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DISCLAIMER

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ABSTRACT

Using the state-of-the art computer program, DOE 2.1A, the hourly and resulting seasonal coefficients of performance of four residential heat pumps were simulated for a typical house in one city in each of the four major climate zones in the U.S. For purposes of comparison a conventional gas furnace and air conditioner system were also analyzed. This study shows that heat pump systems could be developed with seasonal thermodynamic efficiencies of over 200 % in resource energy. In addition, sizing the heat pumps according to a building's thermal integrity and operating conditions maximizes seasonal efficiencies.

KEYWORDS

residential heat pump systems; space conditioning; computer simulation; heat pump sizing; energy efficient heat pump designs.

INTRODUCTION

This paper examines the seasonal performance of four different heat pump systems used or proposed to be used in residential buildings. These four heat pump systems are the only ones, at the present time, applicable to residential buildings. Systems considered include both those at the demonstration level of development such as gas-fired absorption and Stirling/Rankine heat pumps in addition to commercially available electric air source and water source heat pumps.

A typical one story detached single family house with each of the above space conditioning systems is modelled on DOE 2.1A, a state of the art computer program. The program simulates the yearly hour-by-hour performance of a building for any location for which weather data (temperature, solar insolation, wind speed and direction, and humidity) are available. Since DOE 2.1A performs hourly calculations, it is able to model the degradation of the performance of a heat pump due to cycling, defrosting, and auxillary resistance heating.

Our analysis was carried out for cities within the four major climatic regions of the United States (Minneapolis: cold; Phoenix: hot-arid; Houston: hot-humid; New York: temperate). In order to aid in making comparisons, we also modeled a conventional space conditioning system comprised of a gas furnace and an electric air conditioner.

Computer simulations covering the spectrum of current equipment sizes in residential buildings help to determine the effects of sizing on seasonal performance. Two locations and a sequence of heat pump capacities serve as the parameters for this study. Finally, additional test simulations compare the predicted seasonal efficiencies against measured efficiencies for an electric air source heat pump in the hot-humid climate zone.

METHODOLOGY AND SYSTEMS DESCRIPTION

The prototype house used is a one story building with 143 sq. m (1540 sq. ft.) of floor area and 14.3 sq. m (154 sq. ft.) of window area. It represents a typical single family house built in the U.S. at the present time [1]. Foundations are either basements (Minneapolis, New York City) or slabs on grade (Houston, Phoenix) in accordance with regional practices [2]. The levels of thermal integrity used in the simulation studies are given in Table 1. The house is kept at 21.1 °C (70 °F) from 7 a.m. to 12 midnight and at 15.6 °C (60 °F) from midnight to 6 a.m. during the heating season. During the cooling season the thermostat is set at 25.6 °C (78 °F) throughout the day. Opened windows increase indoor comfort whenever outdoor temperature and humidity conditions allow [2].

The analytical tool used is the DOE 2.1A computer code developed by the Lawrence Berkeley Laboratory (LBL) and the Los Alamos National Laboratory (LASL) with U.S. Department of Energy funding. Extensive testing against actual building performance has validated this code which is considered by many as one of the most advanced building energy analysis programs. Because DOE 2.1 uses an empirical curve fit to determine the hour by hour equipment efficiencies, there is some concern that it may not model equipment performance exactly. However, a more detailed analysis taking into account the physical characteristics of the particular equipment may not necessarily lead to more accurate results.[3]

Using Test Reference Year (TRY) weather data, the program determines the building heating and cooling loads on an hourly basis. The energy required to meet the hourly space conditioning load is then determined by multiplying the heat pump's capacity (full load COP*) by the ratio of the building load to the equipment capacity (part load COP). For each of the four heat pump types examined, these two parameters are determined as discussed below.

A high performance conventional 10.55kW (36,000 Btuh) electric air source heat pump simulated a base case. Steady state performance data supplied by the manufacturer do not include performance degradation from the accumulation of ice (from ambient air moisture) on the system's evaporator coils. The build-up of frost, commencing at about 10 ^{o}C (50 ^{o}F) and becoming rather significant below 4.4 ^{o}C (40 ^{o}F), will reduce the heat transfer from the outside air to the refrigerant fluid and result in lower performance. A manufacturer of relatively efficient heat pumps provided unpublished data on these conditions. [4]. These data agree with experiments, as yet unpublished, from the National Bureau of Standards (NBS) which show approximately a 15% decrease in performance at 1.7 ^{o}C (35 ^{o}F) [5]. In contrast, the steady state performance data (w/o

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*The COP's given in this paper are expressed in terms of resource energy. Electricity is converted using a factor of 10.8 MJ (10,239 Btu) per kWh. This conversion factor ignores transmission losses of approximately 10%. Similarily, a 10% storage, production, and transmission loss factor for natural gas is also ignored.

defrost degregation) are plotted as a function of the ambient temperature in Figure 1. In the cooling mode, no degradation of a similar nature takes place. Cooling performance depends only on the outdoor and indoor dry bulb and wet bulb temperatures. The cooling COP in Figure 1 assumes a 19.4 ^{o}C (67 ^{o}F) indoor wet bulb temperature. To define the system's actual performance, we incorporated the defrost-degrade performance curve into DOE 2.1A. As DOE 2.1A only performs hourly calculations, it does not compensate for cycling within each hour. To incorporate this effect, the system's performance was degraded by using a part load curve expressing the coefficient of performance as a function of the part load. The part load is the ratio of the building's heating (or cooling) load to the heat pump's capacity for a given temperature. Shown in Figure 2, part load heating and cooling degradation curves are the result of test data taken at NBS and assume an average cycling rate of 1.0 cph [6]. As in this case, new buildings with a well insulated envelope and reduced infiltration are expected to have a cycling rate of 1.0 cph. The unit modelled is a split system with the compressor located outdoors. A thermostatically controlled crankcase heater is automatically activated whenever necessary [4] [7]. If the capacity of the heat pump is not adequate, an electric resistance heater supplies the remaining heat.

The second heat pump modelled on DOE 2.1A is an electric 10.55 kw (36,000 Btuh) water source system using ground water as the heat source/sink. Ground water is an excellent source/sink for a heat pump system because it is relatively constant, plentiful, and moderate in temperature (7.2 ^{o}C (45^{o}F) to 23.9 ^{o}C (75 ^{o}F) throughout most of the United States). This analysis assumes a closed loop system with a multiple-shallow (30.5 m (100 ft deep)) heat exchanger [8]. As the temperature of the ground water is usually closer than that of ambient air to the temperature of the refrigerant, COP's of water source heat pumps are inherently higher than those of air source heat pumps. In addition, there is no loss of performance due to defrosting in water source heat pumps. Finally, this type of a heat pump requires no crankcase electric heater as it is a unitary system with the compressor located indoors. However, moving the water through the multiple-shallow heat exchanger requires extra pump power. Ground water temperatures serve as constant temperature heat sources/sinks in DOE 2.1A. The same part load correction factors used with the air source system were also used here [9]. Two water source systems are modelled: one for water temperatures higher than 15.6 °C (60 °F), and another for water temperatures between 7.2 °C (45 °F) and 15.6 °C (60 °F). The water flow rate is 0.38 l/s (6 gpm) for the regular unit and 0.44 l/s (7 gpm) for the low temperature model. Electric resistance heating supplies that portion of the load that the heat pump cannot provide.

The third system we modeled is a gas fired air-source absorption heat pump. We considered two possible system designs: one with a heating mode only and another with both heating and cooling modes. As with other systems in this study, the capacity is 10.55 kW (36,000 Btuh). A conventional air conditioner supplements the heating only system. Although the dual mode gas fired absorption heat pump has not yet reached the commercial stage, experimental data are available. Arkla Industries developed a heating only ammonia-water system. The unit has a 14.65 kWh (50,000 Btuh) capacity at 8.3 °C (47 °F) ambient with a COP of 1.25; at -8.3 °C (17 °F) its COP is 1.12. The parasitic electric power requirement for this system is estimated at 34 W per kW (0.1 kW per 12,000 Btuh) [10] [11]. A similar organic fluid heat pump developed by Allied Chemical, supplies both heating and cooling. It operates at a heating capacity of 23.7 kW (81,000 Btuh) at 19.4 ^{o}C (47 ^{o}F) with the same heating COP and parasitic power requirements as the Arkla version. Its cooling capacity is 10.55 kW (36,000 Btuh) at 35 ^{o}C (95 ^{o}F) with a COP of 0.6. The parasitic power requirements while cooling are estimated at 92 W per kW (0.27 kW per 12,000 Btuh) [10] [12]. While gas absorption systems operate at lower COP's than electric systems, their overall COP in "resource energy" terms is higher in the heating mode. This is due to the fact that conventional heat pumps use electricity produced at an average efficiency of a little over 30%. As with a conventional air source system, evaporator coil frosting and part load effects lead to a decrease in performance. No such data, to our knowledge, exist on these effects in gas absorption heat pumps. Thus, the percentage efficiency decrease from defrosting was taken to be the same as that for a conventional air source heat pump. Figure 1 includes a performance curve derived from the manufacturer's data. The part load curve used for the first two systems is also used for this system. An electric resistance heater supplements the heat pump whenever necessary.

The fourth heat pump system studied is a gas fired air source Stirling/Rankine system with 10.55 kW (36,000 Btuh) heating and cooling capacities. Stirling/Rankine gas fired heat pumps achieve high "resource energy" efficiencies because the thermodynamic cycle of a Stirling engine approximates that of a Carnot cycle. Utilizing waste thermal energy from the engine further enhances the system's efficiency. As part of the standard Stirling engine, exhaust air is used to preheat incoming combustion air. This rise in combustion air temperature reduces fuel consumption, thereby increasing the thermal efficiency of the engine. An added benefit of this process is that the preheated incoming air does not allow the accumulation of frost on the evaporator coils. While other gas driven heat pumps could use exhaust air to keep frost from building up on the evaporator coils, this is not currently being done. Because of this and because of the Stirling/Rankine's higher proportion of energy input going to the preheater (30%), the Srirling/Rankine cycle was the only heat pump simulted without defrost degradation. In addition, the Stirling engine's coolant, which contains approximately 45% of the thermal energy input, circulates through the indoor air handler, providing extra heating.[13] No commercially available units exist yet, but at least two manufacturers are attempting to develop such a system. General Electric has tested a 10.55 kW (36,000 Btuh) device with both heating and cooling modes. This system uses a spring-mass resonating free piston Stirling engine/compressor. A gas-fired Stirling/Rankine heat pump using a diaphram free-piston Stirling engine/compressor assembly is under development by Mechanical Technology Incorporated (MTI). These manufactures/developers have targeted a heating COP of 2.31 at 8.3 ^{o}C (47 ^{o}F) with an estimated paracitic electricity consumption of 68 W per kW (0.2 kW per 12,000 Btuh) of capacity. The target cooling COP is 1.13 at 35 ^{o}C (95 ^{o}F) with estimated paracitic power requirements of 68 W per kW (0.2 kW per 12,000 Btuh) [10] [14] [15]. Figure 1 shows heating and cooling steady state performance curves. These full load curves along with the part load effect curves of Figure 2 define the system's hourly performance as modelled on DOE 2.1A. An electric resistance heater supplements the heating capacity of this unit as well. Note that the gas absorption and Stirling/Rankine heat pumps are classified as heat actuated heat pumps (HAHP).

The final system consists of variations on a 11.72 kWh (40,000 Btuh) gas furnace combined with a 10.55 kW (36,000 Btuh) electric air-conditioner. A regular furnace with electronic ignition and tight dampers or a pulse combustion furnace provides heating. An air-conditioner with the same performance as the air source heat pump and another air conditioner with a higher steady state EER serve as the cooling part of this system. Figure 1 shows the full load COP of these systems. (A 3% degradation of furnace performance due to fan energy is not shown.) The part load curves (with the exception of those for the furnace) are those in Figure 2.

Table 2 gives the steady state performances of all the systems analyzed in this study. Note that the absorption unit has either both heating and cooling modes or only a heating mode with the cooling provided by either of the two A/C models modelled in conjunction with the furnace. The resistance heater has a 9 kW capacity and is turned on at any outdoor temperature (up to 21.1 ^{o}C (70 ^{o}F)) so as to maintain uniform indoor comfort conditions. Below an outdoor temperature of -26.1 ^{o}C (-15 ^{o}F) the resistance heater takes over the heating load for the air source heat pumps. (At this temperature, the site energy COP falls below 1).

For all air source units, heating COPs are given at 8.3 ^{o}C (47 ^{o}F) and cooling COPS are at 35 ^{o}C (95 ^{o}F). The COP's of the low temperature water source units are at 10 ^{o}C (50 ^{o}F) while for the regular temperature units, they are at 18.3 ^{o}C (65 ^{o}F). All COP's, electric or gas, include parasitics (fans, pumps, etc.) so that an equal comparison of all units is possible.

VARATIONS IN EQUIPMENT SIZING

In addition to the 10.55 kW (36,000 Btuh) systems already discussed, we analyzed 5.27 kW (18,000 Btuh), 15.82 kW (54,000 Btuh) and 21.10 kW (72,000 Btuh) units for selected systems and locations. The same house was analyzed. Note that in new residential buildings with well insulated envelopes and low infiltration, small systems (10.55 kW (36,000 Btuh) or less) are more than adequate to meet building heating and cooling loads [16]. System oversizing can result in lower seasonal efficiencies due to short "on-times" when there is a load. For the purposes of this analysis, New York City represents a heating climate and Phoenix a cooling climate. This sizing sensitivity examines two heat pump systems, an electric air source system and a gas air source absorption system. These systems represent two different types of full load curves, one with a slope of almost 45° and the other with very little or no slope (see Figure 1).

VALIDATION OF RESULTS

Only limited data of adequate quality are available to validate analytical models describing the performance of heat pumps. The measured performance of two electric air source heat pumps installed in unoccupied residences compares with the model used in this paper. Detailed measurements carried out by the Oak Ridge National Laboratory (ORNL) at a test site near Knoxville showed a 36 % decrease in the seasonal heating COP from the manufacturer's steady state COP at 8.3 °C (47 °F). The heating COP was 10 % to 20 % less than the manufacturer's steady state COP at 1.7 °C (35 °F). [16] Simulations for Nashville, comparable in climate to Knoxville, were made on DOE 2.1A for a house similar to the test house. The DOE 2.1A input incorporated both the defrost-degrade performance curve and the part-load effect curve. Compared against the manufacturer's steady state data, the results showed a 35 % decrease in the seasonal heating COP at 8.3 °C (47 °F); and a 20 % decrease in the seasonal cooling COP at 35 °C (95 °F).

Earlier measurements on the seasonal performances of heat pumps in ten residences (seven single family houses and three apartment units) also show a degradation of the same magnitude [18]. The ten units were located in a 2200 to 2800 degree day, base 18.3 °C, (4000 to 5000 degree days, base 65 °F) climate. The manufacturer's steady state COP was 2.5 at 8.3 °C (47°F) while the average measured seasonal performance was 1.40 for the houses and 1.69 for the apartments. This difference is due to different duct efficiencies and thermal integrities. Given the uncertainty in climate and duct losses, the resulting degradation is within 5% of that predicted by our model (see Appendix A).

Although this valadation is minimal in extent, it provides some confidence that this analysis is on the right track. Additional validation will be necessary for the electric air source model and the other models as well.

RESULTS OF THE SIMULATION

Results of simulations for the 8.75 kW (3 ton) systems are given in Tables 3 (heating) and 4 (cooling). Annual energy consumption is expressed in resource energy and all COP´s are resource energy equivalent efficiencies. Tables 5 (heating) and 6 (cooling)summarize results of the equipment oversizing simulations. The seasonal COP´s are given for each equipment size including the base case size from Tables 3 and 4. Note that all COP´s are resource energy equivalent efficiencies.

CONCLUSIONS AND RECOMMENDATIONS

The results of the base case simulations reveal a number of general trends. The electric air source heat pump has the lowest heating resource energy seasonal COP. An electric water source heat pump and a gas air source absorption heat pump are roughly equivalent in terms of seasonal COP´s. However, in cold climates the relatively low ground water temperatures diminish the capacity of water source heat pumps. At a water temperature of 7.2 °C (45 °F), their capacity is lower than air source gas absorption units at an ambient air temperature below -12.2 °C (10 °F). The Stirling/Rankine gas fired heat pump is clearly the most efficient of all systems, achieving a resource energy seasonal COP of well over 1.0 in all climate zones. It´s efficiency is least (but still over 100%) in cold climates because of the added strip resistance heating necessary. By comparison, the seasonal efficiency of a regular gas furnace is slightly higher than that of an electric air source heat pump, while the seasonal efficiency of an advanced gas furnace is slightly less than that of either an electric water source or a gas air source absorption heat pump. Second generation gas absorption heat pumps under development by Allied Chemical are expected to have a COP of

1.50 at 8.3 °C (47 °F). Furthermore, existing units have demonstrated cycling efficiencies of over 90% at 50% part load [19]. Both improvements would increase the seasonal heating COPs by about 25% from their calculated values in Table 3. This would result in net efficiencies in excess of 100% everywhere in the U.S. One must bear in mind that electric heat pumps and gas furnaces are commercially available products whereas the gas heat pumps are not. The steady state efficiencies of the latter, if and when they become commercially available, may turn out to be less, equal, or even higher than those assumed in this analysis.

Of the various cooling systems, the gas absorption heat pump is quite inefficient with seasonal cooling COP's on the order of 0.35 to 0.40. An electric air source heat pump and an average electric air-conditioner are next in terms of ascending efficiency, each having roughly twice as high a seasonal COP as that of the gas absorption unit (0.65 to 0.75). A Stirling/Rankine gas fired heat pump has a somewhat higher cooling seasonal COP (between 0.70 and 0.80). An advanced electric air conditioner achieves relatively high seasonal efficiencies (0.80 to 0.95). Finally, the electric water source heat pump outperforms all other systems in all cities except Houston. The ground water temperature in this climate zone approaches the low 20's °C (mid 70's °F). In the same climate zone, an advanced electric air conditioner operates at a slightly better seasonal cooling COP because of moderate daily air temperature variations (5.5 °C (10°F) to 8.3 °C (15°F). In addition, the presence of a larger latent load allows the heat pump's capacity to be better utilized and thus increases its overall performance. In the other cities, the cooling seasonal COP of the electric water source heat pump is between 100 and 125 %.

Varying the equipment size has quite different effects for the heating mode than it does for the cooling mode. In the heating mode, a 10.55 kW (36,000 Btuh) heat pump has the highest seasonal COP. In addition, the seasonal COP in a heating climate (New York) is lower than that in a cooling climate (Phoenix). The first effect is a result of the daily 5.5 ^{o}C (10 ^{o}F) setback. This causes the building's load to range from 8.8 to 11.7 kW (30 to 40 kBtuh) during the morning hours of the heating season in either location. The second effect is due to the higher hourly demand in the heating climate which increases the number of hours during which the resistance heater must help meet part of the heating load. The performance of the gas absorption unit decreases as equipment size increases. This is due to higher part load losses. The unit has a rather flat full load curve and consequently a relatively higher capacity than that of an air source electric unit. In the cooling mode, the smallest unit has the highest seasonal efficiency. This is because the hourly cooling demand is about one half that for heating (0 to 4.4 kW (15 kBtuh)). Because the hourly cooling demand is higher in the cooling climate, and, thus, part load losses are less, the cooling seasonal COP is highest in this climate. The performance of the gas absorption unit is independent of sizing. Because of its very low cooling COP, even 21.1 kW (72,000 Btuh) units lead to a few undercooled hours a year.

The above discussion leads to some recommendations as to future systems development. First, unless substantial improvements are made in the cooling performance of gas absorption units, these units should concentrate only on heating. Second, electric water source, gas air source absorption and gas air source Stirling/Rankine heat pumps constitute very efficient heating systems. Third, gas air source Stirling/Rankine, advanced electric air source air conditioners, and water source electric heat pumps are very efficient cooling systems with the third one being close to or over 100 % efficient everywhere. Fourth, the development of a gas fired water source Stirling/Rankine heat pump could achieve first law thermodynamic heating efficiencies of over 200% in all climate zones with similar cooling efficiencies in at least 3/4 of the country. Fifth, oversizing heating and cooling systems results in a 5 to 10 % degradation of the respective seasonal COP's. Finally, whenever temperature setbacks are present, the equipment size must be adjusted (oversized) to pick-up the extra hourly demand. In other words the potential part load losses due to oversizing are more than offset by the inherent resource energy inefficiency of the added resistance heating necessary with lower capacity. Consequently, for a heat pump without advanced controls operating with setbacks, the system should be sized to handle morning start-ups.

Even though the purpose of this study was to evaluate the absolute and relative performances of heat pumps, it is instructive to conclude by addressing some non-technical, but pertinent, issues. Second and third generation heat activated heat pumps will undoubtedly exceed the performance levels indicated in this study. However, such systems may be expensive to develop. Furthermore, their mechanical complexity may result in higher maintainance costs than those of conventional units. One would expect the economic viability of gas driven heat pumps will depend largely on the relative costs of gas and electricity. [20] [21] However, a recent analysis of the penetration of conventional electric heat pumps reveals a consumer preference for such systems over gas furnaces even where first and operating costs may dictate otherwise. [22]

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APPENDIX A

DOE 2.1A was used to simulate the seasonal heating and cooling COP's of conventional electric air source heat pumps in 50 locations throughout the U.S. [16]. The ORNL heat pump study indicates that the relative seasonal degradation of the COP from its steady state COP is independent of the steady state COP itself [17]. The ratio of the seasonal COP to the steady state (rated) COP is plotted against degree days base 18.3 ^{o}C (65 ^{o}F) for both heating and cooling in Figures A1 and A2. As long as equipment is sized properly, this ratio can be virtually independent of buiding thermal integrity and operating conditions [16]. Ratios for a gas furnace and a conventional water source heat pump are also shown in Figures A1 and A2. For a furnace a band indicates the greater variability of this ratio in gas heated houses. This variability is due to the location of the furnace with respect to the building living space and the resulting impact on internal loads and infiltration. For an electric water source heat pump only a few data points have been calculated -- the rated COP's are those at the water temperatures indicated in Table 2. Selected cities, characterized by degree days, are displayed in Figures A1 and A2. For a heat pump in the heating mode higher performances occur in climates with less than 2800 heating degree days, base 18.3 ^{o}C (5000 degree days, base 65 ^{o}F). Also, relatively small, 8.3 ^{o}C, (15 ^{o}F) daily temperature swings result in slightly higher seasonal performances. The higher the cooling degree days the higher the performance of a heat pump or of an air conditioner. Below 550 cooling degree days, base 18.3 eC, (1000 degree days base 65 ^{o}F) the seasonal performance degrades rather rapidly as parasitic electricity consumption overrides compressor electricity consumption. As humidity increases cooling becomes more efficient due to better utilization of the equipment's latent capacity and lower daily temperature swings.

Each of the two sets of data points was fit with a cubic curve so that a location's heating or cooling COP can be calculated from the steady state COP. The following equations were calculated. Coefficients are given for centigrade degree days (and farenheit).

Heating Seasonal COP (HSCOP):

$$HSCOP = HCOP \times (a + b \times HDD + c \times (HDD)^2 + d \times (HDD)^3)$$

$a = 0.6358$

$b = 0.2137 \times 10^{-4}$ (0.1187×10^{-4})

$c = - 0.3998 \times 10^{-8}$ (-0.1234×10^{-8})

$d = - 0.1375 \times 10^{-11}$ $(-0.2357 \times 10^{-12})$

Cooling Seasonal COP (CSCOP):

$$CSCOP = CCOP \times (e + f \times CDD + g \times (CDD)^2 + h \times (CDD)^3)$$

$e = 0.4032$

$f = 0.5920 \times 10^{-3}$ (0.3289×10^{-3})

$g = - 0.3039 \times 10^{-6}$ (-0.9379×10^{-7})

$h = 0.5341 \times 10^{-10}$ (0.9158×10^{-11})

where HCOP is the steady state COP at 8.3 ^oC (47 ^oF); CCOP is the steady state cooling COP at 35 ^oC (95 ^oF) outdoor dry bulb and 19.4 ^oC (67 ^oF) indoor wet bulb; HDD is heating degree days and CDD is cooling degree days base 18.3 ^oC (65 ^oF). Depending on other climatic variables such as daily temperature swings and humidity the actual heating and cooling seasonal COP's may deviate as much as 10 % in either direction from the COP's predicted by the above equations.

TABLE 1. Building Envelope Thermal Integrity

Location	Wall	Ceiling	Foundation	Glazings
Minneapolis	R-4.8	R-8.6	R-1.8; 2.5m-B*	3
New York	R-3.3	R-6.7	R-1.8; 2.5m-B	3
Houston	R-3.3	R-5.3	R-.9; 0.6m-S	2
Phoenix	R-3.3	R-5.	R-.9; 0.6m-S	2

(*) B:Basement; S:Slab-on-grade.

All R values in $m^2 \times {}^oC/W$;
Infiltration = 0.7 air changes per hour.

TABLE 2. Summary of Systems With Their Respective Performance Characteristics Used in This Analysis

System	Heating COP (fuel)	Cooling COP (fuel)
Air source HP (ASHP)	3.10 (electric)	2.70 (electric)
Water source HP (WSHP)	3.28 LT 3.24 RT (electric)	4.23 LT 3.59 RT (electric)
Absorption air source HP (HAHP ABS)	1.15 (gas)	0.49 (gas)
Stirling/Rankine air source HP (HAHP ST/RA)	1.97 (gas)	0.96 (gas)
Furnace with A/C (FURNACE-AC)	0.80 R 0.95 S (gas)	2.70 R 3.43 S (electric)

LT: low temperature; RT: regular temperature
R: regular; S: super efficient

TABLE 3. Annual Heating Energy Consumption and Seasonal COP

LOCATION Load (GJ / yr)	ASHP	WSHP LT	WSHP RT	HAHP ABS.	HAHP ST/RA	FURNACE-AC R	FURNACE-AC S
	Energy (GJ/yr) and Seasonal COP						
MINNEAPOLIS							
56.5	103.6	61.2		58.7	51.3	80.6	64.2
----	0.55	0.92		0.96	1.10	0.70	0.88
NEW YORK							
28.4	45.4	32.8		34.2	21.1	43.6	34.7
----	0.67	0.93		0.89	1.44	0.70	0.88
HOUSTON							
10.1	14.3		11.0	11.3	6.7	14.4	11.4
---	0.70		0.91	0.88	1.50	0.70	0.88
PHOENIX							
7.4	10.3		8.2	8.3	4.6	10.5	8.3
---	0.71		0.89	0.89	1.57	0.70	0.88

TABLE 4. Annual Cooling Energy Consumption and Seasonal COP

LOCATION Load (GJ / yr)	Energy (GJ/yr) and Seasonal COP						
	ASHP	WSHP		HAHP	HAHP	FURNACE-AC	
		LT	RT	ABS	ST/RA	R	S
MINNEAPOLIS							
8.0	12.5	6.2		23.0	11.7	12.5	9.9
---	0.63	1.27		0.34	0.68	0.63	0.80
NEW YORK							
8.4	13.3	7.4		24.6	12.4	13.3	10.5
---	0.63	1.15		0.34	0.68	0.63	0.80
HOUSTON							
38.5	50.5		41.2	93.2	46.8	50.5	39.7
----	0.76		0.94	0.41	0.82	0.76	0.97
PHOENIX							
43.2	60.0		43.8	110.0	56.1	60.2	47.3
----	0.72		0.99	0.39	0.77	0.72	0.91

TABLE 6. Cooling Seasonal COP vs. Equipment Size

Capacity (tons)	New York ASHP	New York HAHP ABS	Phoenix ASHP	Phoenix HAHP ABS
1.5	0.67	0.35	0.77	0.39
3.0	0.63	0.34	0.72	0.39
4.5	0.62	0.34	0.70	0.39
6.0	0.61	0.33	0.69	0.39

TABLE 5. Heating Seasonal COP vs. Equipment Size

Capacity (tons)	New York ASHP	New York HAHP ABS	Phoenix ASHP	Phoenix HAHP ABS
1.5	0.65	0.82	0.69	0.81
3.0	0.67	0.89	0.71	0.89
4.5	0.65	0.70	0.68	0.70
6.0	0.63	0.67	0.66	0.68

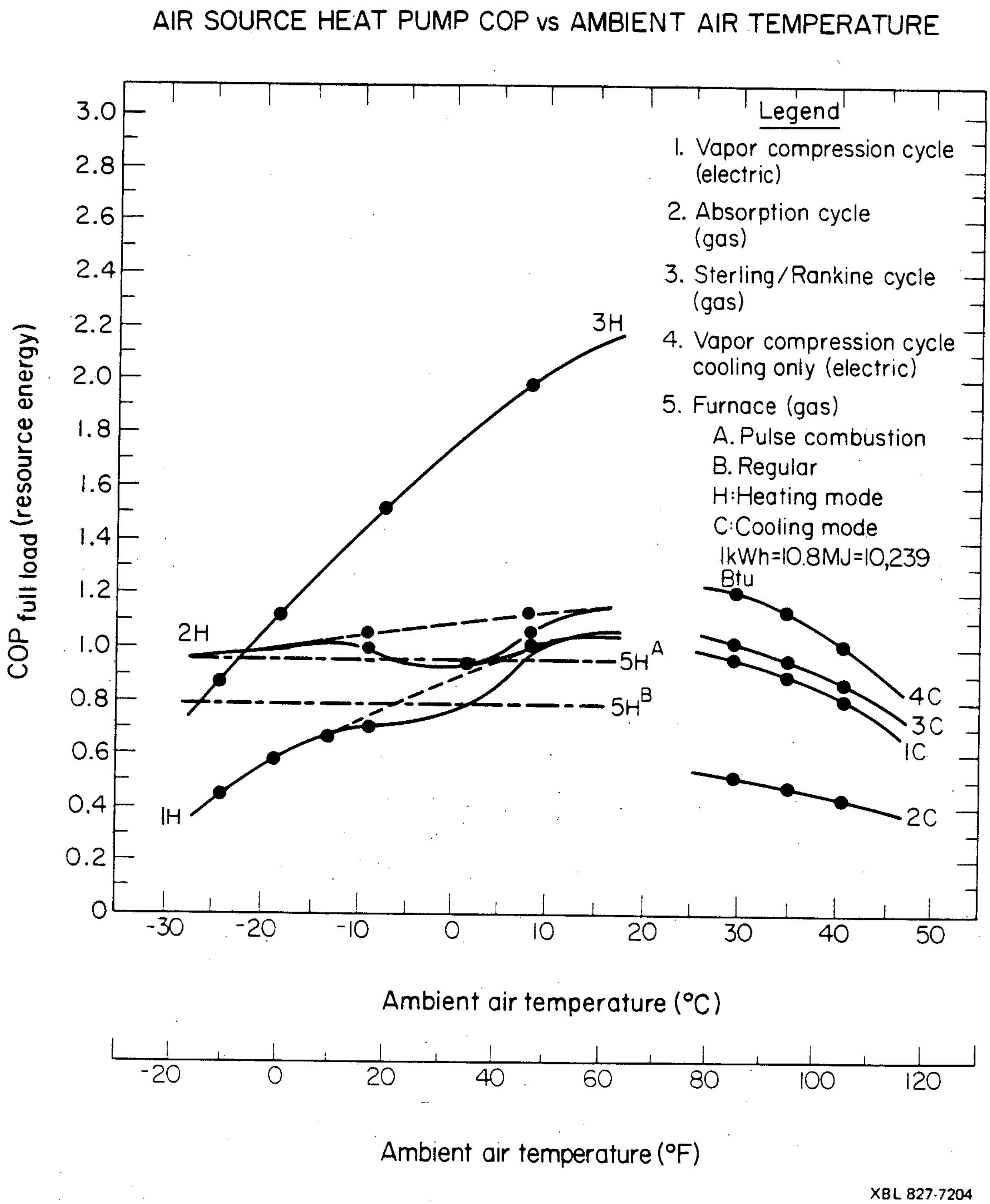

Figure 1: Air Source Heat Pumps Full Load COP vs. Ambient Temperature

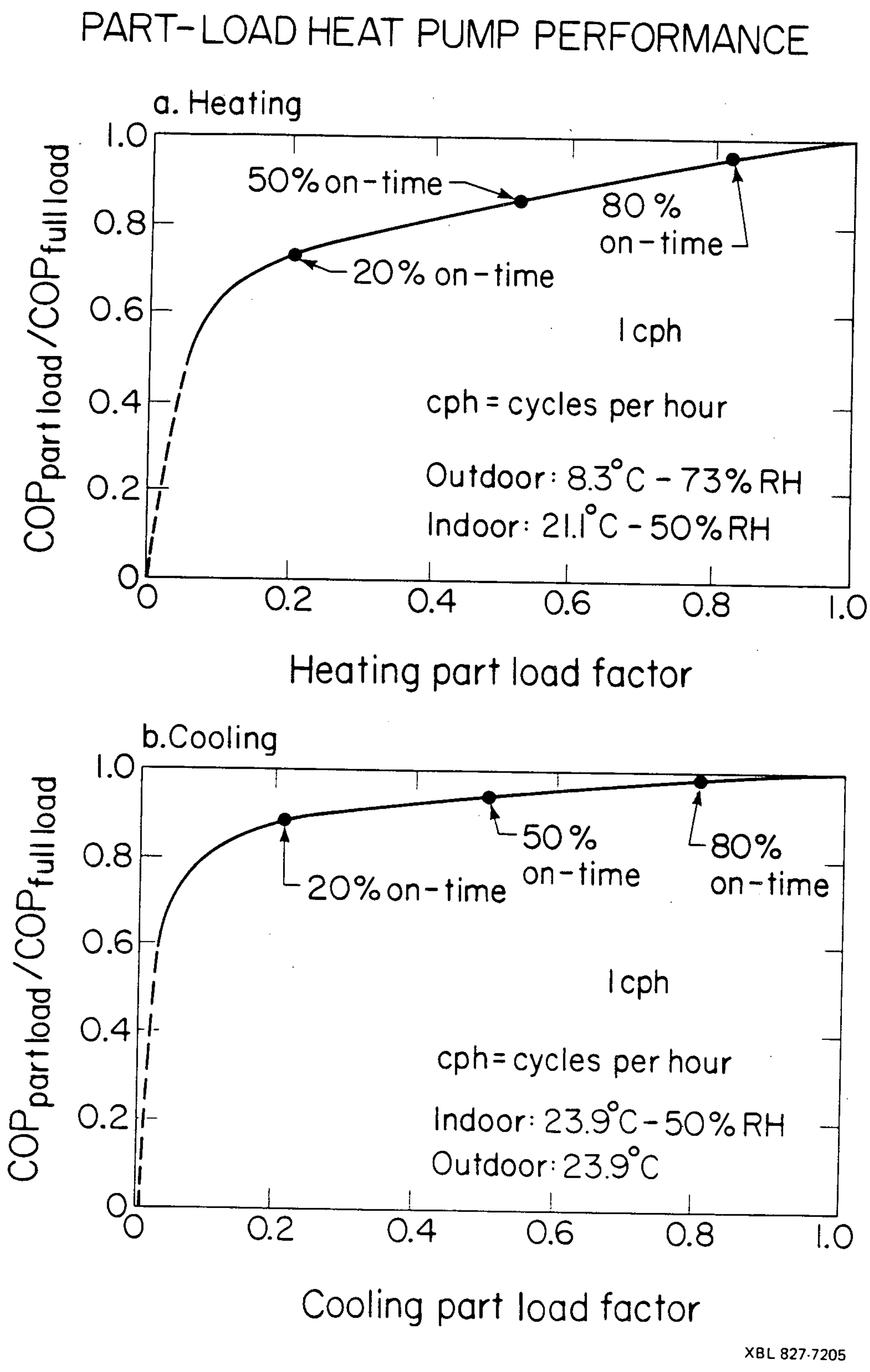

Figure 2: Heating and Cooling Heat Pump Part Load COP vs. Part Load

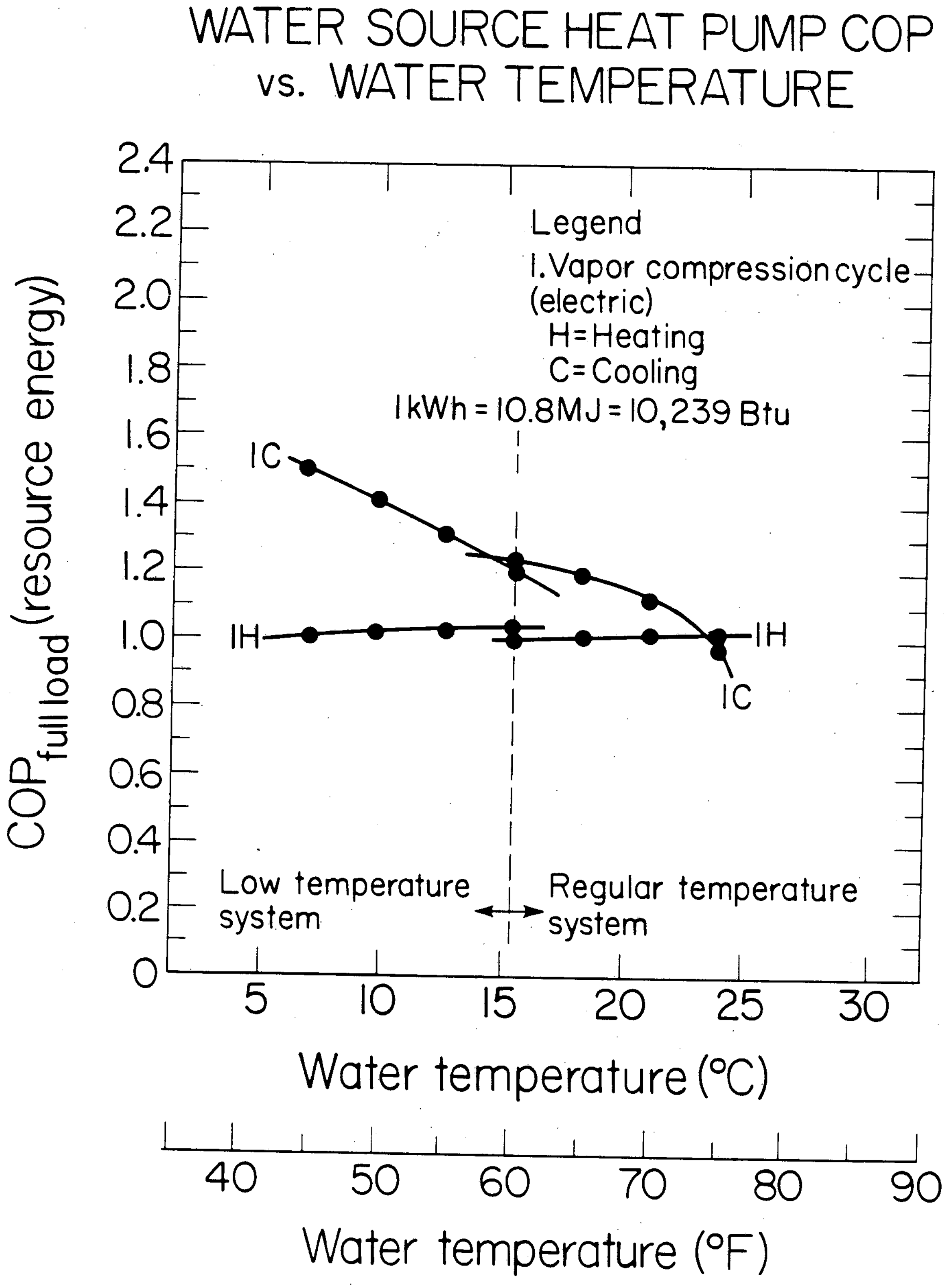

Figure 3: Water Source Heat Pump Full Load COP vs. Water Temperature

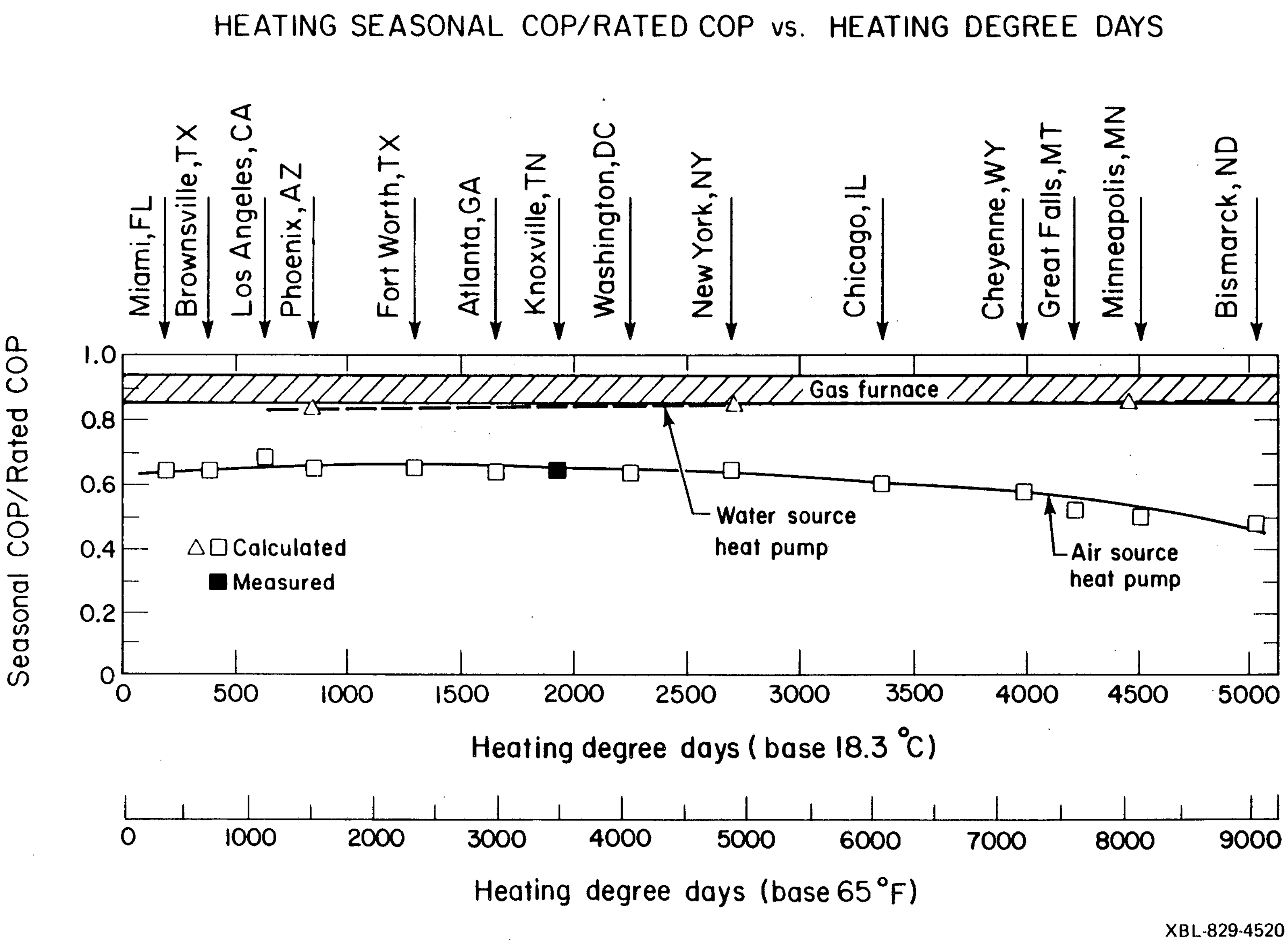

Figure A1: Heating Seasonal COP/Rated COP vs. Degree Days

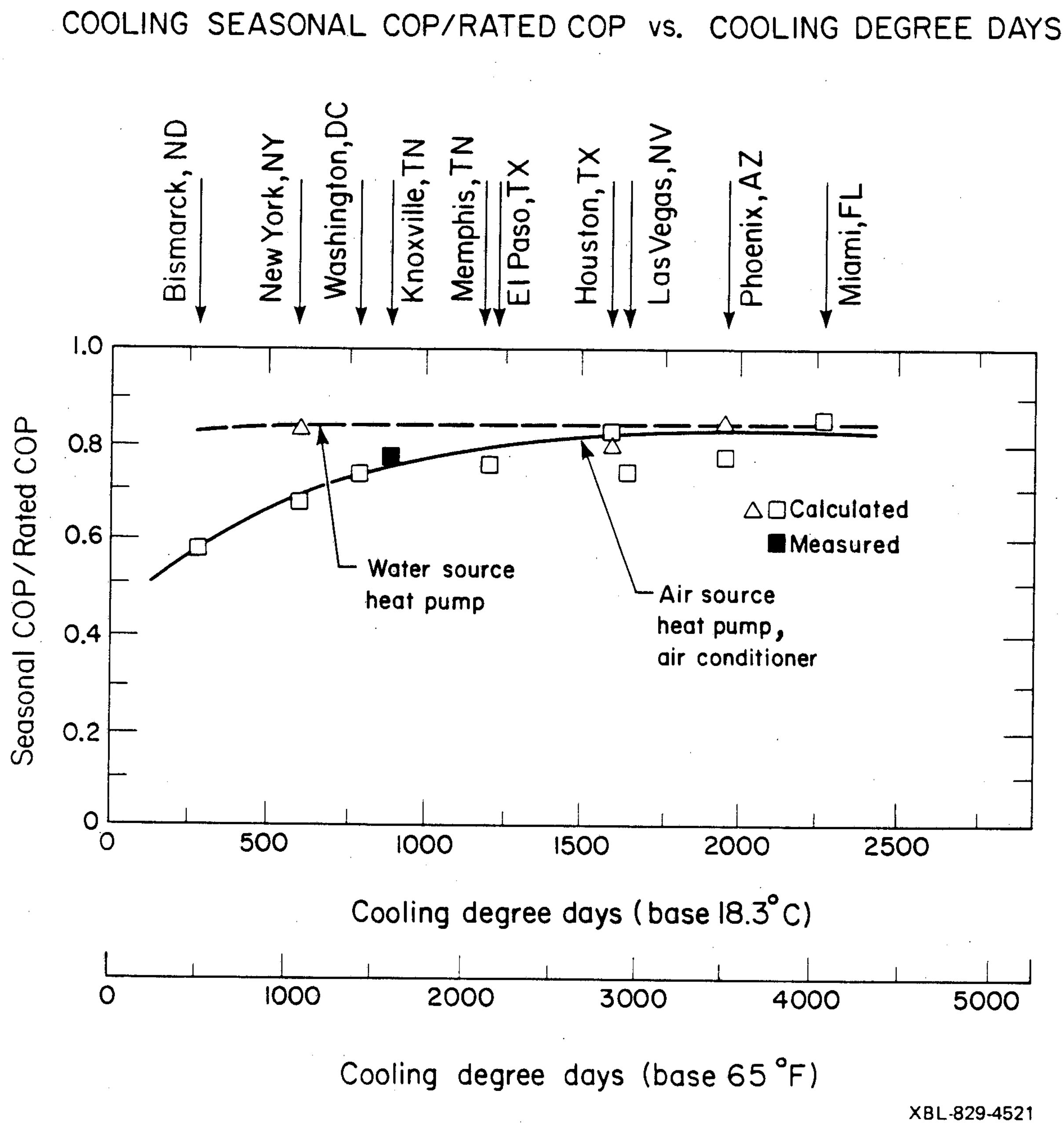

Figure A2: Cooling Seasonal COP/Rated COP vs. Degree Days