ANALYSIS OF A WATER SOURCE HEAT PUMP SYSTEM ON A MUNICIPAL WATER SYSTEM

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A water-source heat pump system was connected to a municipal water system and monitored. The purpose of monitoring was twofold: determine field performance COPs, and determine the probable impacts of system operation on water main temperatures. COPs have been calculated for the system operating in the heating-only and cooling-only modes. Field performance COPs, including water pump energy use were 2.46 (cooling) and 2.99 (heating). Omitting water pump energy, COPs rise to 2.97 and 3.32, respectively. These latter values are 90% and 87% of the equipment's ARI rating values, which do not include pumping energy use. Possible causes of the shortfall from rated performance are discussed. Pump sizing and control are important to achieving good performance. The thermal effects of the heat pump system on the municipal water system are discussed. Depending on capacity, roughly one such system per block appears acceptable from a water temperature impact standpoint. Siting and design considerations for these systems are also discussed.

INTRODUCTION

In the spring of 1989, Lambert Engineering installed monitoring equipment for Pacific Power & Light on a system consisting of three water-source heat pumps. The heat pump system heats, cools, and ventilates a new library for the City of Hermiston, Oregon. Hermiston experiences 5123 heating degree-days and 738 cooling degree-days, on average. Winter (97.5%) and summer (2.5%) design drybulb temperatures are 8°F and 96°F, respectively, for the nearby Umatilla Army Depot.

The main objectives of the monitoring project were twofold: (1) to verify that the HVAC system operates at desirable efficiencies, and; (2) identify the environmental impact of the HVAC system on the city water. Since the system discharges water back to the water main, the effect on the city water temperatures was a primary focus. Feasibility of installing additional heat pump systems on the city water system was of interest.

Three water-source heat pump units are connected in parallel to the municipal water system. Two of the heat pumps are 5-ton (Model #CM 814060) units. The third unit is a 3-ton (Model #CM 814036). None of the heat pumps are equipped with resistance heat. A single-speed water pump circulates water to all three heat pumps any time one or more of the units operates in a mode other than "fan-only" (i.e., when a compressor runs).

All three heat pumps operate in "fan-on" mode (with continuous fan operation) during occupied periods. All three heat pumps operate in "fan-auto" (i.e., the fans run only when heating or cooling is required) mode during vacant times. Ramp recovery type setback thermostats allow recovery to occupied conditions to begin earlier on days when heating or cooling loads are greater.

An 8-channel proprietary data logger was installed, and data collection began 6/19/89. Measurement points included: electrical energy consumption of each heat pump and the water pump, the inlet and outlet water temperature to the heat pump system, and the water flow through the system.

Power measurements were made using proprietary watt transducers, accurate to ± 2% of reading. Temperatures were measured using AC2626 water insertion temperature probes. Absolute accuracy of the temperature measurements is ± 1°F, of better.

The flow measurement was made with an orifice plate inserted into the inlet water stream. The orifice plate was interfaced to a P-3061 Delta-P transducer, which, in turn, was interfaced to the data logger. The data logger calculated flow measurements from the differential pressures reported by the transducer (in inches of water).

Flow measurements were conditionally averaged during periods of pump operation only. All transducers were sampled at least once per second; the data logger stored hourly data summaries. Lotus 1-2-3 was used for data analysis.

HEAT PUMP PERFORMANCE ANALYSIS

This study posed two basic questions: (1) Do such systems have favorable energy efficiency compared to air-to-air heat pumps? and (2) Do city water impacts permit significant use of the concept? Hermiston Library data analysis required two main phases. The first phase involved gaining a thorough understanding of the building's HVAC system. Operating patterns, energy use, thermal interactions with the water source, and COP determinations were extracted from the monitored data. In the second phase, knowledge of the building's HVAC performance was applied to preliminary analysis of city water impact. The objectives of the first phase were to verify that the system achieved an advantageous COP, and determine thermal interaction patterns with the water source. Second phase objectives were to gain a preliminary understanding of siting and city water "environmental impact" concerns. Estimates were made, based on observed heat rejection and extraction data, of temperature effects on downstream water users. These effects would consist of temperature rise during summer, and temperature fall during winter, as a result of heat pump operation. No survey of user tolerances for city water temperature changes was made. Instead, a ± 10°F maximum tolerable change in city water temperature was arbitrarily assumed for purposes of analysis. In short, is the concept worth using, and can the concept be applied to a significant extent?

HVAC Efficiency--Field COP Determination

Verification of favorable operating energy efficiency relied primarily on field COP determinations. However, field COPs are not directly comparable to COPs determined for product rating purposes. Rating COPs are suitable for laboratory repeatability and fair product comparisons. For cost reasons, rating tests do not simulate all aspects of field operation. Different fan energy use, different operating patterns, part-load effects, and other real-world factors produce lower field COPs. Most importantly, ARI standard 320-86 does not include water pumping energy use in rating COPs. This feature of the rating method causes us to refer to two different types of COP. "Field COP" includes water pump energy use. "Pumpless COP" omits pump energy use, and is primarily useful for comparison to rating COPs. The field COPs are akin (but not identical to) rating COPs for air-to-air heat pumps. It is useful to examine field COPs, but it must be recognized that they are likely to be lower than rating COPs. While it is not practical to normalize for all field effects on COP, some can be normalized out or identified. To do so requires identification of field operating characteristics.

HVAC Operating Characteristics

"Daytyping" and examination of raw data were used to determine HVAC operating characteristics. Figure 1 is a sample daytype plot; it shows that:

- Heat pump 3 operated on essentially the same schedule Monday through Thursday in July 1989.
- HP3 started continuous "fan-on" mode about 8 am
- The heat pump switched off very shortly after 8 pm
- The fan, by itself, operates at 984 watts

Examination of similar plots, and the raw data, also established that:

- The heat pumps run in fan-on mode during daytime.

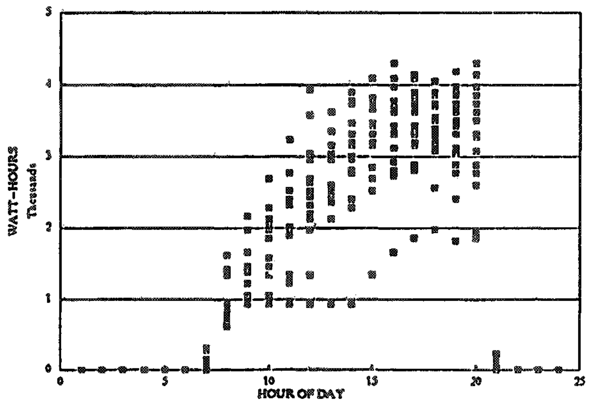

Figure 1. Sample Daytype Plot HP3 Operation, Monday through Thursday, July 1989

- During vacant times, fans run only when heating or cooling is required.
- The water pump runs whenever a compressor operates.
- Intermittent compressor operation is more common than continuous compressor operation.

These operating characteristics represent logical and efficient control and operation. However, they differ from rating COP test conditions.

Heat Transfer Versus Ventilation

Rating COPs address heat transfer performance; energy used solely for ventilation is not a factor. Energy used by the Hermiston system solely for ventilation is not necessary to the heat transfer process. "Fan-only" energy use obscures heat transfer performance, and should be excluded from COP analysis.

Fan-only mode energy use, during fan-on mode operating hours, varies according to heat pump compressor duty cycle. At zero compressor duty, all heat pump energy use is fan energy; at 100% compressor duty, no energy use is attributable to fan-only operation. Separating out fan-only energy therefore required knowledge of three factors--"fan-on" operating schedules, fan-only power levels, and compressor duty cycles.

The fan-on mode hours were extracted from daytype plots. For fractional hours of fan-on mode operation (morning and evening) there was probably some error in determining fan-on mode duty cycle. However, whole hours where heat pumps operated in "fan-on" mode are identified with a high degree of certainty. Fan-on mode duty cycle estimates from daytype plots were used to construct hourly "fan schedules" for the two months analyzed, for each heat pump. The fan power levels were also determined from daytype plots, with the exception of HP1 for December operation. Its fan-only power was not discernable from December data, due to a high compressor duty cycle. The December value was assumed to be the same as July's.

Heat pump compressor duty cycle was assumed to vary linearly from zero to 100% as hourly heat pump power level varied from fan-only levels to maximum, for hours of continuous fan-on operation. Since entering water temperatures were quite steady, assuming a constant value of compressor power is reasonably accurate. Maximum power levels were observed from daytype plots, in most cases. Judgmental corrections were applied, where July maximums did not clearly show "full bore" compressor operation. Compressor duty cycles for all periods of fan operation were estimated to be:

$$\text{Compressor Duty Cycle} = \frac{\text{Heat pump energy - (fan power * fan-on duty cycle)}}{\text{Compressor power * fan-on duty cycle}} \qquad (1)$$

where compressor power is maximum power less fan power. The portion of each heat pump's hourly energy use attributable to heat transfer operation was then calculated as:

$$\text{Heat Transfer Energy Use} = \text{Total HP Wh} - \text{(1-Compressor Duty Cycle)*(Fan Watts)*(Fan-on Duty)} \qquad (2)$$

Values used for computations are shown in Table 1. The resulting "cooling mode" or "heating mode" hourly energy use was used in subsequent heat transfer calculations. Figure 2 shows an example plot of total heat pump energy use versus heat transfer energy use: the left line is "fan-auto" mode hours; the right is fan-on.

Table 1.

	UNIT	FAN WATTS	MAXIMUM WATTS	COMPRESSOR WATTS
July 1989	HP1	460.5	2640	2180
	HP2	590	5300	4710
	HP3	984	5000	4016
Dec 1989	HP1	460.5**	2650	2190
	HP2	590	4875	4285
	HP3	748*	4400	3652

* Lower than July due to a filter change

** Assumed identical to July

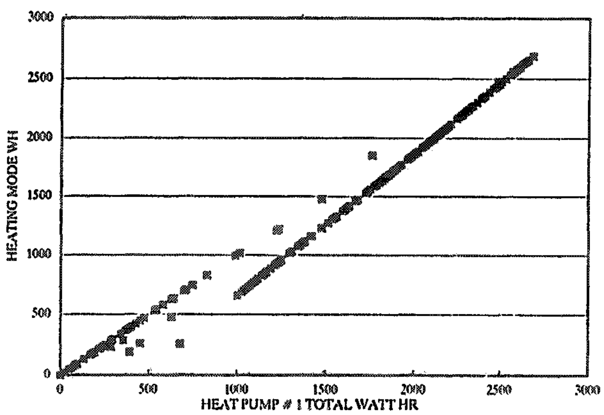

Figure 2. HP1 Heating Mode Watt Hours vs. Total Watt Hours, December 1989

Water Pump Operation

Water-to-air heat pump ratings per ARI 320-86 do not include water pump energy use. The Hermiston system uses a single water pump sized to serve all three heat pumps. As a consequence, hourly field COPs are not only different from rating COPs, but vary depending on how many heat pumps operate. When only one or two heat pumps are running, pump energy use is relatively high. This effect is shown in Figure 3. Pump energy use per heat pump heating watt-hour is significantly higher at light loading conditions. The scatter in the left-hand portion of Figure 3 represents different combinations of heat pumps operating. For example, when the three ton heat pump operates by itself, pump watts per heat pump watt will be quite high. But when two or three heat pumps run simultaneously for part of the hour, pump watts per heat pump watt will be much lower. This effect accounts for a major portion of system COP degradation at light loadings,as will be shown later.

Other Part-Load Effects

As noted, the Hermiston system does not operate at 100% compressor duty cycle, most of the time. This is a natural consequence of sizing for maximum expected loads. Rating tests are typically performed using steady 100% compressor operation. This results in additional COP shortfall, relative to rating test COPs.

Other Field Conditions

Heat pump #3 suffered a dirty filter in July. All filters were changed in August 1989. Although HP3 is the only unit with a demonstrably higher fan energy use due to a dirty filter, this same condition may have existed for the other units. Also, the data do not provide any assurance that either the fans or the water pump were optimally sized.

Water-Side Heat Transfer

Field COPs also required calculating waterside heat transfer. Given perfect measurements, calculating

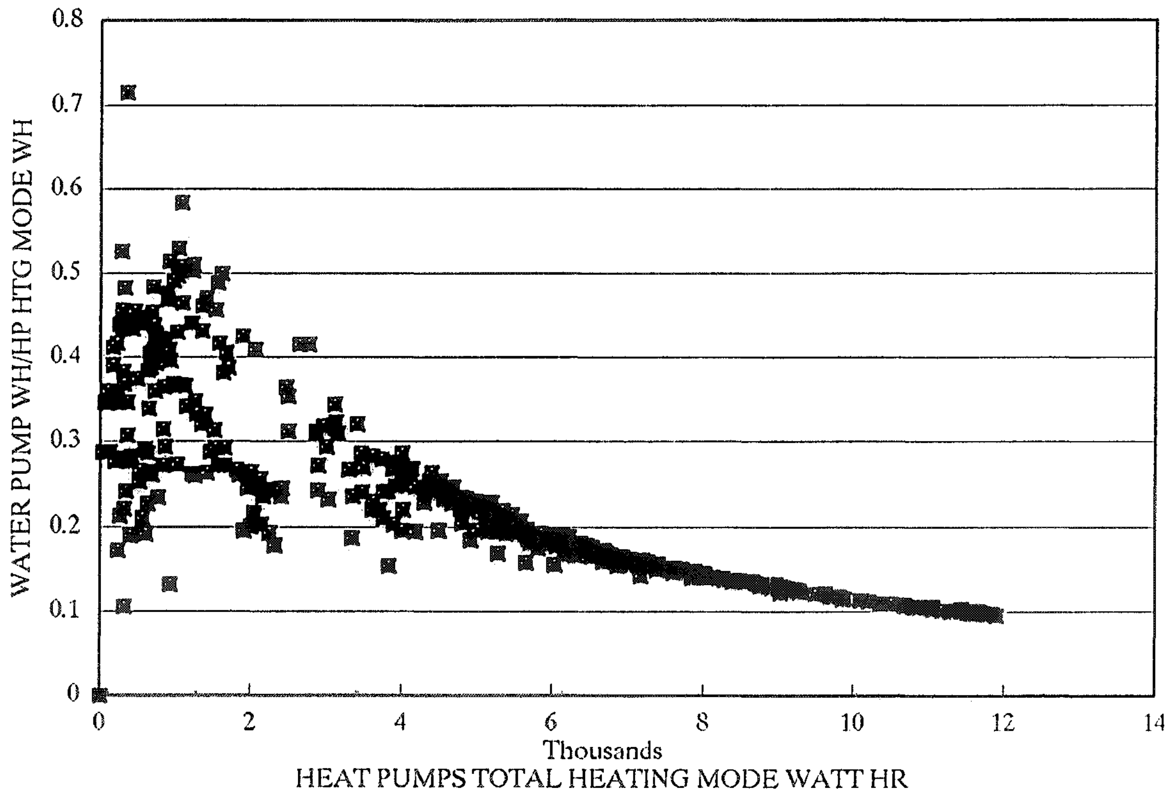

Figure 3. Water Pump Energy Use as a Fraction of Heat Pump Heating Mode Energy Use, vs. Hourly Heat Pump Energy Use, December 1989

heat transfer with the city water would be straightforward. However, the measured inlet and outlet water temperatures are subject to small errors of two kinds. First, there are slight differences in calibration between temperature sensors. Second, the inlet and outlet water temperatures are influenced by mechanical room ambient conditions. During intermittent pump operation, these ambient effects alter the apparent temperature difference across the water loop.

Figure 4 shows these two effects, for July data. The magnitude of a "perfect" Delta T (inlet temperature-outlet temperature) would increase very nearly linearly with increasing cooling energy use, starting from zero. Instead, the "raw" Delta T starts near -2.7°F. Room ambient effect is noticeable at low cooling energy use, which corresponds to low pump duty cycle. Sensor calibration difference is also apparent. The portion of the curve at high cooling energy use (essentially 100% pump duty cycle) does not project backward precisely to zero Delta T, but to a value near -1°F.

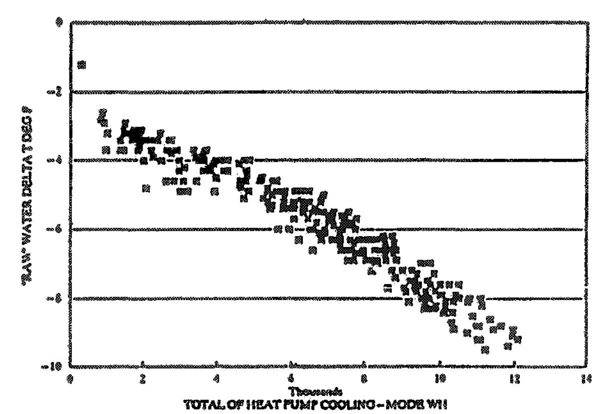

Figure 4. Raw Waterside Delta T vs. Heat Pump Cooling-Mode Watt Hours, July 1989

Small differences (<1°F) in the temperature response of the two water temperatures sensors can be expected; this difference can vary with absolute temperature. Fortunately, there are boundary conditions that can be used to identify and correct for small sensor errors. Using error analysis, it can be shown that:

$$\text{Adjusted Delta T} = DT - E_o + DT_o\,(PDC-1) \tag{3}$$

where Adjusted Delta T = is the corrected value of inlet-outlet water temp, for pump-on conditions;
DT = is the "raw" value of inlet-outlet water temp;
E_o = is the calibration offset error, °F;
DT_o = is the room-ambient effect (°F) with pump off;
PDC = is the water pump duty cycle

The portion of the Delta T versus heat pump cooling (or heating) watt hours which represent full-time water pump operation should intersect the Delta T axis at the value of offset error E_o. Regressions of "raw" Delta T versus heat pump watt-hours, using only values of heat pump watt hours above 8000 watt-hours, give the desired results. For July Data (inlet water temperature near 64°F) the resulting value is -.92°F. For December (entering water temperature near 57°F) the value is -.66°F.

The value of $(DT_o + E_o)$ may be similarly determined. In this case, the regression was run for *low* values of heat pump energy use. Under these conditions, the intercept is the limit of the error $(DT_o + E_o)$ as pump duty cycle approaches zero from the positive side.

The value of $(DT_o + E_o)$ obtained for July, using values of 1000 to 3000 watt-hours for the regression, is -2.67°F. This gives a value of -1.75°F for DT_o, the room ambient effect on $T_{in} - T_{out}$. This effect diminishes with increasing pump cycle (PDC). PDC was calculated using PDC= pump watt-hours/1138. Applying these corrections to the "raw" Delta T gives a July temperature difference versus heat pump cooling watt-hours curve as shown in Figure 5. Similar processing of December data gave a December value of DT_o = 1.18°F. The corrected values of Delta T were used in the heat transfer calculations.

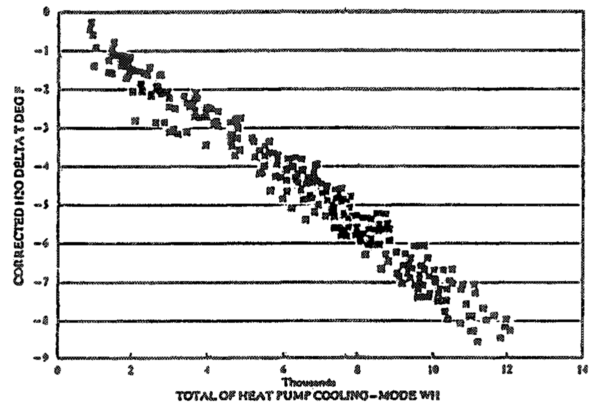

Figure 5. Corrected Waterside Delta T vs. Heat Pump Cooling-Mode Watt Hours, July 1989

Hourly heat transfer was computed as the product of hourly water flow, in pounds, times corrected Delta T. These values were used to compute hourly COP values.

COP Calculations

Hourly COPs were calculated using:

$$\text{Cooling COP} = \frac{\text{Heat Rejected - Cooling Energy Use}}{\text{Cooling Energy Use}} \tag{4}$$

$$\text{Heating COP} = \frac{\text{Heat Extracted + Heating Energy Use}}{\text{Heating Energy Use}} \tag{5}$$

Cooling and heating energy use both include water pump energy use as well as heat pump heat transfer mode energy use. For both July and December, hourly COP values at very low values of heat pump energy use show high scatter. The scatter results from imperfections in both the waterside Delta T corrections, and the fan schedule determinations for partial hours of fan-on operation. Also, the waterside Delta T corrections are necessarily based on average conditions, and did not work as well for atypical conditions. Cooling and Heating COP plots versus loading are shown in Figures 6 and 7. Linear fits for these data give:

$$\text{Cooling COP} = 1.976 + 6.23 \times 10^{-5} \times \text{cooling Wh} \tag{6}$$

$$\text{Heating COP} = 2.86 + 1.407 \times 10^{-5} \times \text{heating Wh} \tag{7}$$

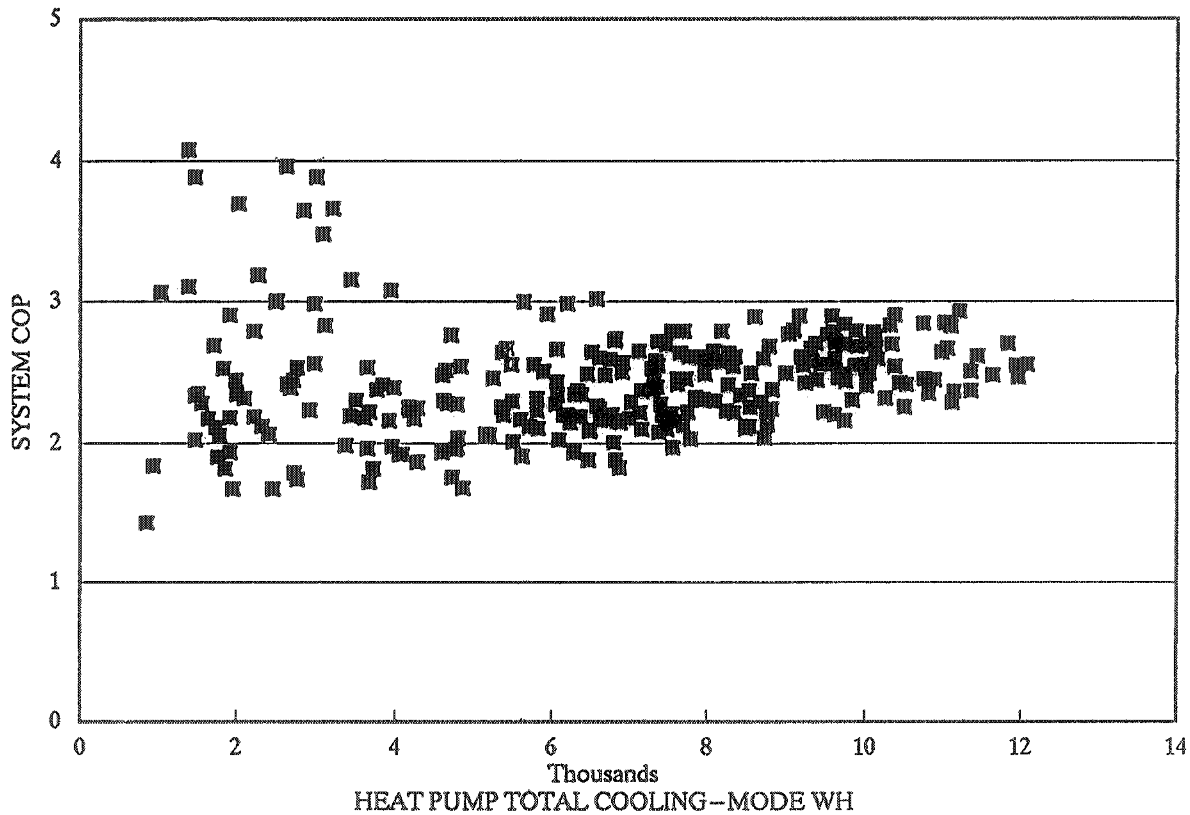

Figure 6. Hourly Field COP vs. Heat Pump Total Cooling-Mode Watt Hours, July 1989

These hourly Field COP plots show COP reduction due to disproportionately high pump energy use at part load, particularly for cooling data. For purposes of comparison between field performance and rating COPs, it is useful to compute COP of the Hermiston system without considering water pump energy use. Figure 8 shows a plot of "Pumpless COP" for July data. As one would expect, the hourly COPs are higher than those shown in Figure 6. The variation in pumpless COP with hourly cooling mode energy use is reduced, compared to the Field Cop; a regression shows that the data are nearly trendless in this respect. If there is a residual part load effect, it is probably due to intermittent compressor operation.

Because of low end "scatter" in heat transfer data, the above fits were used to generate whole-month values for heat rejection (July = 2.53×10^7 BTU) and heat extraction (December = 1.87×10^7 BTU). Whole-month values for both Field Cop and Pumpless COP were computed based on these values. They are shown in Table 2, with ARI rating values for comparison.

The previously noted dirty filter depressed the cooling field COP by 0.07. Also, use of a single pump, rather than three smaller pumps, adversely affected both Field COPs. Had pump energy per heat pump watt-hour remained constant at 0.1 (the value at the right of Figure 3), the Field COPs would have been 2.64 (cooling) and 3.12 (heating).

Comparison of these COPs to typical air-to-air values is difficult, since the ARI values are not directly comparable. Field COP data for air-to-air units are scarce. Comparisons of measured versus rated seasonal COP values in (Brewster 1987) suggest air-to-air units typically achieve about 85% of seasonal values derived from ARI ratings. This

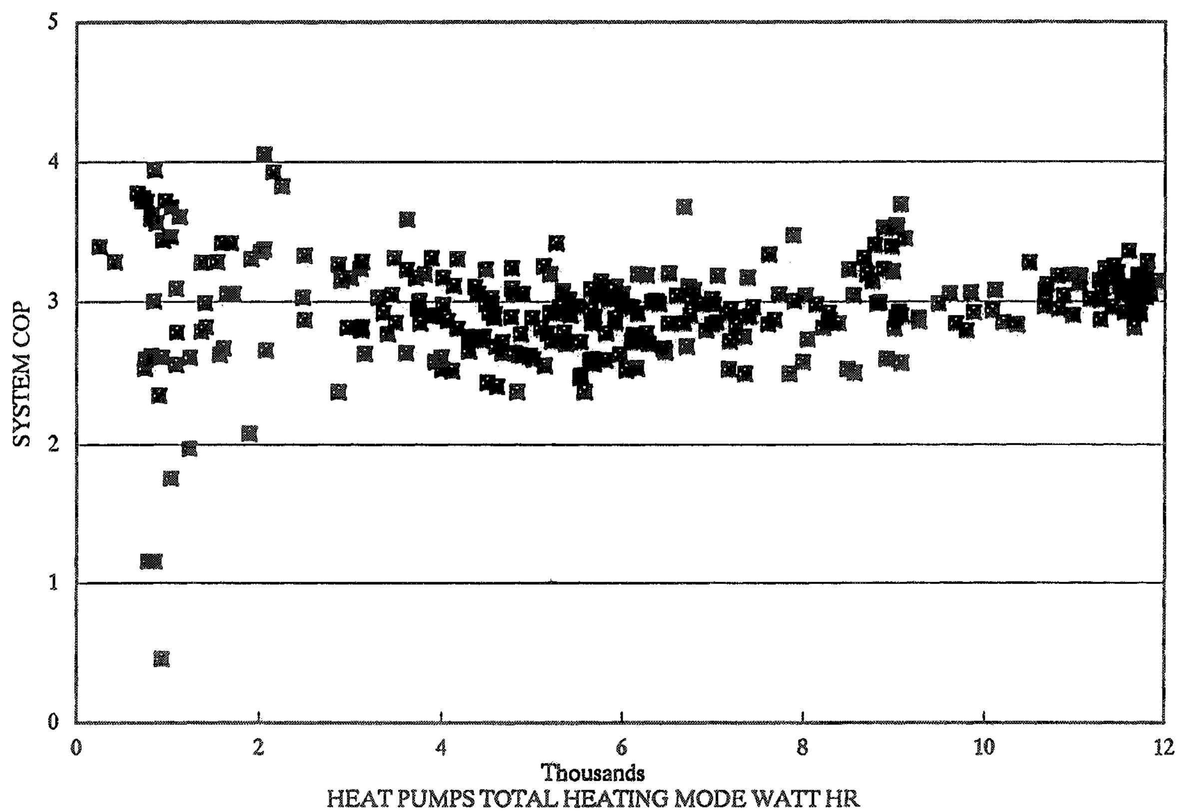

Figure 7. Hourly Field COP vs. Heat Pump Total Heating-Mode Watt Hours, December 1989

implies that an air-to-air unit would have to have a rated heating COP of 3.51 to equal the measured heating performance of the Hermiston system in December. This leads us to tentatively conclude that the Hermiston system operates at a favorable heating COP, compared to typical air-to-air units. Cooling performance (Field EER=8.4) is slightly less than typical rated values for air-to-air units. The lack of cooling Field COP or EER data for air-to-air units prevents a direct comparison.

SYSTEM IMPACT ON WATER MAIN TEMPERATURES

The Hermiston HVAC system discharged its "source" water back to the city water main. Without dilution of discharge water, by mixing with flow to downstream water users, the temperature effects of such a system could be unacceptable. Siting and extent of use of such systems must account for these temperature effects.

City water main temperatures and flow downstream of the Hermiston Library were not monitored, since there was no means of establishing whether the Hermiston water main flows were meaningfully representative of such installations. Instead, an engineering analysis, using Hermiston Library waterside heat transfer data, and metered water consumption data was performed. The City of Bend Oregon provided metered water use data.

CITY WATER SYSTEM CHARACTERISTICS

City water main flows may be roughly categorized into three situations: (1) Transmission mains, serving large numbers of downstream users, with high average water flows; (2) "Grid" pipes, serving local users in the same block or neighborhood, with moderate average water flows; (3) "End of Line" pipes, with few or no downstream users, and lower than average flows.

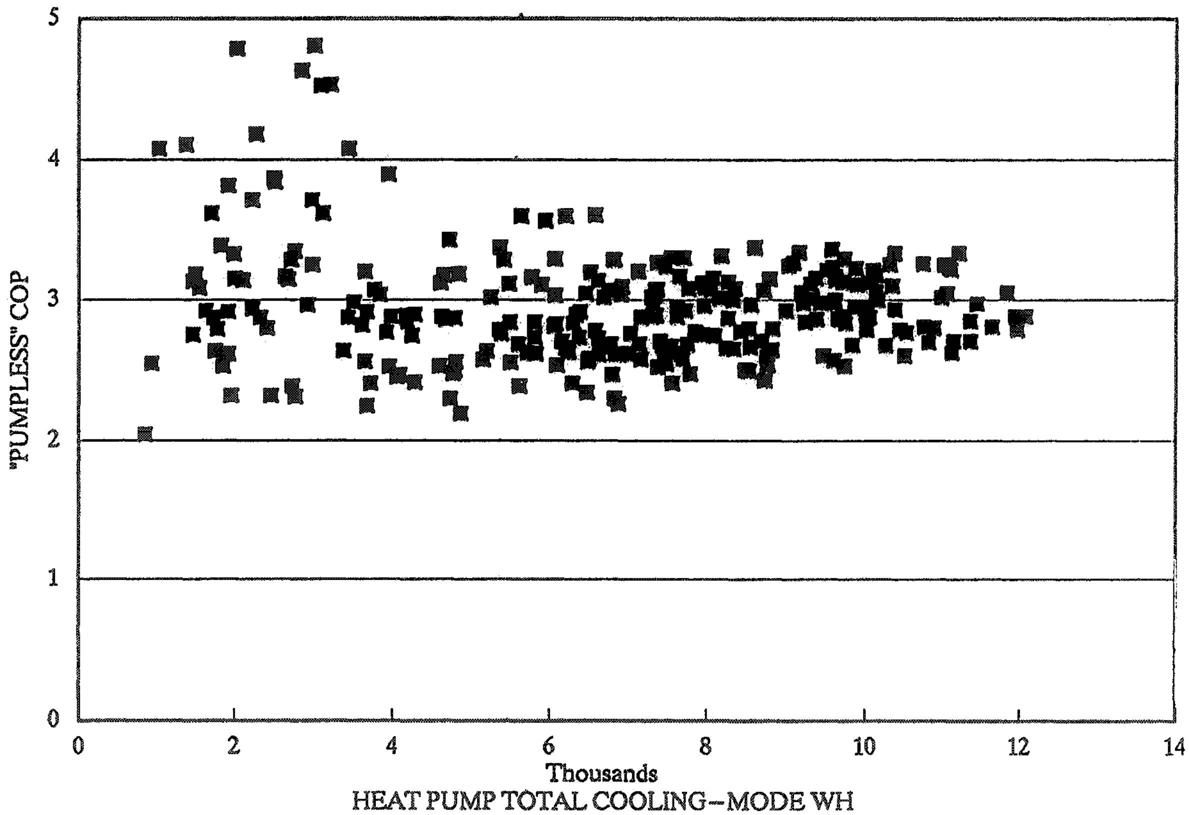

Figure 8. Hourly "Pumpless COP" vs. Heat Pump Total Cooling-Mode Watt Hours, July 1989

Table 2. Summary of COP Data

Mode & Period	Field Cop	Pumpless Cop	ARI COP
Cooling-July 89	2.46	2.97	3.29
Heating-Dec 89	2.99	3.32	3.82

These represent diverse ranges of "heat sinking" capability. The goal is to identify the heat pump tonnage that can be installed without adverse effects. Number of "downstream users" differentiates the above three situations. This analysis therefore addresses allowable heat pump tonnage per downstream user. To do so, the water use of "downstream users" needs characterization.

METERED WATER CONSUMPTION DATA

Water use will vary according to the type of user, and by season for some users. In addition, water use varies substantially from day to night. The City of Bend Water Department provided metered water use data for (1) Downtown Commercial Area -

(2 winter months); (2) Residential Properties - (2 winter months); (3) Residential Properties - (3 summer months).

The "Downtown Commercial Area" data is a street one block long, including buildings on both sides of the street. Water users include offices, retail dry goods establishments, a theater, and restaurants. The area sampled has predominantly two-story buildings. There are 18 metered customers in the sample.

The "Residential" sample is for 8 buildings with a total of 14 occupied living units. They are not on a contiguous block. All but one of the duplexes has lawn areas that are watered in the summer.

Commercial water use averaged 62.94 cubic ft (2053 lb) per customer-day with a standard deviation of 117.94 cubic ft (3847 lb) per customer-day. This is assumed to be relatively non-seasonal.

Residential water use in winter averaged 23.77 cubic ft (775 lb) per unit-day with a standard deviation of 6.57 cubic ft (214 lb) per unit-day. Residential summer use averaged 76.79 cubic ft (2505 lb) per unit-day with a standard deviation of 43.42 cubic ft (1416 lb) per unit-day.

The commercial daily use pattern is assumed to be primarily during business hours, with some use extending into early evening. This is an excellent match with the Hermiston Library's heat pump operating schedule. The residential use pattern is assumed to be mostly from 6 am to 10 pm, also a reasonably good match.

WATER MAIN TEMPERATURE EFFECTS

Holding temperature effects experienced by downstream water users within acceptable limits will require a diluting flow, generated by these same downstream users. Hermiston Library's HVAC system has a nominal 13 ton capacity. The summer heat rejection rate for July 1989 was 62,846 BTU/ton-day. The winter heat extraction rate for December 1989 was 46,330 BTU/ton-day.

We neglected ground effects, and assumed a ± 10°F maximum change in water main temperature is acceptable. For these conditions, the "downstream users" required per ton of installed capacity are either 3.06 "average" commercial users, or 6 residences.

The limiting condition for commercial downstream users is summertime heat rejection. (Wintertime maximum temperature change experienced by commercial "downstream users" would be about -7.4°F) The limiting condition for residential downstream users is wintertime heat extraction. Summertime maximum temperature change experienced would be about +4.2°F, with 100% residential downstream users.

Several comments and qualifications apply to these conclusions. First, the actual maximum effects on city water, experienced by downstream users, will likely be less than those stated. This is due to heat transfer between the water main and surrounding soil. With roughly one or two systems per block, there is significant ground-coupling of the water main, per system. Second, the temperature effects stated are associated with seasonal extremes; most of the year, temperature effects will be significantly smaller. Third, use of night setback thermostats on such systems, as was done at the Hermiston Library, is recommended. Primarily daytime heating and cooling will serve to optimize the "match" of heat rejection and extraction patterns to water use patterns of downstream users. Fourth, the variability of water use among commercial users is high. Where the number of downstream commercial users is limited, some consideration should be given to whether their use is likely to be above or below average, according to business type.

CONCLUSIONS

The system operated in cooling mode with a Field COP of 2.46; heating mode Field COP was 2.99. Cooling mode COP was reduced by a dirty filter. Cooling and heating COPs were both depressed by high water pump energy use at part load. Field COPs would have been about 2.71 (cooling) and 3.12 (heating) without these effects. "Pumpless COP" values indicate that system field performance achieved was 90% (cooling) and 87% (heating) of ARI rating.

Available Field COP data for air-to-air heat pumps suggest the Hermiston system's field performance in heating mode was probably better than a typical air-to-air unit. No air-to-air field cooling COP data was found for a cooling comparison.

The system showed significant reductions in Field COP when only partially loaded. This was due primarily to the water pump configuration. Increased COPs may be realizable with a different pump arrangement, such as using three smaller pumps, one for each heat pump unit.

The environmental effects of the heat pump system on the municipal water system are less certain. Scoping calculations indicate that potential penetration rates may be limited--approximately one system per city block. Additional modeling and/or monitoring is required.

Other environmental effects should be considered when siting these types of systems. Ground coupling of the city water lines may reduce the thermal loading of heat pump systems to the city water system, and increase potential penetration rates. Penetration rates will vary depending on the siting of systems along transmission lines versus branch circuits. Short circuiting of the inlet and discharge lines during periods of low flow in the city water pipes may occur if too close to each other. Computer models may be able to optimize the location and number of heat pumps throughout a water system.

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