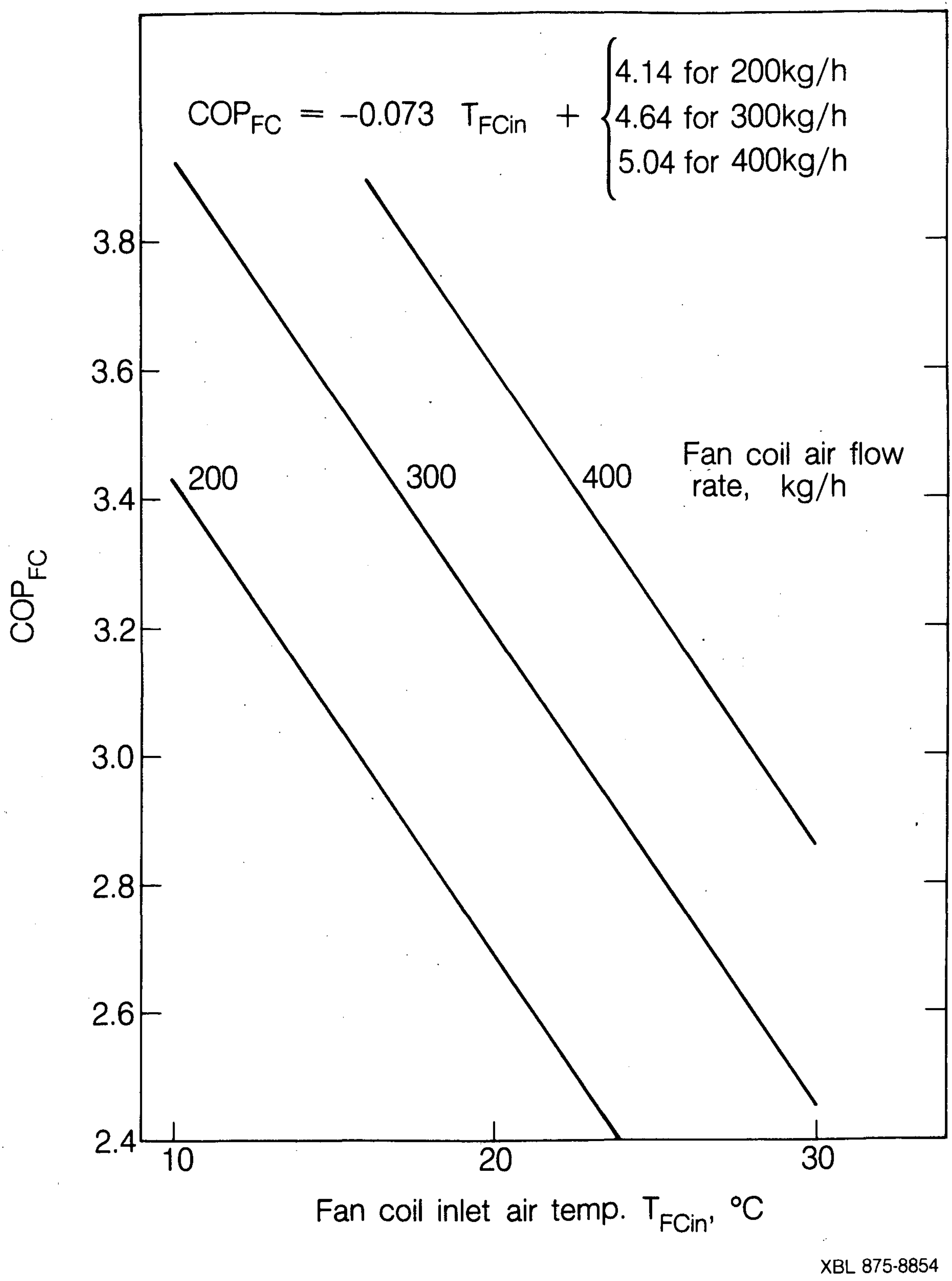

Figure 11. Unit A heat pump coefficient of performance for space heating (COP_{FC}). The exhaust air has a flow rate of 200 kg/h, temperature of 21°C, and relative humidity of 50%.

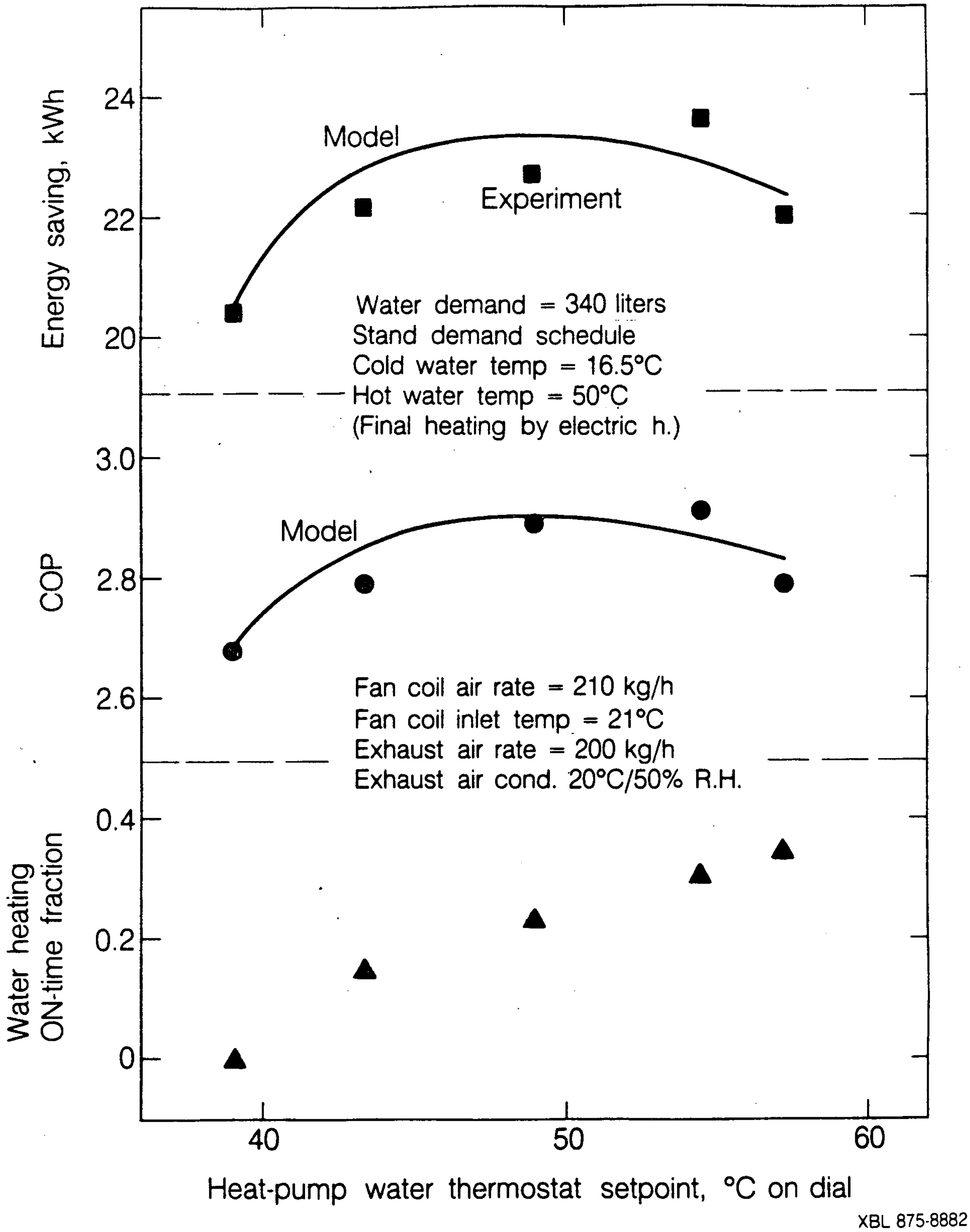

Figure 12. Unit A heat-pump performance in combined water-space heating mode with a 24-hour ON time.

$$COP_{combi} = x_{FC} \cdot COP_{FC} + x_W \cdot COP_W \tag{11}$$

where x_{FC} = Fan coil ON time fraction
x_W = Water heating ON time fraction.

For the case of Fig. 12, the compressor ran continuously so that $x_{FC} + x_W = 1$. The calculated energy savings are given by the integrated form of Eq (11):

$$\text{Max. Energy Savings} = (COP_{combi} - 1) \cdot E_{Comp} - \int^{24h} P_{Efan} \cdot dt \tag{12}$$

where the 24-hour compressor energy consumption E_{Comp} is approximately a constant equal to 13.2 kWh.

For optimal applications of the heat pump in combined mode, the objective is to maximize the first term of the expression for the energy savings which is the energy recovered by the heat pump, where:

$$\text{Energy Recovered} = \int^{24h} (COP - 1) \cdot P_{Comp} \cdot dt \tag{13}$$

The foremost task in maximizing energy recovery expressed in Eq (13) is to keep the compressor ON as much as possible, i.e., keep P_{Comp} from dropping to zero. This objective is met by using the heat pump in fan-coil mode when there is demand for space heat, and deferring any conflicting water heating demand until later. The second task, which is of much lesser importance, is to maximize COP in Eq (13). This again favors space-heating most of the time because the water tank must be quite cold before water-heating COP can compete with the high space-heating COP.

These arguments speak against the current Unit A control system that has water-heating priority. Instead, space heating should have priority over water heating until the water temperature falls below an override water temperature (below which water heating gets priority). Such a control system could be implemented by two thermostats in the water tank: one low in the tank with a single set point for low-priority water heating (set at the desired tank temperature and controlling the compressor only when the fan-coil thermostat is satisfied), and a second thermostat high in the tank. This second thermostat would have two set points: one at the desired hot water temperature and controlling the electric-resistance heating element, and the other set at a lower temperature and acting to switch priority to water heating (override action).

UPDATE OF TRNSYS MODEL AND REGIONAL ASSESSMENT

As mentioned in the introduction of this report, preliminary evaluations of the ventilation rates obtained with EAHP systems, their impact on total residential energy consumption, and their cost effectiveness have been completed [1,2]. The preliminary evaluations were made using a modified version of the TRNSYS (Transient System Simulation) computer program to model the house and EAHP system. The modeled EAHP system has a configuration similar to Units A and B except its condenser is external to the water tank and water is pumped from the bottom of the water tank through the condenser, and back into the tank near its top. In addition, the optional fan coil in the modeled EAHP system uses hot water from the tank as the heat source, in contrast to the fan coil of Unit A which is heated directly by the refrigerant. The configuration of the modeled EAHP system was based on the only EAHP for which substantial performance data could be obtained.

The results of the preliminary evaluations for the Pacific Northwest (Preliminary Regional Assessment), were found to depend substantially on the thermodynamic performance (e.g. COP) of the EAHP, for which very limited data were available. In addition, we were not confident that the simple TRNSYS model of water within the hot water tank (water-tank model) would accurately predict the temperature of water that passed through the condenser. Errors in this prediction could also substantially affect the predicted overall energy performance of the EAHP system.

The experimental data obtained indicates that the TRNSYS program underestimates the degree of thermal stratification in the water tank. For example, TRNSYS simulations typically indicated a temperature difference between hot water delivery temperature and tank bottom temperature of only 3.5°C. The corresponding temperature difference from our experimental evaluations of Unit B, which is more similar to the modeled EAHP than Unit A, is 9.8 °C. Thus, an improved water tank model is required to model some EAHP configurations. Alternately, empirical models of EAHP/water tank systems, such as presented here, could be utilized within the TRNSYS program. However, for Unit A, an inaccurate prediction of thermal stratification is of little importance because Unit A performance can be correlated to the average temperature of water within the tank.

Since the existing TRNSYS water-tank model is adequate for simulations of Unit A, a fairly accurate update to the preliminary predictions of energy savings can be made by simply increasing the COP of the simulated EAHP system by 30 percent to coincide with the measured COPs of Unit A. This procedure is conservative since it does not account for the higher COPs of Unit A when it is used with the fan-coil condenser to heat air. Table 5 provides revised estimates of yearly energy savings and costs of conserved energy (CCE) for houses with an EAHP and fan coil built to meet the Model Conservation Standard used in the Bonneville Power Administration area. The energy savings are approximately 15 percent larger than previously estimated and CCE values are decreased by approximately 13 percent. These energy savings and CCE values are based on comparing EAHP-ventilated houses that have a winter-average ventilation rate of 0.5 air changes per hour (ach) to houses with natural infiltration at a winter-average rate of 0.7 ach. The justification for comparing houses with different average ventilation rates has been provided previously [2], and is primarily based on the variability of natural infiltration rates compared to the stable rate of ventilation provided by EAHPs. Because the revised CCE values are comparable to, and in some cases smaller than, the current electricity prices in the cities for which the predictions are made, the use of EAHP systems seems to be relatively attractive. However, performance data and costs obtained in actual field studies are required for definitive conclusions.

Larger increases in energy savings could be obtained if EAHP control systems are modified as previously suggested so that priority is given to providing heat to the fan coil whenever water tank temperatures are above a specified value. Based on an analysis of data from previous TRNSYS simulations, we estimate that the compressor on time fraction could be approximately 0.9 during the heating season with an optimal control system-- this compares to a 0.6 on time fraction predicted with current control systems. Such an increase in compressor operation would result in an additional energy savings of approximately 1,000 kWh in Portland and 1,500 kWh in Spokane and Missoula. Consequently, improved control systems appear to be a very worthwhile goal.

CONCLUSIONS

One of the EAHP systems studied experimentally has a COP that is approximately 30 percent greater than the COP used in a preliminary modeling effort. This improved COP is expected to result in approximately a 15 percent increase in energy savings and 13 percent decrease in cost of conserved energy compared to previous predictions.

The experimental data also indicate that an improved water tank model for the TRNSYS computer program, which yields more accurate predictions of the degree of thermal stratification in the water tank, is needed for accurate physical simulations of some EAHP configurations. However, empirical models of EAHP/water tank systems, such as presented in this paper, may be adequate for many applications.

Perhaps most importantly, the experimental data and analysis provided in this report indicate that large increases in energy savings (1,000-1,500 kWh per year) might be obtained if EAHP control systems were modified so that more heat is supplied to the indoor air and greater advantage is taken of the energy storage capabilities of the water tank.

ACKNOWLEDGMENTS

This work was supported by the Bonneville Power Administration under contract No. DE-AI79-86BP60326 and the Assistant Secretary for Conservation and Renewable Energy, of the Office of Building and Community Systems, Building Systems Division of the U.S. Department of Energy,under Contract No. DE-AC03-76SF00098.

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1. P.H. Wallman, B.S. Pedersen, R.J. Mowris, W.J. Fisk and D.T. Grimsrud, "Assessment of Residential Exhaust-Air Heat Pump Applications in the United States", *Energy - The International Journal* **12**, No. 2 (1987).

2. P.H. Wallman, W.J. Fisk and R.J. Mowris, "Preliminary Assessment of Residential Exhaust-Air Heat Pump Applications in the Pacific Northwest", LBL-Report 22234, Lawrence Berkeley Laboratory, Berkeley, CA (1986).

3. *Fluid Meters: Their Theory and Application*, p. 201, American Society of Mechanical Engineers, N.Y. (1971).

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, liters	260	220
Avg. Compressor Power, Watts	570 (base run)	750 (base run)
Exhaust Fan Power, Watts	72 (fixed speed)	120 (max, variable speed)
Resistance Heater Power kW	1.7	1.2
Fan Coil Fan Power, Watts	72	-

Table 2. Instrumentation Used in Heat Pump Experiments.

PARAMETER	INSTRUMENT OR METHOD
<u>Temperatures</u>	
Cold Water Inlet Hot Water Outlet Exhaust Air Inlet to Evaporator Exhaust Air Outlet from Evaporator	By Omega RTDs
Fan Coil Air Inlet Fan Coil Air Outlet	By Instrulab High Precision Platinum RTDs + RS232 Link
Tank Water Temperature, Level 1 Tank Water Temperature, Level 2 Tank Water Temperature, Level 3 Tank Water Temperature, Level 4 Tank Ambient Temperature, Level 1 Tank Ambient Temperature, Level 2	By Type T Thermocouple
<u>Flow Rates</u>	
Cold Water Hot Water	By Kent Displacement Flow Meters + Counters (Different Sensitivities)
Exhaust Air Fan Coil Air	By orifice plates + Validyne Pressure Transducers
<u>Volume</u>	
Cold Water Feed	By Manual Reading of Sightglass on Water Supply Tank
Evaporator Condensate	By Manual Reading of Weight on Scale
<u>Humidity</u>	
Exhaust Air Inlet to Evaporator	By Vaisala Humidity Probe and manual Wet Bulb and Dry Bulb Thermometer Readings
Exhaust Air Outlet from Evaporator	By EG+G Dew Point Analyzer
<u>Power/Energy</u> (Watt/Watt-hour)	
Compressor Exhaust Fan Auxiliary Resistance Heater Fan Coil Fan	By OSI Watt/Watt-hour Transducers

Table 3. Keithley 500 Data Acquisition System.

Hardware	
KEITHLEY 500	Mainframe & Interface Card for IBM AT Computer
AIM 1	8-Channel Analog Input Module
ADM 2	14-Bit D/A Converter Module
AIM 6	4-Channel RTD Analog Input Module
AIM 7	16-Channel TC Analog Input Module
PIM 1	Pulse-Counting Module
Software	
SOFT 500	Version 4.0

Table 4. Software - Integrated Parameters.

<u>Running Sum</u>

Time
Compressor ON Time
Water Flow ON Time
Fan-coil ON Time
Water demand mass
Water demand enthalpy change
Exhaust air mass
Exhaust air enthalpy change
Fan-coil air mass
Fan-coil air enthalpy change
Tank heat loss

<u>Running Sum and Sum of Squares</u>

Cold water inlet temperature (when water is flowing)
Hot water outlet temperature (when water is flowing)
Tank average temperature (when compressor is ON and fan coil is OFF)
Tank bottom temperature (when compressor is ON and fan coil is OFF)
Fan-coil air inlet temperature (when fan coil is ON)
Fan-coil air outlet temperature (when fan coil is ON)
Exhaust air temperature (when compressor is ON)
Exhaust air humidity (when compressor is ON)

Table 5. Updated yearly savings and cost of conserved energy realized with exhaust-air heat pump system plus fan coil in a house built to the Model Conservation Standard with a winter-average ventilation rate of 0.5 air changes per hour relative to the same house with natural infiltration at a winter-average rate of 0.7 air changes per hour.

Location	Yearly Energy Saving* kWh	Cost of Conserved Energy ¢ / kWh
Portland	5,760	4.6
Spokane	6,870	3.8
Missoula	8,090	3.5

* The following yearly energy savings are obtained by a simple exhaust-fan (no heat recovery) with a constant ventilation rate of 0.5 ach: 870 kWh in Portland, 1390 kWh in Spokane, and 2290 kWh in Missoula.

APPENDIX A

The equipment used for conditioning both the exhaust air and the cold water is described as follows: room air was heated, blown by an auxiliary fan first across a humidifier, then through a constant-temperature cooling coil and finally through an electric heating element. The constant-temperature cooling coil controlled the dew point of the exhaust air by removing excess moisture from the air stream. The coil was maintained at desired temperature by means of a standard chiller unit that recirculated a water-glycol mixture through this coil. The final electric heating element reheated the air to the temperature desired at inlet of the heat-pump's evaporator. A simpler system was used to condition the water delivered to the heat pump's hot water tank: water was supplied from two 400-liter tanks maintained at the desired pressure by pressure-regulated air and at the desired temperature by recycling the water through a chiller. A water recycle loop branched off at the inlet to the heat-pump water tank so that constant temperature water was available to the heat pump even after periods of no water demand.

APPENDIX B

This Appendix contains the primary experimental results obtained in the experimental investigations. Figures B1-B12 refer to Unit B and Figures B13-B31 refer to Unit A. A brief description of each figure follows.

Figures B1, B2: Effect of Water Demand and Demand Schedule.

Water demand is a most important parameter. Both energy savings and COP rise steeply with increasing water demand, but the curves flatten out as the capacity of the heat pump system is approached and the auxiliary resistance heater picks up additional load.

The National Solar Data Network (NSDN) hot water demand schedule (Figure 3 in the report) yielded essentially the same results as the standard schedule. However, a constant demand schedule, (23 liters/hr from 6:00 to 22:30) yielded significantly poorer results. Note that the constant draw does not result in a steady-state experiment because the compressor goes through as many as 7 ON/OFF cycles in a 24-hour period (only 5 on/off cycles occur for the standard demand schedule in Fig. 4).

Figure B3: Steady-State Experiments (Compressor ON Continuously).

This figure shows that the heat pump is capable of delivering a steady stream of warm water, e.g. 44°C at a high COP (3.1 corresponding with the 44°C outlet temperature). Note, however, that a rising inlet water temperature has a negative effect on COP (discussed later).

Figure B3 also shows the effect of mixing the tank (with the aid of a small pump that draws water from the top and injects it together with the cold feed water into the bottom of the tank). As expected, COP suffers significantly when the tank is mixed.

Figure B4: COP Correlation with Condenser Water Temperature.

This figure is based on a compilation of all runs with variable water conditions at the stated fixed inlet air conditions. Since essentially all points fall on the drawn line, water temperature at the condenser determines the COP (once the evaporator temperature has been fixed by the air conditions). The only point falling significantly off the line is from a steady-state experiment with mixing of water within the tank. It is possible that the mixing process increases the water velocity at the condenser and, thus, enhances the condenser performance.

Figures B5, B6: Heat Pump Thermostat Set-Point Effects.

The COP decreases slowly with increasing heat pump set point (always keeping the resistance-heater set point 10°C lower). However, energy savings increase slowly with set point due to increased total hot-water heating requirement.

Raising the auxiliary resistance heater set point to the same level as the heat-pump set point does not alter COP nor the average temperature of the hot water. However, energy savings are significantly reduced. The standard deviation of the hot water temperature is also reduced, as shown in Figure B6. Consequently, a homeowner can maximize his energy savings by not using the resistance heater and accepting somewhat larger "swings" in delivered hot water temperature.

Figures B7, B8: Effect of Water Inlet Temperature.

In the relatively narrow temperature range studied, COP increases somewhat with falling feed-water temperature. The effect on energy savings is larger because the falling feed-water temperature also increases the total amount of energy required for water heating (when hot-water set point is kept constant).

Figures B9, B10: Effect of Air Flow Rate.

Air flow rate is seen to increase COP as is expected on the basis of an increase in average evaporator temperature with air flow rate. The energy savings, however, go through a flat maximum because the increased air flow rate also increases the energy required for 24-hour operation of the fan. This term is shown on the bottom of Figure B9. Note that the energy savings depicted on Figure B9 concern water heating only, i.e., the fact that ventilation rate has an impact on house heating or cooling is ignored.

Figures B11, B12: Effect of Inlet Air Conditions.

Both temperature and humidity have the expected effect on COP: increases in either parameter increase COP.

Figures B13, B14: Effect of Water Demand and Demand Schedule.

Both energy savings and COP rise steeply with water demand until saturation occurs at very high water demands. The National Solar Data Network (NSDN) schedule, with its lower draw rates (Fig. 3. of report), produces essentially the same results as the "standard" draw schedule with its intermittent flow (Fig. 4 of report).

Figure B15: Steady-State Operation (Compressor ON Continuously).

These results show that mixing of the tank (by an external pump) has an insignificant effect on Unit A COP, implying that tank stratification is not important for obtaining high COPs with the heat pump. This is because the large condenser transfers heat along almost the entire tank height.

Figure B16: COP Correlation with Average Tank Temperature.

This figure is based on a compilation of all runs with the stated fixed evaporator conditions and variable water conditions. The conclusion from the excellent fit provided by the drawn line is that heat pump performance is fully determined by the average water temperature in the tank, once the evaporator temperature is fixed by the inlet air conditions. Only the run in standby operation with no water draw falls significantly off the line, possibly due to significant radial temperature differences within the tank.

Figures B17, B18: Heat Pump Thermostat Set Point Effects.

Although energy savings increase with increasing setpoint temperature, due to an increased load, the COP decreases quite steeply. Therefore, low hot-water temperatures are favorable from a total energy usage point-of-view.

Also shown in Figures B17, B18 is the effect of raising the setpoint for the resistance heater: Although the COP does not suffer appreciably (because only the water at the very top of the tank is heated electrically), energy savings drop significantly. On the positive side, supplemental electric heating is shown to reduce the standard deviation of the hot water delivery temperature.

Figures B19, B20: Effect of Water Inlet Temperature.

The COP is not affected appreciably, but energy savings increase with decreasing feed water temperature (due to increased load).

Figures B21, B22: Effect of Exhaust Air Flow Rate.

An increase in air flow rate increases both the COP and energy savings. As shown by Figure B21, fan power is almost independent of flow rate because the flow rate is adjusted by a slide valve, not by a variable speed fan. The energy savings depicted on Figure B21 concern water heating only, i.e. the fact that the ventilation rate has an impact on the space heating or cooling load is ignored.

Figures B23, B24: Effect of Exhaust Air Conditions.

The COP increases with both temperature and relative humidity.

Figures B25, B26: Fan Coil Energy Savings and COP for Several Fan Coil Air Flow Rates.

The assumption of 24-hour fan coil operation is emphasized: the energy savings in Figure B25 are only realized if the fan coil runs continuously. If the fan coil cycles ON and OFF during the day, energy savings will be directly proportional to ON time.

Figures B25 and B26 show that high air flow rates through the fan-coil condenser are very favorable; both COP and energy savings increase steeply with increasing air flow rate. However, the energy savings curve levels off before the COP curve implying that the point of diminishing return is around 400 kg/h. This saturation occurs because the compressor power decreases significantly at the high air flow rates (which in turn is due to the reduced refrigerant pressure that accompanies high heat fluxes in the fan coil). Note that fan-coil fan power ends up as useful space heat and is therefore not a debit on the energy savings.

Figures B27, B28: Effect of Fan Coil Inlet Air Temperature.

The inlet air temperature affects COP much as tank-average temperature affects water heating COP. In practice, however, the inlet air temperature will be in the range 19 - 21°C most of the time, consequently the high sensitivity to air temperature is less important.

Figure B29: Energy Savings, COP and Water Heating ON-Time Fraction for Combined Water and Space Heating.

For these results, the assumption of 24-hour heat pump operation has been made; if the fan coil is used less, the energy savings (and to some extent the combined COP) will be affected.

Figure B29 shows the effect of shifting from 100% fan-coil operation (the left-most point on the x-axis) to a heat-pump loading (right-most point) such that the hot water demand is satisfied by the heat pump only (without electric-resistance heat) and remaining time is devoted to fan-coil operation. For the intermediate points, electric-resistance water heating increases to the left, and becomes insufficient at the left-most point for the water demand of 340 liters (the Unit A electric-resistance water heater uses 1.7kW and located near the top of the tank).

The main conclusion from the results in Figure B29 is that heat pump load shifting between the fan coil and water tank has little effect on the combined COP and energy savings, and that a simple model matches the experiments quite well. The model for the combined COP is simply the fan-coil COP multiplied by the fan-coil ON time fraction plus the water-heating COP times the water-heating ON time fraction. The water-heating COP is obtained from Figure B16 based on the measured tank average temperature. Model energy savings are then calculated according to Eq. (13).

The reason for the flat maximum in COP (and energy savings) in Figure B29 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 (Figure B16). For a high fan-coil air flow rate (such as 400 kg/h), there would be no such peak in COP (nor in energy savings), but rather a monotonic decline as the water heating load is increased. However, even in this case, energy savings (and combined COP) are relatively insensitive to load shifting between water heating and space heating.

Figures B30, B31: Runs with an Over-Charged Refrigerant System.

The results in Figure B30 parallel the results in Figure B29 with the exception of refrigerant charge: the refrigerant charge was increased between the two series of runs, and accidentally too much was added. Fan coil COP was not affected appreciably, but the water heating COP was reduced by about 20%. This is the cause of the divergence between model and experiment toward the right side of Figure B30.

The results shown in Figure B31 (the model results because the experimental results are too low due to the over-charging of refrigerant) demonstrate the large energy savings that are obtained with a fan coil that is not heat-sink limited. Figure B31 can be compared with Figure B13 which depicts enery savings without fan coil operation. With fan-coil operation (and space as a heat sink), the energy savings are quite insensitive to hot water demand.

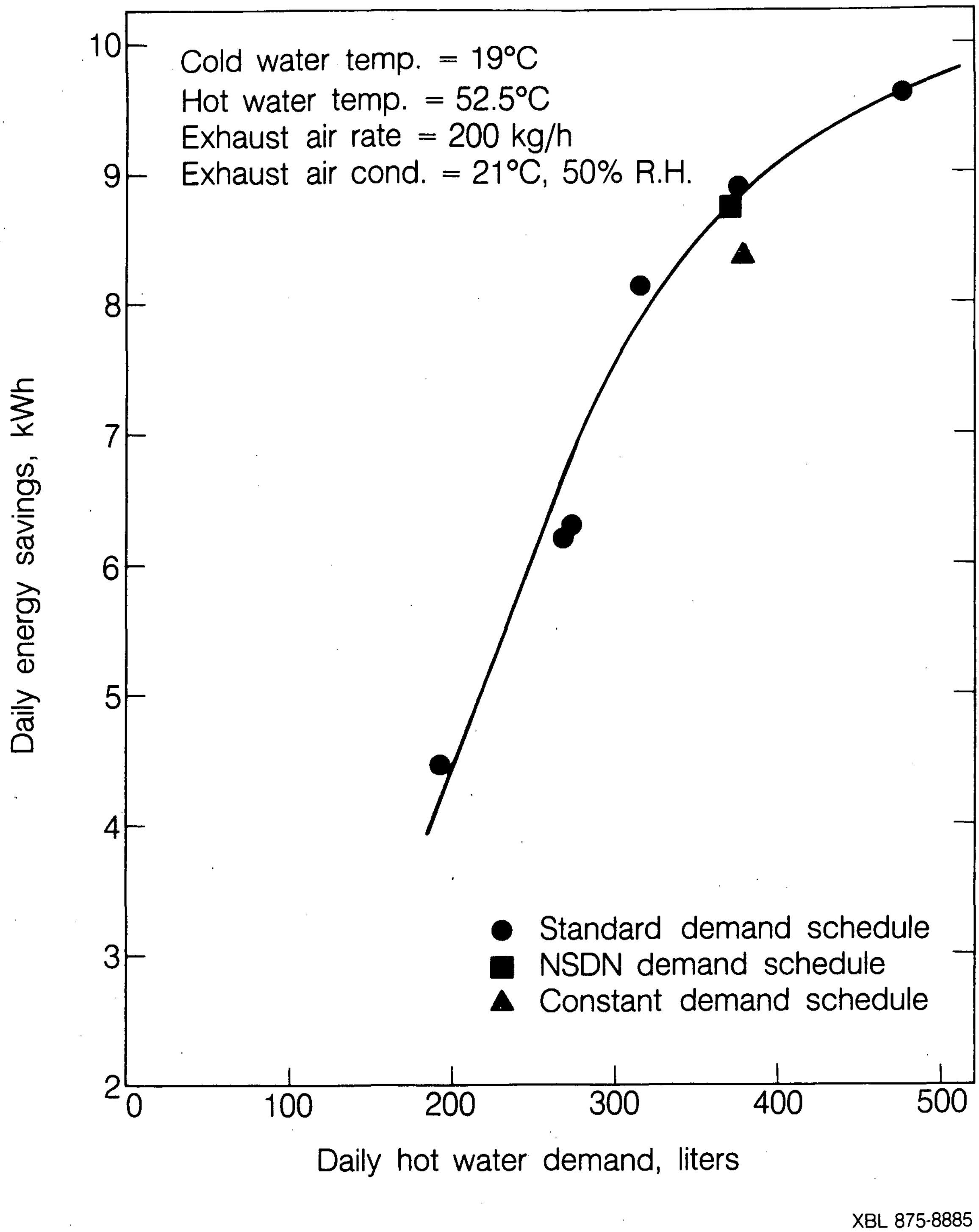


Figure B1. Energy savings with Unit B heat pump relative to electric resistance water heater as function of water demand and demand schedule.

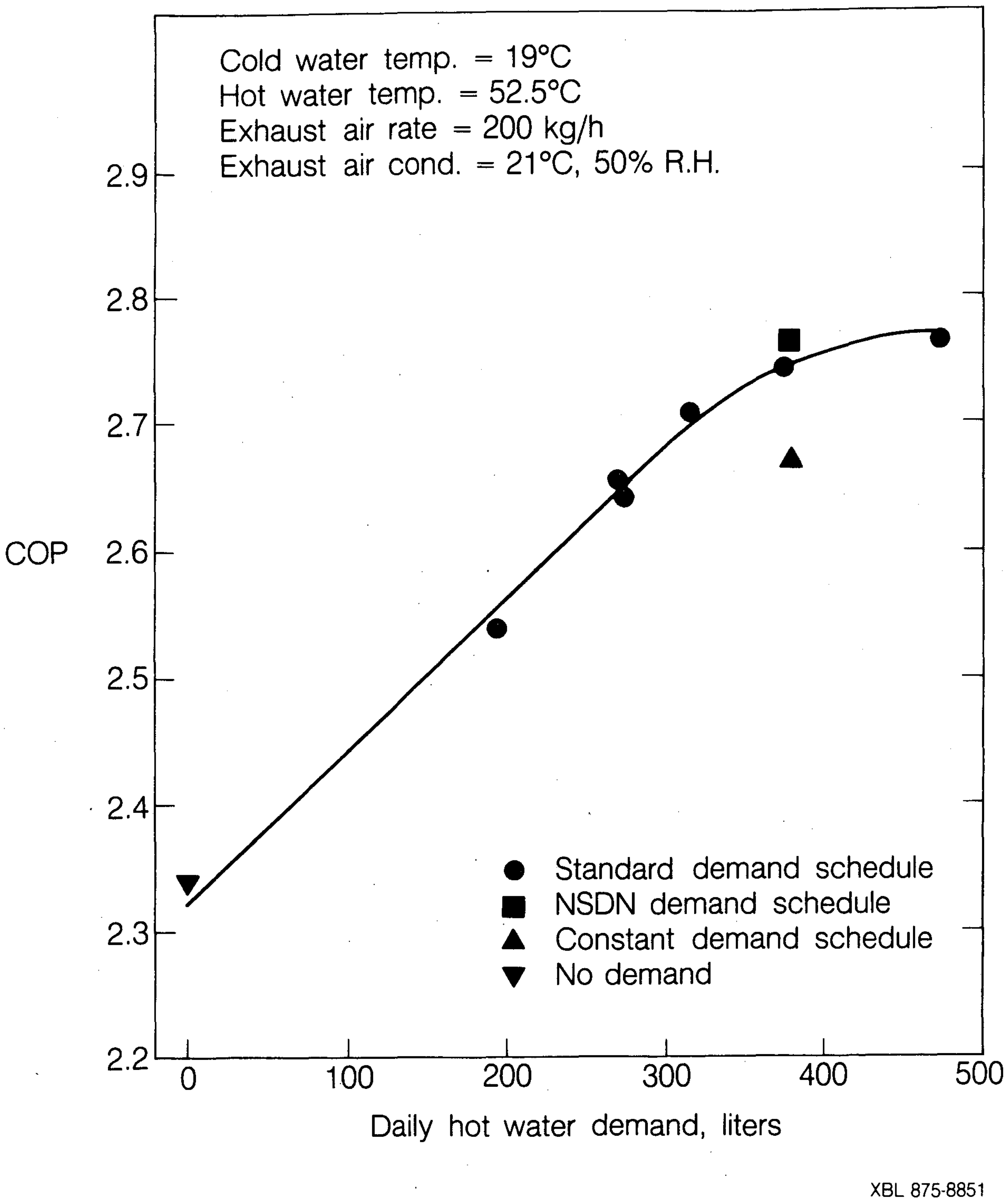


Figure B2. Unit B heat-pump COP as a function of water demand and demand schedule.

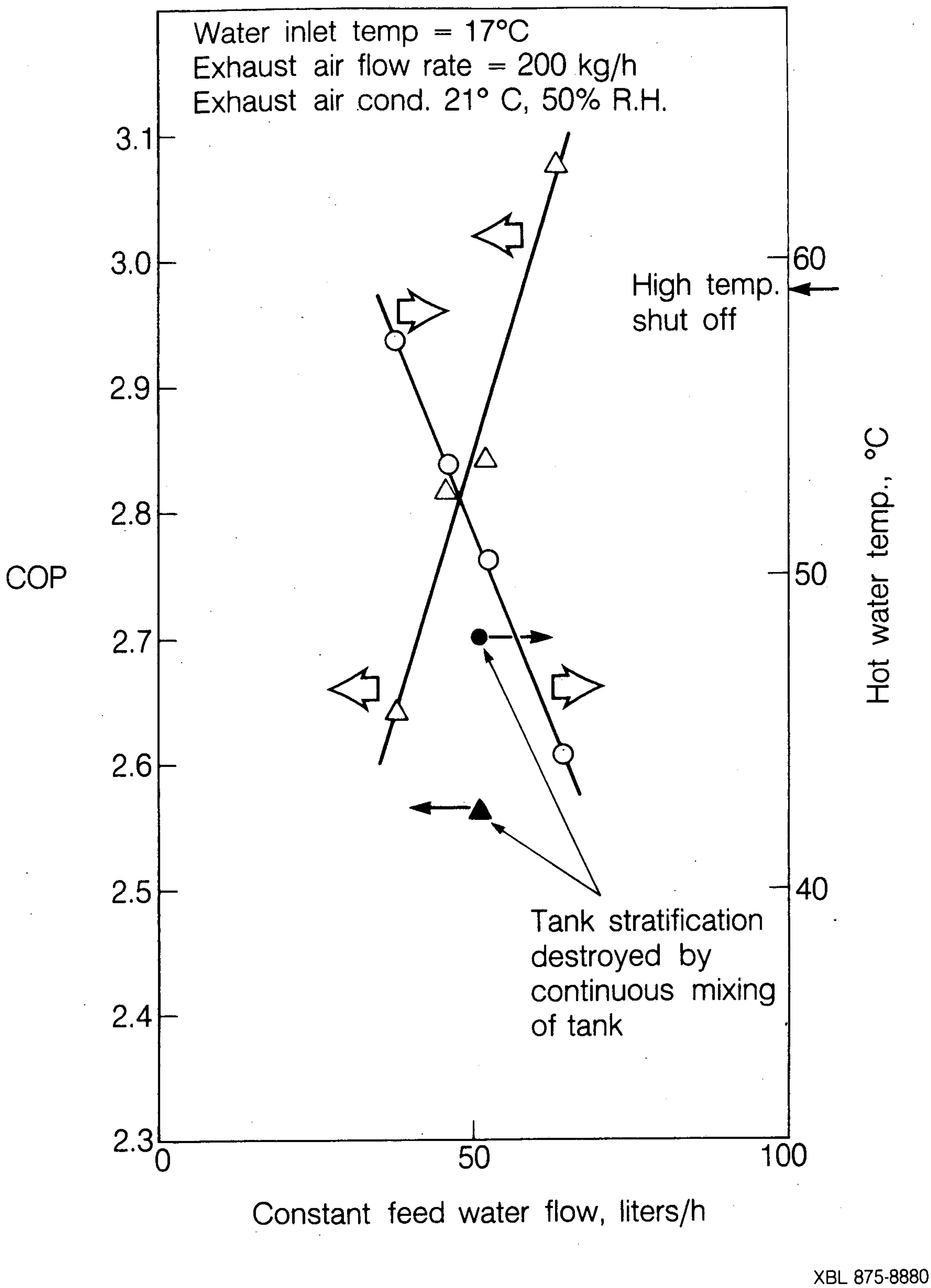


Figure B3. COP and hot water temperature obtained with continuous steady-state operation of Unit B heat pump.

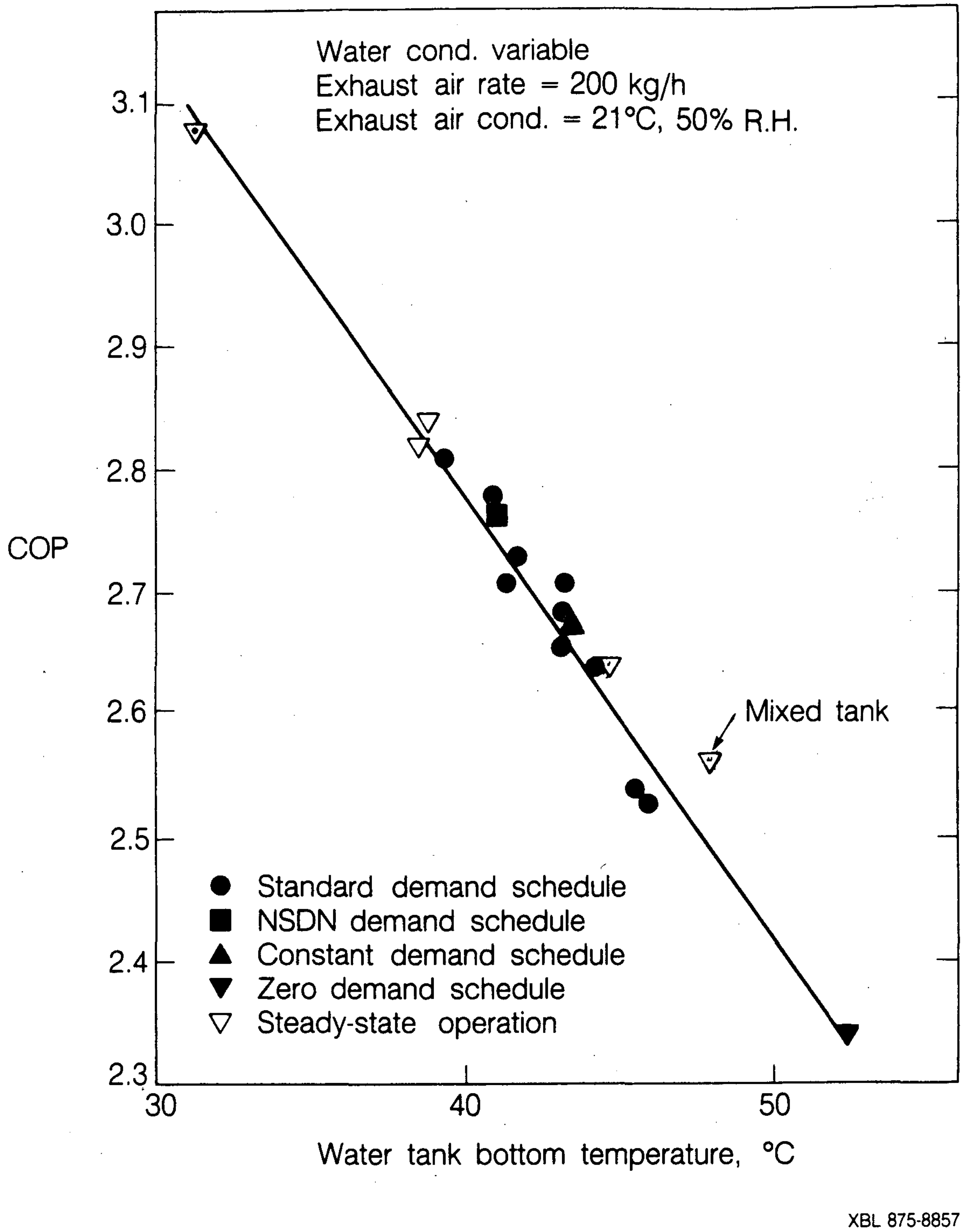


Figure B4. Unit B heat-pump COP as function of water temperature at the bottom of the tank (mean temperature for unsteady-state operation).

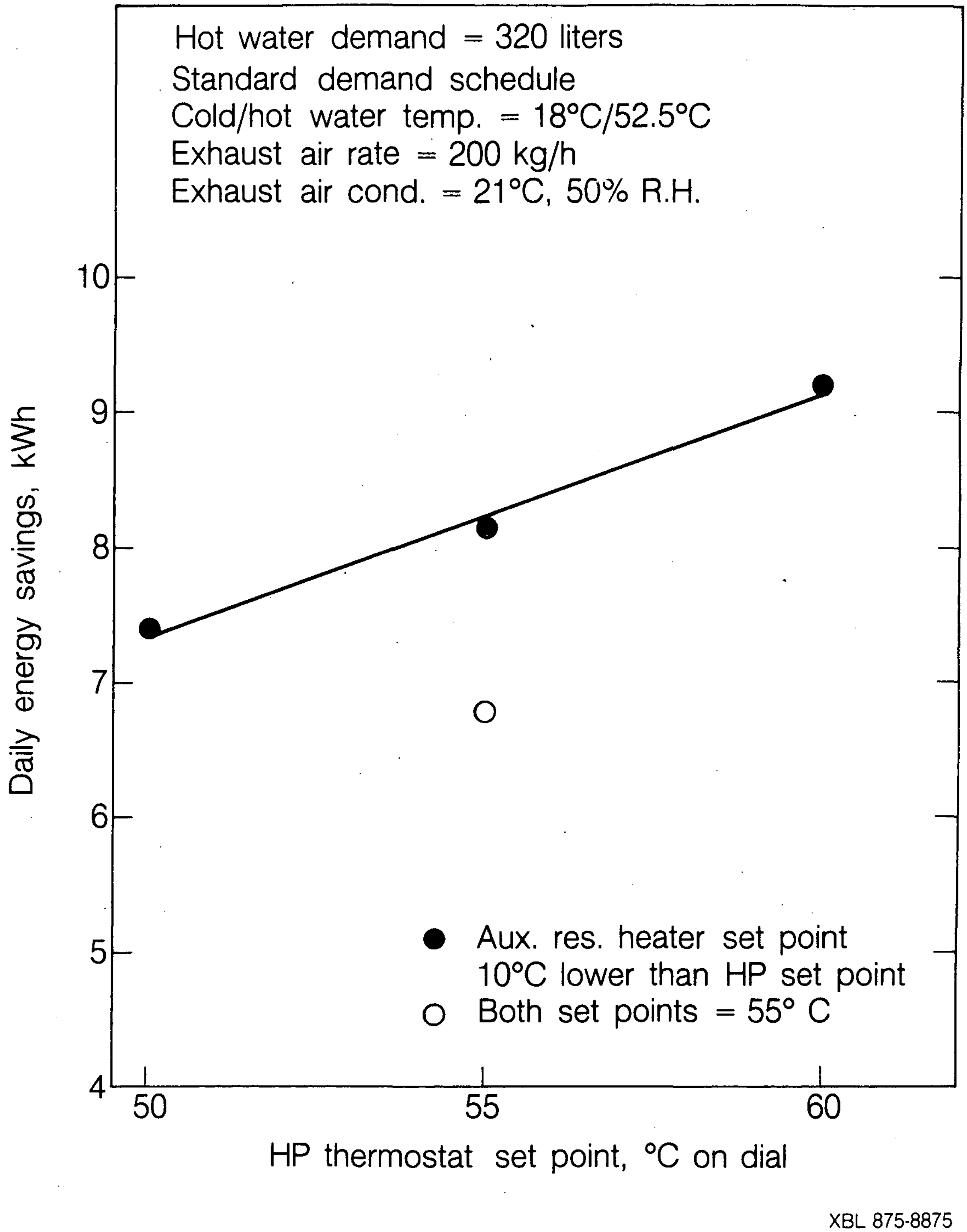


Figure B5. Energy savings with Unit B heat pump relative to electric resistance water heating as function of heat-pump thermostat set point (and auxiliary resistance heater set point).

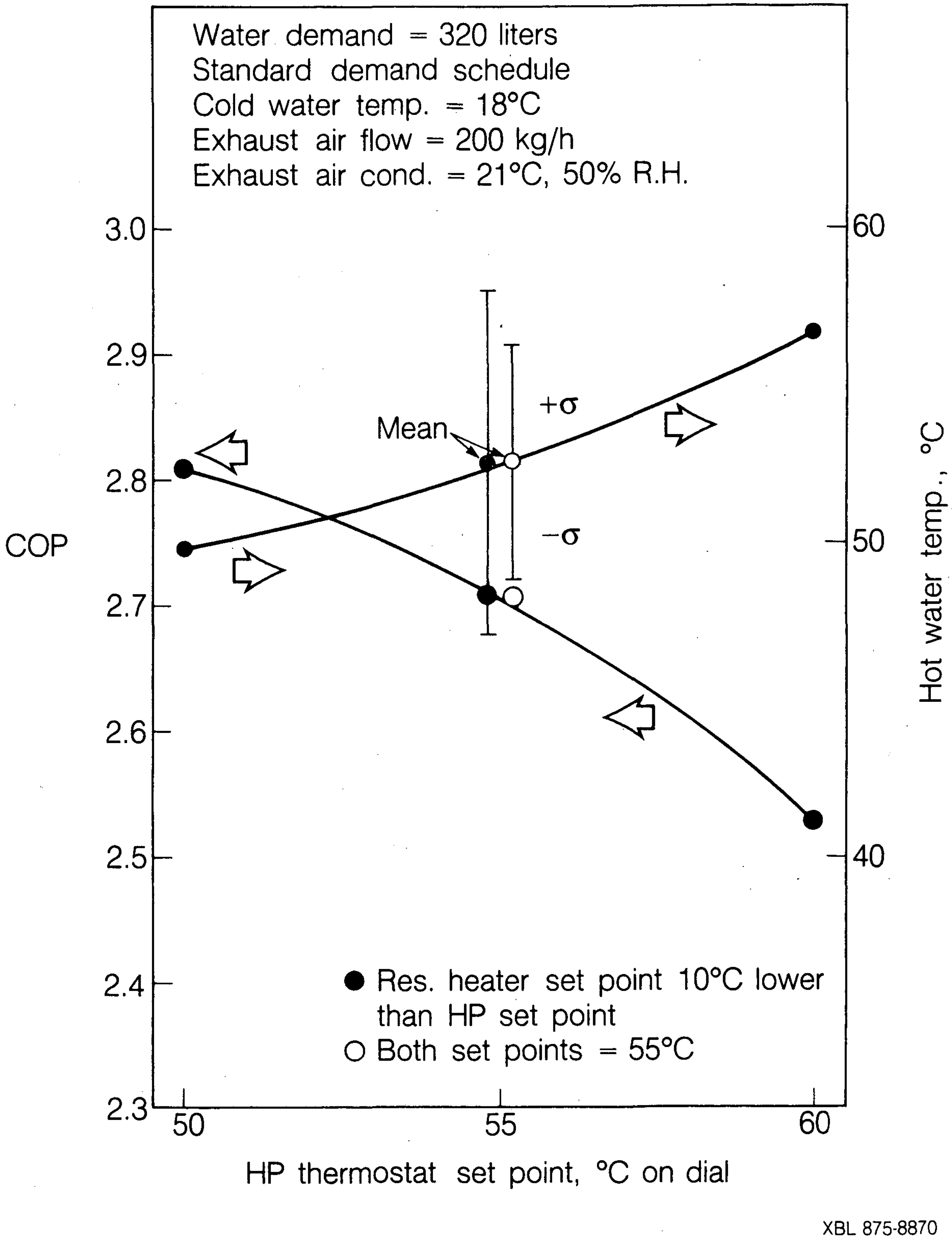


Figure B6. Unit B heat-pump COP and hot water temperature as function of heat-pump set point and resistance-heater set point.

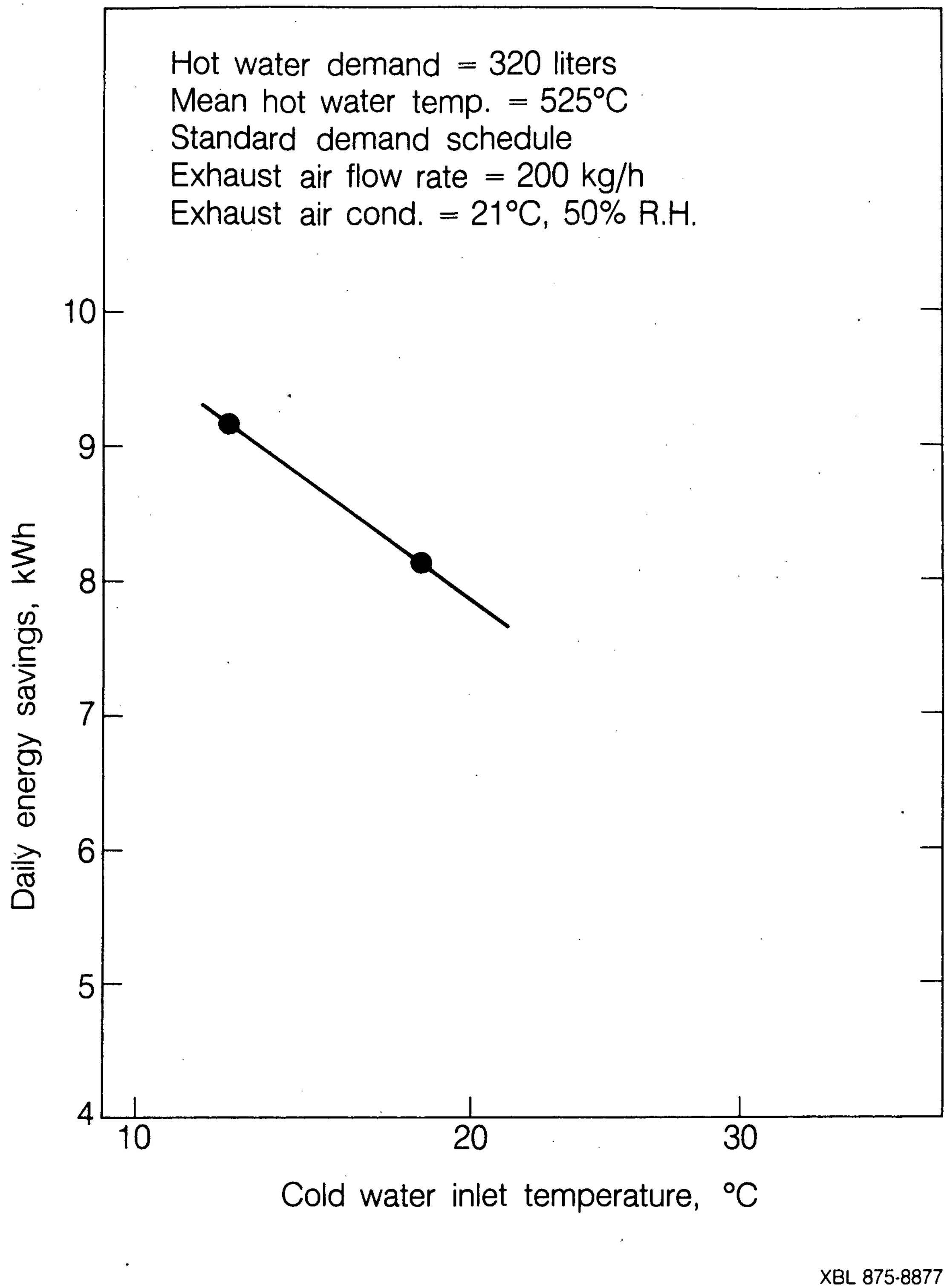


Figure B7. Energy savings with Unit B heat pump relative to electric resistance water heating as function of cold-water inlet temperature.

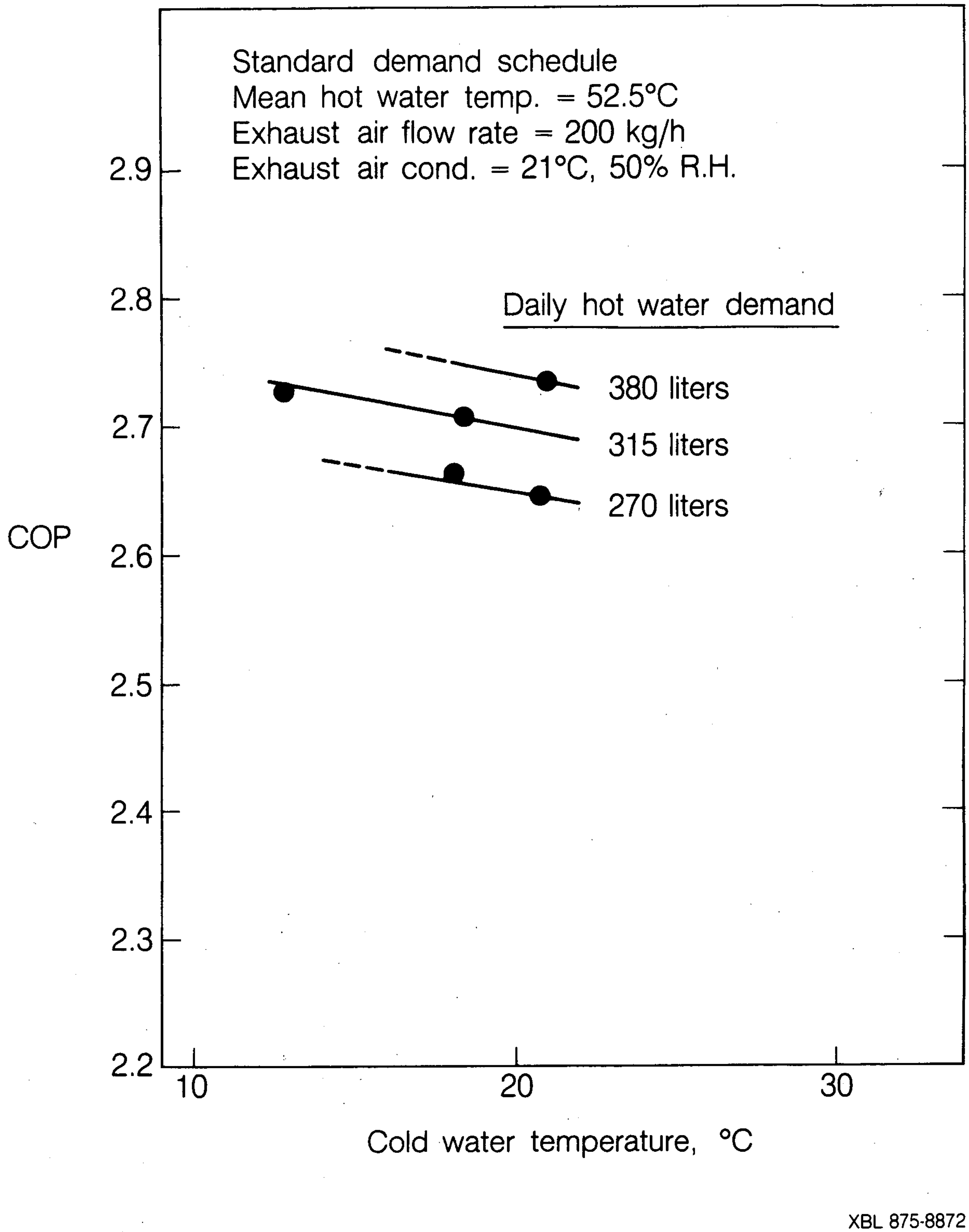


Figure B8. Unit B heat-pump COP as function of cold-water inlet temperature and water demand.

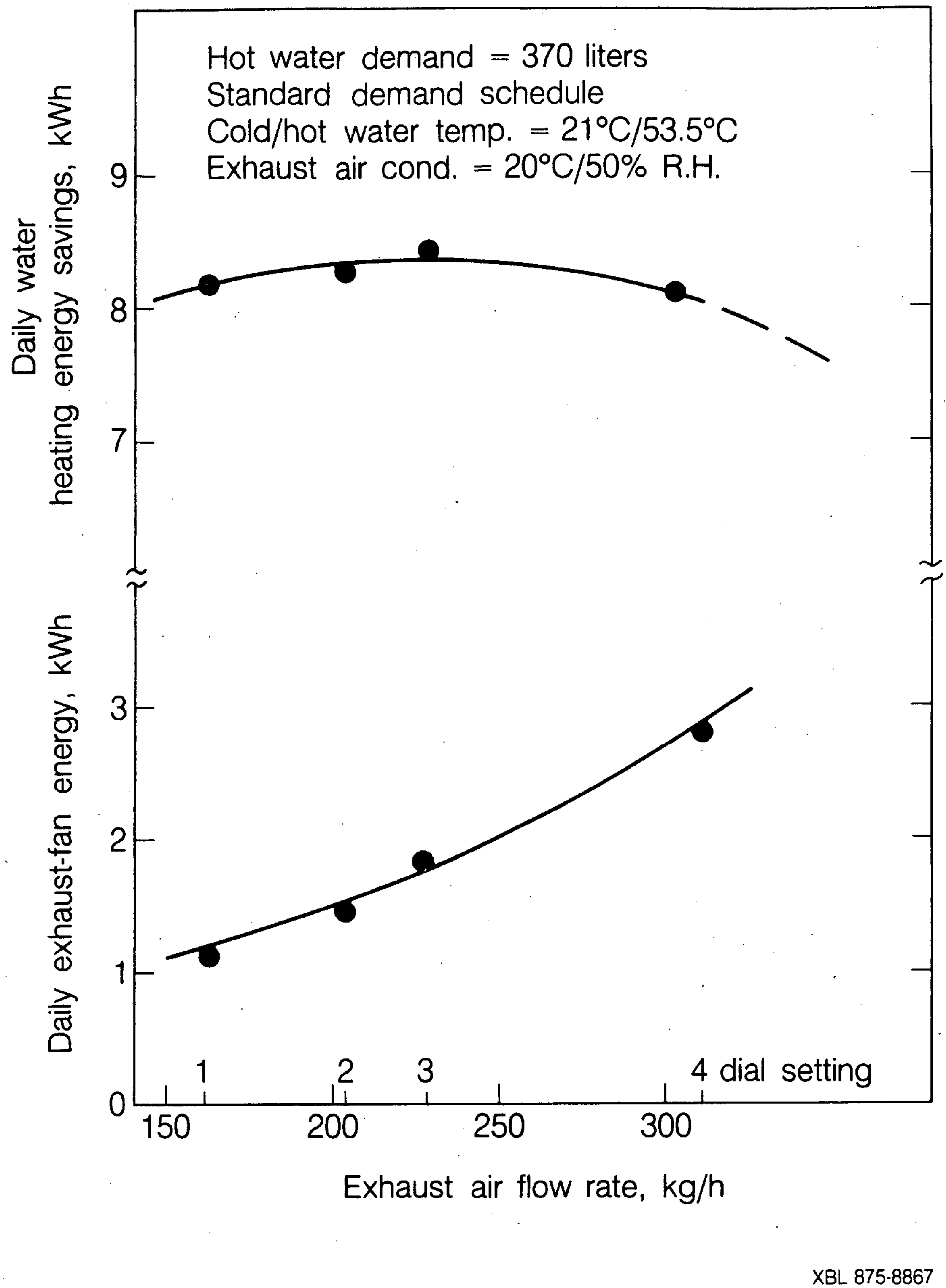


Figure B9. Energy savings with Unit B heat pump relative to electric resistance water heating as function of exhaust air flow rate. The heat pump energy savings have been debited with the energy required for 24-hour operation of the fan.

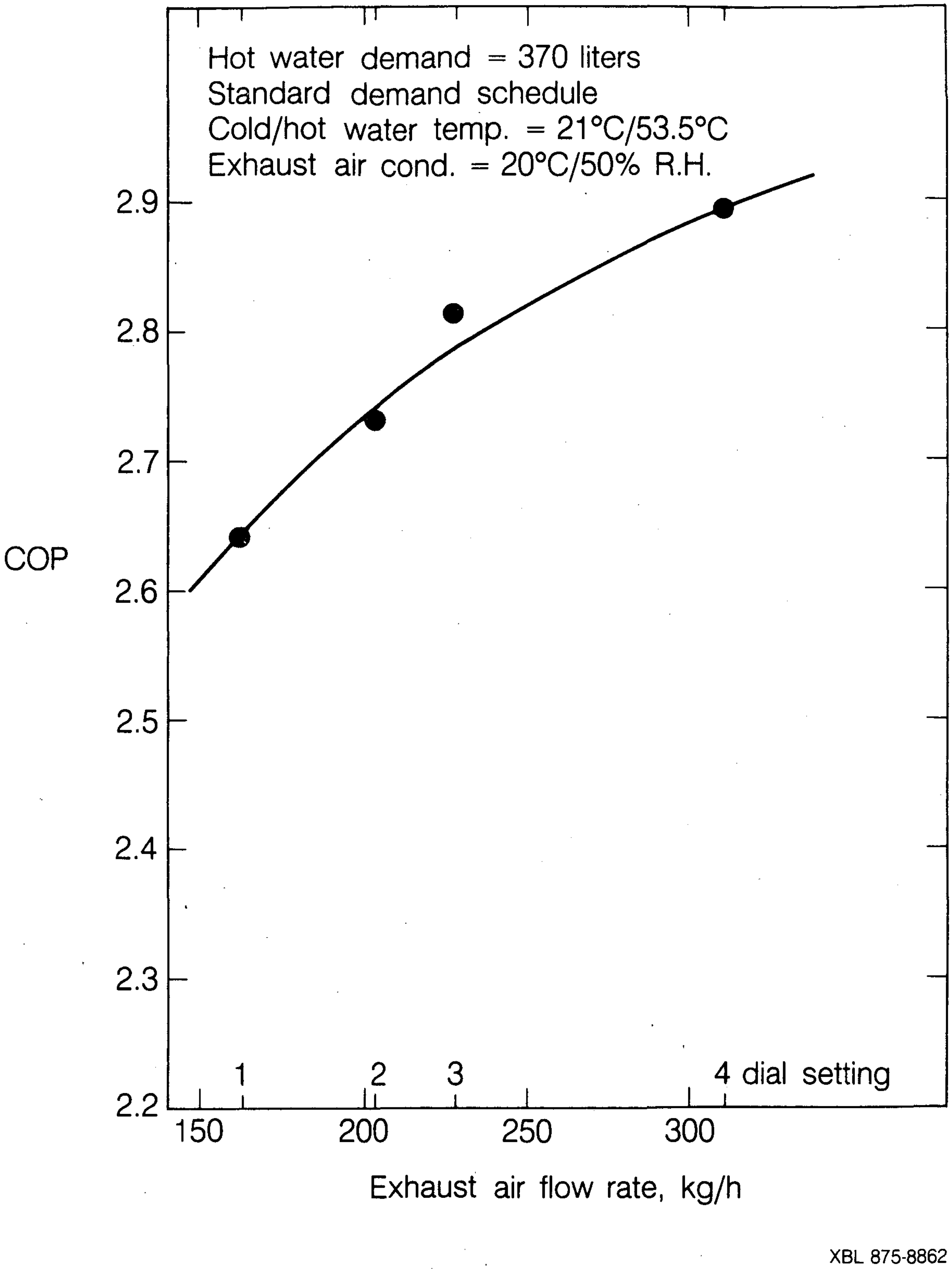


Figure B10. Unit B heat-pump COP as function of exhaust air flow rate.

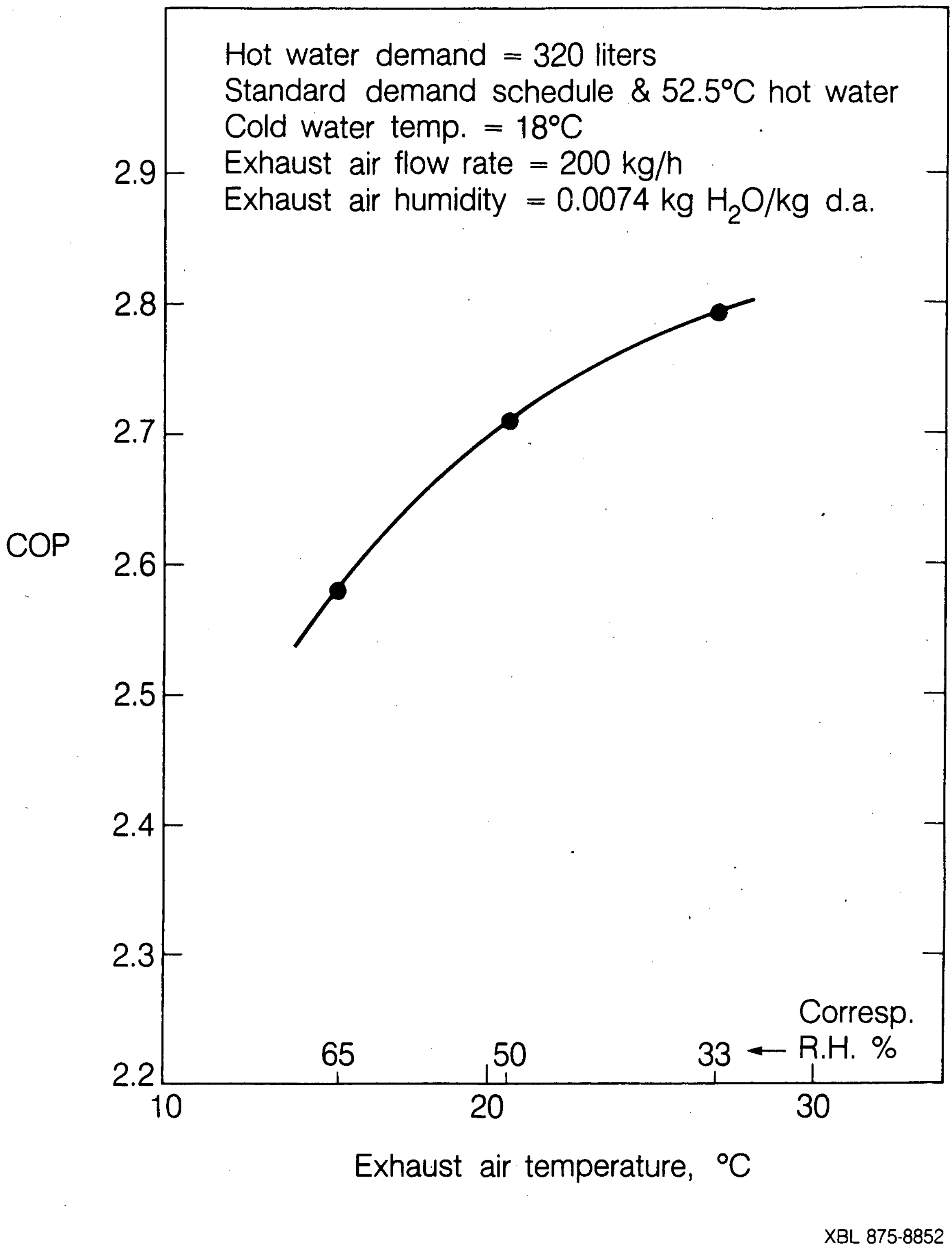


Figure B11. Unit B heat-pump COP as function of exhaust air temperature (inlet air temperature to evaporator).

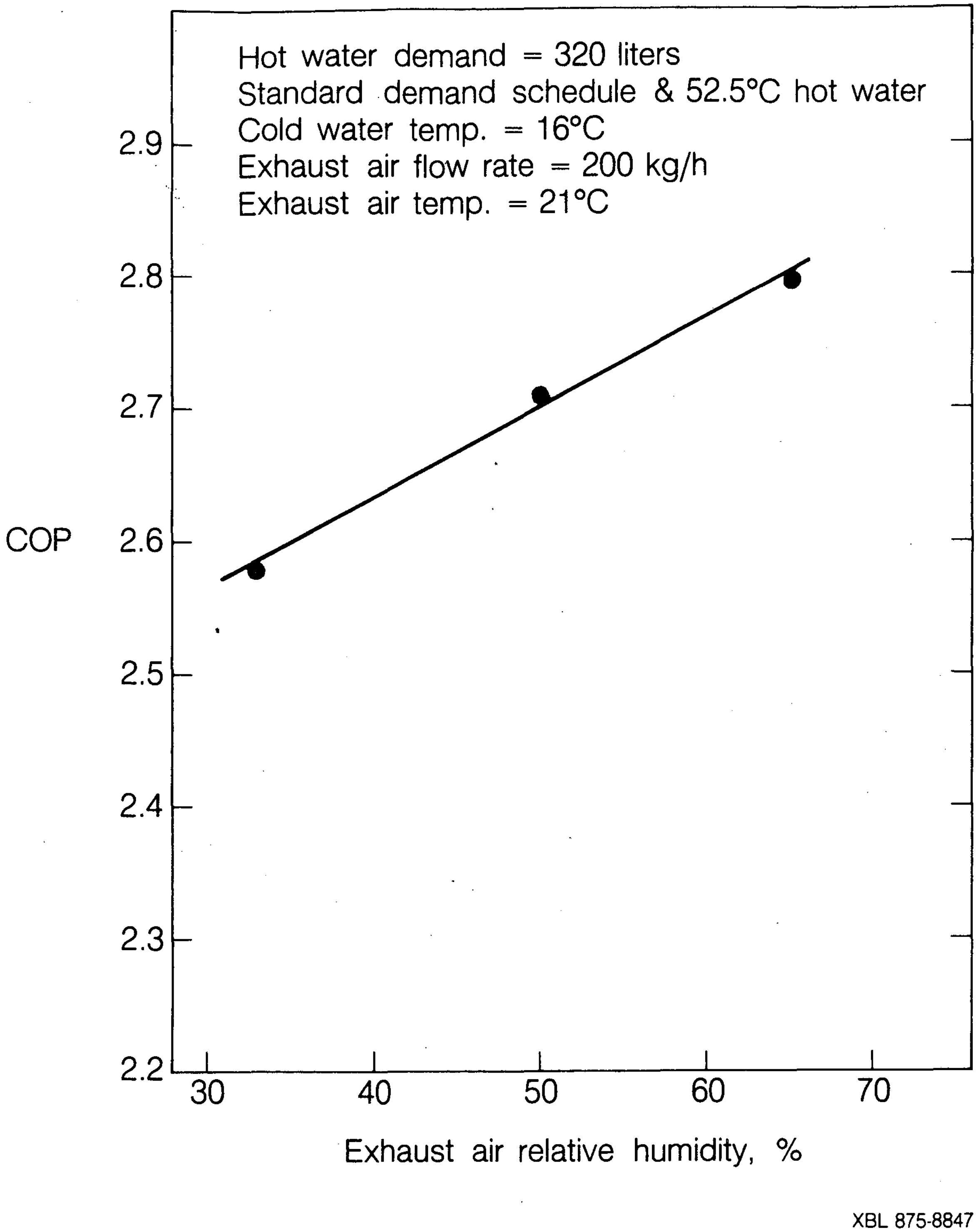


Figure B12. Unit B heat-pump COP as function of exhaust air humidity (inlet air humidity to evaporator).

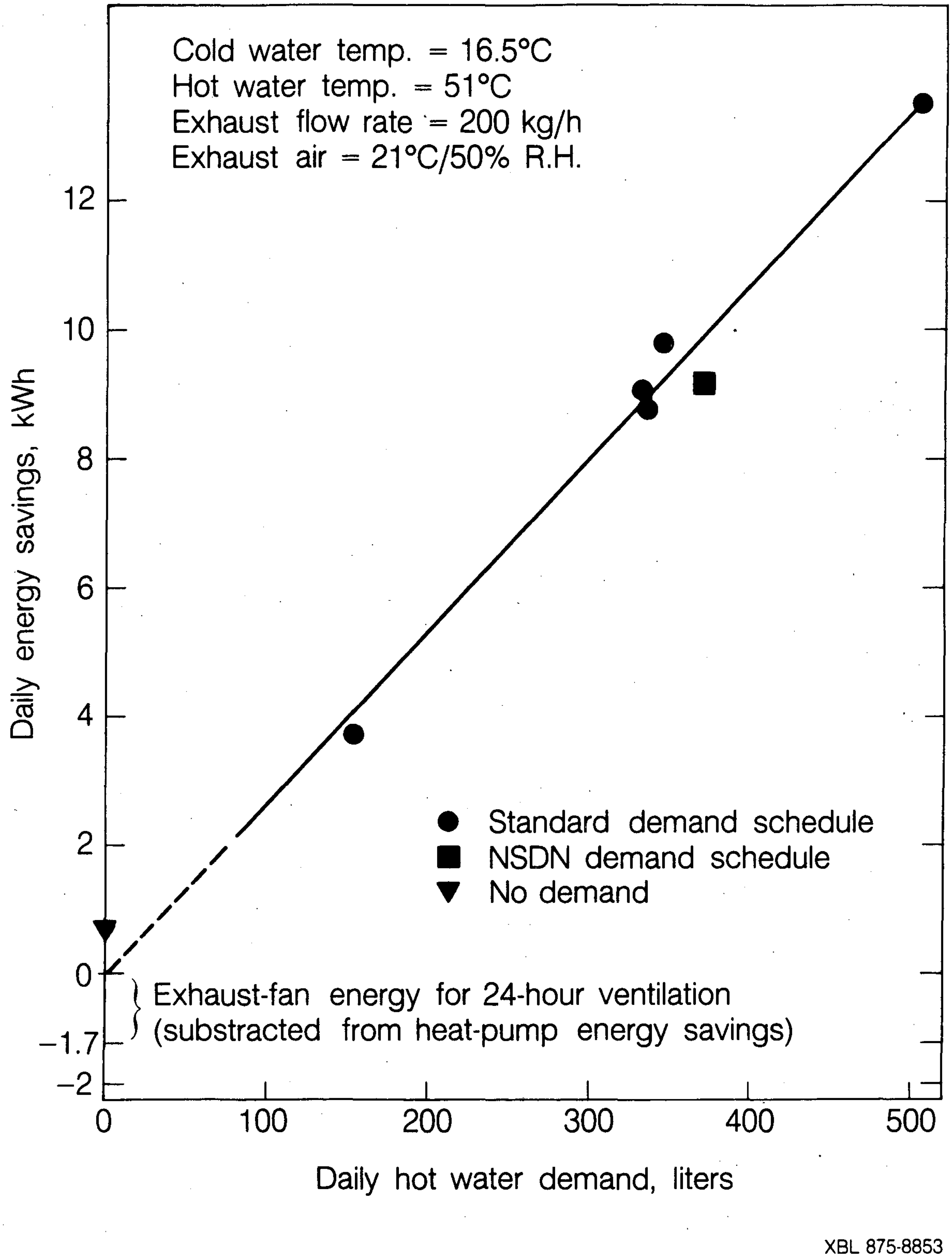


Figure B13. (Water heating only)
Energy savings with Unit A heat pump relative to electric resistance water heating as function of water demand and demand schedule.

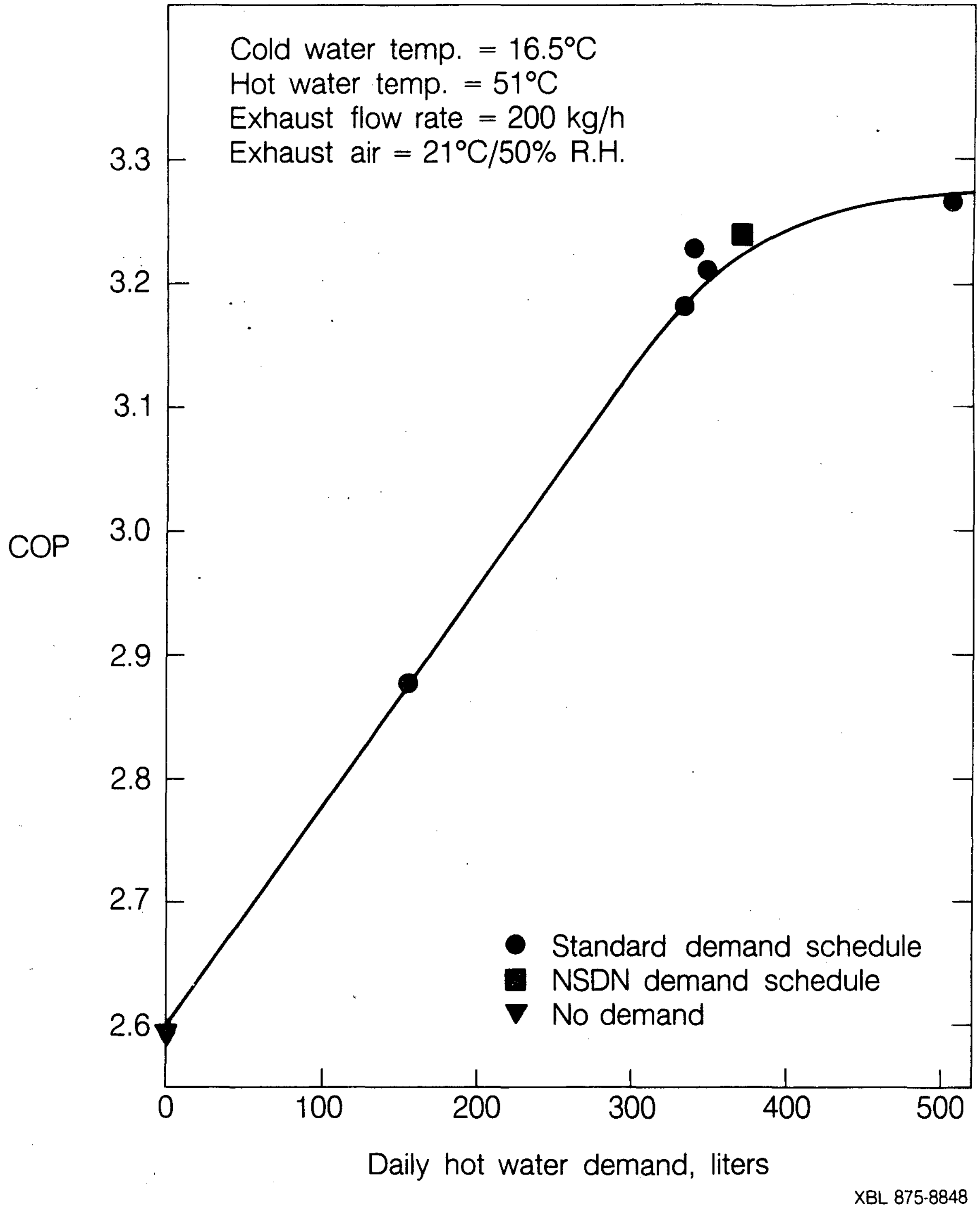


Figure B14. (Water heating only)
Unit A heat-pump COP as function of water demand and demand schedule.

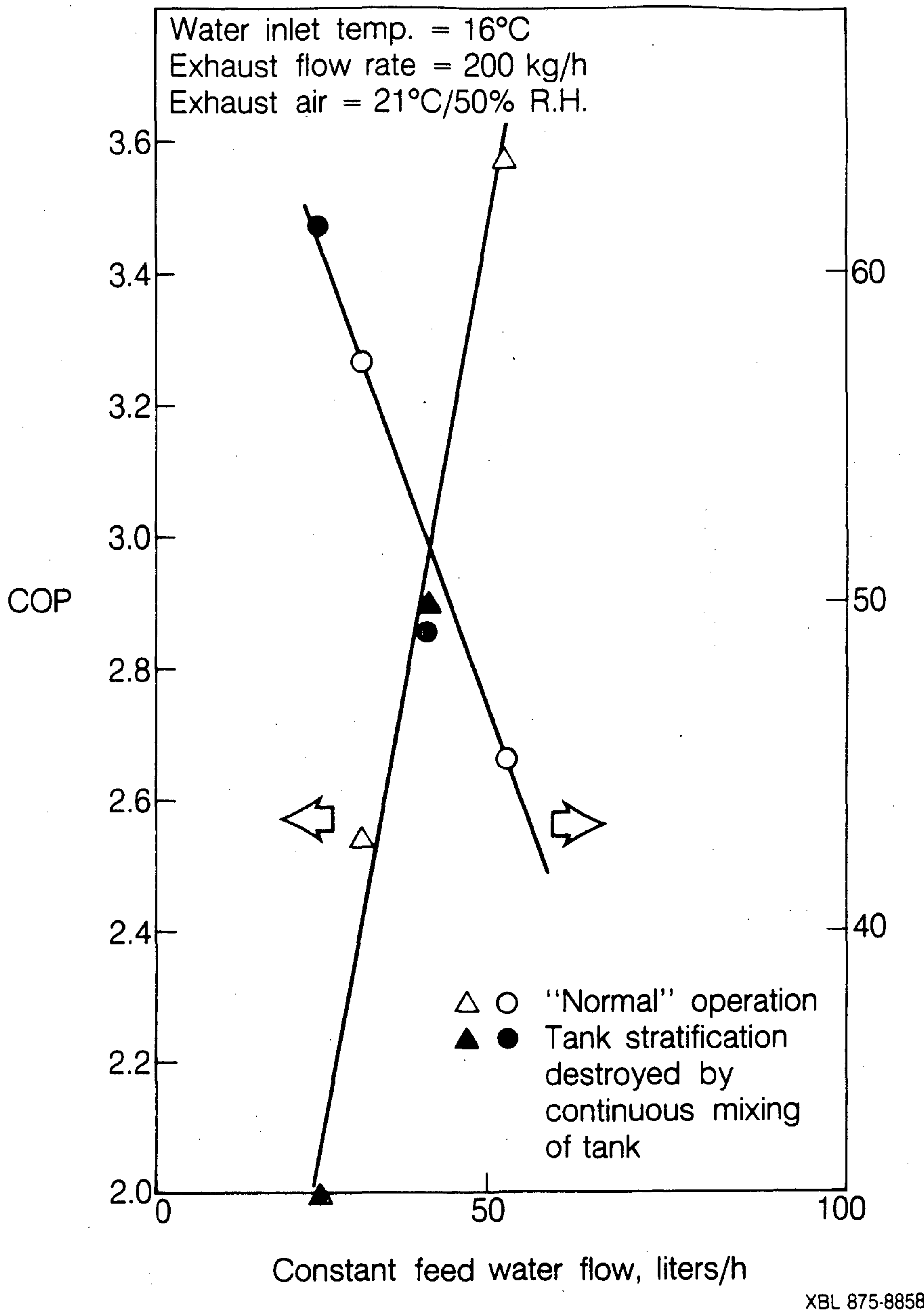


Figure B15. (Water heating only)
COP and hot water temperature obtained with continuous steady-state operation of Unit A heat pump.

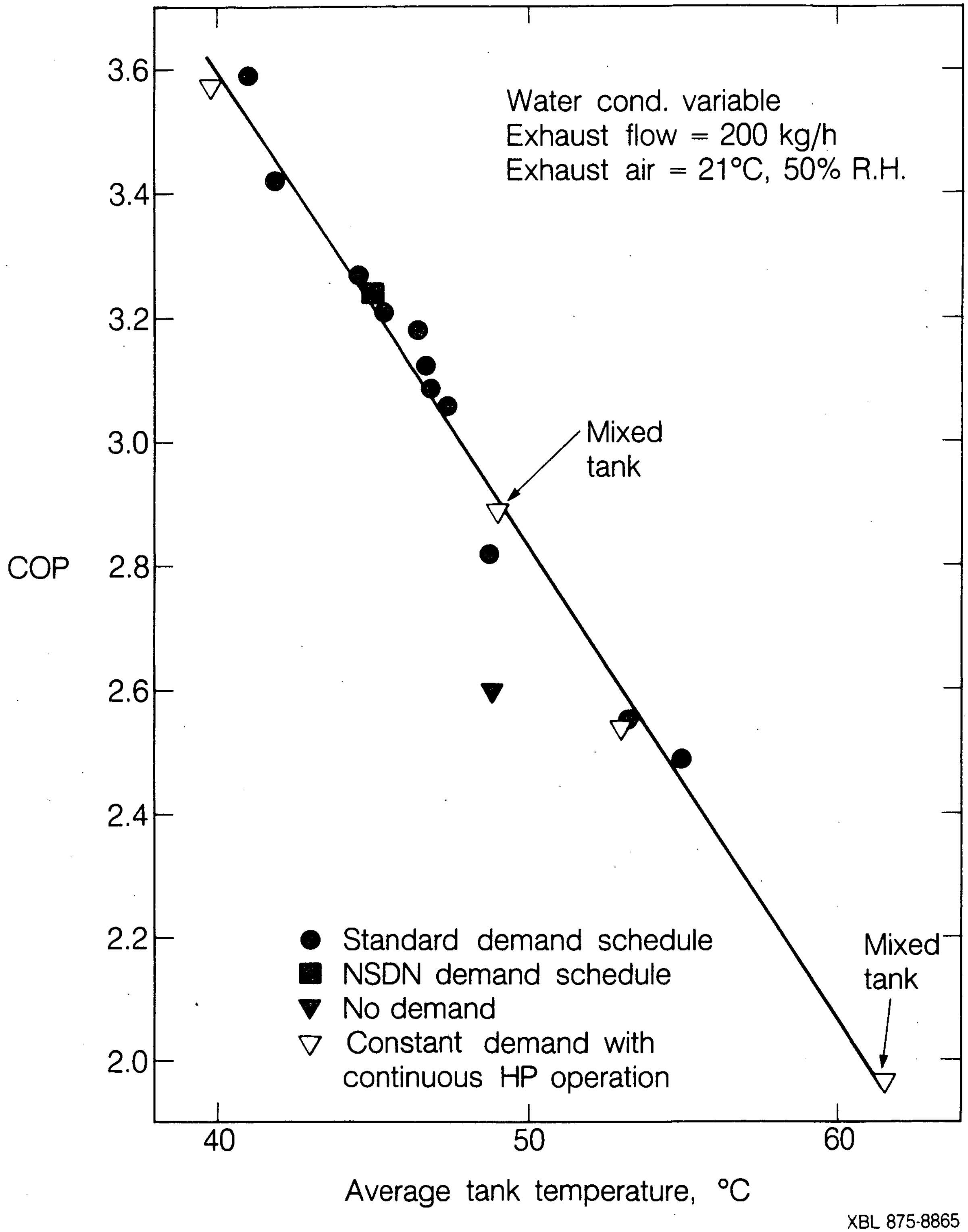
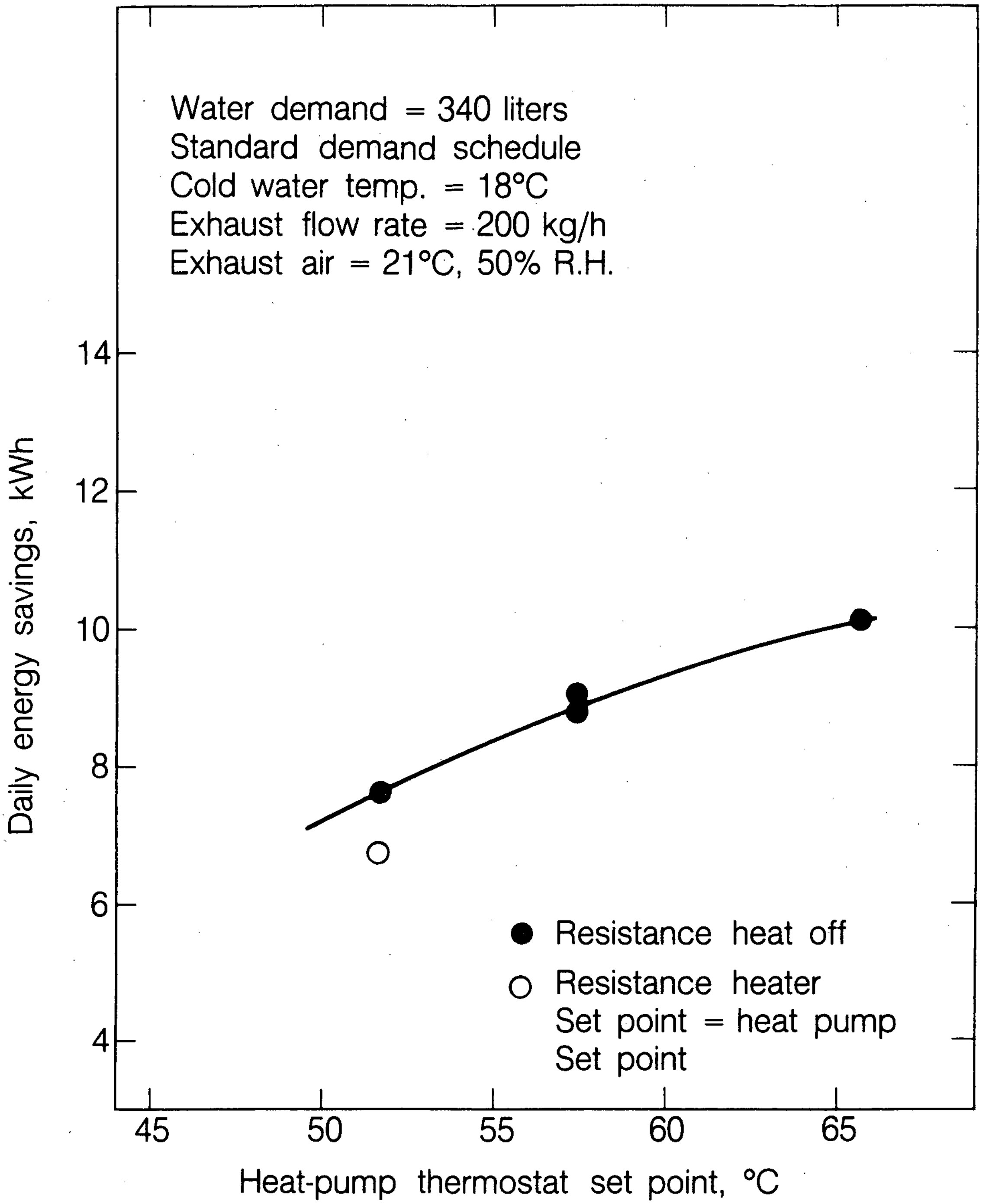


Figure B16. (Water heating only)
Unit A heat-pump COP as function of average temperature of water in tank (average in space and time).



XBL 875-8860

Figure B17. (Water heating only)
Energy savings with Unit A heat pump relative to electric resistance water heating as function of heat pump thermostat set point (and auxiliary resistance-heater temperature set point).

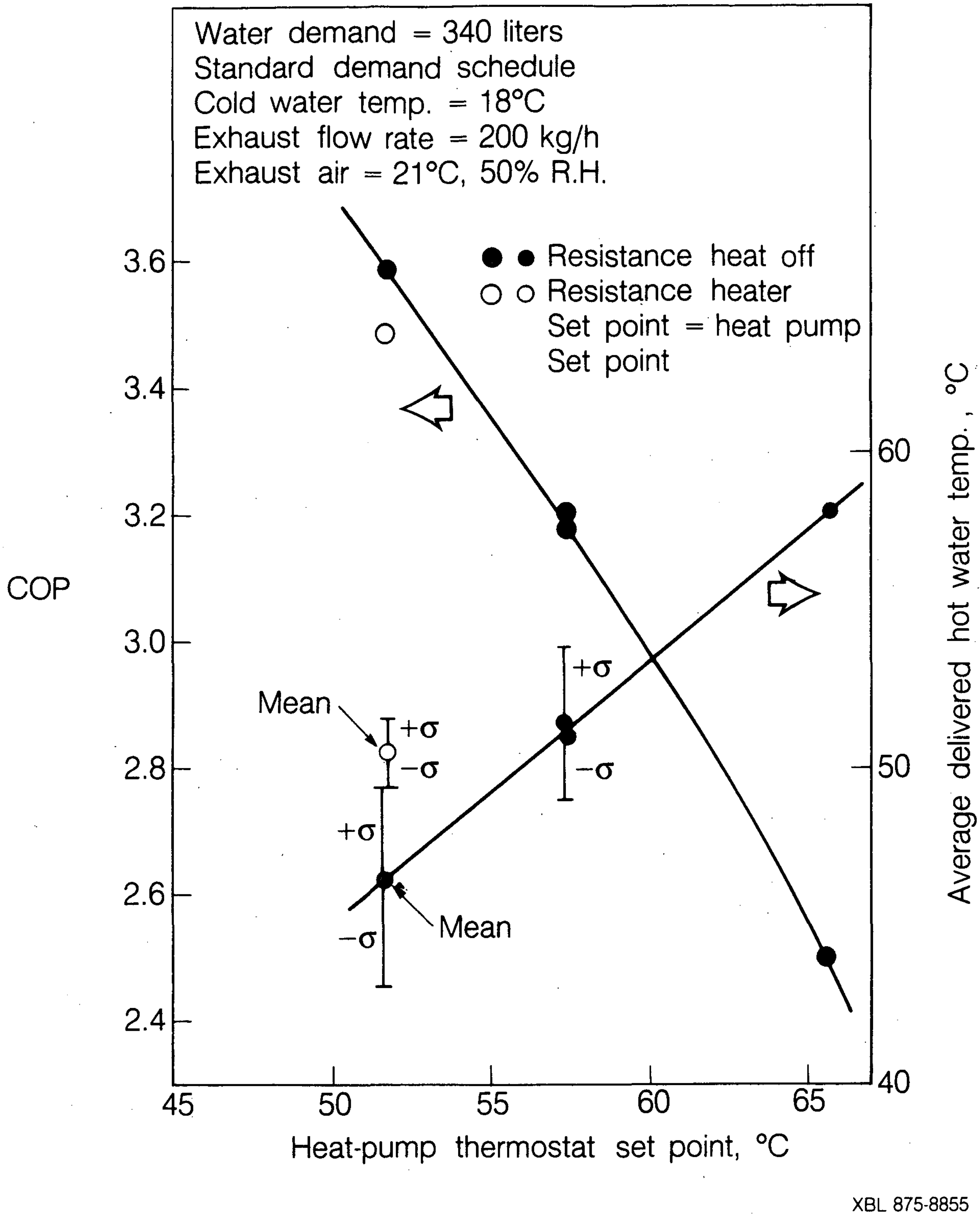
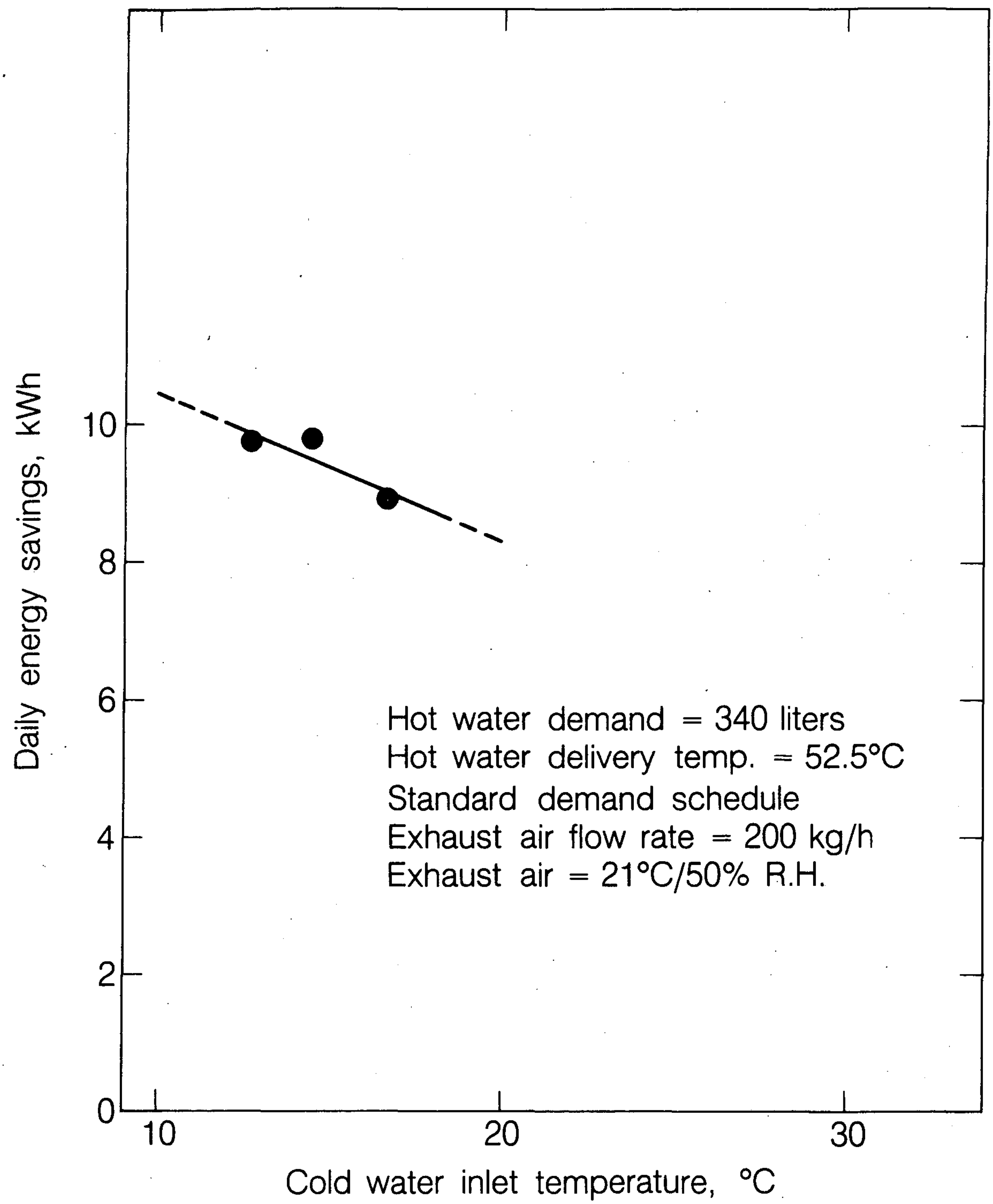
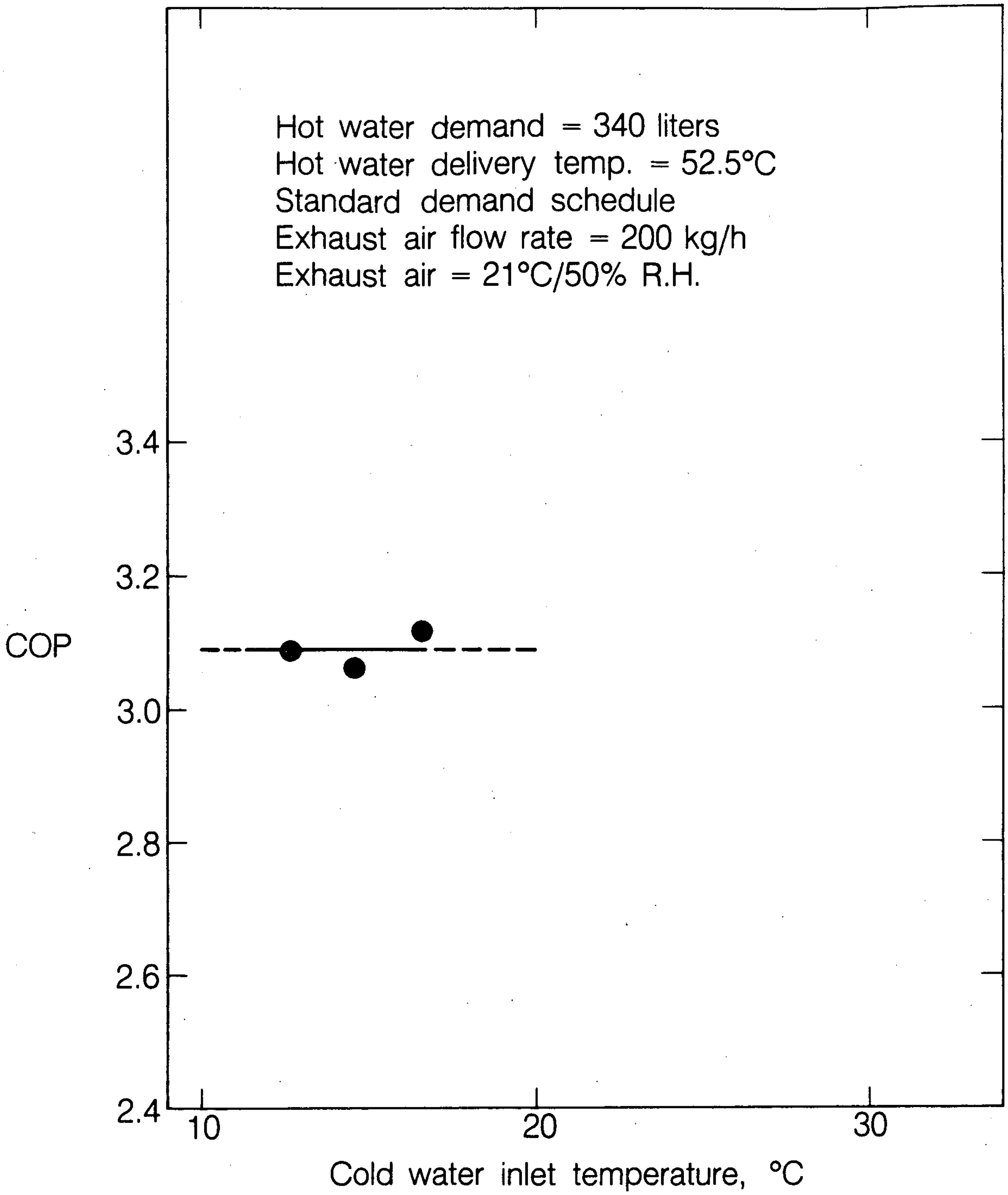


Figure B18. (Water heating only)
Unit A heat-pump COP and hot-water delivery temperature as a function of heat-pump and resistance-heater temperature set points.



XBL 875-8850

Figure B19. (Water heating only)
Energy savings with Unit A heat-pump relative to electric resistance heating as function of cold-water inlet temperature.



XBL 875-8849

Figure B20. (Water heating only)
Unit A heat-pump COP as function of cold-water inlet temperature.

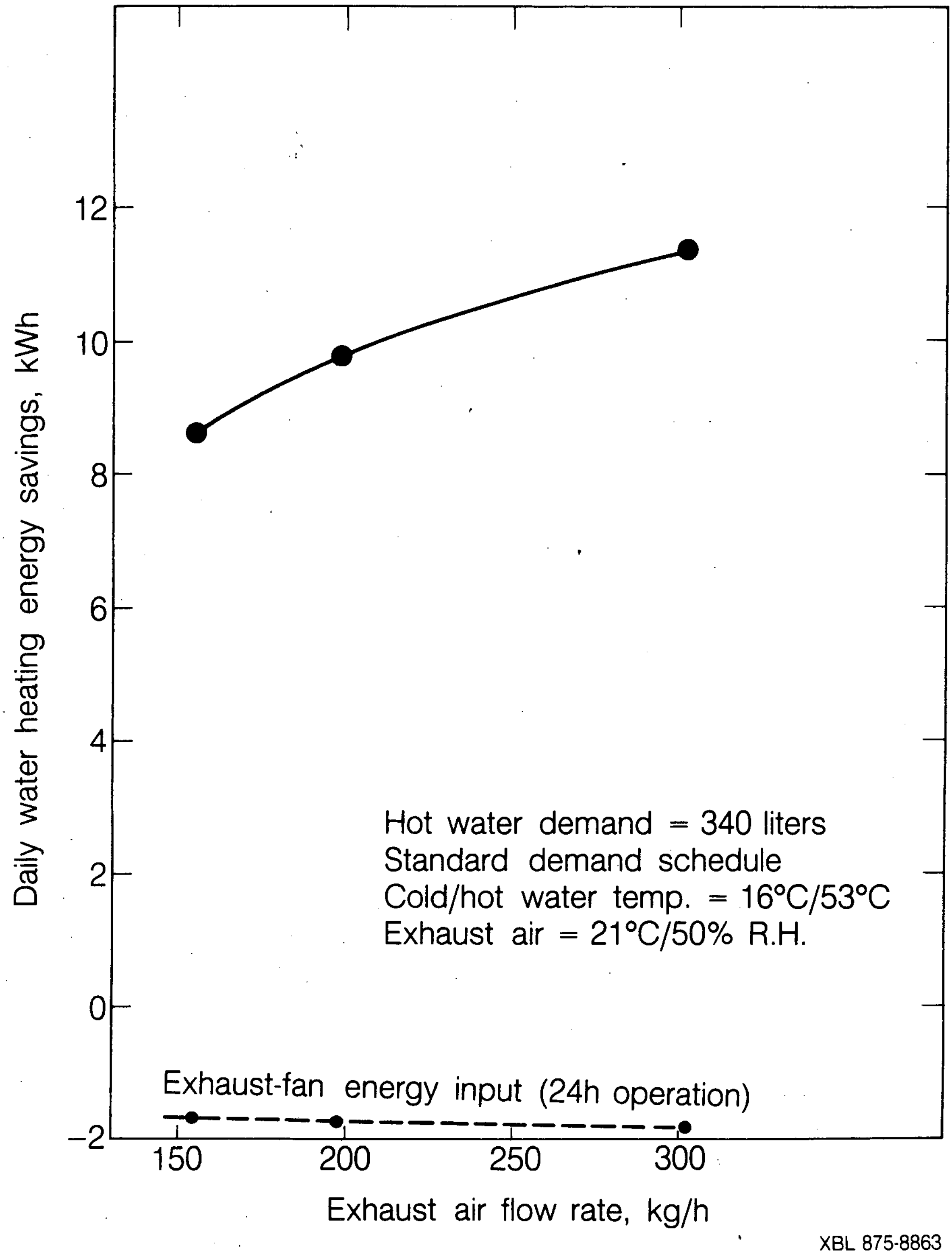


Figure B21. (Water heating only)
Energy savings with Unit A heat pump relative to electric resistance water heating as function of exhaust air flow rate. The energy savings have been debited with the energy required for 24-hour operation of the exhaust fan.

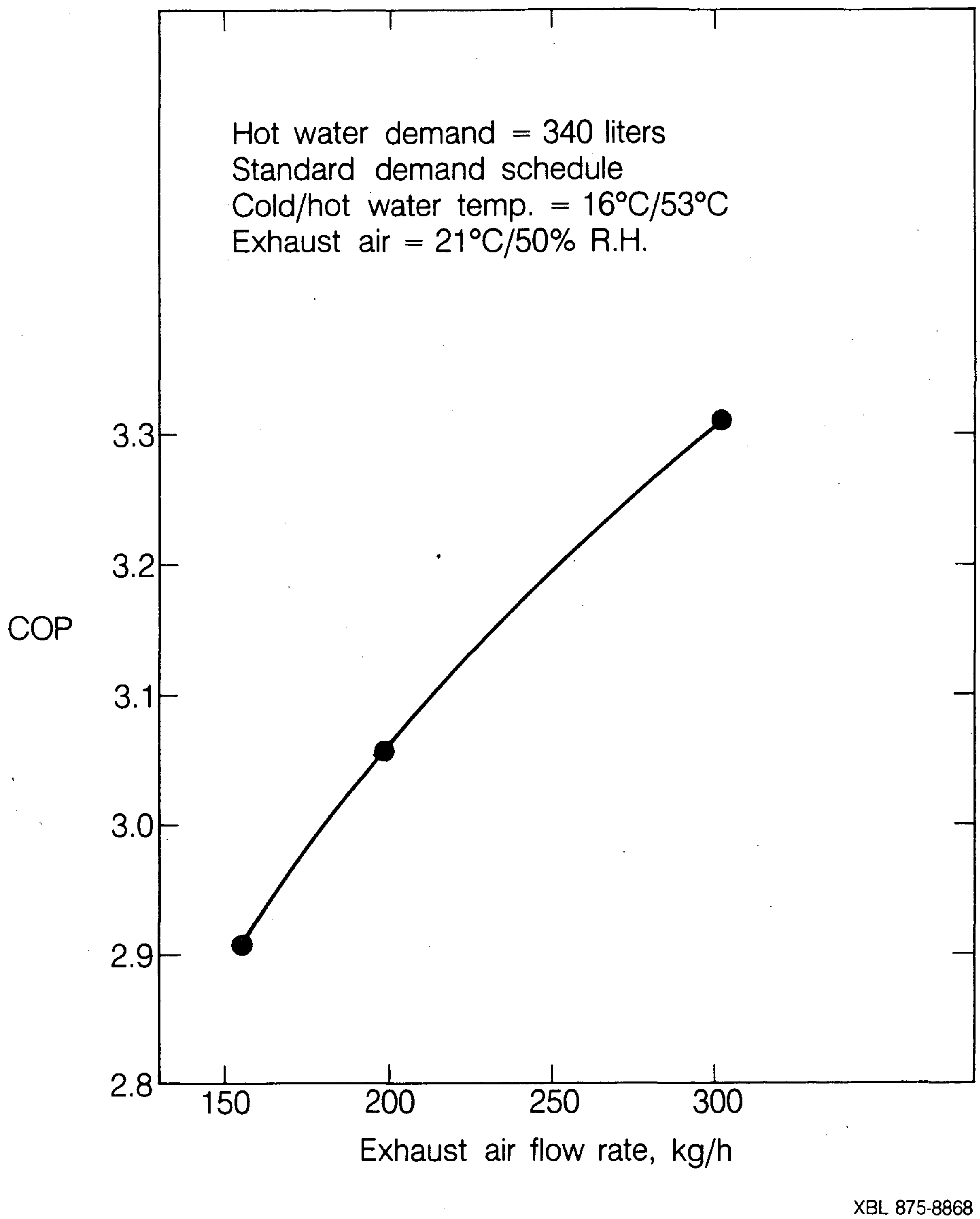


Figure B22. (Water heating only)
Unit A heat-pump COP as function of exhaust air flow rate.

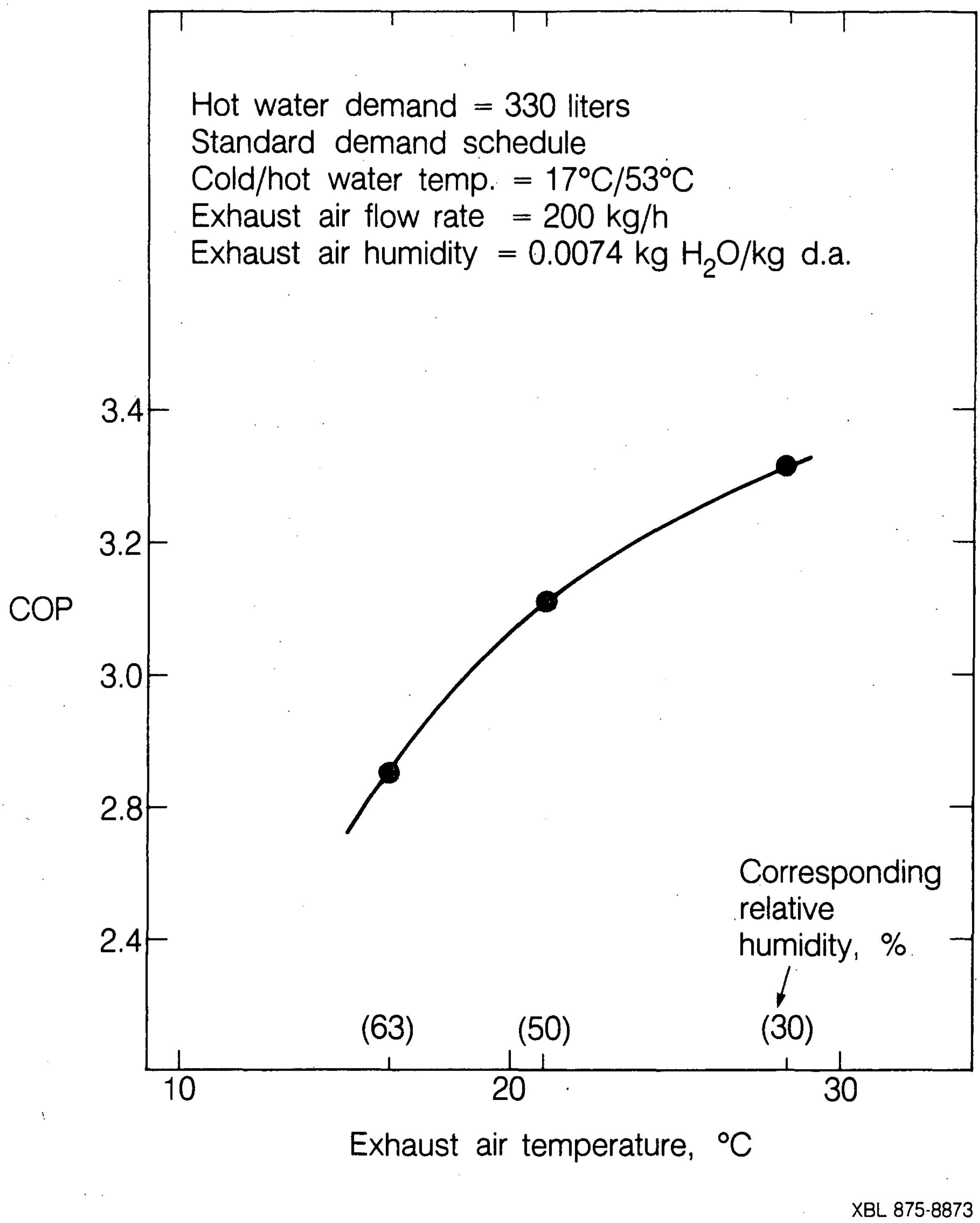


Figure B23. (Water heating only)
Unit A heat-pump COP as function of exhaust air temperature (evaporator inlet temperature).

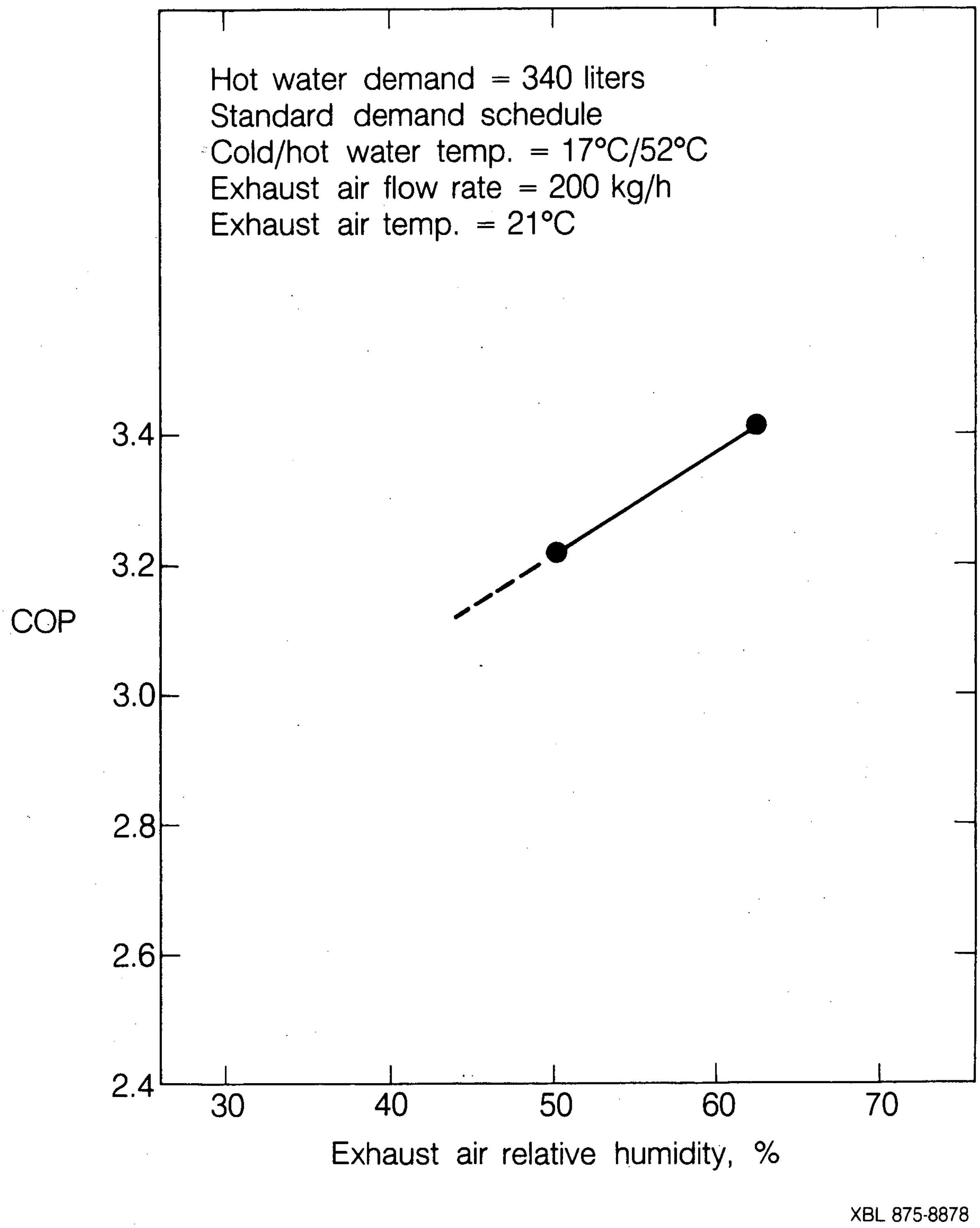


Figure B24. (Water heating only)
Unit A heat-pump COP as function of exhaust air humidity (evaporator inlet condition).

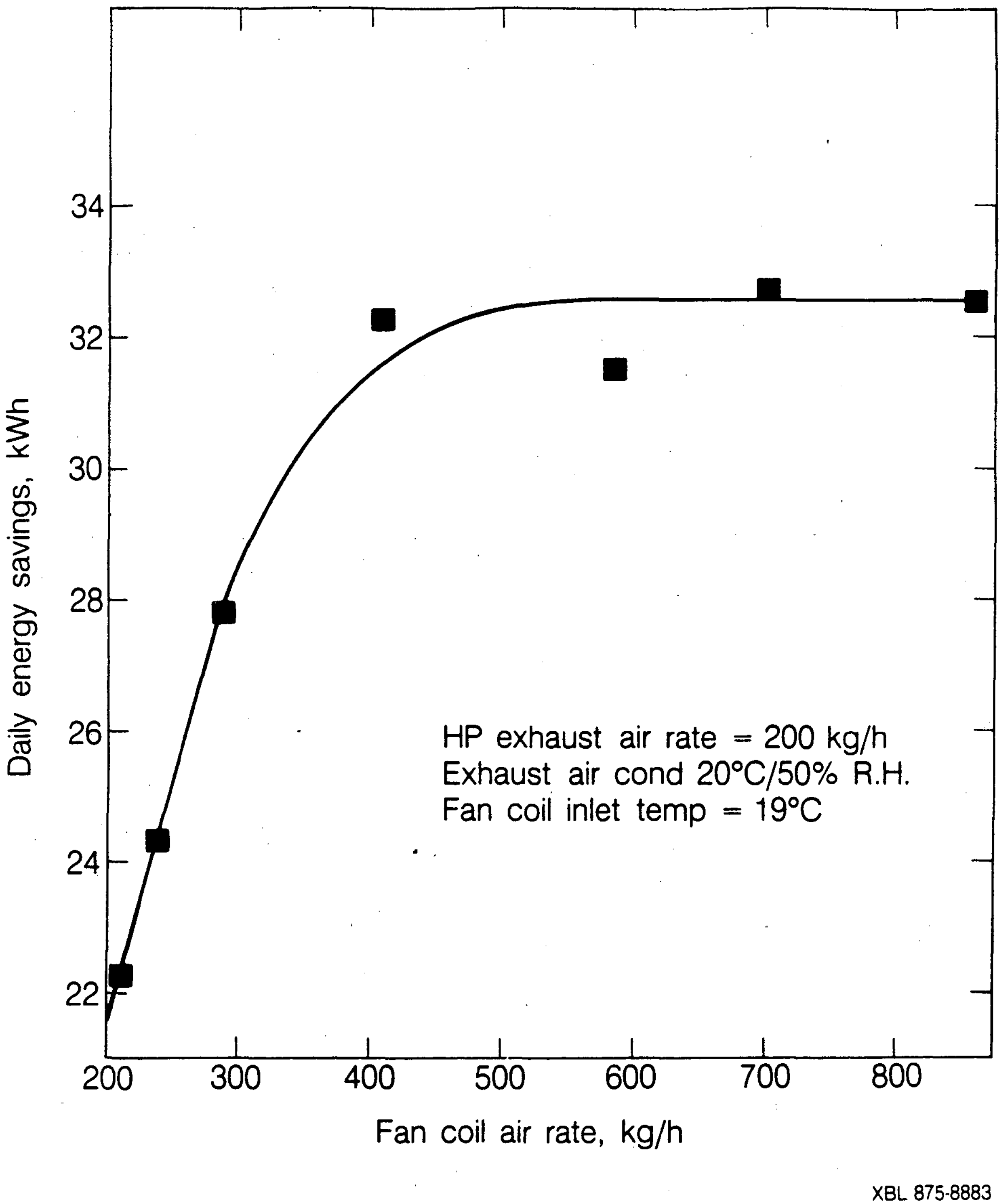


Figure B25. (Space heating only)
Daily energy savings obtained by 24-hour operation of Unit A fan-coil unit relative to electric resistance heating as function of fan-coil air flow rate. The energy savings have been debited with the energy required to operate the exhaust fan (1.7 kWh), but not with the energy required to operate fan-coil fan.

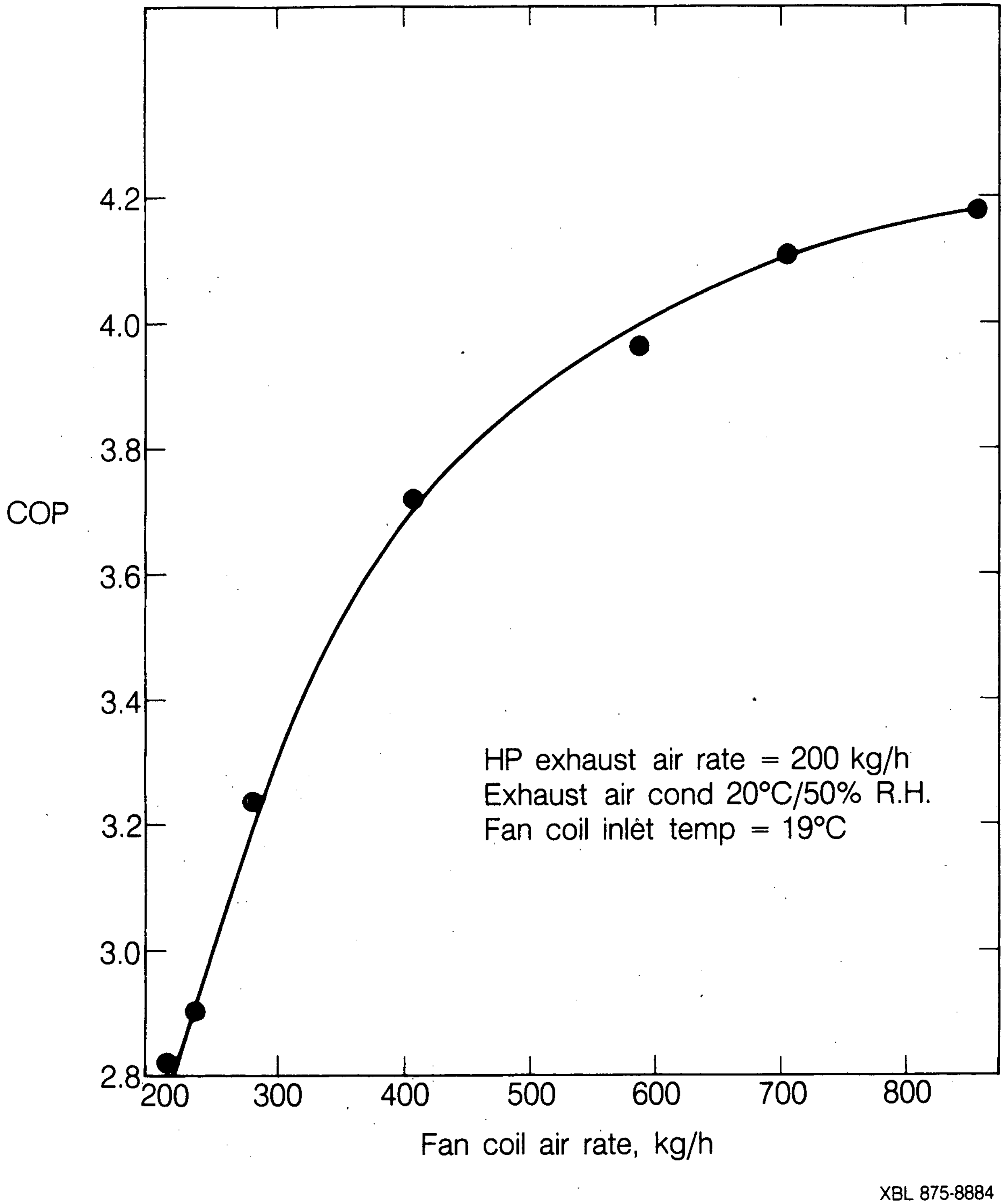


Figure B26. (Space heating only)
Unit A heat-pump COP for operation with the fan-coil condenser as function of fan-coil air flow rate.

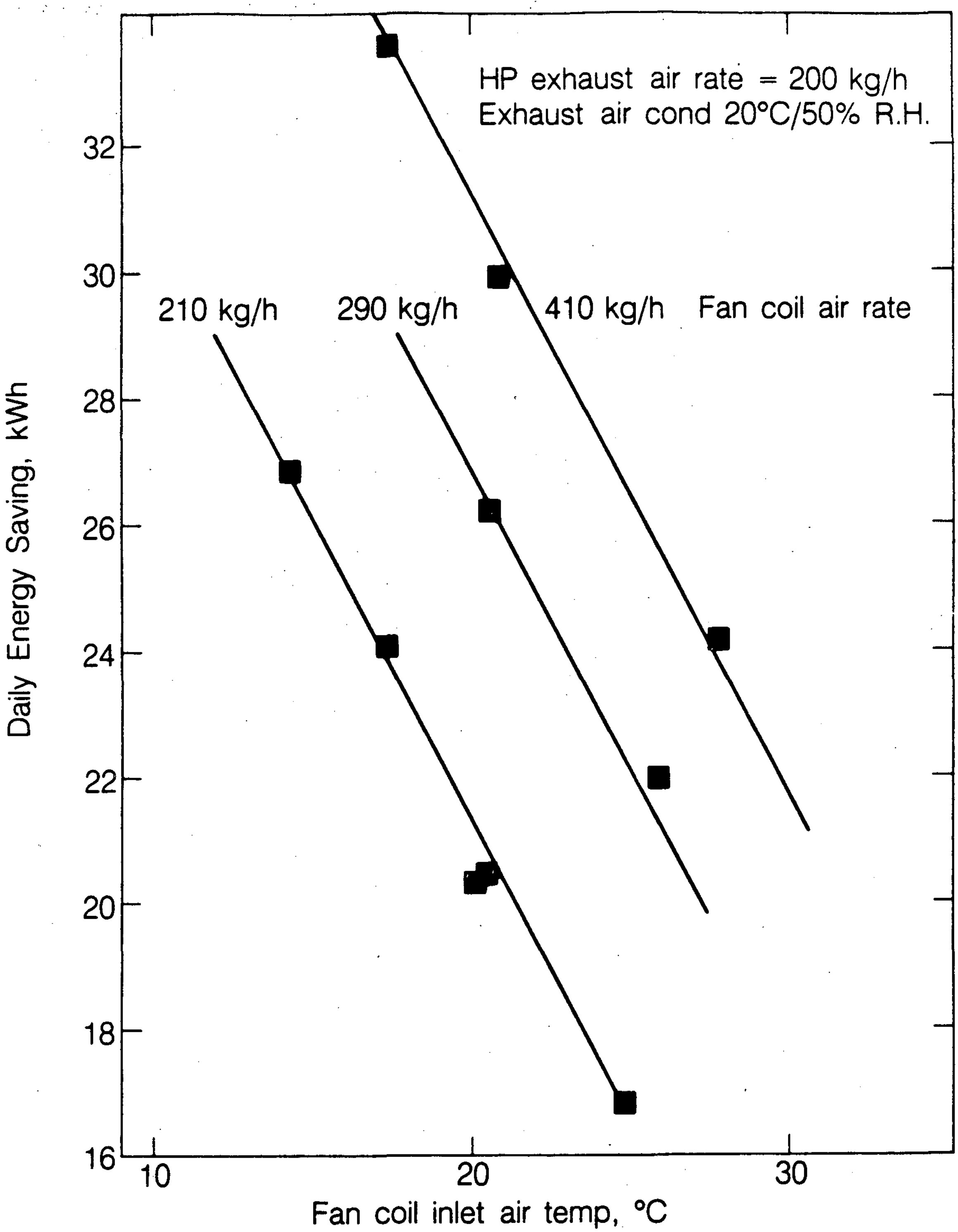


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Figure B27. (Space heating only)
Daily energy savings obtained by 24-hour operation of Unit A fan-coil unit relative to electric resistance heating as function of fan coil inlet air temperature and flow rate. The energy savings have been debited with energy required to operate the exhaust fan (1.7 kWh).

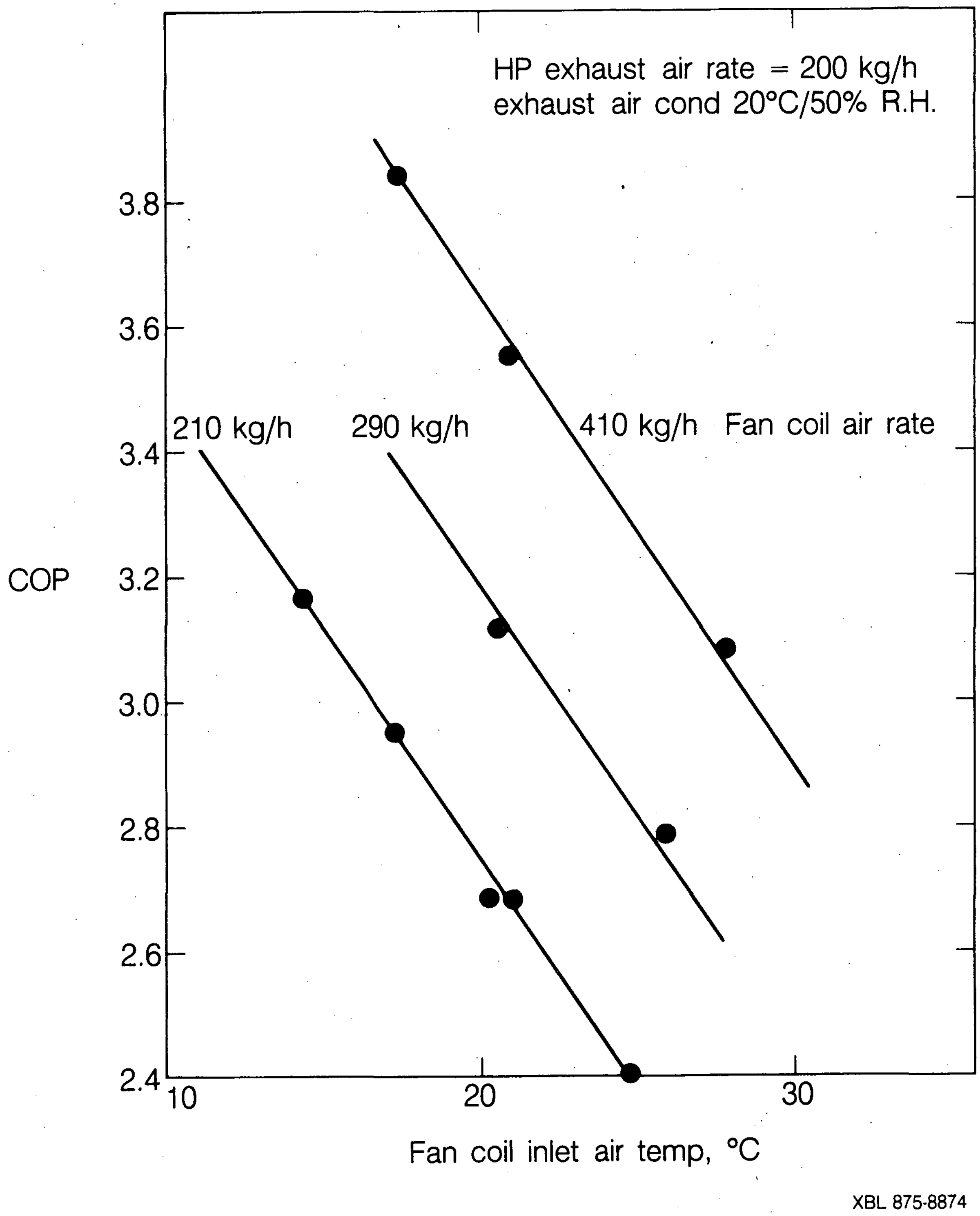

Figure B28. (Space heating only)
Unit A heat-pump COP for operation with the fan coil condenser as function of fan-coil inlet air temperature and flow rate.

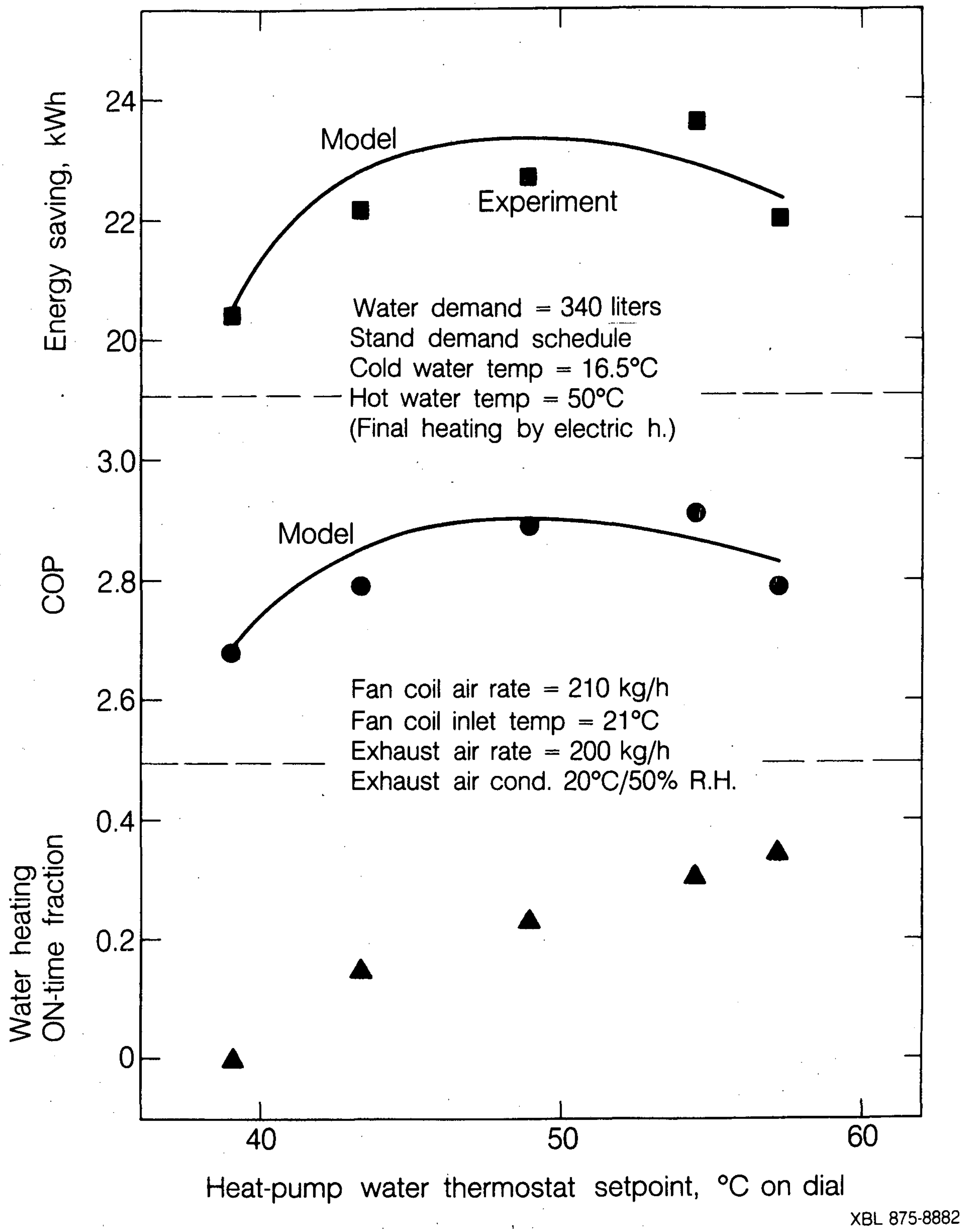

Figure B29. Unit A heat-pump performance in combined water-space heating mode with a 24-hour ON time.

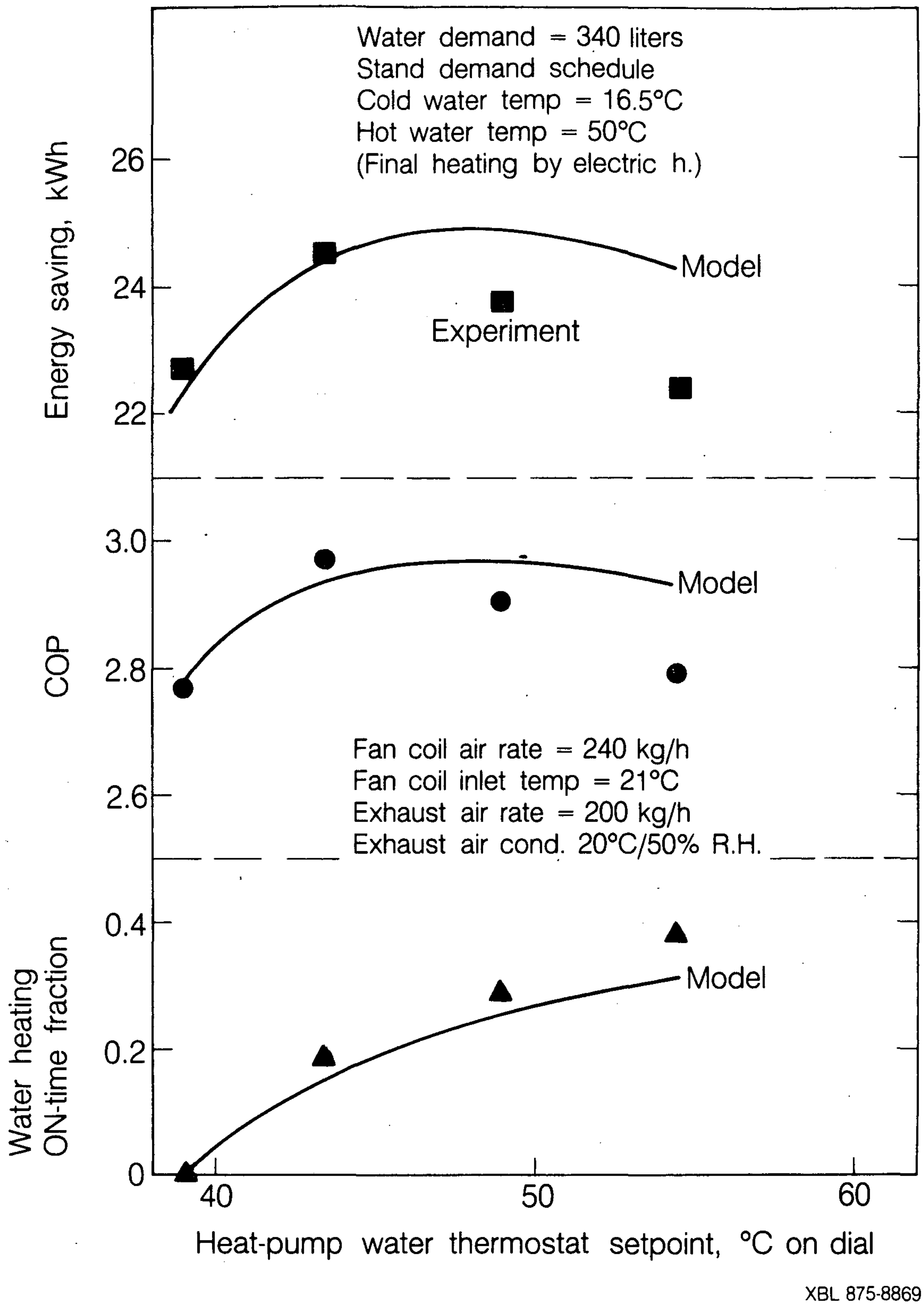

Figure B30. Unit A heat-pump performance in combined water-space heating mode as a function of heat-pump thermostat set point with 24-hour ON time and an excessive charge of refrigerant.

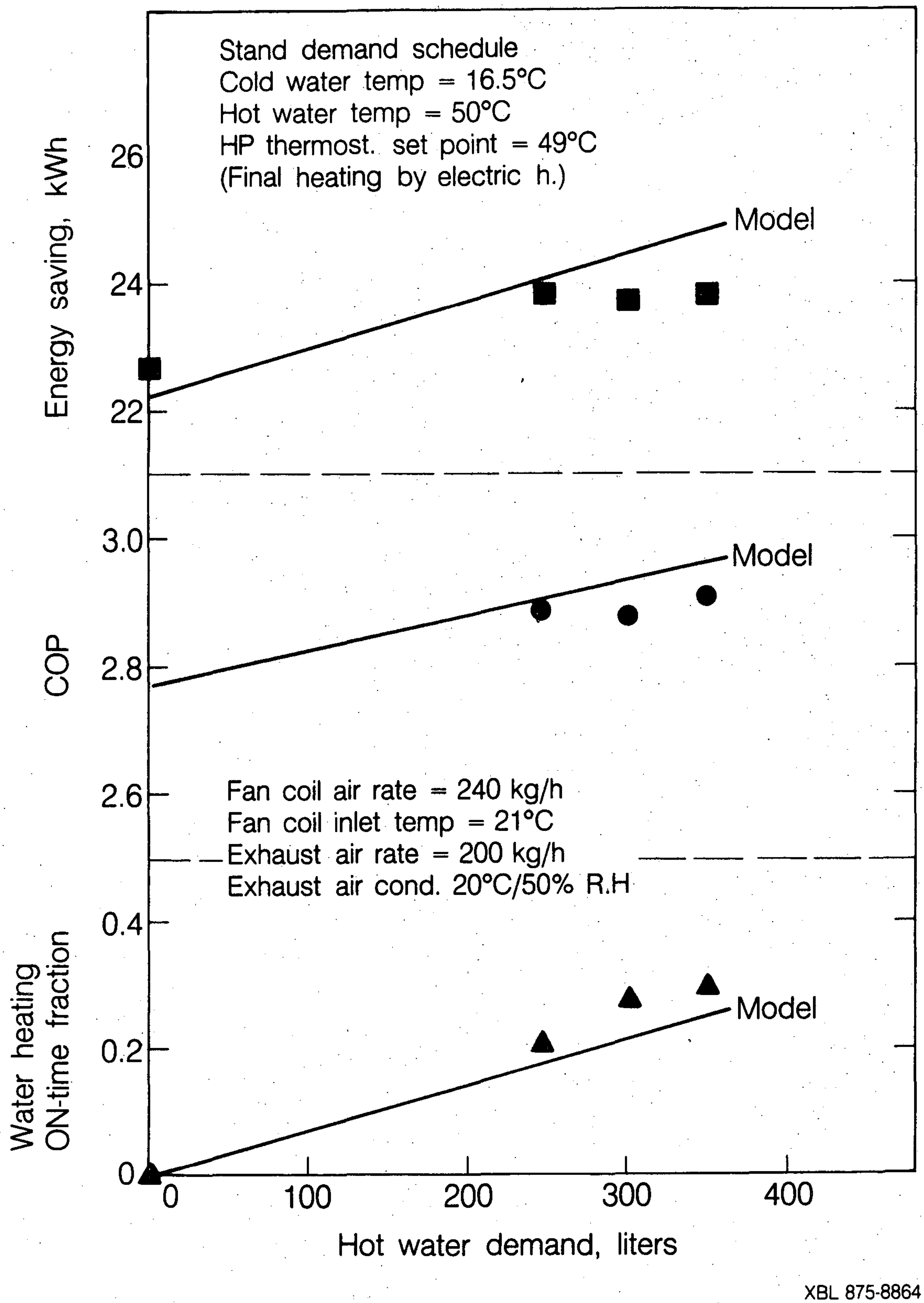

Figure B31. Unit A heat-pump performance in combined water-space heating mode as a function of hot water demand with 24-hour ON time and an excessive charge of refrigerant.

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