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J. B. BRIGGS C. J. SHAFFER

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EG&G Idaho, Inc.

IDAHO NATIONAL ENGINEERING LABORATORY

DEPARTMENT OF ENERGY

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ABSTRACT

An analysis of seasonal heat pump performance in a typical northern United States environment (neighborhood of 8000 degree-days of heating) was performed for several typical commercially available air- and water-source heat pumps. The typical commercially available heat pumps were chosen from an evaluation using certified performance data published by the Air-Conditioning and Refrigeration Institute (ARI). The seasonal heat pump performance was determined in terms of energy savings over the use of electrical resistance heating alone.

SUMMARY

An analysis of seasonal heat pump performance in a typical northern United States environment (neighborhood of 8000 degree-days of heating) was performed for several typical commercially available air- and water-source heat pumps. The typical commercially available heat pumps were chosen from an evaluation using certified performance data published by the Air-Conditioning and Refrigeration Institute (ARI). The seasonal heat pump performance was determined in terms of energy savings over the use of electrical resistance heating alone.

The evaluation of the ARI-certified performance data showed that the majority of the commercially available heat pumps have a capacity under 60000 Btu/h and that there are considerably more air-source than water-source heat pumps available. This implies that large industrial sized heat pumps capable of utilizing industrial waste heat are somewhat limited. A general trend towards a slightly better performance for larger capacity heat pumps was found from the performance data, but this is not necessarily true of the products of an individual manufacturer.

The seasonal heat pump performance evaluation was performed using a computer model named HTPUMP which was developed for this purpose. The computer model evaluates the heat pump performance for a given environment on an hourly basis, and then the seasonal heating and cooling performance is determined from the hourly performance data. Parametric studies were performed to determine the seasonal performance under a variety of conditions.

The heat pump seasonal performance was evaluated in a typical northern United States environment which was defined for the purposes of this study as an environment having approximately 8000 degree-days of heating and 500 degree-days of cooling. Atmospheric data on an hourly basis obtained from the National Oceanic and Atmospheric Administration for a location with 8290 degree-days of heating and 630 degree-days of cooling were used as the typical environment.

The seasonal performance results of these typical commercially available heat pumps indicates that an air-source heat pump system can save on the order of 45% of the energy required to heat using electrical resistance

heating alone and that a water-source heat pump using 60 and 80°F water can save the order of 64% and 67%, respectively. Significant water pumping power requirements can reduce the water-source heat pump energy savings.

SI CONVERSION

The International System (SI) of units is being used in all reports initiated by EG&G Idaho, Inc., after April 28, 1977 . The following report was initiated in December, 1976, and most of the analysis was completed using standard air-conditioning units before April 1, 1977. Data which were used in the analysis were extracted from publications which also use standard air-conditioning units. This report has also been prepared using standard air-conditioning units to maintain consistency throughout.

The following conversion table may aid in making any necessary conversions from air-conditioning units to SI units.

Multiply Air-Conditioning Units	By	To Obtain SI Units
British thermal units (Btu)	1.055×10^{3}	joules (J)
hours (h)	3.6×10^{3}	seconds (s)
Btu/h	2.93×10^{-4}	kilowatts (kW)
gallons per minute (gpm)	6.309×10^{-5}	metre^3 per sec (m^3/s)
pounds (lbm)	2.2046	kilograms (kg)
cubic feet (ft^3)	2.8317×10^{-2}	cubic metre (m^3)

Temperature Conversion

$$°C = \frac{9}{5}(°F-32)$$

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I. INTRODUCTION

A seasonal evaluation of water-source heat pumps in comparison to air-source heat pumps was performed for space heating and cooling applications, industrial or otherwise, in a typical northern United States environment. The evaluation was done for several typical commercially available water- and air-source heat pumps to determine the conditions under which the water-source heat pumps perform better than the air-source heat pumps on the basis of energy savings. The water-source heat pumps utilizing low temperature industrial waste heat can perform considerably better than the air-source heat pumps. A heat pump can be reversed to provide space cooling during the summer months when spacing heating is not needed. The evaluation was restricted to the space heating and cooling applications due to the commercially available heat pump data.

Heat pump performance is sensitive to the environmental conditions in which it operates, and is especially sensitive to the energy source temperature (i.e., water, air, etc.). Air-source heat pumps are particularly sensitive to the energy-source temperature since both the heat pump operational characteristics and the space heating load vary with the atmospheric dry bulb temperature.

This report briefly describes the heat pump operation, demonstrates the availability of commercial heat pumps, discusses heat pump seasonal performances and documents the computer calculation model.

II. HEAT PUMP DESCRIPTION

A heat pump is a thermodynamic device that operates in a cycle that requires work and that accomplishes the objective of transferring heat from a low-temperature body to a high-temperature body. The heat pump cycle is identical to a refrigeration cycle in principle but is different in that the primary purpose of the heat pump is to supply heat rather than remove it from an enclosed space.

The heat pump cycle (shown schematically in Figure 1) consists of four basic operations: evaporation, compression, condensation, and expansion. A heat pump accomplishes its task of transferring energy from a low-temperature source to a high-temperature receptacle through the use of a secondary fluid (usually Freon-22) which has a boiling point several degrees below 0°F. Space heating (in the winter) can be accomplished by transferring energy from the low-temperature outside air to the even lower-temperature secondary fluid in the liquid phase. The secondary liquid is evaporated in an outdoor heat exchanger and is converted to a cool secondary vapor. Work is then done on the secondary vapor through the use of a compressor. Following the compression process the secondary vapor is at high pressure and at a corresponding high temperature as defined by the equation of state for the fluid. The vapor temperature at this point is higher than the temperature of the indoor air. By condensing the secondary vapor with an indoor heat exchanger and fan, energy is transferred from the high-temperature secondary vapor to the indoor air thus heating the indoor air. The secondary fluid then moves to the outdoor unit at which point the high pressure is relieved by capillary tubes or an expansion valve, and the secondary fluid returns to the evaporator in its liquid state at very low temperature, and the cycle continues. The net result is that heat is transferred from a low-temperature source to a high-temperature receptacle with the addition of a work input to the cycle. The advantage of this type of a heating system is that more energy is made available for space heating than there is work required to operate the heat pump.

A heat pump cycle can be reversed to provide space cooling during the summer months. That is, most heat pumps provide a four-way valve as shown

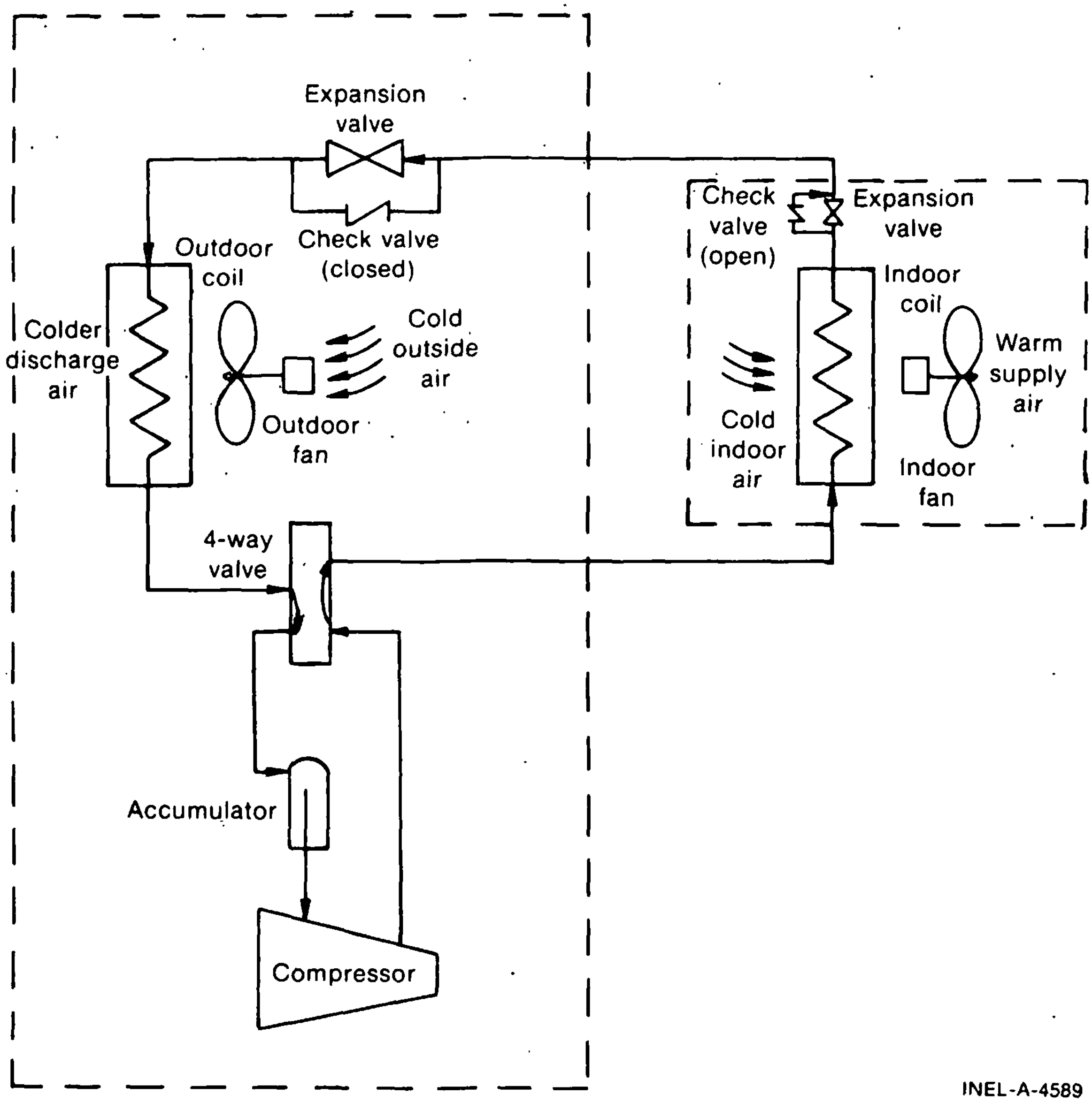

Figure 1. Heat pump schematic in the heating mode.

in Figure 1 which effectively switches the indoor and outdoor heat exchangers so that the indoor exchanger becomes the evaporator and the outdoor exchanger becomes the condenser. The heat pump then operates normally except that heat is removed instead of supplied to the space.

Some heat pumps have been designed to operate utilizing a water source instead of an air source simply by designing the outdoor heat exchanger to operate between the heat pump working fluid and water instead of between the working fluid and air. These so called water-to-air heat pumps have advantages over the air-to-air type if a relatively warm source of water is available which does not require an excessively large amount of pumping power. In particular, industrial waste heat might be used.

Several heat pump configurations can be visualized utilizing a seemingly inexhaustible number of energy sources. Some of these energy sources are outside air, sensible heat from stream or well water, latent heat of fusion from water (ice formation), warm discharge effluents from industry, fireplace waste heat, and heat generation in sewage. Most of these energy sources are not widely available to the general public.

Four types of heat pump systems are in common use today: (1) single-package heat pumps using an air source, (2) split-system heat pumps using an air source, (3) single-package heat pumps using a water source, and (4) split-system heat pumps using a water source. Single-package heat pumps have all the essential components contained within a single unit while split-system heat pumps house the essential components in two separate units (one unit outdoors and one unit indoors).

Heat pump performance for heating at a given set of operating conditions is usually described by a coefficient of performance (C.O.P.) defined by Equation (1):

$$\text{C.O.P.} \equiv \frac{\text{(Btu/h) output}}{\text{(Btu/h) input}} \equiv \frac{\text{(kW) output}}{\text{(kW) input}} \qquad (1)$$

Heat pump cooling performance at a given set of operating conditions is usually described by an energy efficiency ratio (E.E.R.) defined by Equation (2).

$$\text{E.E.R.} \equiv \frac{\text{(Btu/h) Heat rejection}}{\text{(watts) Heat input}} \qquad (2)$$

The coefficients of performance and energy efficiency ratios are useful in comparing heat pump sizes and manufacturers (see Section III), but do not provide the proper data required to compute seasonal energy savings. A seasonal performance factor for heating (S.P.F.) may be defined by Equation (3).

$$\text{S.P.F.} \equiv \frac{(\text{HPE}) + (\text{AE})}{3413[\text{HPP}+\text{AP}]} \qquad (3)$$

where

HPE = energy supplied by the heat pump during the heating season (Btu)

AE = energy supplied by auxiliary electric heat during the heating season (Btu)

HPP = energy required by the heat pump during the heating season (kWh)

AP = energy required by auxiliary electric heat during the heating season (kWh)

1 kW = 3413 Btu/h.

Seasonal performance factors make it possible to estimate seasonal energy saving assuming that the space would normally have been electrically heated. The estimate is made by dividing the energy required to heat the space using electrical resistance heating only during the heating season by the heat pump seasonal performance factor. The result is the energy required to heat the same space using a heat pump system. A comparison of the two heating requirements will yield the seasonal energy savings. For example, a heat pump with a seasonal performance factor of two would yield a fifty percent energy savings over electrical resistance heating.

Frost formation on the outdoor coils is an inherent problem encountered with air-to-air heat pumps whenever the air temperature drops below about 40°F. Frost formation reduces the capacity of the heat pump if allowed to remain on the coils. For this reason all air-source heat pumps use an automatic defrost cycle for frost removal. The method of frost removal varies with different manufacturers, but most use either a timed cycle or one based on a pressure drop across the outdoor coils. When in the defrost mode, the heat pump cycle will reverse and remove heat from the enclosure and melt the frost on the outdoor coils. An auxiliary heating unit turns on during the defrost cycle to maintain the enclosure at the desired temperature. A timed defrost cycle does this at predetermined time intervals and continues for a set period of time. A defrost cycle based on the pressure drop across the outdoor coils initiates when a pressure differential is detected as a result of blockage of the coils by a predetermined amount of frost. Ice formation on the evaporator coils of a water-to-air heat pump operating in the heating mode can also be a problem when the water temperature entering the heat pump coils drops much below 50°F.

A seasonal performance factor for cooling (S.C.F.) may be defined by Equation (4).

$$\text{S.C.F.} = \frac{HR}{HPC} \qquad (4)$$

where

HR = energy removed by the heat pump during the cooling season (Btu).

HPC = energy required by the heat pump during the cooling season (kWh).

The cooling seasonal performance factor for a heat pump system should be approximately the same as for a regular air-conditioning unit in that both operate on the same principle. However, the heat pump has the advantage that it can provide both heating and cooling.

III. COMMERCIAL HEAT PUMP AVAILABILITY

An evaluation was made of the availability of commercial heat pumps using data published by the Air-Conditioning and Refrigeration Institute (ARI) which sponsors and administers a "Directory of Certified Unitary Air Conditions, Unitary Heat Pumps, Sound-Rated Outdoor Unitary Equipment, and Control System Humidifiers"[1]. Unitary heat pumps of many different manufacturers were tested by the ARI under common laboratory conditions. Details of the ARI directory and the laboratory testing conditions, procedures, data, and evaluation results are found in Appendix A of this report.

In summary, four performance tests were made on each certified pump model: (1) maximum operating conditions, (2) low-temperature operation, (3) insulation efficiency, and (4) condensate disposal. A heat pump model passing these performance tests was then rated using established standard rating conditions (listed in Table A-I of Appendix A).

Basically, these rating conditions for a water-source heat pump were to produce 70°F indoor heating and 80°F indoor cooling from a 60°F water-source for heating and an 85°F water-source for cooling. The rating conditions for an air-source heat pump require the heat pump to produce 70°F indoor air temperatures using outdoor air temperatures as low as 17°F, and to produce indoor air temperatures of 80°F using outdoor air temperatures as high as 95°F. The certified ratings are given in terms of the coefficient of performance (COP) for heating and the energy efficiency (EER) for cooling.

The COPs and EERs were plotted (Figures A-1 through A-8) for comparison as a function of the respective heating and cooling capacities. Each manufacturer was given a different symbol so that one manufacturer could be compared with another. The ARI data evaluated in Appendix A contains data for both single-package and split air-source systems, and for single-package water-source systems. (There were no split water-source system data in Reference 1.)

The following results and conclusions were drawn from the evaluation described in Appendix A:

The data are considerably scattered with very few general trends. The COPs for air-source heat pumps tend to bunch in the general neighborhood of 2.0 to 2.5 and 1.2 to 1.8 for air temperatures of 47 and 17°F (dry bulb), respectively, and in the general neighborhood of 2.3 to 3.0 for water-source heat pumps using 60°F water. The EERs tend to bunch in the general neighborhoods of 6.0 to 7.0 and 7.5 to 9.0 Btu/Wh for air-source and water-source heat pumps, respectively. There is a general trend towards a slightly better performance for larger capacity heat pumps but this is not necessarily true of the products of an individual manufacturer; that is, the performance of some manufacturers' units will decrease with increasing capacity or will vary randomly.

Heat pump quality and probably cost appear to be factors affecting the performance and therefore contributing to the data scatter. Some of the manufacturers have more than one certified model of the same capacity which seems to imply a difference in cost; otherwise the poorer performing model would surely be removed from the market. Some manufacturer's heat pumps seem to perform consistently higher than the majority for all sizes of heat pumps.

An overall general evaluation of these data seem to imply that a majority of heat pumps could be better designed and in fact some are of quite poor design. Heat pump performance is especially low for the smaller capacity air-source units operating at the lower rating temperature of 17°F, and yet some of the heat pumps still manage to operate with a COP near 2.0 at this low temperature. Furthermore, there is no thermodynamic reason why a heat pump of larger capacity should perform better than a lower capacity pump, implying that better engineering design could increase the capacity of the smaller unit. Perhaps better component or secondary fluid selection and matching or standardization of good designs could significantly improve the performance of the majority of the certified heat pumps.

2. Examination of the data shows that the majority of the commercially available heat pumps have a capacity under 5 tons (60 000 Btu/h) and that there are considerably more air-source than water-source heat pumps available. This implies that availability of large industrial-sized heat pumps capable of utilizing industrial waste energy is somewhat limited.

3. A comparison of the air-source heat pump data does not show any obvious and significant differences between the single-package units and the split-system units so that the selection of one type or the other will be based upon the application criteria.

4. A comparison of the water-source data with the air-source data appears to show a higher performance in general for the water-source heat pumps having a COP of the order of 10 to 20 percent greater than COP for air-source heat pumps tested with 47°F air. The water- and air-source heat pumps would probably compare closely if the water-source heat pumps were tested using 47°F water.

   A smaller part of the difference between the water- and air-source data could be caused by differences in outdoor heat exchanger efficiencies; that is, water heat exchangers usually perform more efficiently than air heat exchangers primarily because of the larger heat capacity and the larger convective heat transfer coefficient of water. A more detailed evaluation will be needed to determine how significant the difference in heat exchanger efficiency really is.

   Another point to be made is that the water-source heat pumps may in some applications require very significant pumping powers to pump the water from its source to the heat pump heat exchanger for use. In summary, the application criteria will determine which type of heat pump will perform better at a given source temperature.

IV. HEAT PUMP SEASONAL PERFORMANCE EVALUATION MODEL

A computer model named HTPUMP (Reference 2) has been developed to determine the seasonal performance of a given heat pump in a given application and environment. The heat pump data needed for program input are the heating and cooling capacities, and the heat pump power requirements for heating and cooling as a function of the indoor temperature and the source temperature. An input description to HTPUMP is found in Appendix B. Environmental data are needed to determine both the source temperature for air-source heat pumps and the space heating and cooling load requirements. The heating and cooling load calculations require some specification of the type and size of space. The environmental data consist of hourly averaged values. The computer model evaluates the heat pump performance on an hourly basis and then the seasonal heating and cooling performance is determined from the hourly performance data.

1.0 HOURLY HEATING AND COOLING LOAD MODEL

Detailed calculations for the heating and cooling load involve parameters such as enclosure size, type and amount of insulation, location of windows, number and age of inhabitants, outdoor air temperature, wind velocity, etc. It can be very time-consuming and very expensive to evaluate the heat load for a particular enclosure for every hour for an entire year, and when the calculation is complete, the results are only good for that particular enclosure. This procedure can be generalized and simplified considerably by describing an enclosure by a constant heat-loss factor which has the dimensions of energy per unit time per degree of temperature difference between outdoor and indoor temperatures (Btu/h-°F). Parametric studies can then be made to determine seasonal performance factors as a function of the heat-loss factors for a given heat pump capacity.

Using a constant heat-loss factor the hourly heat load (Btu/h) can be determined as follows:

$$HL = C(T_{in} - T_{out}) - Q \qquad (5)$$

where

HL = heat load (Btu/h)

C = constant heat-loss factor (Btu/h-°F)

T_{in} = indoor temperature (°F)

T_{out} = outdoor temperature (°F)

Q = internal energy generation (Btu/h).

The cooling load may be similarly calculated as:

$$CL = C(T_{out} - T_{in}) + Q \tag{6}$$

where

CL = cooling load (Btu/h).

Equations (5) and (6) assume that the indoor temperature is equal to the desired temperature (the desired heating indoor temperature may not be the same as the desired cooling indoor temperature). If the indoor temperature is less than the desired heating temperature, the heat load (Btu/h) may be approximated by:

$$HL = C(T_{in} - T_{out}) - Q + mc_p (T_{dih} - T_{in}) \tag{7}$$

where

T_{dih} = desired indoor heating temperature (°F)

mc_p = energy storage capacity (Btu/°F).

Similarly if the indoor temperature is greater than the desired cooling temperature, the cooling load (Btu/h) may be approximated by:

$$CL = C(T_{out} - T_{in}) + Q + mc_p(T_{in} - T_{dic}) \tag{8}$$

where

T_{dic} = desired indoor cooling temperature (°F).

The indoor temperature is allowed to fluctuate between the desired heating and cooling temperatures when both heating and cooling are allowed and the outdoor temperature fluctuates about the required levels. If only heating is allowed and the outside temperature rises to the level that cooling is needed, then the indoor temperature can rise to any level. If only cooling is allowed and the outside temperature drops to the level that heating is needed, then the indoor temperature can drop to any level.

The internal energy generation term, Q, depends upon a variety of factors such as body heat, appliances, lights, industrial processes, etc., and can considerably reduce the heating requirements during the heating season and increase the cooling requirements during the cooling season. The internal energy generation rate varies from hour-to-hour, day-to-day, or month-to-month, but was treated as a constant for this analysis and is input into the computer program as a constant.

2.0 HEAT PUMP HEATING AND COOLING MODEL

The heat pump heating and cooling model was developed to determine the seasonal performance of both air- and water-source heat pumps.

2.1 Air-Source Heat Pump Model

Air-source heat pumps were modeled using manufacturers' data to describe the heating and cooling capacities and power requirements of the heat pump as a function of the indoor and outdoor temperatures. (A linear interpolation was used between data points.) Manufacturers' data for heat pump power requirements include power to the compressor, outdoor fan(s), and indoor fan. The outdoor dry bulb temperatures are used to calculate the heating or cooling load which must be supplied to a given enclosure during that hour to maintain the indoor temperature at a desired level. Outdoor dry bulb temperatures are also used to determine the heating or cooling capacity a given air-source heat pump is capable of supplying to an enclosure and the electrical energy required by the heat pump for the same hour. If the heat pump is not capable of supplying the total amount of heating energy required to maintain a constant temperature within the enclosure during a heating period, the difference is made up by auxiliary electrical resistance heating.

Frost formation on the outdoor coils encountered with air-source heat pumps reduces the capacity of the heat pump if allowed to remain on the coils. All air-source heat pumps use an automatic defrost cycle for frost removal which uses either a timed cycle or one based on a pressure drop across the outdoor coils. When in the defrost mode, the heat pump cycle will reverse and remove heat from the enclosure and melt the frost on the outdoor coils, while auxiliary heating maintains the enclosure at the desired temperature.

The performance model developed here uses a defrost cycle based on a pressure differential across the outdoor coils. The maximum allowable pressure differential occurred when 0.5 in of ice had accumulated on a flat surface. Basically, the calculation uses the relative humidity to calculate the humidity ratio of the outside air entering the air handler. The humidity ratio was then determined for the outside air leaving the air handler. The amount of frost deposited (if any) was determined from these two humidity ratios and the surface area of the outdoor coils. No frost formation was assumed for outdoor dry bulb temperatures above 40°F.

The only major differences between the heating and cooling models are: (1) There is no defrost problem involved with the cooling cycle, and (2) there is no auxiliary cooling involved. If the cooling load is greater than the capacity of the heat pump, the indoor temperature is allowed to rise above the desired indoor cooling temperature.

2.2 Water-Source Heat Pump Model

The water-source heat pump model is basically the same as the air-source heat pump model except that the heating and cooling capacities and power requirements of the heat pump are a function of the source-water temperature and flow rate which are assumed to be constant throughout the calculation. The seasonal performance of the heat pump is still a function of the outdoor air temperature in that the performance is dependent upon the heating and cooling loads which are dependent upon the outdoor air temperature. Provided the water temperature is not too low, frost formation is not a problem with water-source heat pumps for either heating or cooling cycles.

3.0 DESCRIPTION OF A TYPICAL NORTHERN UNITED STATES ENVIRONMENT

The heat pump seasonal performance was evaluated in a typical northern United States environment which will be defined for the purposes of this study as an environment in the neighborhood of 8000 degree-days of heating and 500 degree-days of cooling (the degree-days are based on 65°F). The normal seasonal heating degree-days for the United States is shown in Figure 2 (extracted from Reference 3) with the area of the United States between 7000 and 9000 degree-days darkened. The normal seasonal cooling degree-days for the United States is shown in Figure 3 (extracted from Reference 3) with the area less than 500 degree-days darkened. The area of the United States with a seasonal heating requirement between 7000 and 9000 degree-days represents much of the harsher climate (not including some of the severe high mountain climate) of the country. This area of the country has low cooling requirements.

The seasonal performance evaluation was made utilizing an actual environment recorded on an hourly digital basis. These recorded data were obtained from the National Oceanic and Atmospheric Administration (NOAA) Climatic Center in Asheville, N. C. These data were provided to HTPUMP from a magnetic tape (9 track, TDF-14 format) containing wet and dry bulb temperatures, relative humidity, wind velocity, and atmospheric pressure. The seasonal performance evaluation uses the atmospheric dry bulb temperature, the relative humidity and atmospheric pressure. Computer subroutines were developed to read the tapes, decode the required data, and modify the data as necessary such that the required data were returned to the main program in one-day blocks for analytical purposes. Data modification was necessary for conversion to proper units and for filling in missing data. Missing data were filled with the preceding data points.

The data tape obtained from NOAA consists of airways surface observations from the municipal airport of Idaho Falls, Idaho, for the years 1955 through 1964. The year 1962, having 8290 degree-days of heating and 630 degree-days of cooling, was used in this evaluation. The temperature distribution of this environment, shown in Figure 4, has a maximum of 94°F, a minimum of -28°F, and an average of 44°F.

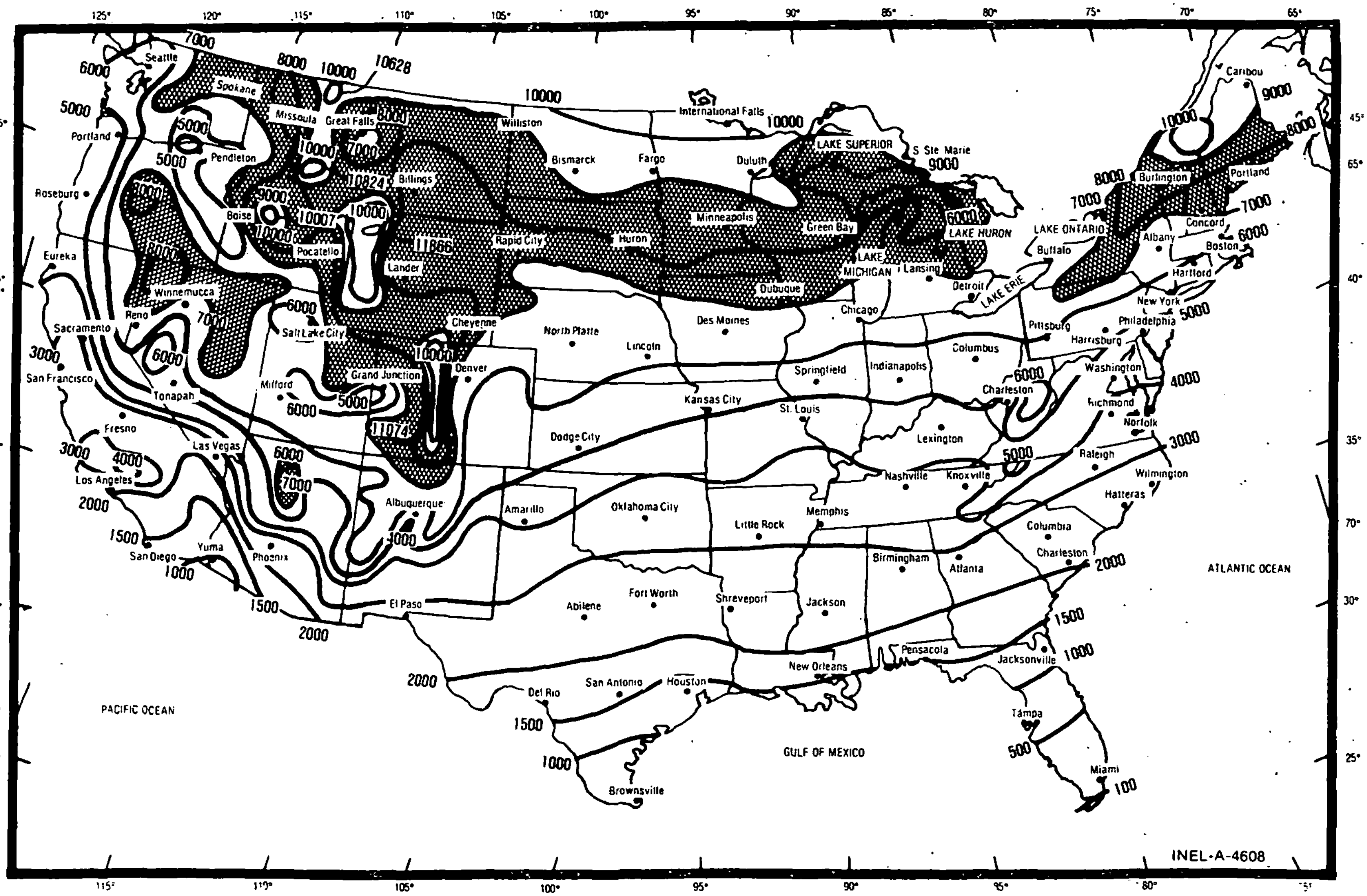

Figure 2. Normal seasonal heating degree days (base 65°F) 1941-1970. (Reference 3) The cross-hatched region represents the regions of the United States which have between 7000 and 9000 degree days of heating.

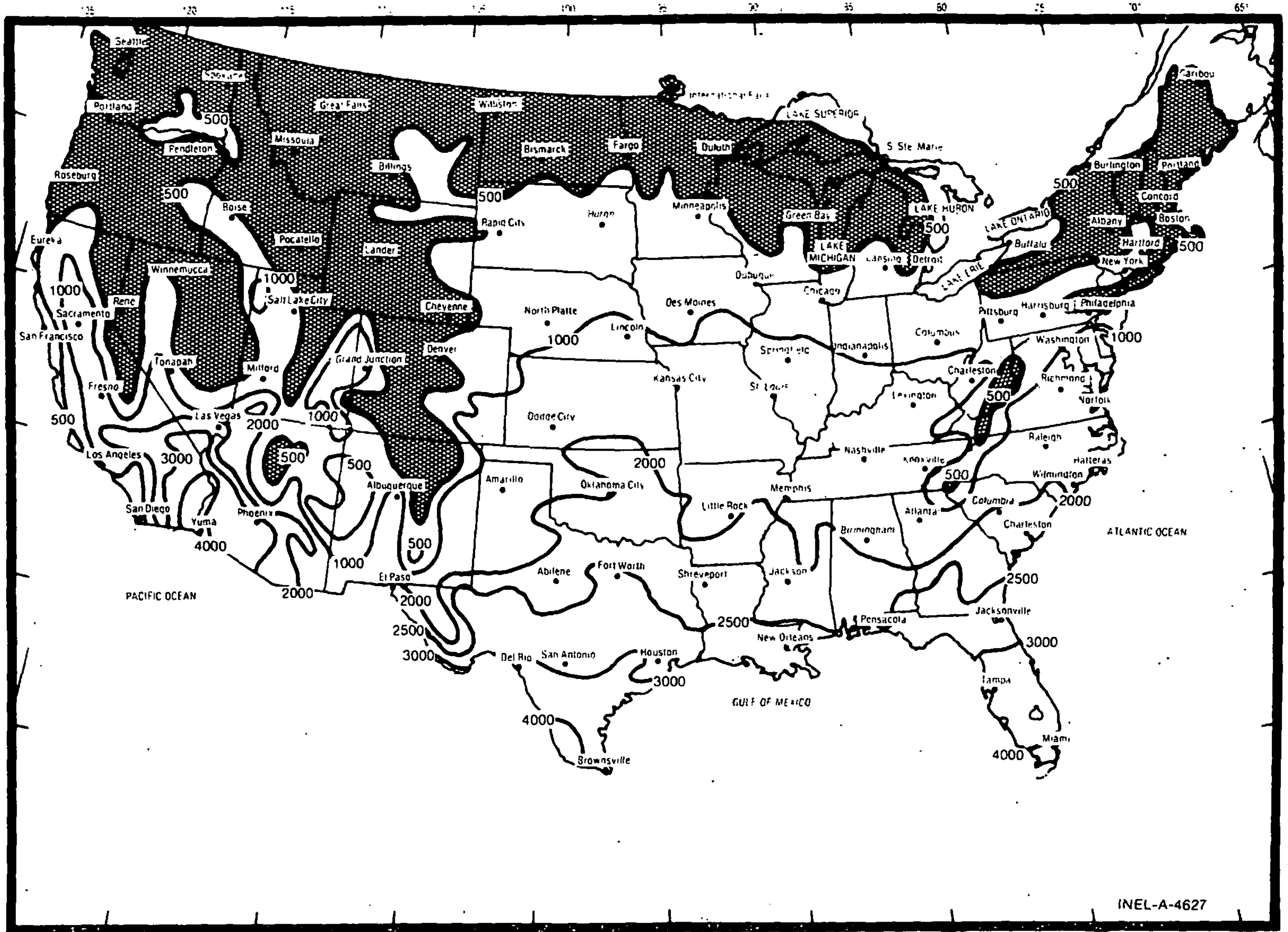

Figure 3. Normal seasonal cooling degree days (base 65°F) 1941-1970. (Reference 3). The cross-hatched regions of the United States which have less than 900 degree days of cooling.

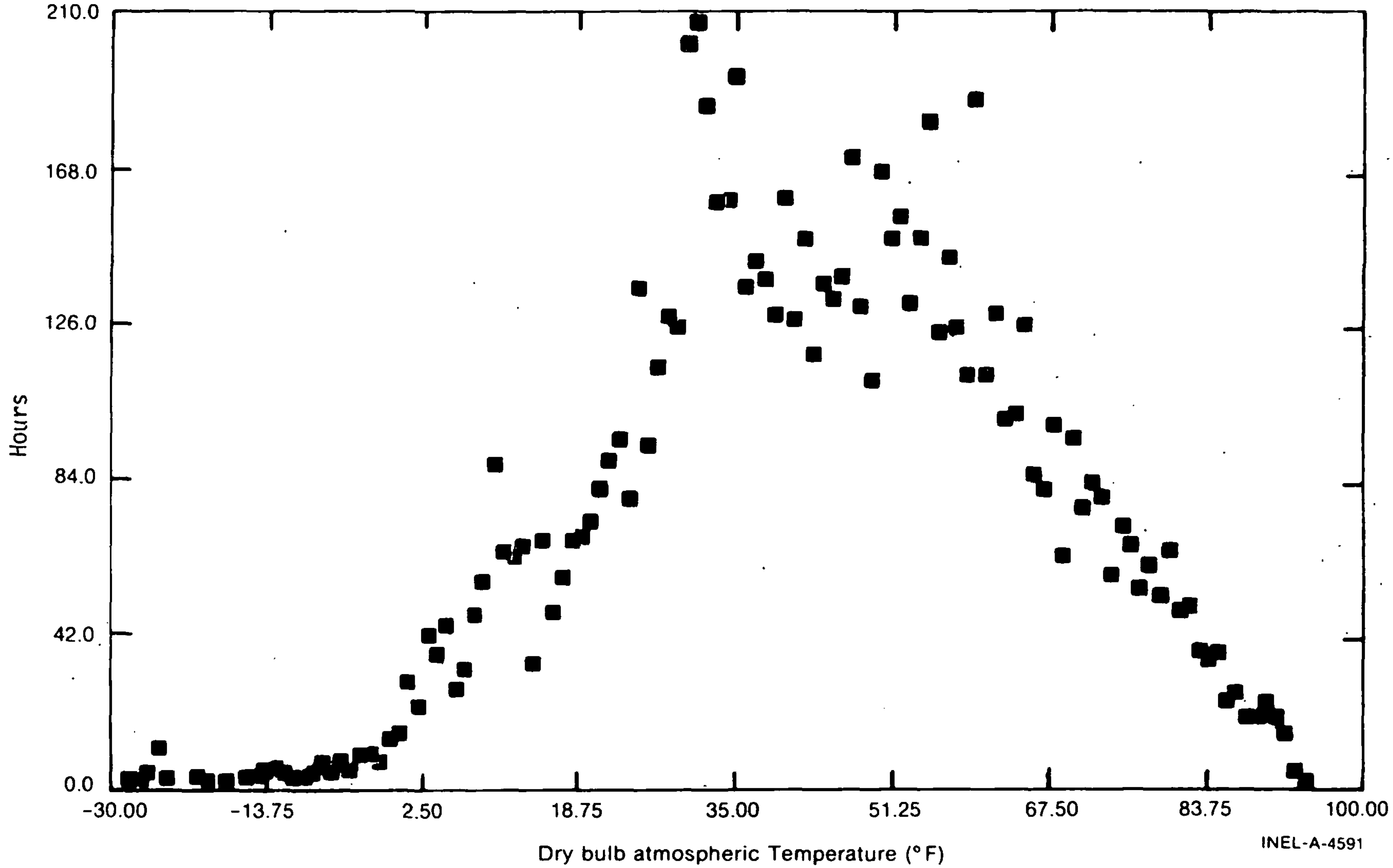

Figure 4. Typical environment seasonal temperature distribution.

V. SEASONAL PERFORMANCE EVALUATION RESULTS

The seasonal performance was evaluated for rather typical air-source and water-source heat pumps of different sizes. The heat pump performance data of these typical heat pumps and the seasonal performance results both are presented in this section.

1.0 TYPICAL HEAT PUMP PERFORMANCE DATA

1.1 Air-Source Heat Pump Performance Data

The performance data of four air-source heat pumps of different sizes from the same manufacturer were chosen as typical data. The manufacturer was chosen on the basis of typical heating and cooling performance as defined by the comparison studies documented in Appendix A and on the basis of availability of detailed data. The performance data for these four heat pumps are shown in Figures 5 through 8.

The heating capacities of the heat pumps are shown in Figure 5 as a function of the outdoor dry bulb air temperature. The four typical heat pumps are nominally sized at 2-, 3-, 4-, and 5-tons of cooling capacity. (One ton of cooling capacity is equal to 12 000 Btu/h of heat removal capability.) These nominal capacities are obtained at an outdoor air temperature of roughly 45°F. As shown in Figure 5, the heating capacity is highly dependent upon the source air temperature and is a nonlinear function which increases with an increasing air temperature. The heat load requirement on the other hand increases with a decreasing air temperature. For example, an enclosure with a constant heat-loss factor of 500 Btu/h-°F would have a heat load of zero at about 65°F air temperature and a heat load of about 42 500 Btu/h (neglecting any internal energy generation) at an air temperature of -20°F. This means that at the higher temperatures the heat pump only has to run part time in order to provide the heat requirement but at the lower temperatures auxiliary heating must be employed to assist the heat pump in providing the heat requirement. The two-ton heat pump for instance will provide the heat load by running full time when the outdoor air temperature is about 30°F

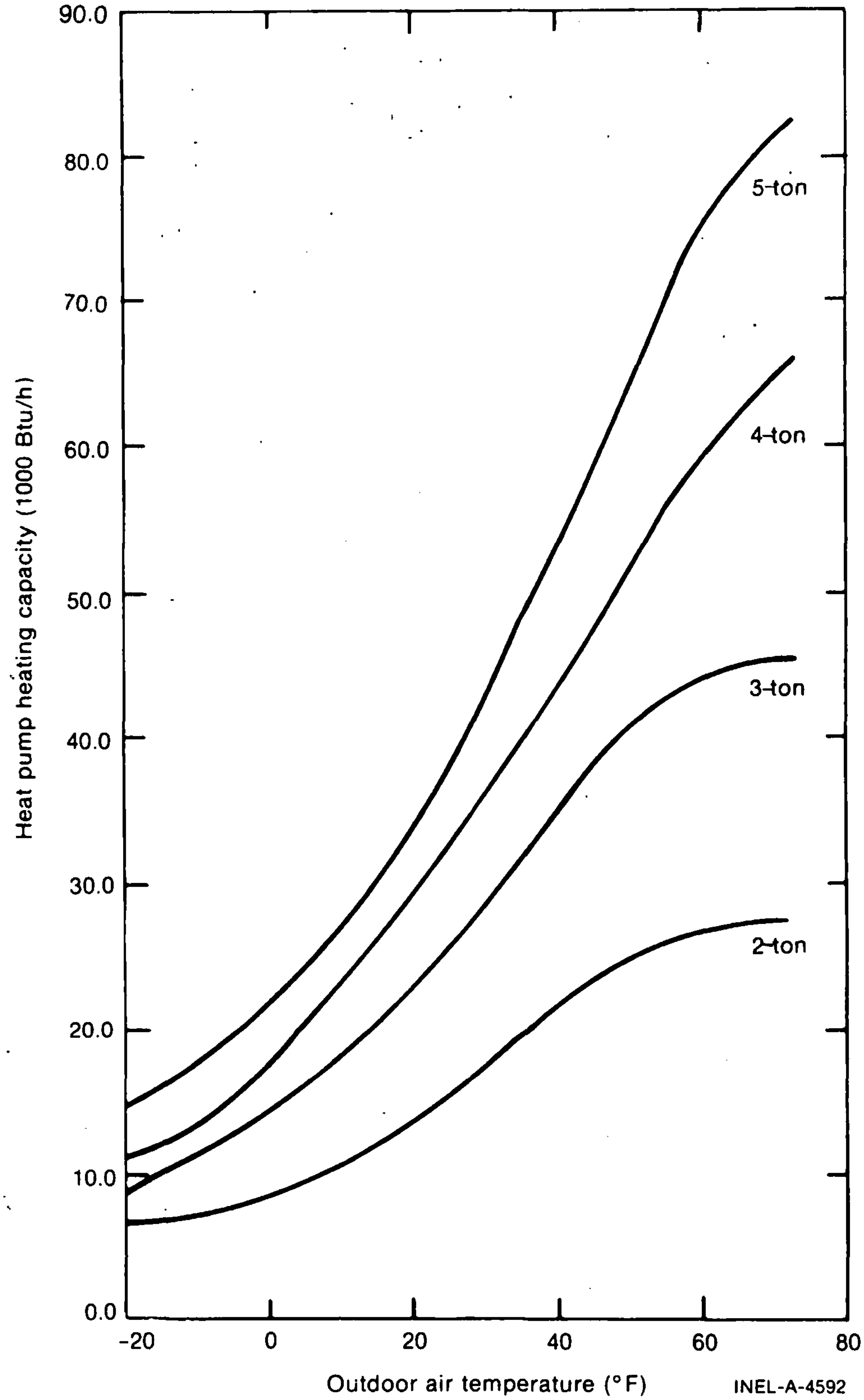

Figure 5. Heating capacities of typical air-source heat pumps

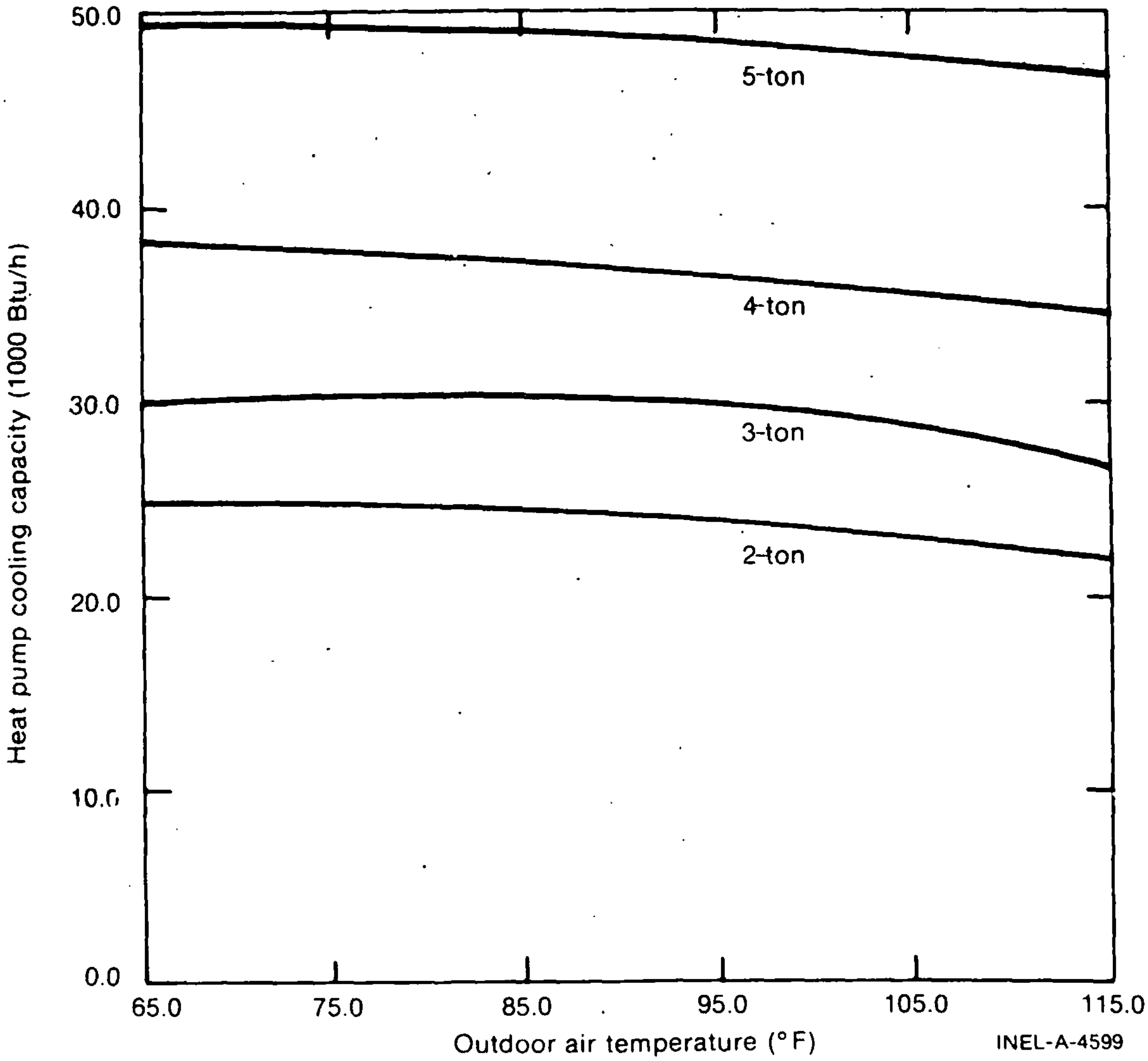

Figure 6. Cooling capacities of typical air-source heat pumps.

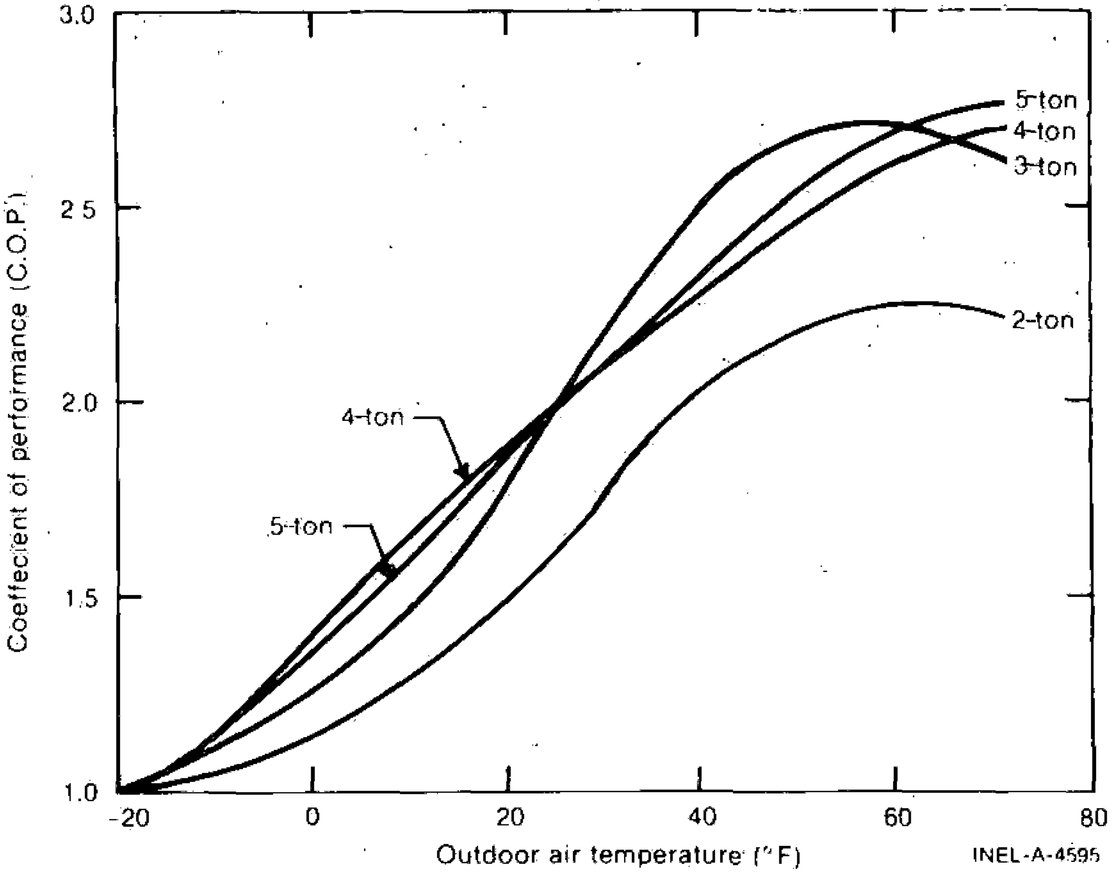

Figure 7. The coefficients of performance of typical air-source heat pumps.

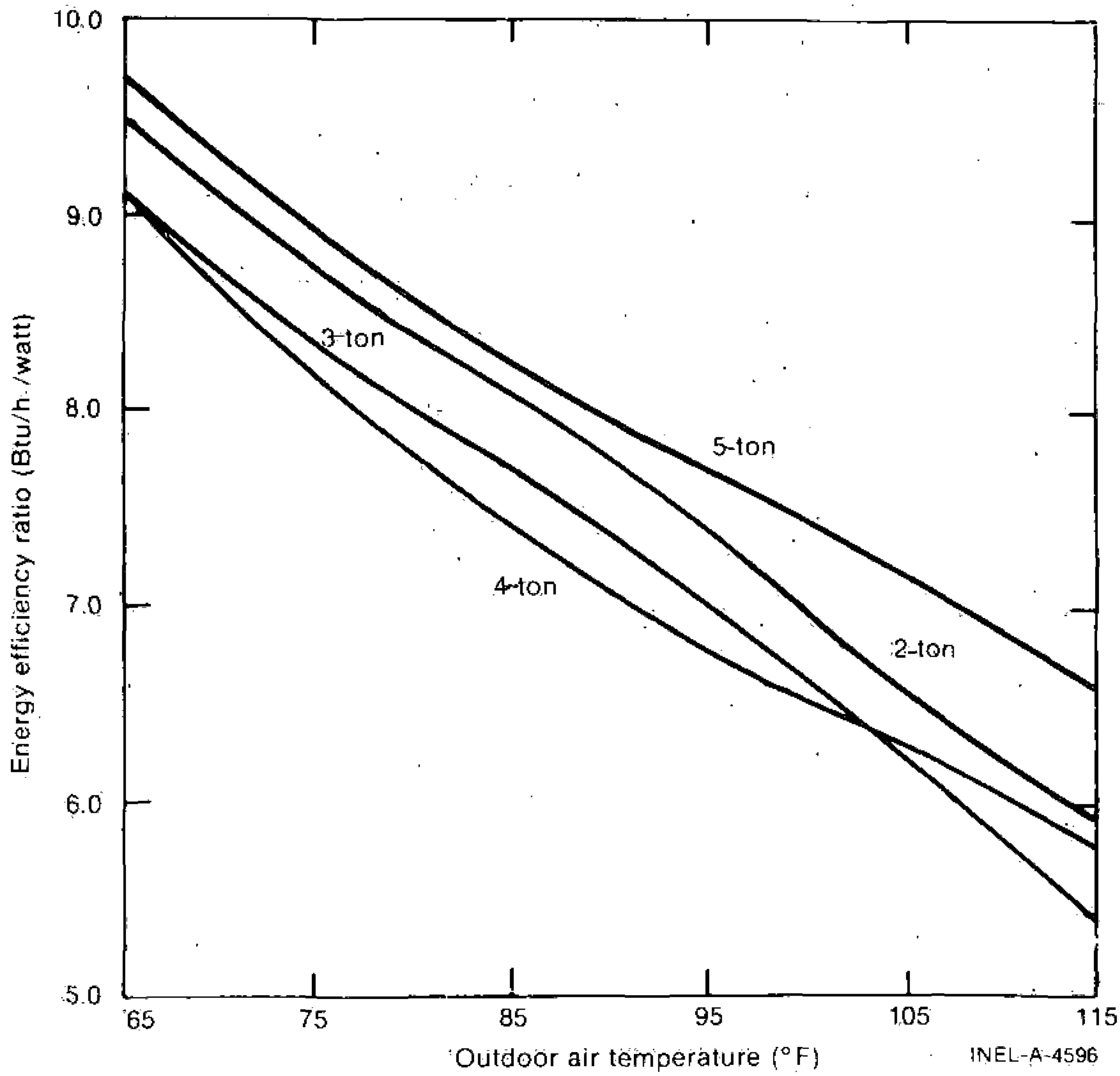

Figure 8. Energy efficiency ratios for cooling of typical air-source heat pumps.

(assuming a constant heat-loss factor of 500 Btu/h-°F) but at any lesser temperature auxiliary heating is required. Similarly, the 3-, 4-, and 5-ton heat pumps can provide the heat loads down to air temperatures of about 20, 15, and 10°F, respectively.

The cooling capacities of the same four heat pumps are shown in Figure 6 also as a function of the outdoor dry bulb air temperature. The cooling capacities decrease somewhat with an increasing air temperature but are much less dependent upon the air temperature than are the heating capacities. The cooling load of an enclosure with a constant heat-loss factor of 500 Btu/h-°F, for example, will increase from a zero cooling load at an air temperature around 65°F to about 25 000 Btu/h at an air temperature of 115°F. There is no auxiliary cooling capability should the heat pump not be able to provide the cooling load.

The heat pump power requirements to provide the heating and cooling capacities (shown in Figures 5 and 6) can be deduced from the coefficients of performance in Figure 7 for heating and the energy efficiency ratios in Figure 8 for cooling. The power requirement at a given air temperature for a given heat pump is obtained by dividing the heating capacity by the coefficient of performance, and similarly for cooling by dividing the cooling capacity by the energy efficiency ratio. The general trend is that the coefficient of performance increases with increasing air temperature while the energy efficiency ratio decreases with increasing air temperature. The coefficient of performance is reduced to a value of 1.0 at low air temperatures meaning that the heating capacity is equal to the heat pump power input. The COPs and EERs vary somewhat from one heat pump to another due to engineering design differences and not because of thermodynamic considerations. Such variations are typical of commercial heat pump data.

1.2 Water-Source Heat Pump Performance Data

The performance data of four water-source heat pumps of different sizes from the same manufacturer (not the same as the air-source heat pumps) were chosen as typical data. Again the manufacturer was chosen on the basis of typical performance as defined by the comparison studies documented in Appendix A and on the availability of detailed data. The water-source heat pumps were also chosen such that the nominal size ratings, i.e., 2-, 3-, 4-, and 5-tons, are the same as the typical air-source heat pumps. The water-source heat pump performance data are shown in Table I.

The heating and cooling capacities, the coefficients of performance, and the energy efficiency ratios for each heat pump are also given in Table I for source-water temperatures of 60, 70, and 80°F. General trends exist in Table I similar to those for the air-source heat pump performance data (i.e., [1] the heating capacities increase with increasing source-water temperature while the cooling capacities decrease, and [2] the COPs increase with the increasing source-water temperature while the EERs decrease).

The water-source heat pumps require less auxiliary heating than the air-source heat pumps if the water temperature is 60°F or greater throughout the season. For example, the smallest heating capacity in Table I is 24 600 Btu/h and, if an enclosure with a constant heat-loss factor of 500 Btu/h-°F is assumed, then auxiliary heating would not be needed until the outdoor air temperature dropped below about 16°F. Similarly, for the heat capacity of 43 050 Btu/h, the auxiliary heating would not be needed until the outdoor air temperature dropped below about -21°F.

2.0 EVALUATION ASSUMPTIONS

Several assumptions were made in the evaluation of typical heat pumps:

1. The indoor heating temperature was set at 70°F and the heating season was January through June and September through December in all cases. There was no heating during July and August even though the indoor temperature occasionally dropped below 70°F during these months.

2. The desired indoor cooling temperature was set at 74°F and the cooling season was June through September in all cases. There was no cooling during the months of January through May and October through December even though the indoor temperature occasionally exceeds 74°F during these months. (Heating and cooling are both available during the months of June and September.)

TABLE I

TYPICAL WATER-SOURCE HEAT PUMP PERFORMANCE DATA

Nominal Size Rating (tons)(1)	Water(2) Flow Rate (gpm)	Source-Water Temperature (°F)	Heating Performance Data		Cooling Performance Data	
			Capacity (1000 Btu/h)	COP	Capacity (1000 Btu/h)	EER (Btu/Wh)
2	3.0	60	24.60	2.32	25.96	9.76
		70	27.85	2.41	24.49	8.65
		80	31.50	2.54	22.53	7.38
3	5.5	60	33.60	2.38	42.85	9.97
		70	36.90	2.40	40.05	8.84
		80	43.05	2.61	37.20	7.62
4	7.5	60	49.00	2.76	52.50	10.44
		70	55.45	2.87	49.50	9.25
		80	62.70	3.02	45.60	8.00
5	8.5	60	55.00	2.68	60.40	9.71
		70	62.05	2.78	57.00	8.61
		80	70.40	2.94	53.00	7.52

(1) One ton of cooling is equal to 12 000 Btu/h.

(2) The lowest source-water flow rate for which data was available is listed.

3. The internal energy generation rate was assumed at a constant 3000 Btu/h in all cases. This rate is an approximation for the sum of the waste heat energy from body heat, appliances and lights for a low occupancy enclosure but does not contain waste heat from an industrial process. The internal energy generation both reduces the heating requirements and increases the cooling requirements.

4. The energy storage capacity (mc_p) for the enclosure was assumed at a constant 6000 Btu/°F which was calculated for a low-occupancy enclosure without an industrial process.

5. The heat pump defrost cycle (for air-source heat pumps) was assumed to operate whenever a frost thickness of 0.5 in has accumulated on the coils and then defrosts for five minutes. It was also assumed that frost formation on the coils could not occur whenever the outdoor air temperature was greater than 40°F and the frost density was assumed to be 52 lbm/ft^3.

6. The auxiliary heating system was assumed to be an electrical resistance heating unit with an energy conversion efficiency of 1.0.

3.0 PARAMETRIC RESULTS

Some parametric studies were performed for air-source and water-source heat pumps.

3.1 Air-Source Heat Pump Parametric Results

The seasonal performance of each of the four typical air-source heat pumps was evaluated for heat-loss factors of 250, 500, 1000, and 1500 Btu/h-°F. The seasonal heating performance results are shown in Figure 9 and the seasonal cooling performance results are given in Table II.

The seasonal heating performance results shown in Figure 9 reflects the heat pump heating capacities shown in Figure 5 and heat pump coefficients of performance shown in Figure 7 applied over an entire heating season. The

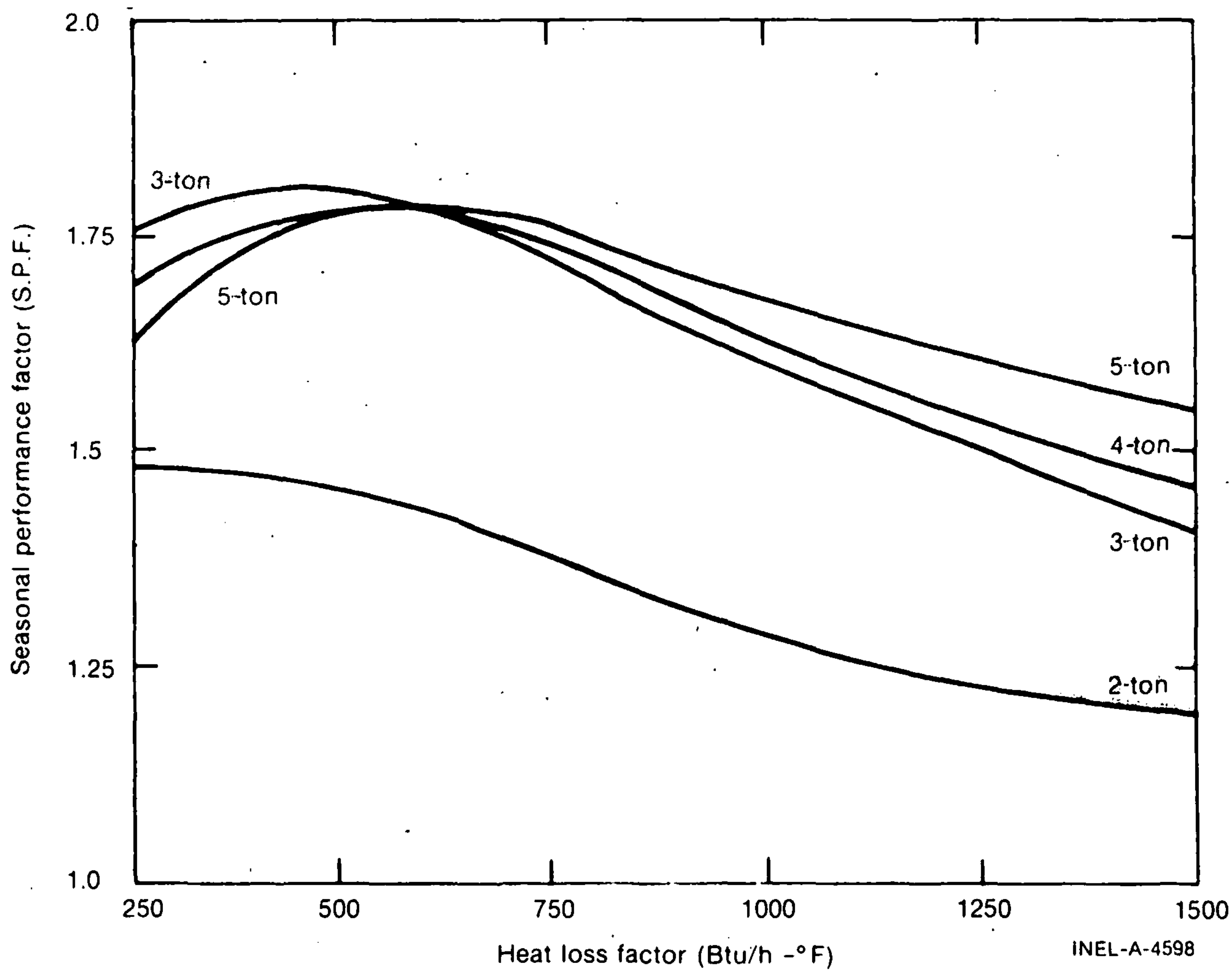

Figure 9. Air-source heat pump seasonal heating performance.

TABLE II

SEASONAL COOLING PERFORMANCE FOR AIR-SOURCE HEAT PUMPS

Heat Pump Size (tons)	Heat-Loss Factor (Btu/h-°F) 250	500	1000	1500
2	8.31	8.14	8.11	8.11
3	7.90	7.72	7.68	7.68
4	7.64	7.45	7.40	7.39
5	8.47	8.31	8.26	8.26

seasonal heating performance is shown as the seasonal performance factor (S.P.F.) which reflects the energy savings of the heat pump system over total electrical heating. For example, a heat pump system having a S.P.F. of 1.75 would require approximately 43% less energy than an all electrical heating system.

The S.P.F.s appear to have a maximum between heat-loss factors of about 500 and 750 Btu/h-°F for the 3-, 4-, and 5-ton heat pumps but the maximum for the 2-ton heat pump is at or less than a heat-loss factor of 250 Btu/h-°F. The S.P.F. decreases for all the heat pumps as the heat-loss factor increases beyond the maximum due to the increased use of auxiliary heating. That is, as the heat-loss factor increases and thus the heat load increases, more auxiliary heating is needed to supply the heat load which causes the S.P.F. to decrease. The S.P.F.s for the 3-, 4-, and 5-ton heat pumps decrease at the low heat-loss factor of 250 Btu/h-°F because of the greater portion of the time which the heat pump runs at a lower coefficient of performance due primarily to the heat generation rate. For example, assume a heat generation rate of 3000 Btu/h and a heat-loss factor of 259 Btu/h-°F. This implies that an indoor-outdoor temperature differential of 12°F is necessary before the heat pump is needed, i.e., the heat pump does not need to be turned on until the outdoor temperature drops below about 58°F. Therefore, the fraction of time that the heat pump runs at lower source temperature increases which decreases the S.P.F.

The seasonal performance factor for the 2-ton heat pump shown in Figure 9 is considerably lower than for the other heat pumps for all heat loss factors evaluated. The low S.P.F. of the 2-ton heat pump is related to the low coefficient of performance curve in Figure 7 which is inherent in the particular design of that heat pump and not of 2-ton heat pumps in general. These results illustrate the non-standardization of commercially available heat pumps.

The seasonal cooling performance for these air-source heat pumps is shown in Table II in terms of the seasonal cooling factor (S.C.F.). The S.C.F.s vary from heat pump to heat pump but vary only slightly with the heat-loss factor which governs the cooling load. The variation from heat pump to heat pump is primarily due to the variation in the energy efficiency ratios as shown in Figure 8. The slight variation with the heat-loss factor is due in effect to the heat generation rate. At the lower heat-loss factors, cooling is needed at lower outdoor temperatures. For example, again assume a heat generation rate of 3000 Btu/h and a heat loss factor of 250 Btu/h-°F which implies that an indoor-outdoor temperature differential of 12°F is necessary to remove the generated heat. Therefore, cooling will be needed at temperatures greater than 12°F less than the desired indoor cooling temperature. If the indoor temperature is to be cooled to 74°F, then cooling must be provided for outdoor temperatures greater than 62°F. Since cooling begins at lower outdoor temperatures for the lower heat-loss factors, the S.C.F. increases slightly due to the higher energy efficiency ratios for cooling at lower outdoor temperatures (see Figure 8).

3.2 Water-Source Heat Pump Parametric Results

The seasonal performance of each of four typical water-source heat pumps was evaluated for heat-loss factors of 250, 500, 1000, and 1500 Btu/h-°F. The seasonal heating performance results are shown in Figures 10 through 13 and the seasonal cooling performance results are given in Table III. The seasonal performance is dependent upon the source-water temperature, source-water flow rate and associated pumping power, and atmospheric air temperature. The coefficient of performance and energy efficiency ratio are dependent upon the source-water temperature, flow rate, and pumping power and the heating and cooling loads are dependent upon the outdoor air temperature. The seasonal performance results presented in Figures 10 through 13 and in Table III assume that the lowest water flow rate allowed by operational limits is used and that the pumping power to pump the water to the heat pump heat exchanger is zero.

The seasonal heating performance for water-source temperatures of 60, 70, and 80°F are shown in Figures 10, 11, and 12, respectively, and reflects the heat pump heating capacities and coefficients of performance shown in Table I applied over an entire heating season. Again the seasonal heating performance is shown as the seasonal performance factor (S.P.F.) which reflects the energy savings of the heat pump system over total electrical heating. The S.P.F.'s appear to reach the maximums at heat-loss factors in ranges of

TABLE III

SEASONAL COOLING PERFORMANCE FOR WATER-SOURCE HEAT PUMPS

Heat Pump Size (tons)	Seasonal Cooling Factors (Btu/kWh) Source-Water Temperature 60°F	70°F	80°F
2	9.76	8.65	7.38
3	9.97	8.84	7.62
4	10.44	9.25	8.00
5	9.71	8.61	7.52

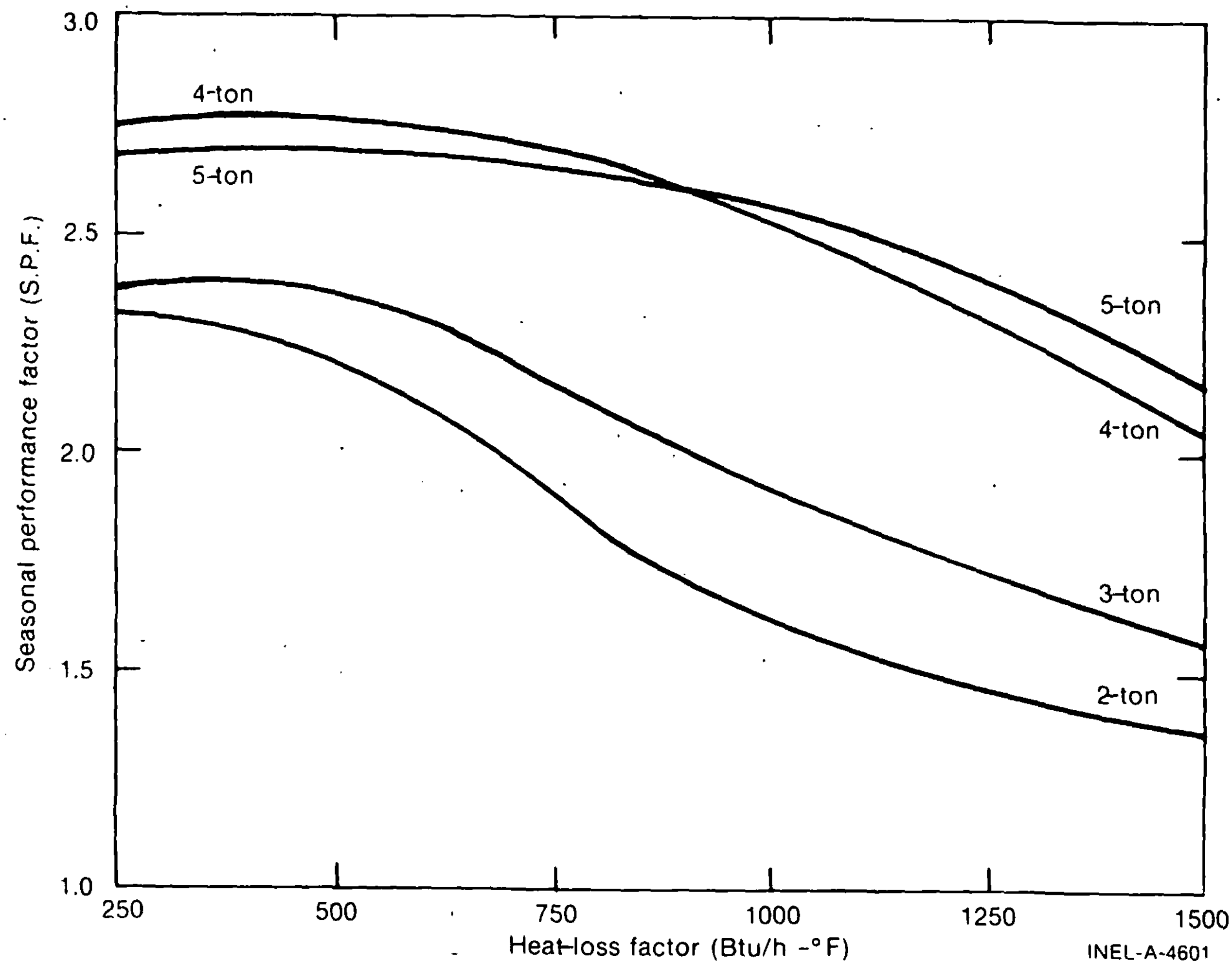

Figure 10. Seasonal performance factor vs. heat-loss factor for water-to-air heat pumps using a 60°F water source and low water flow.

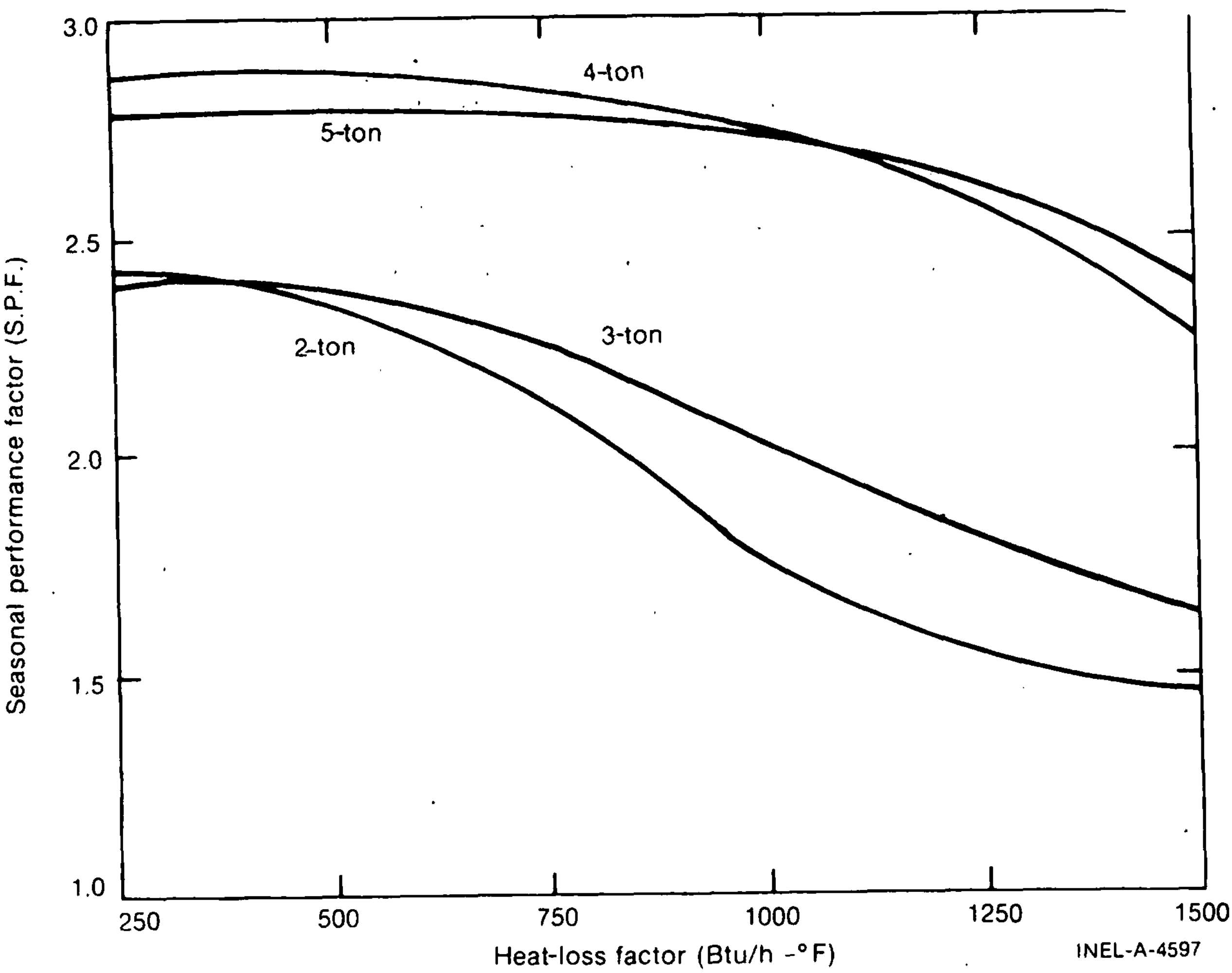

Figure 11. Seasonal performance factor vs. heat-loss factor for water-to-air heat pumps using a 70°F water source and low water flow.

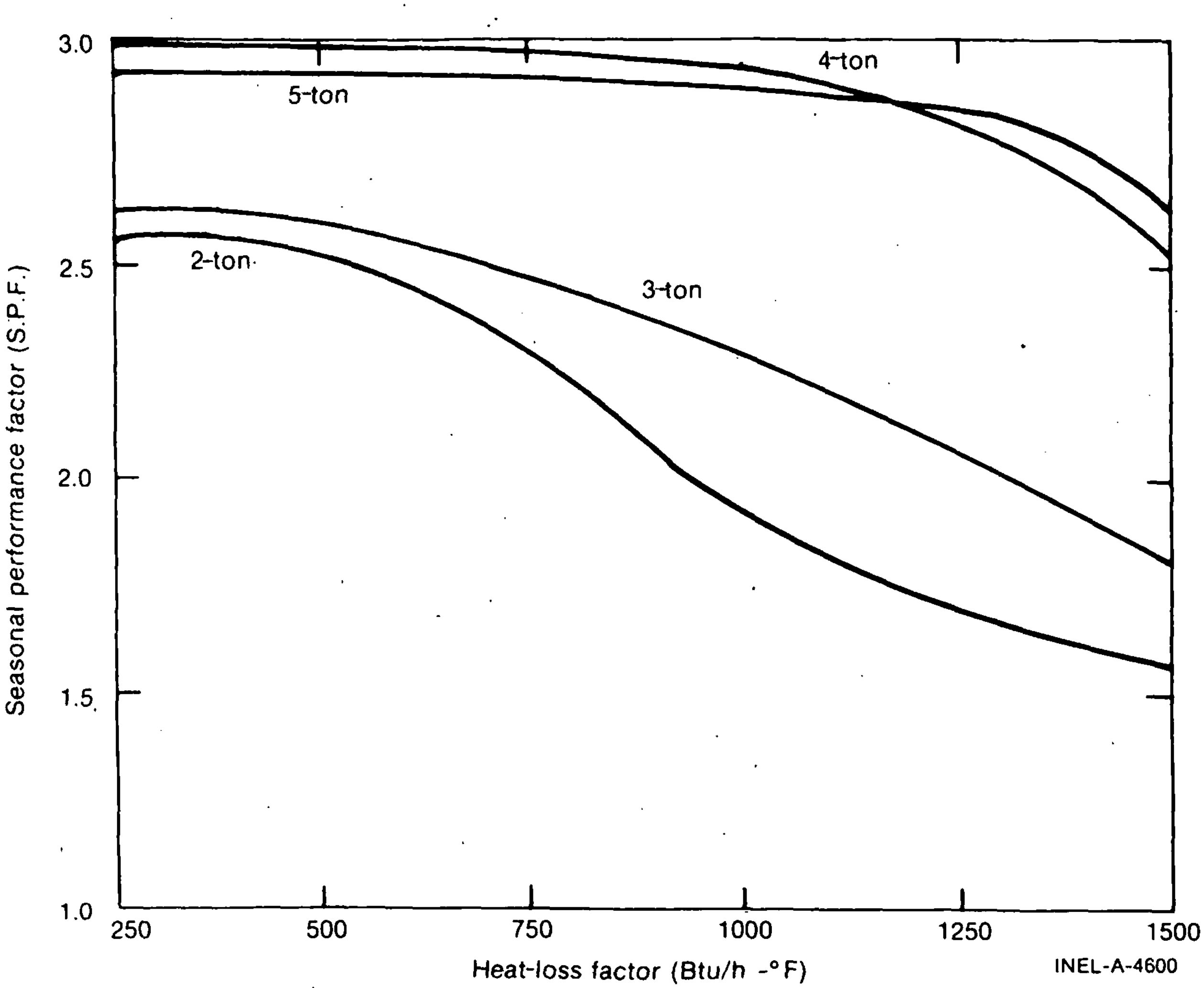

Figure 12. Seasonal performance factor vs. heat–loss factor for water-to-air heat pumps using a 80°F water source and low water flow.

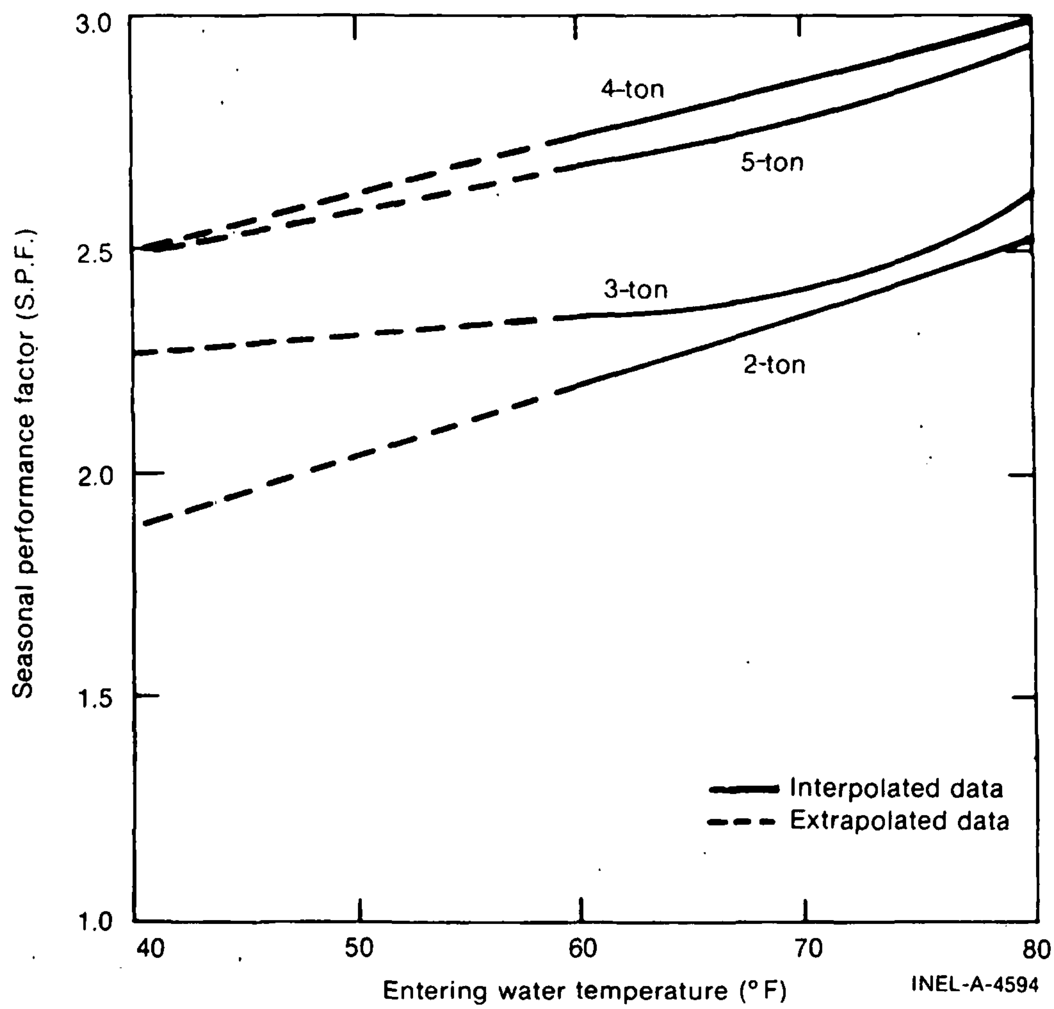

Figure 13. Seasonal performance factor vs. entering water temperature for water-to-air heat pumps using low water flow and a heat-loss factor of 500 Btu/h-°F.

250 to 500 Btu/h-°F and to decrease for higher heat-loss factors. The S.P.F.s decrease at the higher heat-loss factors because of the seasonal increased use of auxiliary heating. The seasonal heating performance dependence upon the source-water temperature is shown in Figure 13 for the heat-loss factor of 500 Btu/h-°F. Figure 13 shows that the S.P.F.s increase with an increasing source-water temperature as would be expected. The plots in Figure 13 were linearly extrapolated below a water-source temperature of 60°F such that the S.P.F.s can be estimated for source-water temperatures a few degrees below 60°F.

The seasonal cooling performance for these water-source heat pumps is shown in Table III in terms of the seasonal cooling factor (S.C.F.). The S.C.F. s vary from heat pump to heat pump and with the source-water temperature but are independent of the heat-loss factor. The S.C.F.s are the same as the EERs shown in Table I. Since the analysis assumes that the water temperature is constant and that there is no auxiliary cooling, then the heat pump cools at a constant EER.

The heat pump seasonal performance of a water-source heat pump is dependent upon the water flow rate through the heat exchanger as is shown in Figure 14. Figure 14 shows the sensitivity of the SPF to the water flow rate for water temperatures of 60 and 80°F. The allowable high and low flow rates for each heat pump are also listed. The SPFs are shown to increase slightly, but it should be noted that this analysis assumes that no pumping power is required to pump the water to the heat pump heat exchanger. If significant pumping power is required for a given application, then the pumping power would increase with an increasing flow rate and could cause the SPF to decrease.

The effect of the pumping power required to pump the water from the source to the heat pump heat exchanger may be estimated using the following equation:

$$SPF_{new} = \frac{(SPF_{old})\ E}{E + (SPF_{old})\ P_p} \tag{9}$$

where

SPF_{new} = new seasonal performance factor adjusted to include pumping energy.

SPF_{old} = seasonal performance factor assuming no pumping other than that required to move the water through the heat exchanger.

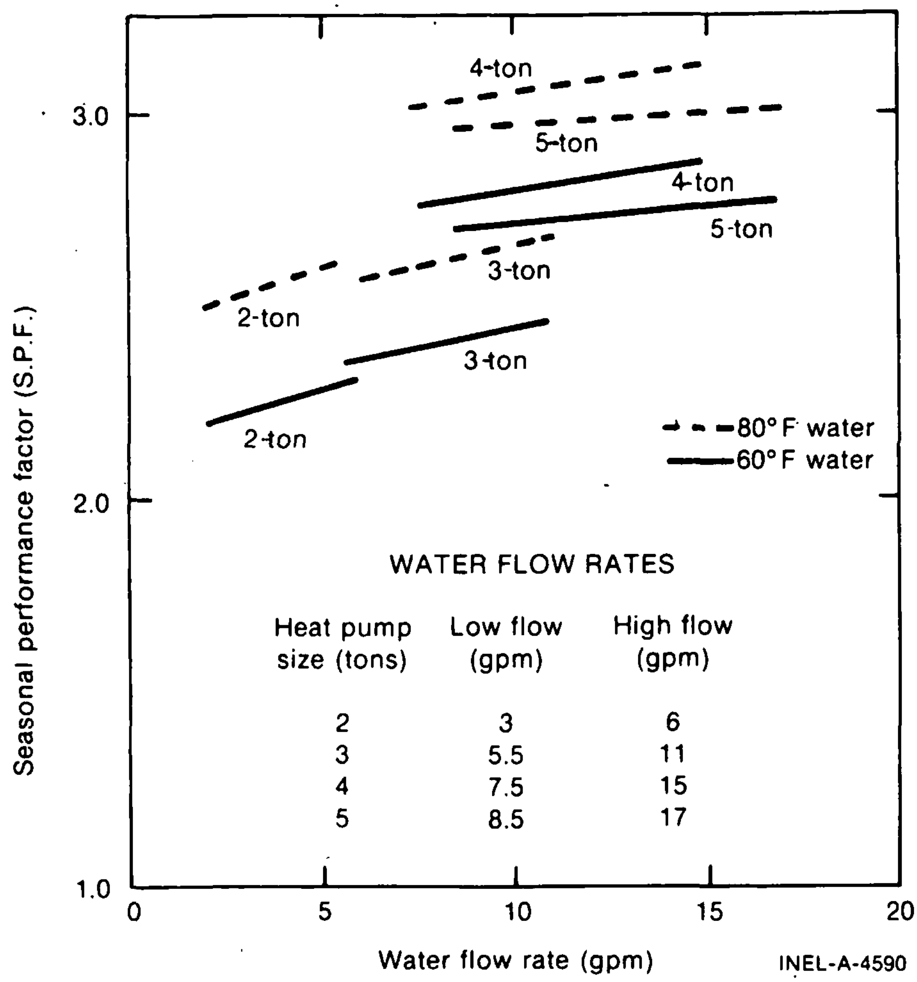

Figure 14. Seasonal performance factor vs. water flow rate for water-to-air heat pumps with a heat loss factor of 500 Btu/h -°F.

E = total energy supplied by the heating system during the heating season (kWh).

P_p = total energy required for pumping during the heating season (kWh).

Use of Equation (9) is illustrated with the following example. Assume a 4-ton water-source heat pump, a heat-loss factor of 500 Btu/h-°F, a water-source temperature of 60°F and a water flow rate of 7.5 gpm. Find the decrease in the SPF if the water must be pumped with a head of 400 feet instead of zero. First, the pumping power is calculated for a flow of 7.5 gpm pumped 400 feet to be 0.7 kW. Then the total pump power is computed from the heat pump run time which can be obtained from Figure 15 for this case. The running time for this problem is 1870 hours which results in 1310 kWh of energy. Next, the total energy supplied by the heat pump system in kWh can also be obtained from Figure 15 which is 24 000 kWh. Inserting these numbers into Equation (9) results in a new SPF of 2.39 whereas the old SPF was 2.75 (see Figure 10) which is about a 13% decrease in the SPF.

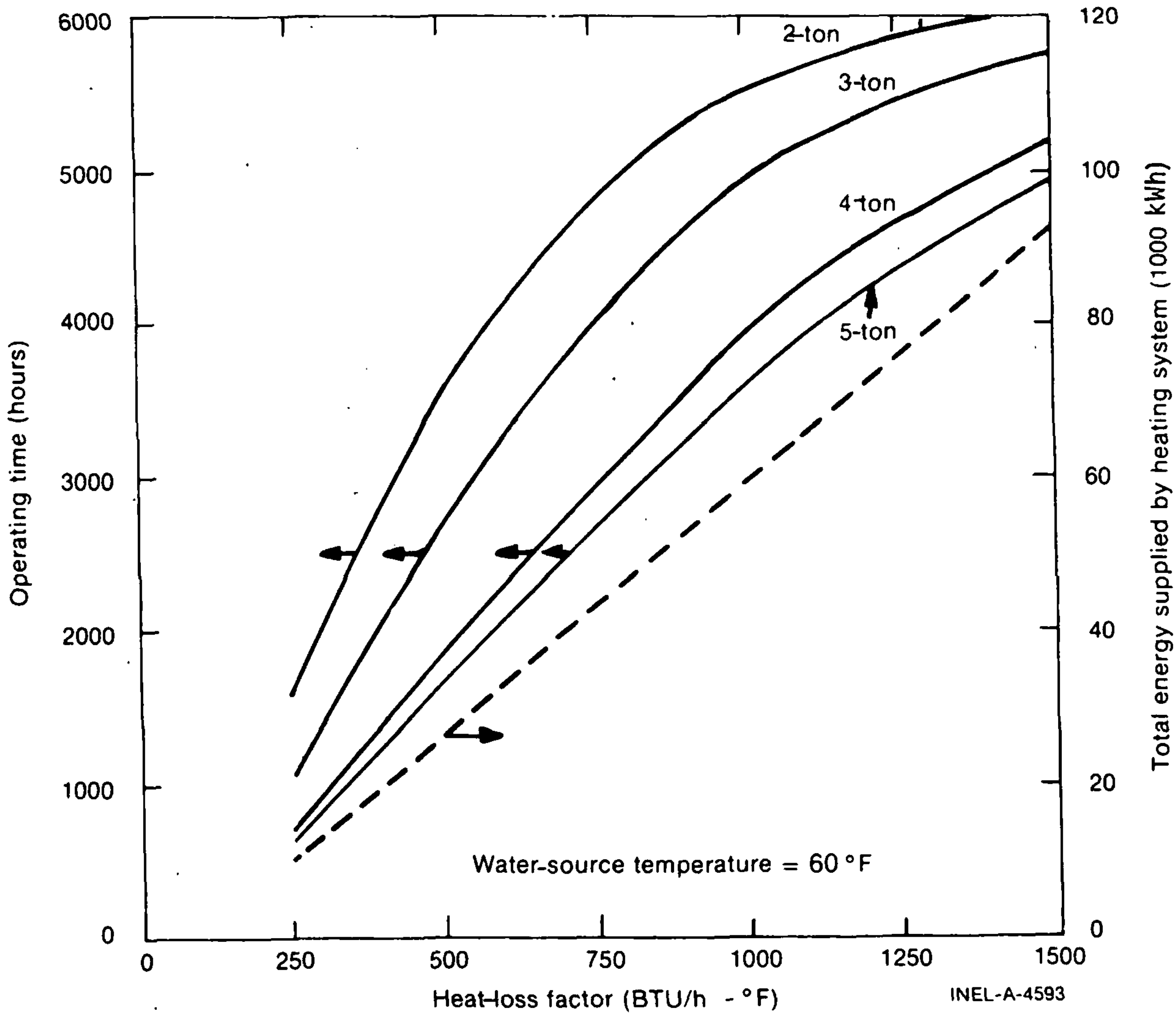

Figure 15. Heat pump seasonal heating operating time and energy supply requirements.

VI. CONCLUSIONS

Several conclusions were drawn about the availability of commercial heat pumps and their seasonal performance in a typical northern United States environment.

1. The performance data on the commercially available heat pumps are considerably scattered with very few general trends. The coefficients of performance for the air-source heat pumps tend to bunch in the general neighborhoods of 2.0 to 2.5 and 1.2 to 1.8 for air temperatures of 47 and 17°F (dry bulb), respectively, and in the general neighborhood of 2.3 to 3.0 for water-source heat pumps using 60°F temperature water. The energy efficiency ratios tend to bunch in the general neighborhoods of 6.0 to 7.0 and 7.5 to 9.0 Btu/Wh for air- and water-source heat pumps, respectively. There is a general trend towards a slightly better performance for larger capacity heat pumps but this is not necessarily true of the products of an individual manufacturer.

2. The majority of the commercially available heat pumps have a capacity under 5 tons (60 000 Btu/h), and there are considerably more air-source than water-source heat pumps available. This implies that large industrial sized heat pumps capable of utilizing industrial waste heat are somewhat limited.

3. The seasonal performance results of these typical, commercially available heat pumps indicate that an air-source heat pump system can save on the order of 45% of the energy required to heat using electrical resistance heating alone and that a water-source heat pump operating using 60 and 80°F temperature water can save the order of 64% and 67%, respectively. Significant water pumping power requirements can reduce the water-source heat pump energy savings.

VII. REFERENCES

1. Air-Conditioning and Refrigeration Institute, Directory of Certified Unitary Air-Conditioners, Unitary Heat Pumps, Sound-Rated Outdoor Unitary Equipment, and Central System Humidifiers, (July 1 to December 31, 1976).

2. J. B. Briggs and C. J. Shaffer, HTPUMP, CCCM file #H00115IA, (1977).

3. National Oceanic and Atmospheric Administration, Heating and Cooling Degree Day Data, C-14, (September 1974), pages 7 and 8.

APPENDIX A

MANUFACTURERS DATA EVALUATION

I. INTRODUCTION

An evaluation of manufacturers data was made using data published by the Air Conditioning and Refrigeration Institute (ARI)[1]. Unitary heat pumps of many different manufacturers were tested by the ARI under common laboratory conditions. A unitary heat pump consists of one or more factory-made assemblies (but when such equipment is provided in more than one assembly, the separated assemblies are to be designed to be used together, and the requirements of rating are based upon the use of matched assemblies) which normally include an indoor conditioning coil, compressor(s) and outdoor coil, or refrigerant-to-water heat exchanger, including means to provide both heating and cooling functions. The ARI standards for unitary heat pump equipment were prepared to establish: definitions and classification; requirements for testing and rating; specifications, literature and advertising requirements; performance requirements; safety requirements; and conformance conditions. The unitary equipment listed in their directory for air conditioners, heat pumps, and humidifiers represents more than 90 per cent of the total U. S. output of these types of equipment falling within the scope of the programs and rated below 135 000 Btu/h cooling capacity for air-conditioners and heat pumps. The scope and standards of the ARI data probably makes the data the most viable means of comparing different manufacturer's heat pumps to determine availability and performance.

II. STANDARDS AND TEST PROCEDURES

Certification standards require passing four performance tests maintaining the conditions indicated in Table A-I throughout each test. The four performance tests are 1) maximum operating conditions, 2) low temperature operation, 3) insulation efficiency, and 4) condensate disposal. In the maximum operating conditions test, tests are run at 90 and 110 percent of rated voltage continuously for a minimum of two hours at the specified temperature conditions without damage to any part of the unit. In the low-temperature operation, the unit will operate continuously or stop and start under an automatic limit device, for six hours without damage, without sufficient frost or ice formation to cut off the air flow and during defrosting, all ice or meltage must be caught and removed by the drain. The insulation efficiency test requires running continuously for four hours after the establishment of specified temperature conditions with controls, dampers, and grilles set for maximum sweating and without condensed water dripping, running or flowing off the unit casing. The condensate disposal test requires all air-source units which reject condensate to the condenser air to operate continuously for at least four hours without condensed water dripping, running, or blowing off the unit casing.

The manufacturers certified ratings are established at the standard rating conditions indicated in Table A-I. The certified ratings are given in terms of the coefficient of performance (COP) for heating (both high and low temperature heating) and the energy efficiency ratio (EER) for cooling. The coefficient of performance is a unitless ratio calculated by dividing the total heating capacity provided by the heat pump system (including the circulating fan heat but excluding the supplementary resistance heat) by the total electrical input. The energy efficiency ratio for cooling is a ratio expressed in units of Btu/h /w calculated by dividing the cooling capacity by the power input. The power input includes the power required to operate all fans or blowers furnished with the model. In the water-source units an allowance for the cooling tower fan motor and circulating water pump motor power input has been added in the amount of 10 watts per 1000 Btu/h cooling capacity.

TABLE A-I[1]

OPERATING TEMPERATURE CONDITIONS FOR STANDARD RATING AND PERFORMANCE TESTS

Test	Indoor Coil Air Entering		Outdoor Surface			
			Water-Source		Air-Source	
	DB* (°F)	WB** (°F)	In (°F)	Out (°F)	DB (°F)	WB (°F)
Standard Rating Conditions, Cooling	80	67	85	95	95[1]	---
Standard Rating Conditions, High Temperature Heating[2]	70	60 (max)	60	---	47	43
Standard Rating Conditions, Low Temperature Heating[2]	70	60 (max)	---	---	17	15
Maximum Operating Conditions, Cooling Cycle	95	71	90	100	115[1]	---
Maximum Operating Conditions, Heating Cycle[4]	80	---	75	---	75	65
Low Temperature Operation, Cooling	67	57	---	70	67[3]	---
Insulation Efficiency	80	75	---	80	80[1]	---
Condensate Disposal Conditions (air-source models only which reject condensate)	80	75	---	---	80	75

1 75 WB maintained if condensate rejected to outdoor coil air.

2 Water flow rate gpm, as determined in Standard Rating, Cooling, test.

3 57 WB maintained if condensate rejected to outdoor coil air stream.

4 Water flow rate gpm, as determined in Maximum Operating Conditions Test, Cooling Cycle.

* DB implies the air dry bulb temperature.

** WB implies the air wet bulb temperature.

Some further clarification of Table A-I may be needed. Table A-I contains the operating temperature conditions for both the standard rating and the performance tests. For example, the standard rating conditions for an air-source heat pump high-temperature heating test are to provide indoor heating at a temperature of 70°F dry bulb (at a maximum of 60°F wet bulb which limits the indoor relative humidity) from outdoor air at 47°F dry bulb and 43°F wet bulb temperatures. The water flow rate of a water-source heat pump is determined by the temperature conditions of the cooling tests, that is, the water flow in the cooling tests must be adjusted until the water outlet temperature reaches the specified outlet temperature using the specified water inlet temperature. The outdoor wet bulb temperature was not specified for the cooling and insulation efficiency tests except under conditions where condensate was rejected to the outdoor coil air and then the wet bulb temperature was set at 75°F.

The ARI has tested and certified different types of heat pumps including both air- and water-source heat pumps and both single- and split-system heat pumps. The published ARI data contains the heating and cooling capacities, the coefficients of performance, and the energy efficiency ratios for those heat pumps tested.

III. HEAT PUMP PERFORMANCE COMPARISON

A comparison of heat pump performance characterisitics has been made on the basis of their coefficients of performance and energy efficiency ratios; that is, the COPs and EERs were plotted as a function of their respective heating and cooling capacities. Each manufacturer was given a different symbol so that one manufacturer can be compared with another. The ARI data used (Reference A-1) in this comparison contains data for both single-package, and split air-source systems, and for single-package water-source systems. There were no split water-source system data in Reference 1.

The performance comparisons for single-package air-source heat pump systems are shown in Figures A-1, A-2, and A-3 for the standard rating high and low temperature heating tests and the standard rating cooling tests, respectively. Similarly, the comparisons for a split air-source system are shown in Figures A-4, A-5, and A-6, respectively. Figures A-1 through A-6 each appear in four parts a, b, c, and d; that is, the quantity of data plotted required four separate plots in order to maintain clarity and so that the data for each manufacturer could be designated by a different symbol. Figures A-7 and A-8 show the heating and cooling comparisons, respectively, for the water-source (single-package) heat pump systems. There is considerably more air-source than water-source heat pump data.

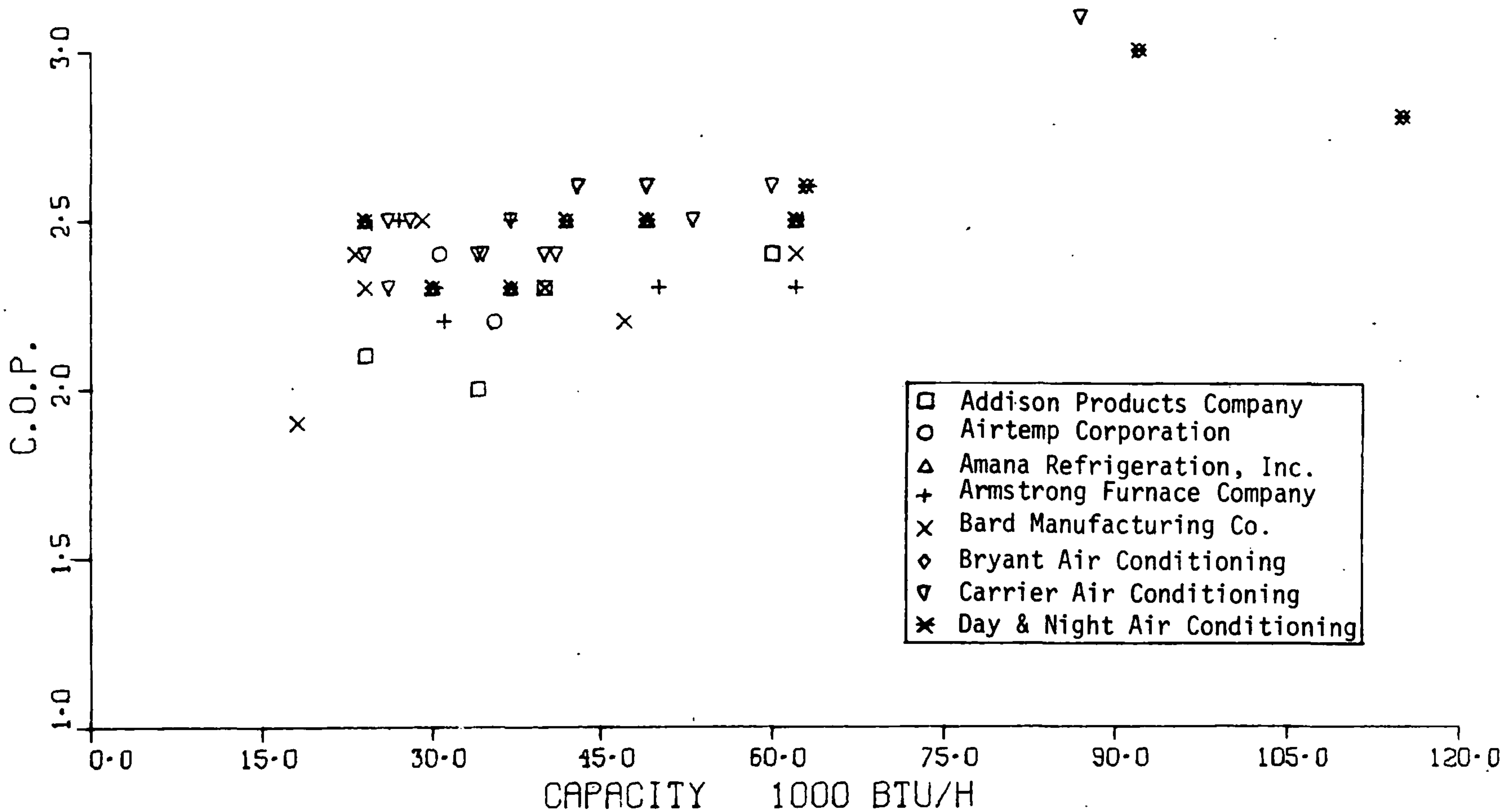

Figure A-1a C.O.P. vs capacity for single package air-to-air heat pumps heating from a 47°F air source.

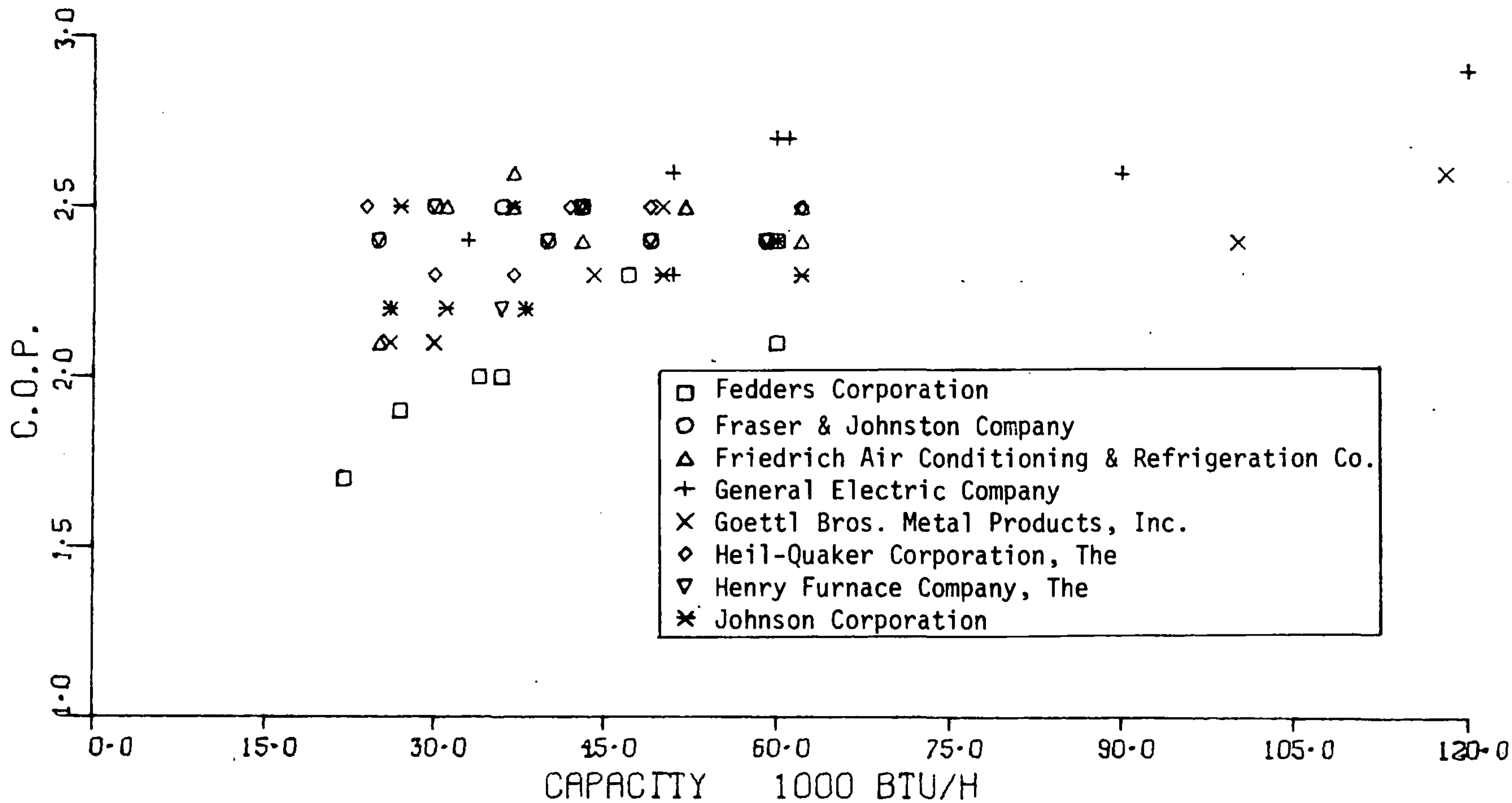

Figure A-1b C.O.P. vs capacity for a single package air-to-air heat pumps heating from a 47°F air source.

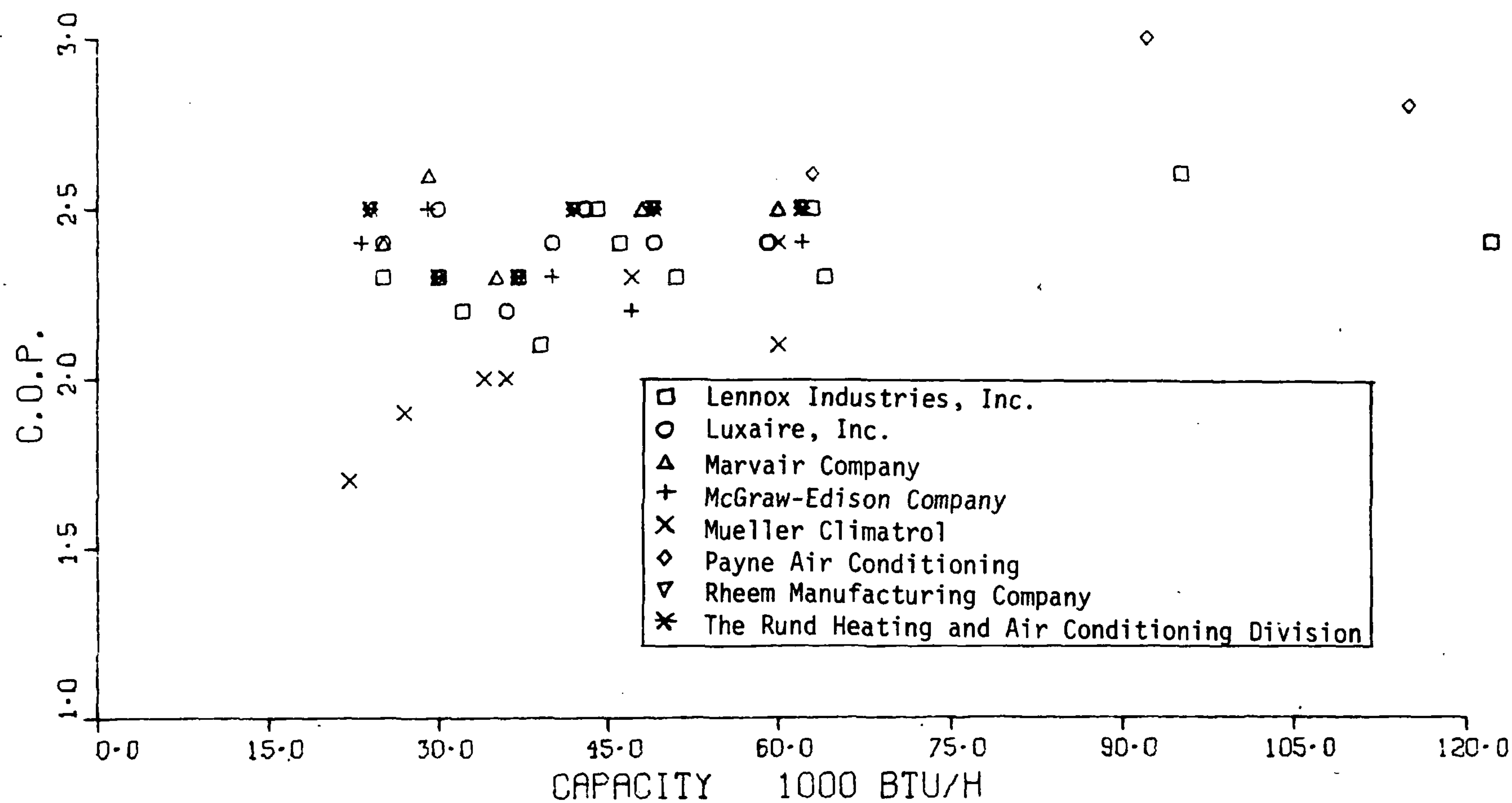

Figure A-1c C.O.P. vs capacity for single package air-to-air heat pumps heating from a 47°F air source.

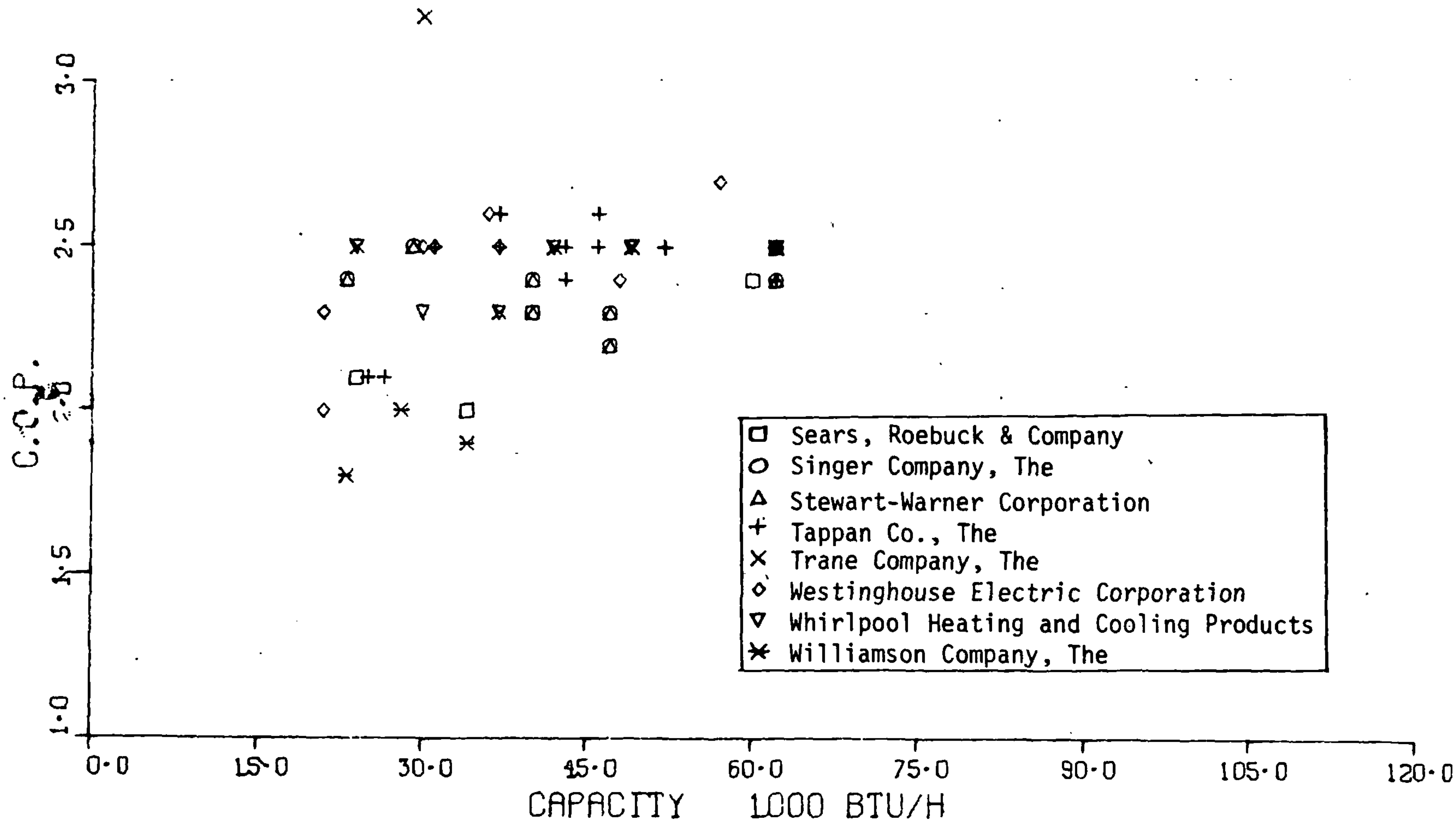

Figure A-1d C.O.P. vs capacity for single package air-to-air heat pumps heating from a 47°F air source.

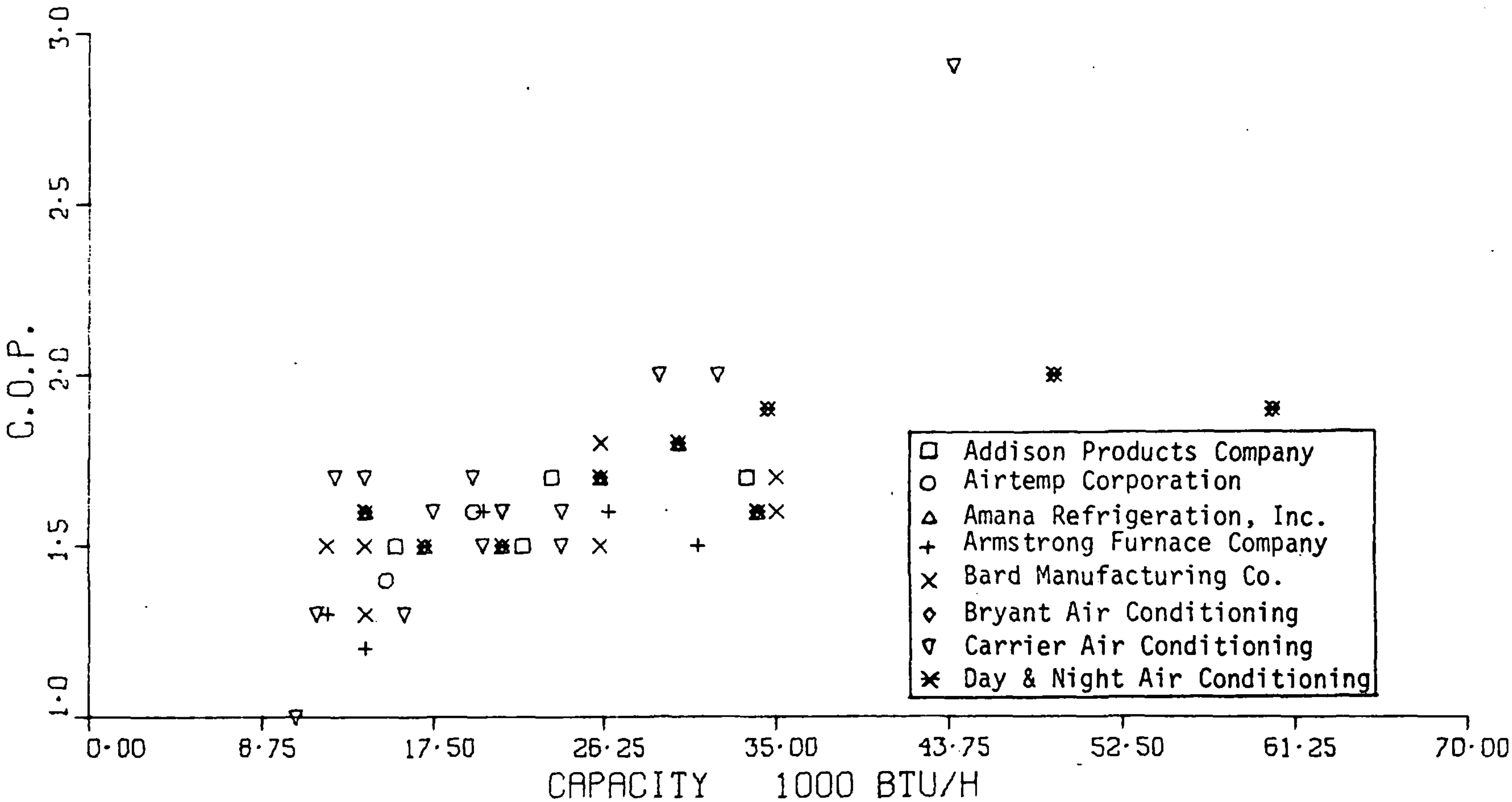

Figure A-2a C.O.P. vs capacity for a single package air-to-air heat pumps heating from a 17°F air source.

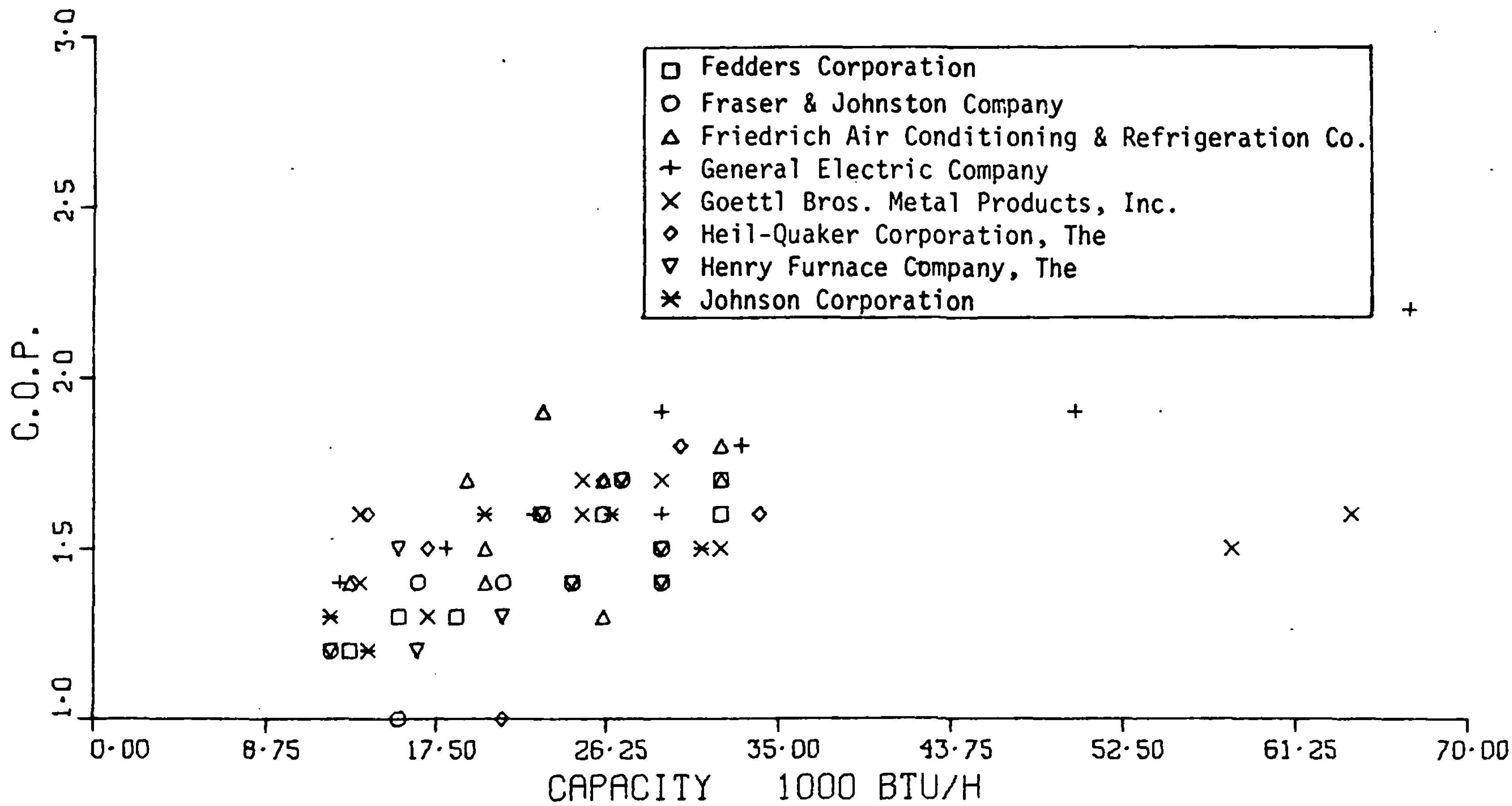

Figure A-2b C.O.P. vs capacity for a single package air-to-air heat pumps heating from a 17°F air source.

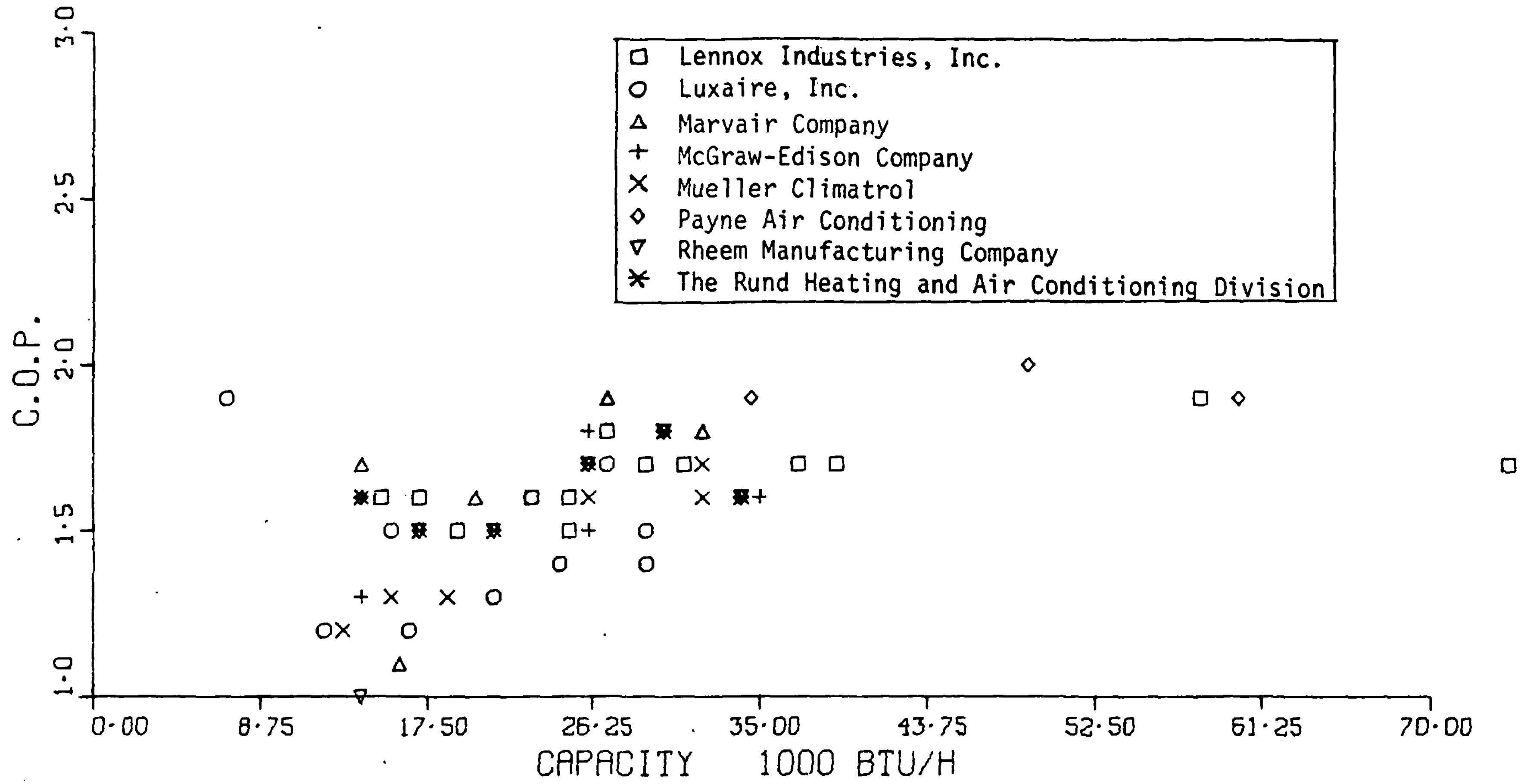

Figure A-2c C.O.P. vs capacity for single package air-to-air heat pumps heating from a 17°F air source.

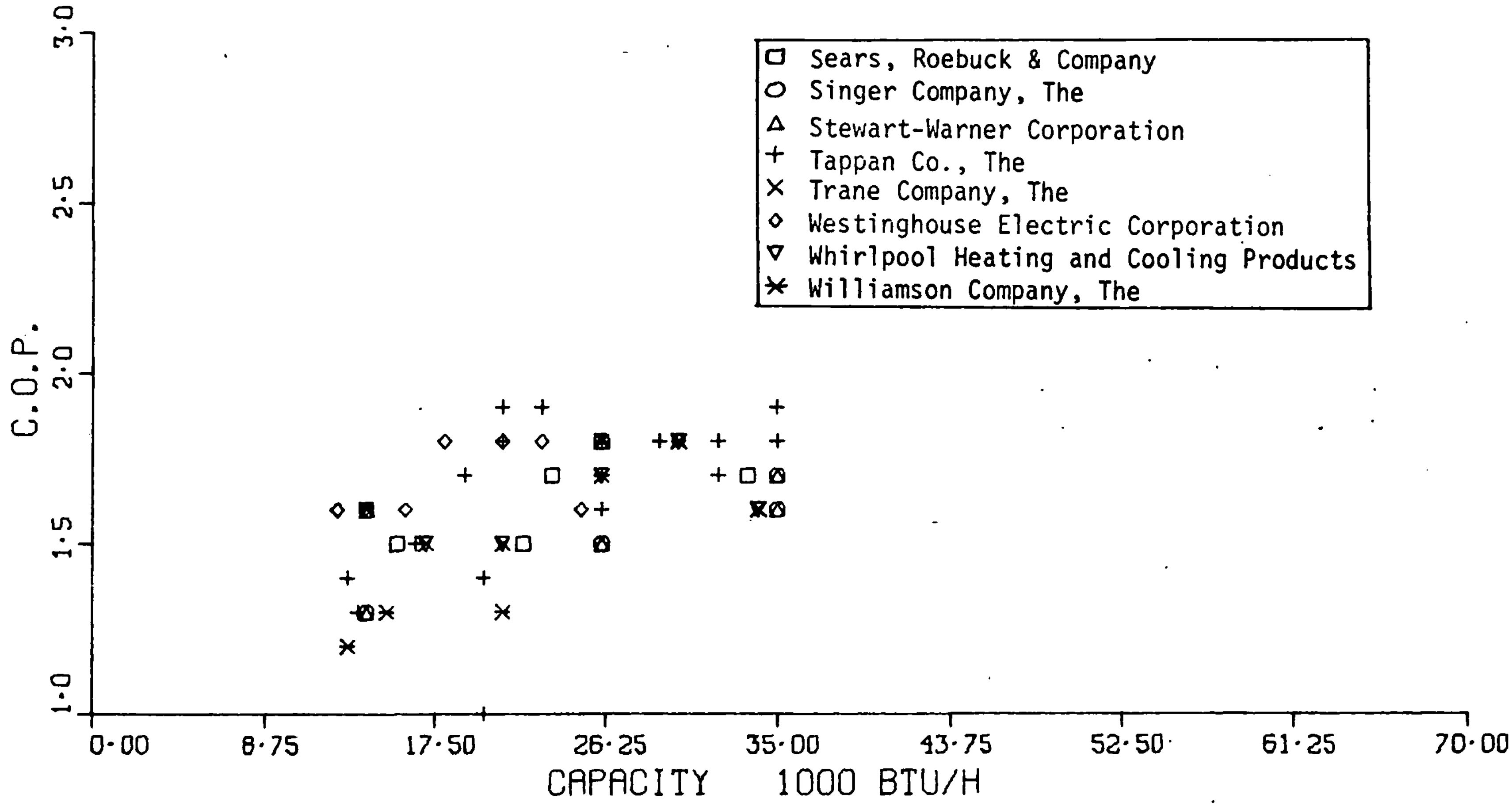

Figure A-2d C.O.P. vs capacity for single package air-to-air heat pumps heating from a 17°F air source.

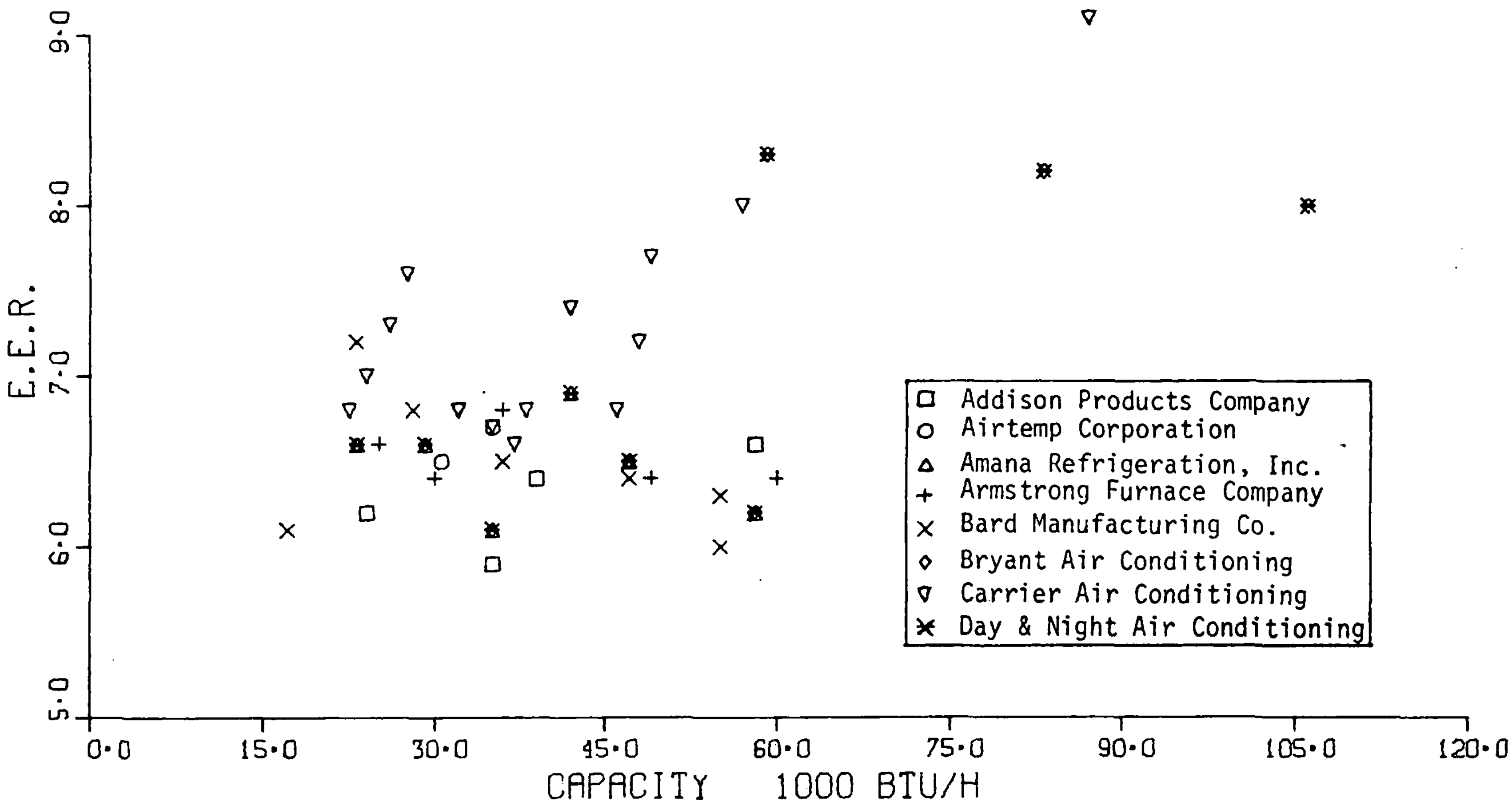

Figure A-3a E.E.R. vs capacity for single package air-to-air heat pumps cooling with a 95°F outdoor air temperature.

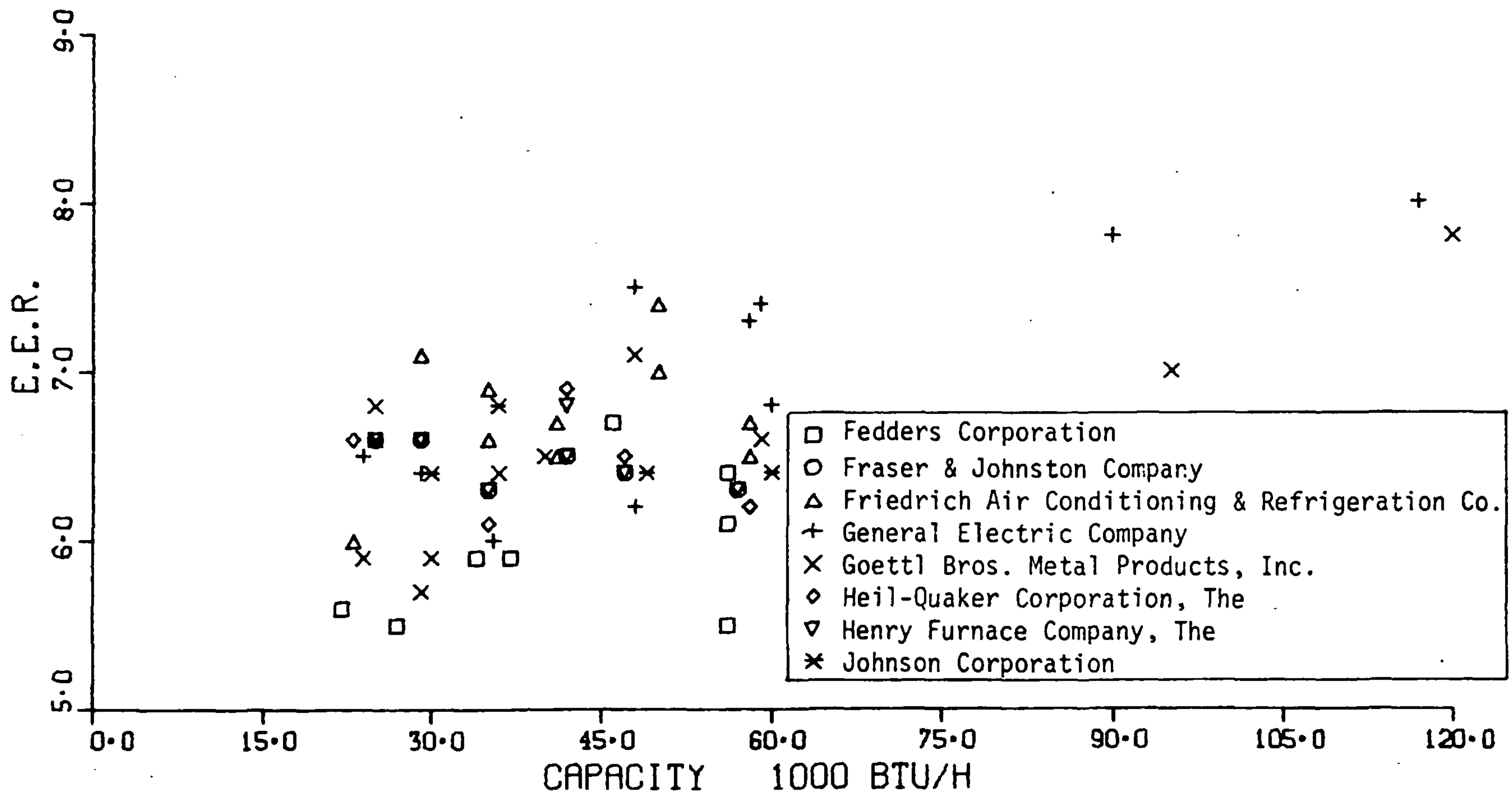

Figure A-3b E.E.R. vs capacity for single package air-to-air heat pumps cooling with a 95°F outdoor air temperature.

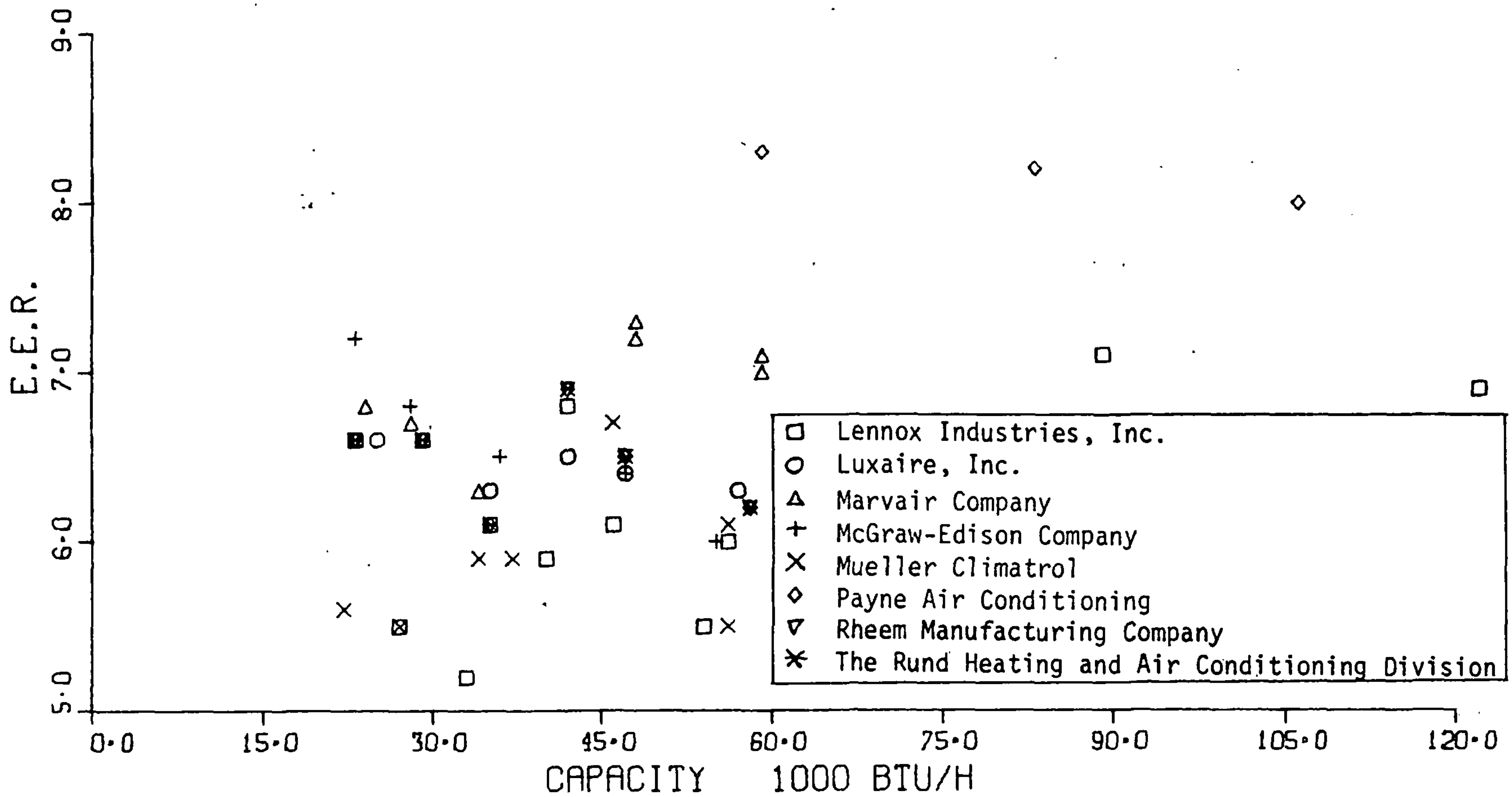

Figure A-3c E.E.R. vs capacity for single package air-to-air heat pumps cooling with a 95°F outdoor air temperature.

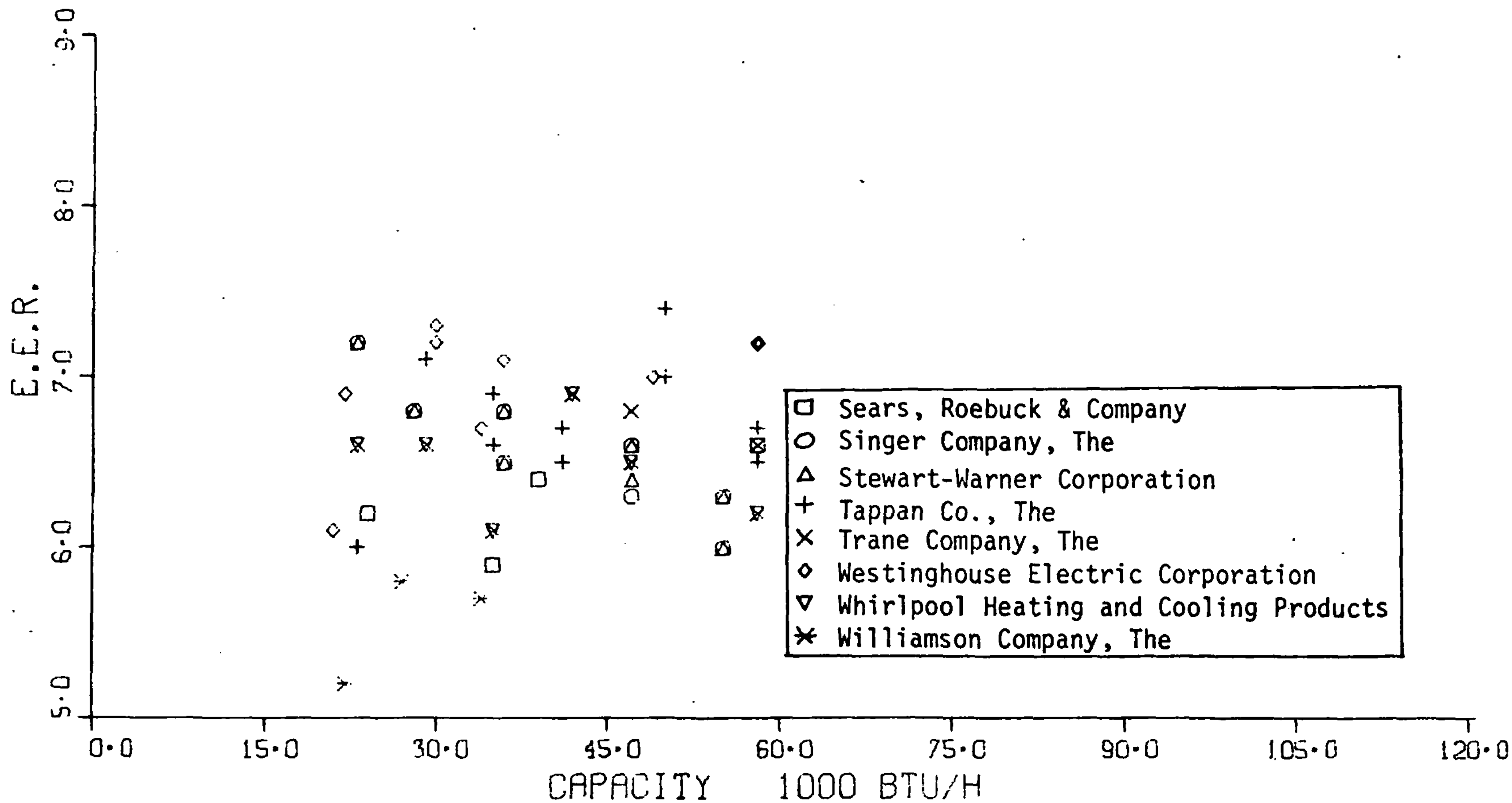

Figure A-3d E.E.R. vs capacity for single package air-to-air heat pumps cooling with a 95°F outdoor air temperature.

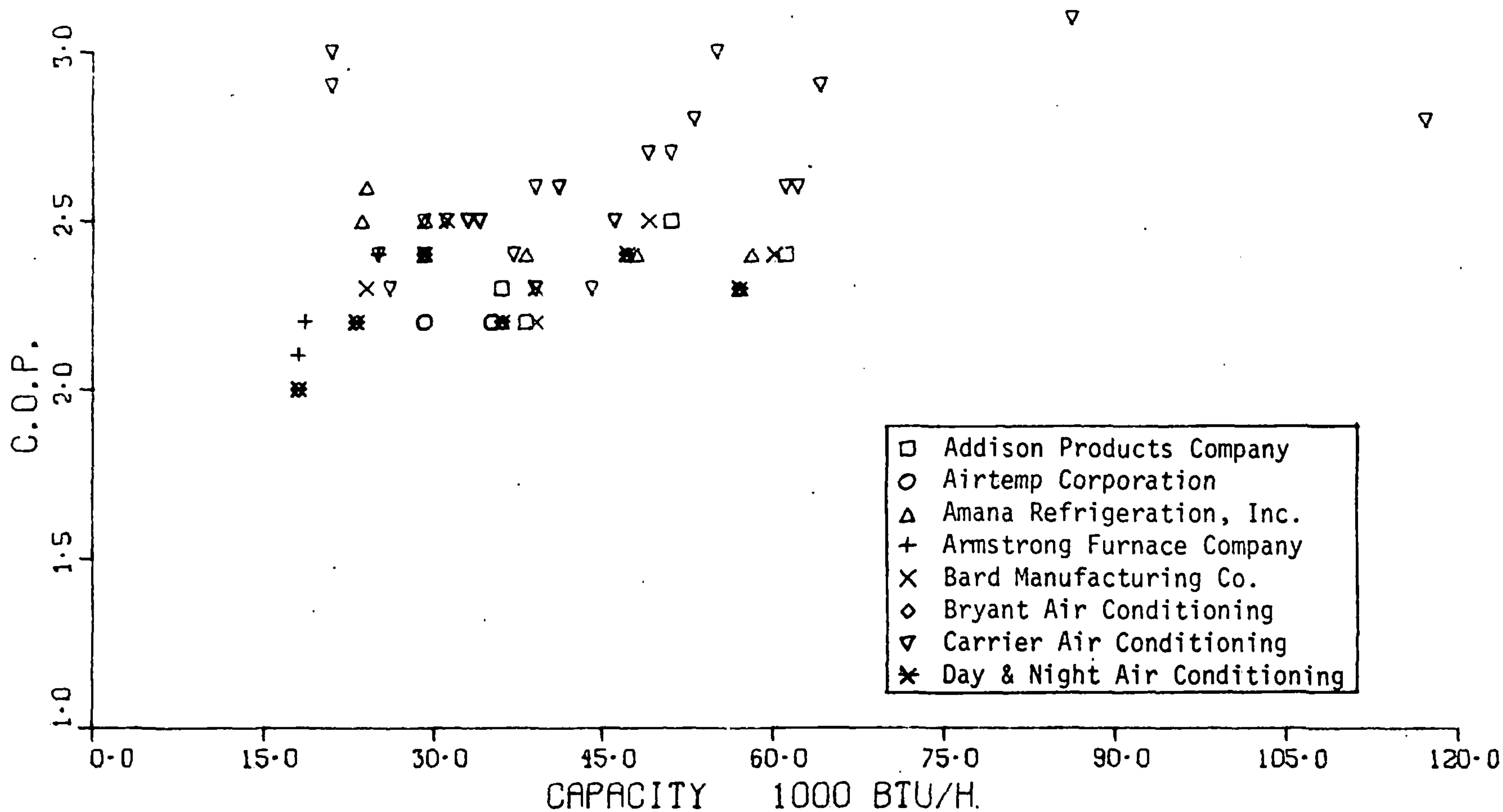

Figure A-4a C.O.P. vs capacity for split system air-to-air heat pumps heating from a 47°F air source.

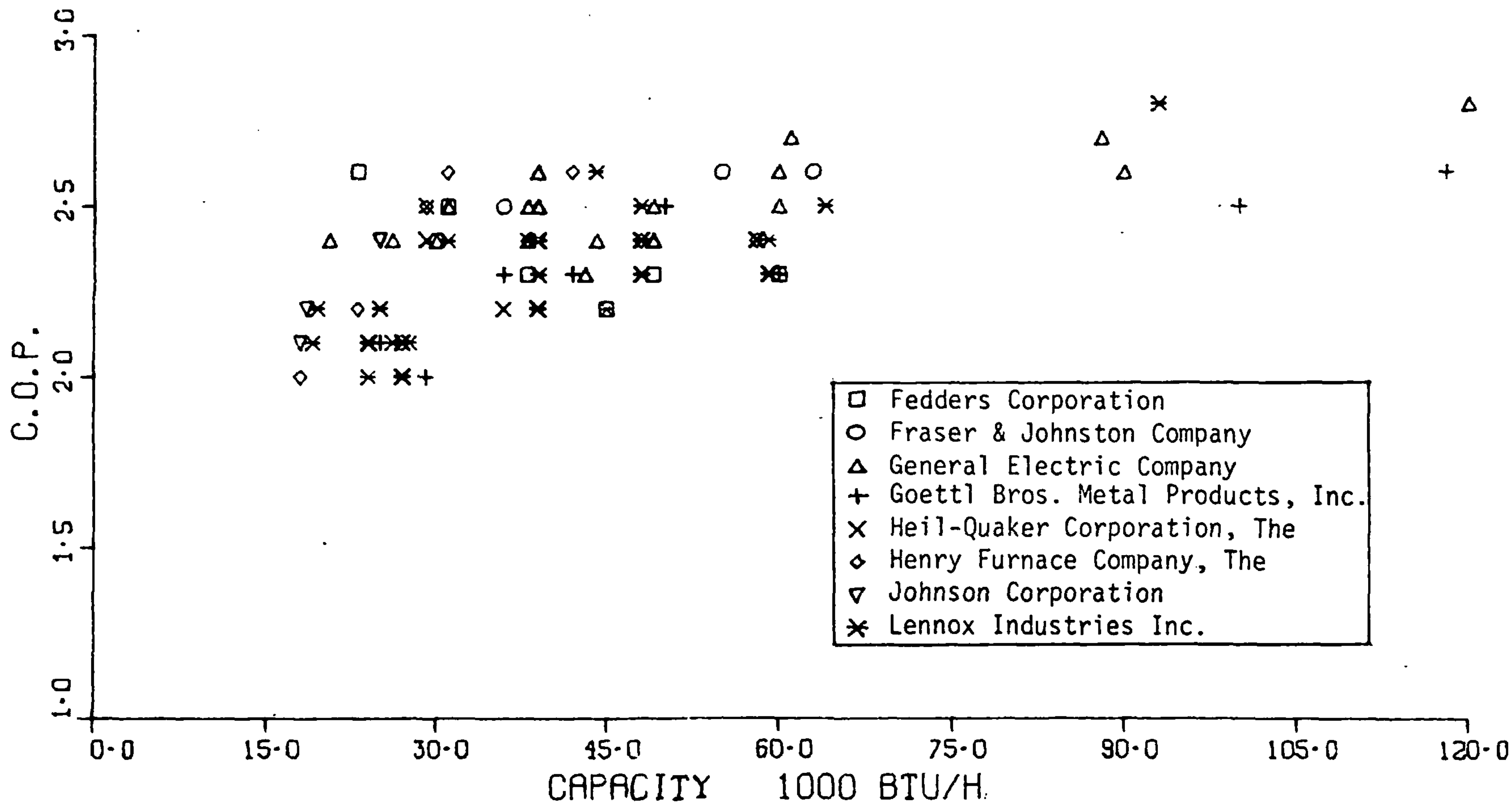

Figure A-4b C.O.P. vs capacity for split system air-to-air heat pumps heating from a 47°F air source.

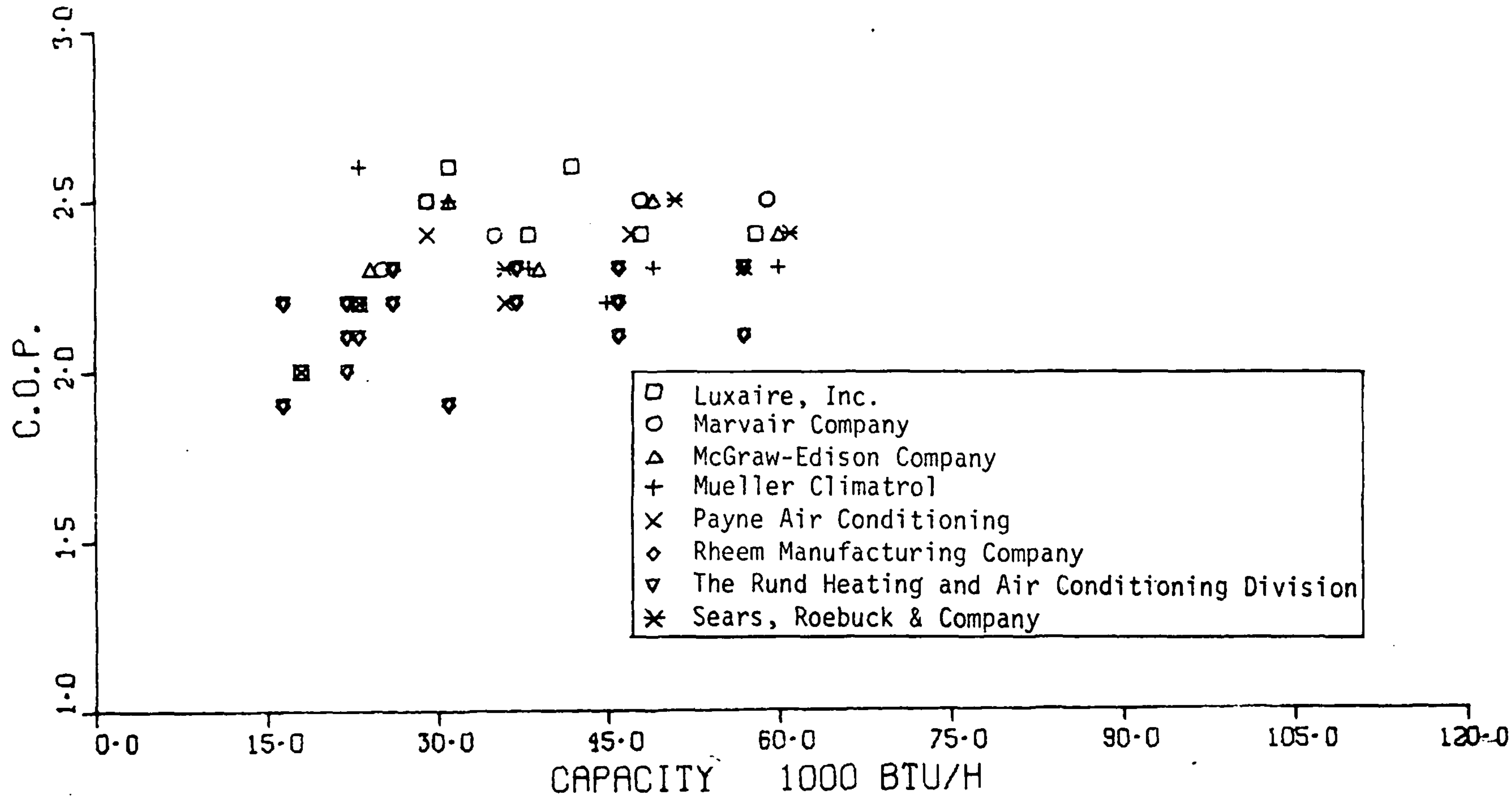

Figure A-4c C.O.P. vs capacity for split system air-to-air heat pumps heating from a 47°F air source.

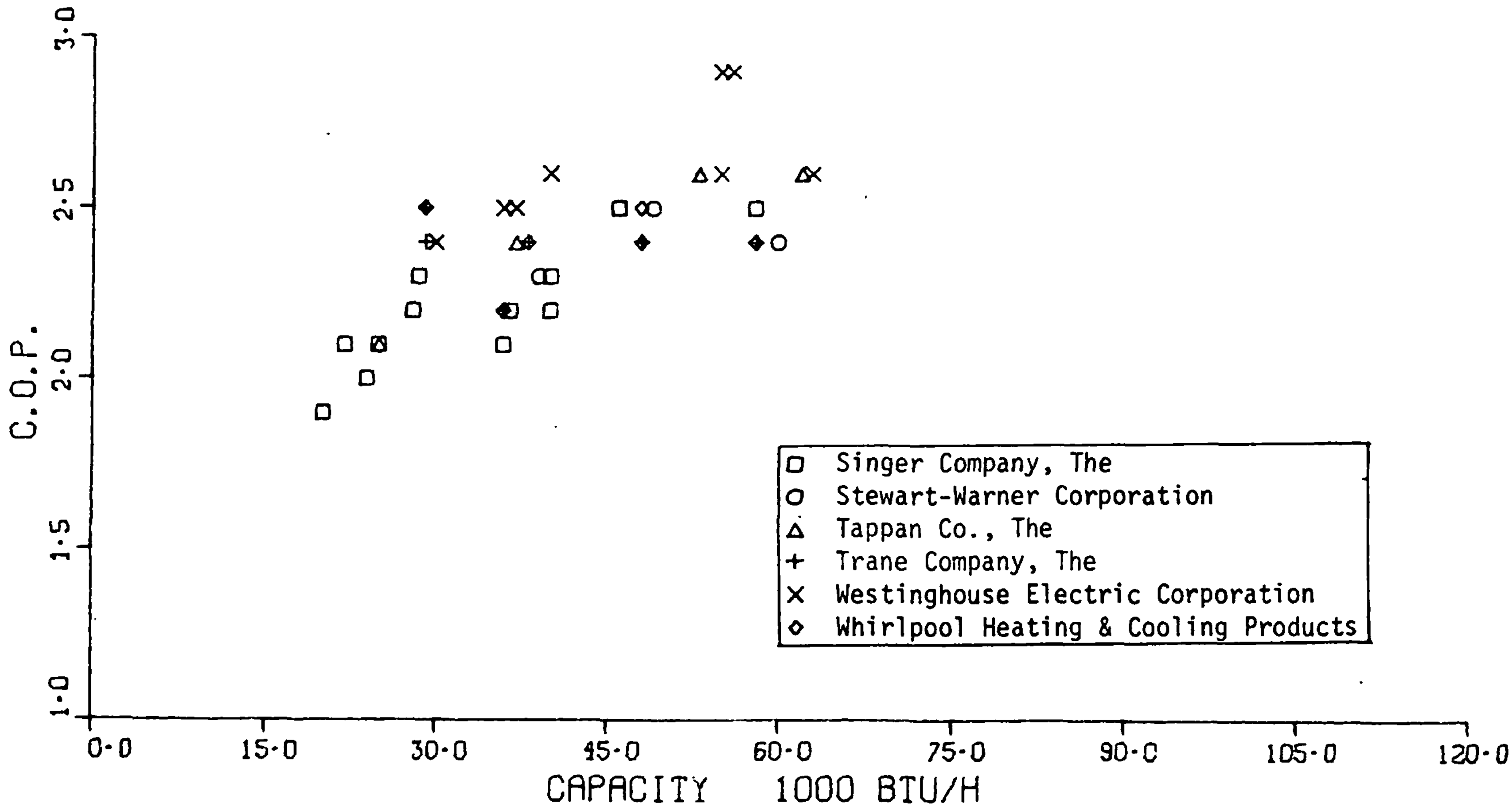

Figure A-4d C.O.P. vs capacity for split system air-to-air heat pumps heating from a 47°F air source.

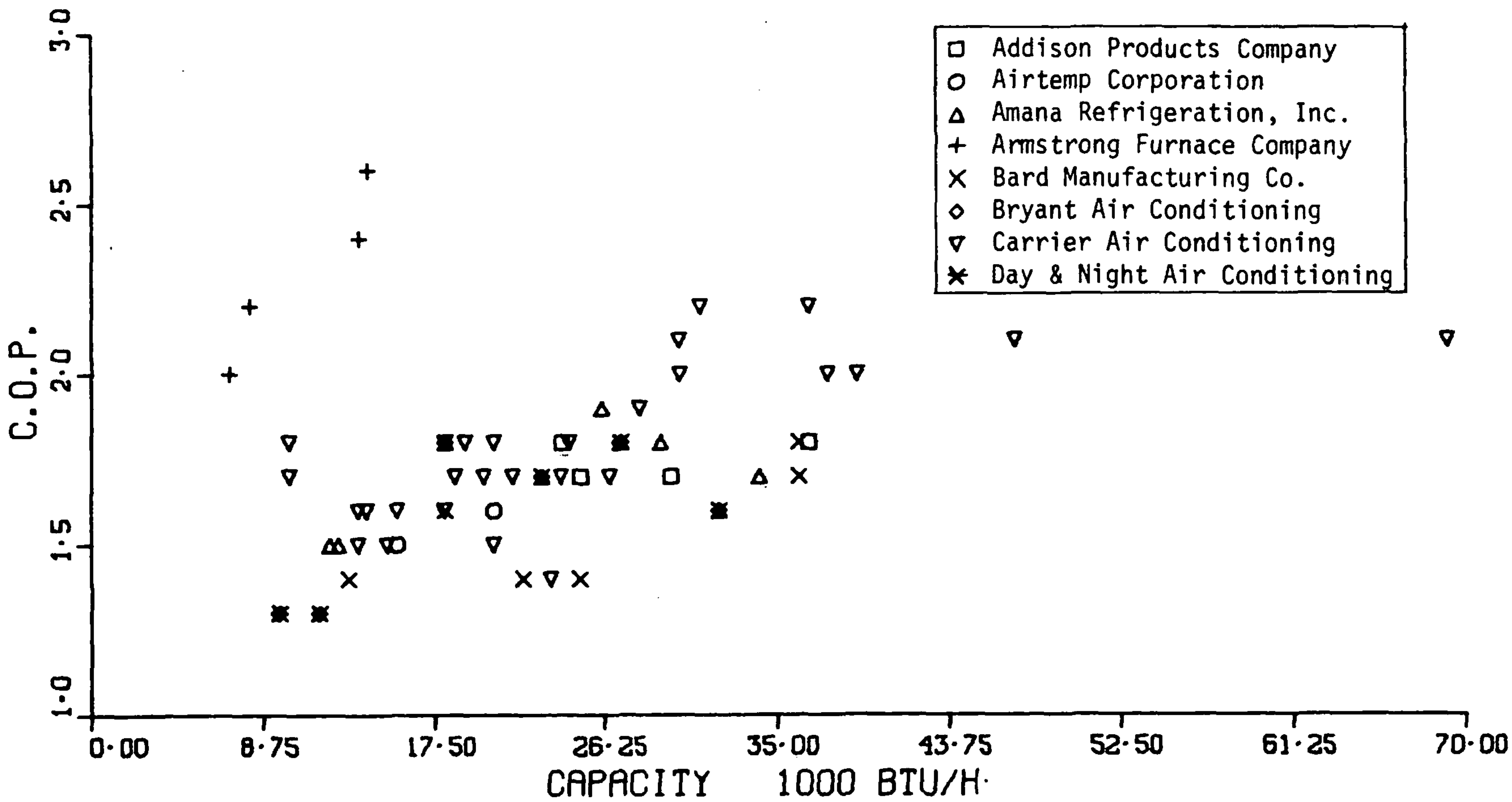

Figure A-5a C.O.P. vs capacity for split system air-to-air heat pumps heating from a 17°F air source.

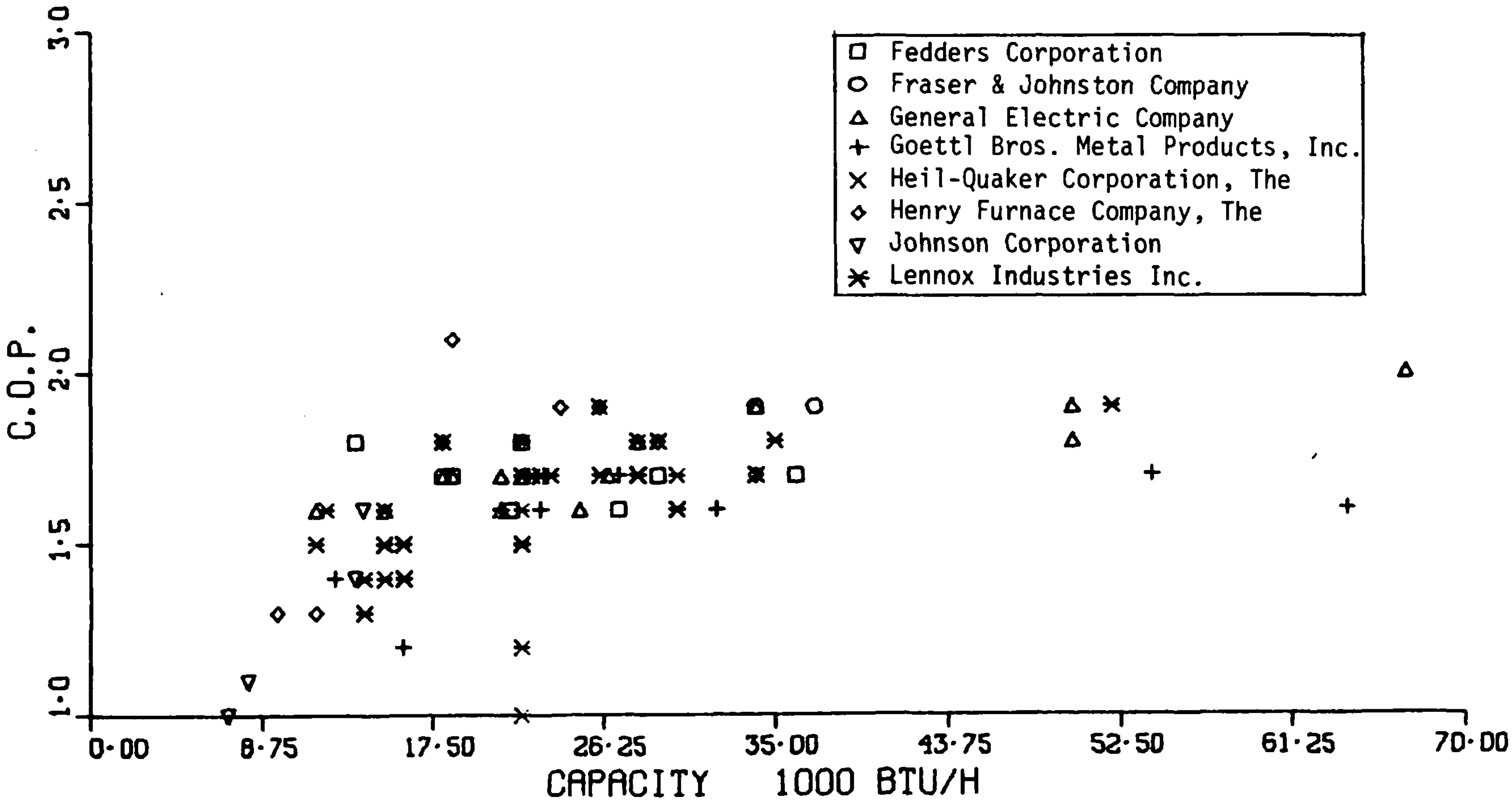

Figure A-5b C.O.P. vs capacity for split system air-to-air heat pumps heating from a 17°F air source.

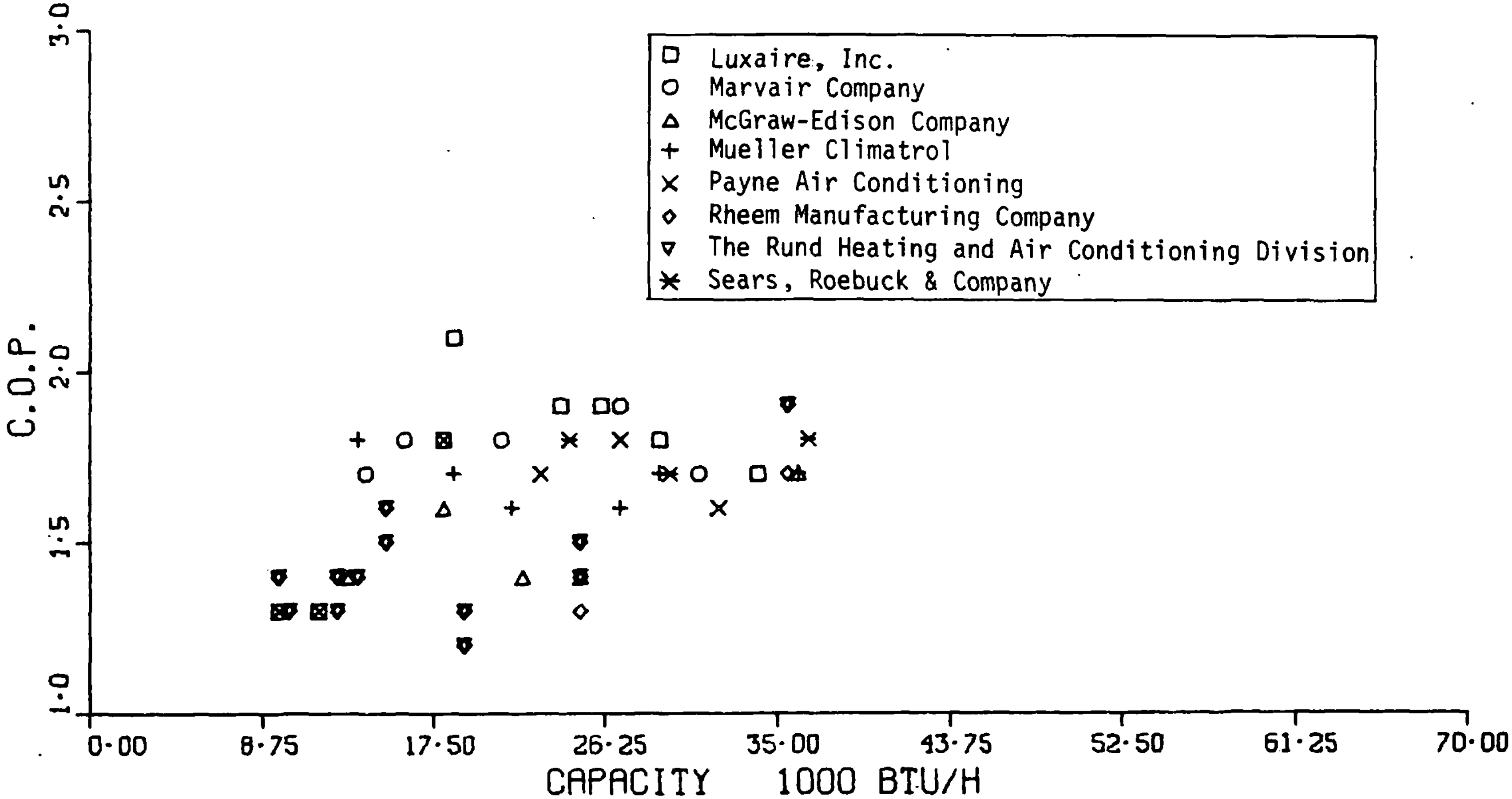

Figure A-5c C.O.P. vs capacity for split system air-to-air heat pumps heating from a 17°F air source.

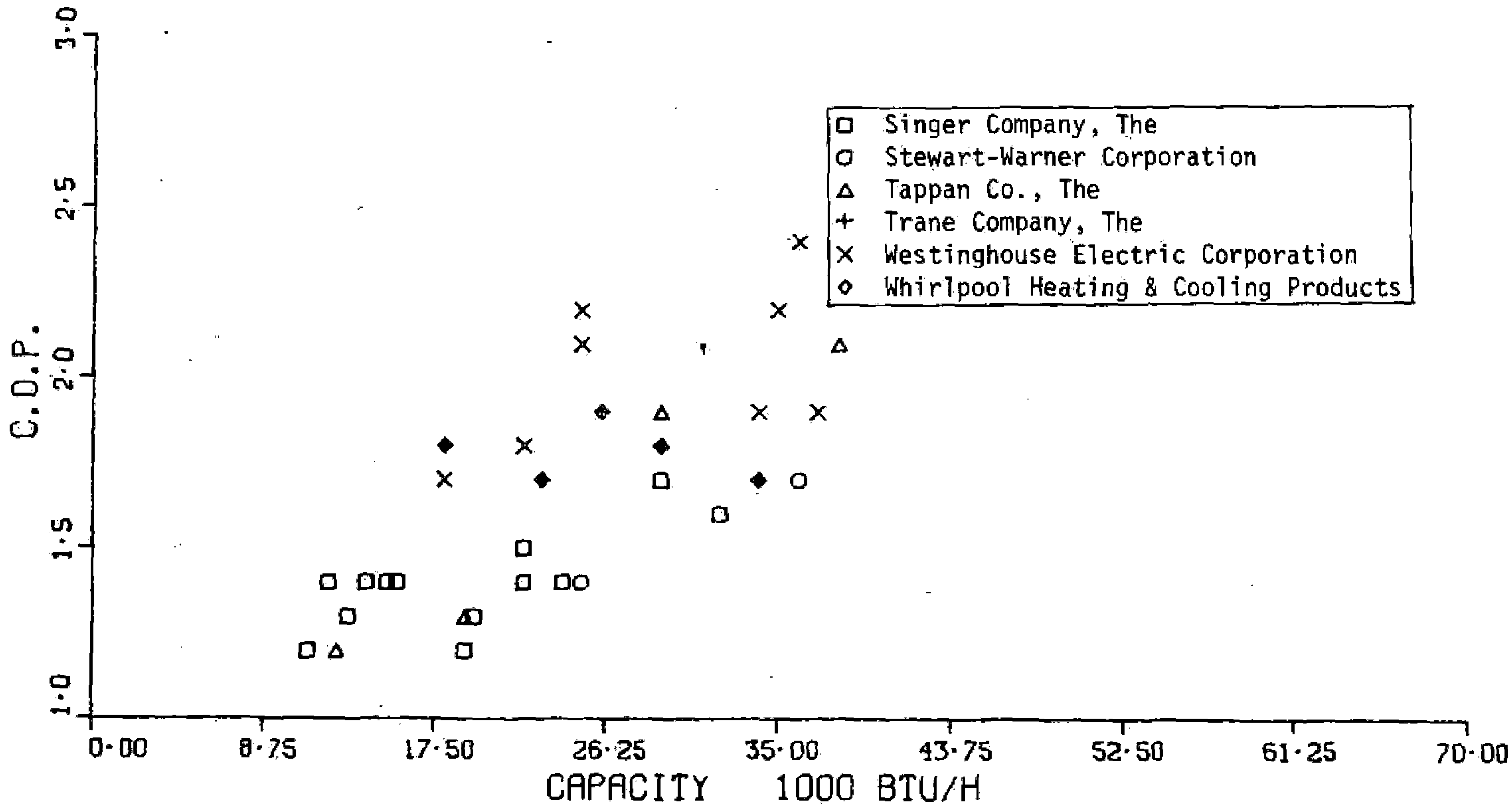

Figure A-5d C.O.P. vs capacity for split system air-to-air heat pumps heating from a 17°F air source.

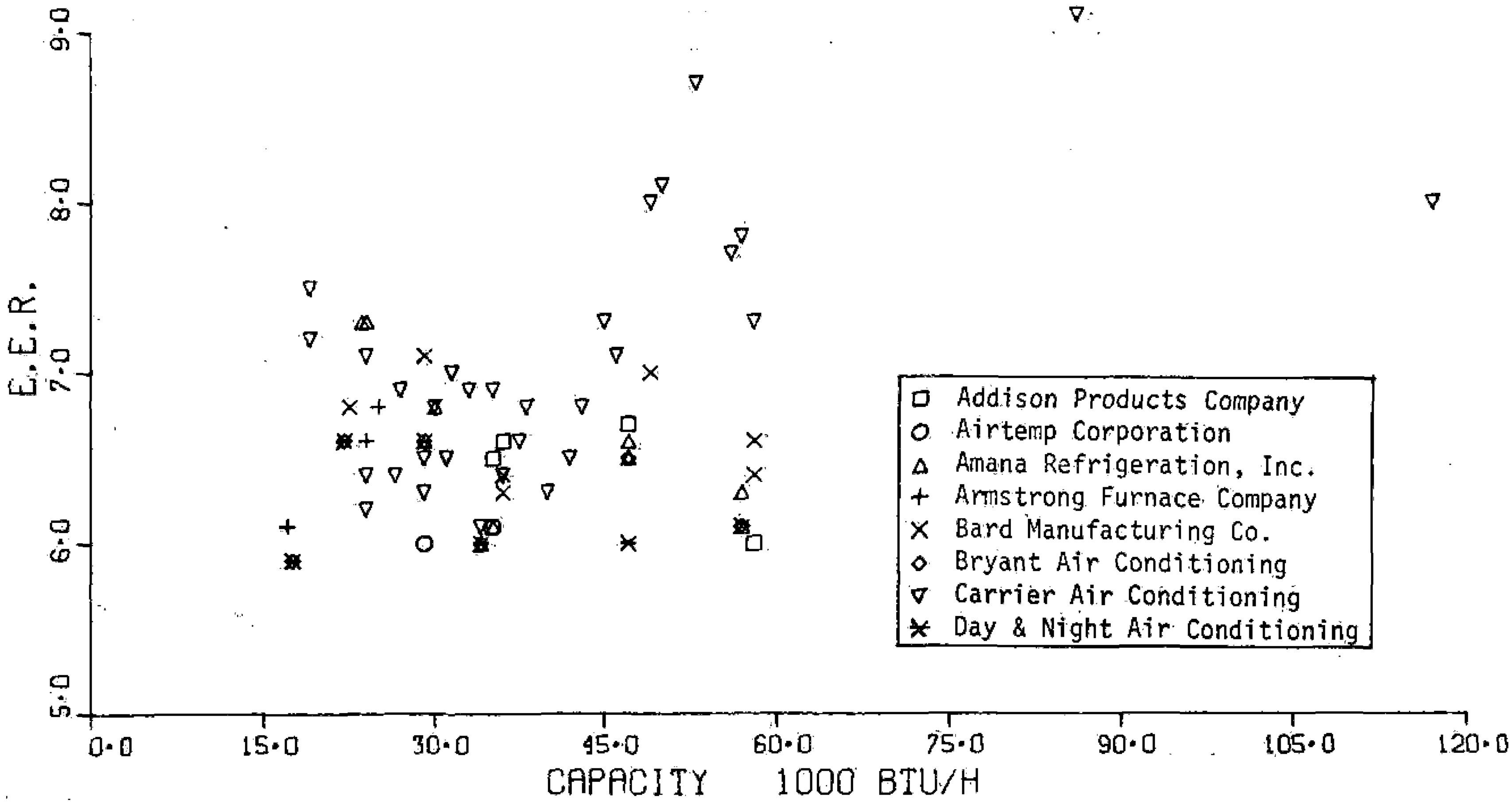

Figure A-6a E.E.R. vs capacity for split system air-to-air heat pumps cooling with a 95°F outdoor air temperature.

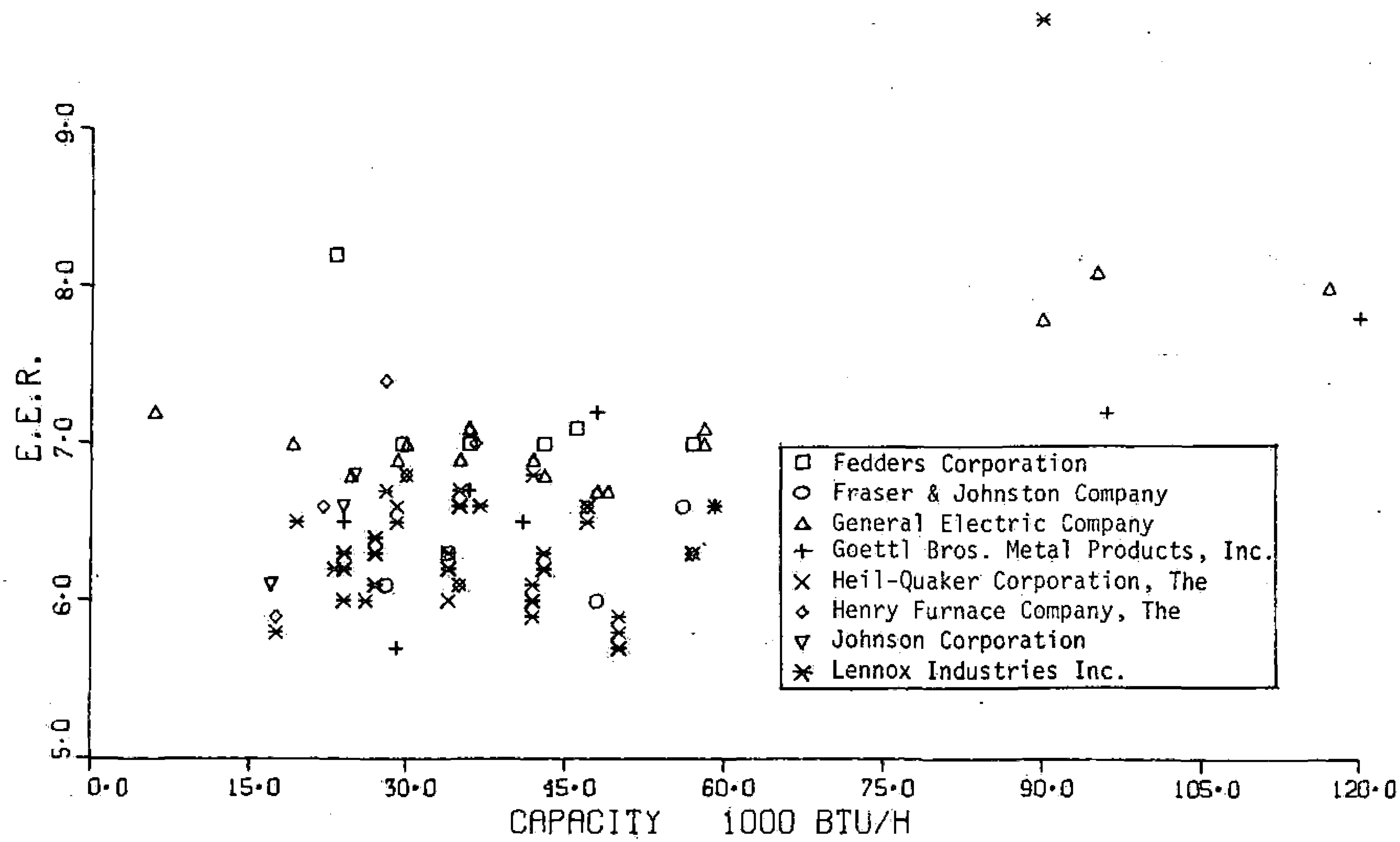

Figure A-6b E.E.R. vs capacity for split system air-to-air heat pumps cooling with a 95°F outdoor air temperature.

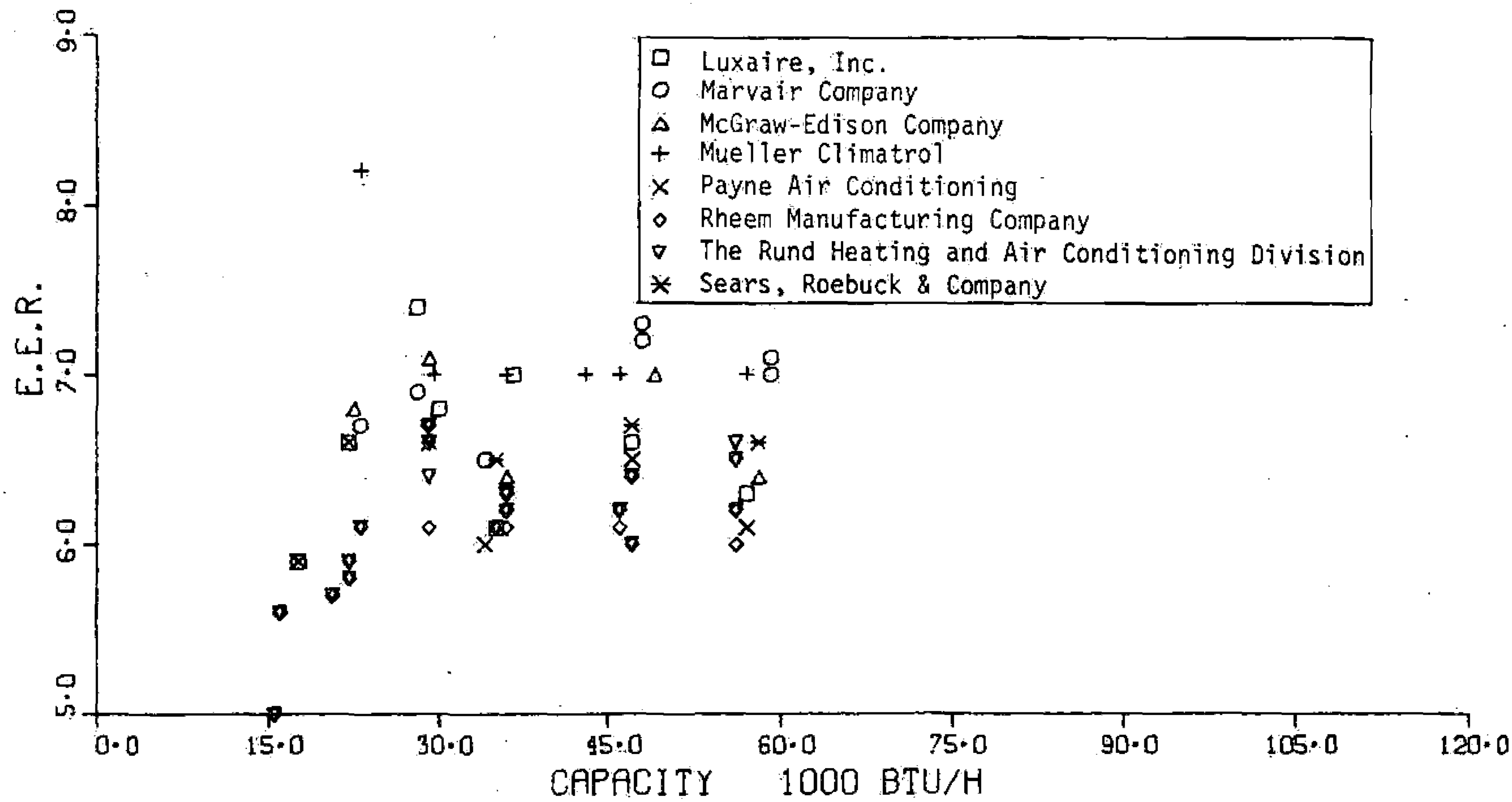

Figure A-6c E.E.R. vs capacity for split system air-to-air heat pumps cooling with a 95°F outdoor air temperature.

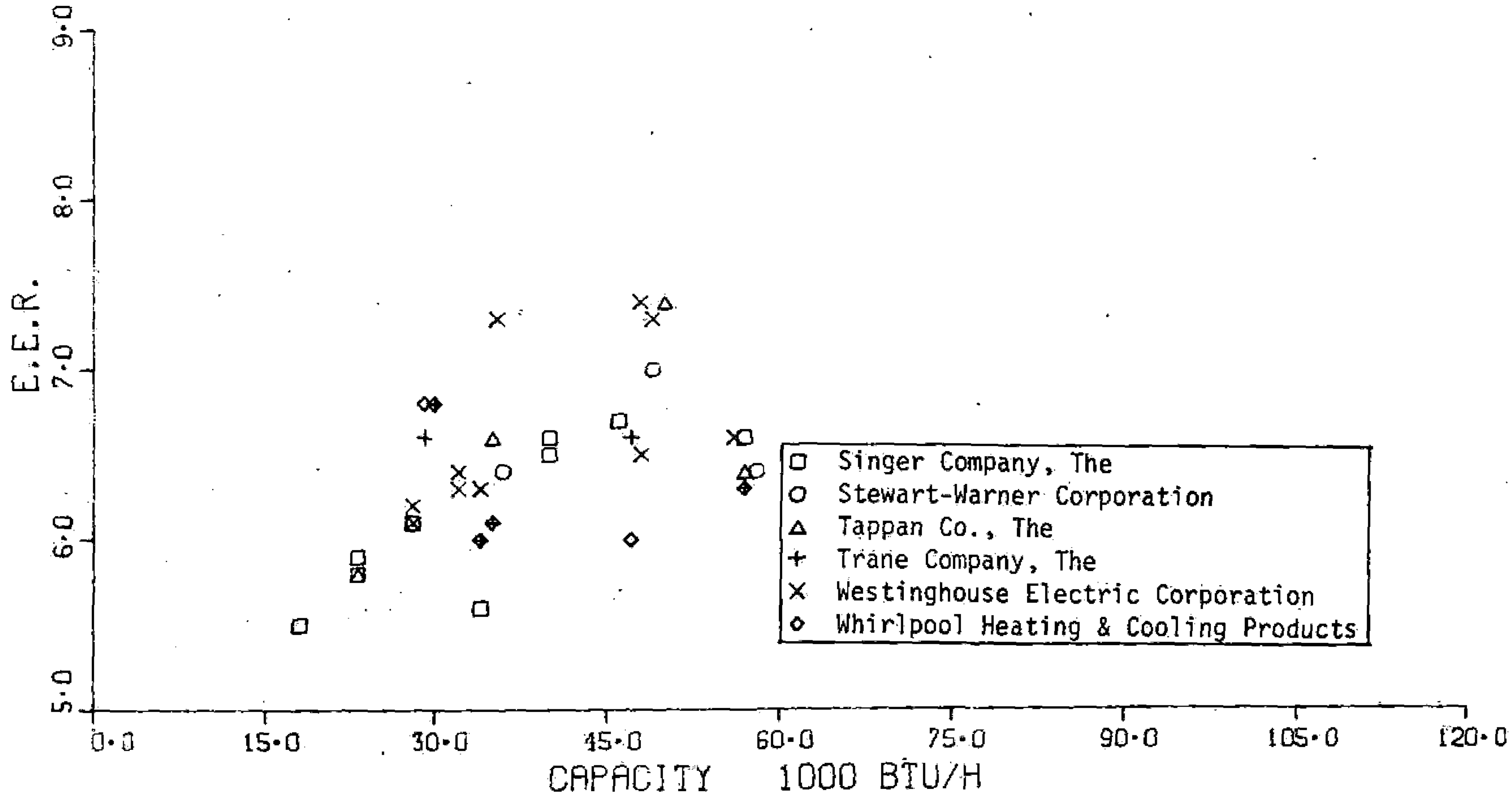

Figure A-6d E.E.R. vs capacity for split system air-to-air heat pumps cooling with a 95°F outdoor air temperature.

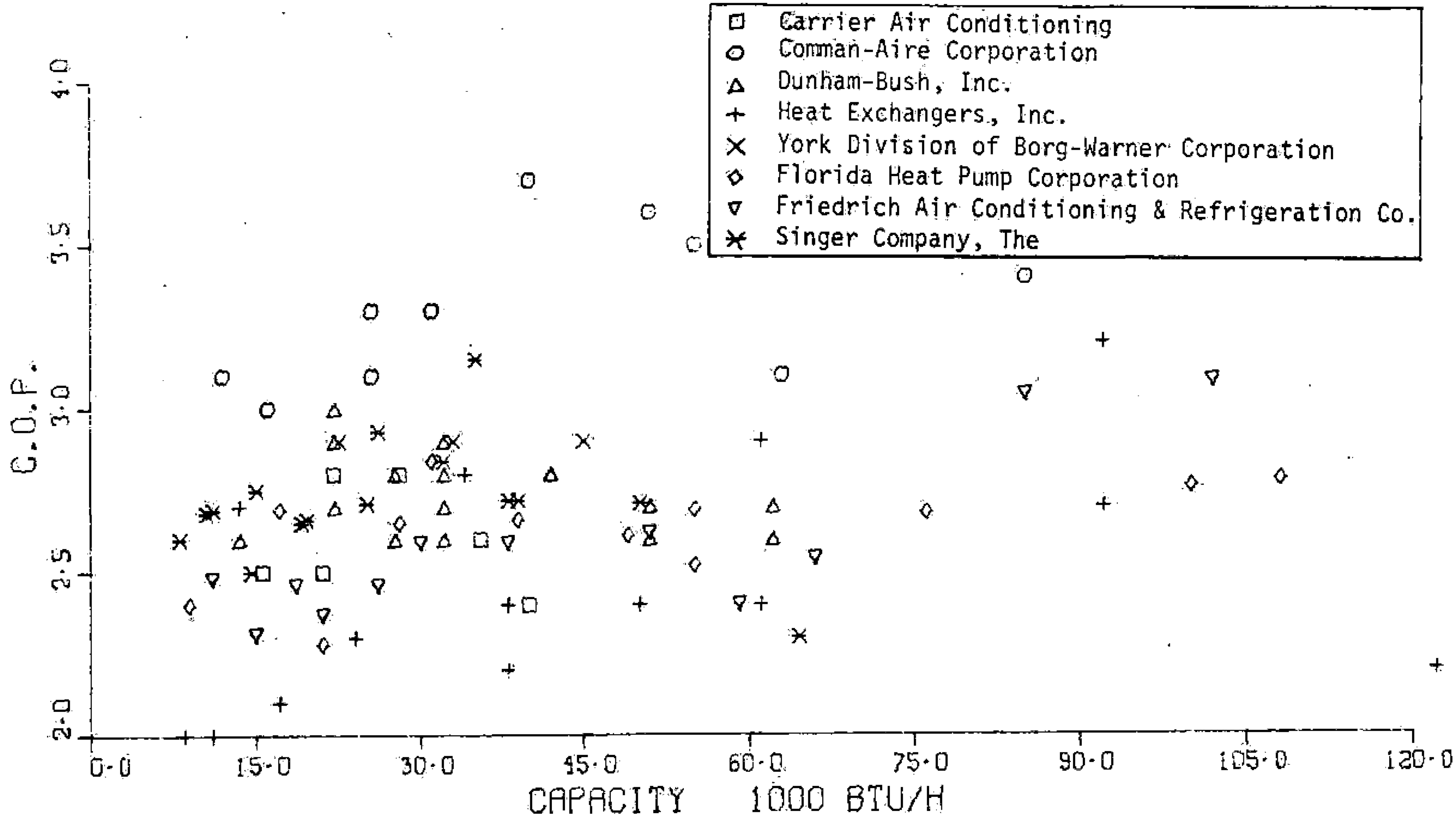

Figure A-7. C.O.P. vs capacity for water-to-air heat pumps heating from a 60°F water source.

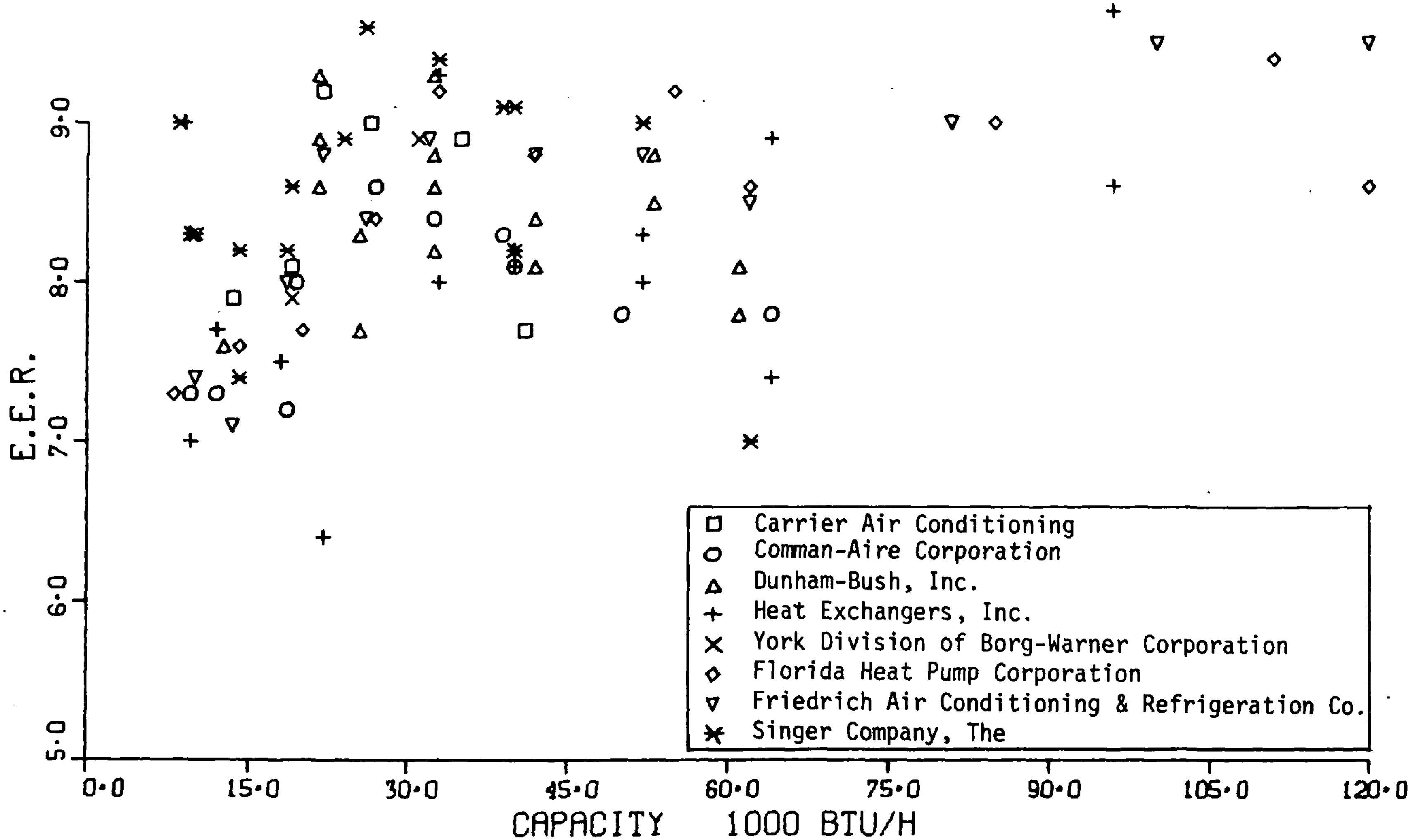

Figure A-8. E.E.R. vs capacity for water-to-air heat pumps cooling with an 85°F water temperature.

APPENDIX B

HTPUMP INPUT DESCRIPTION

HTPUMP INPUT

Card 1: Format 4I2

Cols	LEAP	=	0 Not Leap Year 1 Leap Year
Cols	IYR	=	Year of Data Tape
Cols	NOMO	=	0 All months 1-11 number of months wanted
Cols	Months	=	The number of months (1=Jan., 2=Feb.,.....)

Card 2: Format 9I5

Cols 1-5	IFIX(6)	=	# of heating data pairs
Cols 6-10	IFIX(7)	=	# of cooling data pairs
Cols 11-15	IFIX(9)	=	Heat if month $\leq$ IFIX(9)
			OR
Cols 16-20	IFIX(10)	=	Heat if month $\geq$ IFIX(10)
Cols 21-25	IFIX(11)	=	Cool if month $\geq$ IFIX(11)
			AND
Cols 26-30	IFIX(12)	=	Cool if month $\leq$ IFIX(12)
Cols 31-35	IFIX(13)	=	# of hours of printout desired
Cols 36-40	IFIX(14)	=	0 Air-to-air 1 Solar-water-air 2 Water-to-air
Cols 41-45	IFIX(15)	=	0 No solar calculation 1 Solar augmentation (Heating only)

Card 3: 7 F 10 4

Cols 1-10	B1(1)	=	Desired indoor heating temperature (°F)
Cols 11-20	B1(19)	=	Desired indoor cooling temperature (°F)
Cols 21-30	B1(4)	=	Constant heat-loss factor (Btu/h -°F)

Cols 31-40	B1(7)	=	Constant internal heat generation (Btu/h)
Cols 41-50	B1(3)	=	MC_p for house (Btu/°F)
Cols 51-60	B1(2)	=	Specific heat for air (Btu/lbm-°F)
Cols 61-70	B1(5)	=	Electrical heating efficiency

<u>Card 4: 7 F 10.4</u>

Cols 1-10	B1(9)	=	Upper temperature limit for defrost cycle operation (°F)
Cols 11-20	B1(8)	=	Mass flow rate of air through outdoor unit (lbm/h)
Cols 21-30	B1(11)	=	Surface area for frost formation ($in.^2$)
Cols 31-40	B(10)	=	Frost density (lbm/ft^3)
Cols 41-50	B1(12)	=	Frost thickness required to engage defrost cycle (in.)
Cols 51-60	B1(13)	=	Defrost time per cycle (h)
Cols 61-70	B1(22)	=	Cooling capacity during defrost cycle (1000 Btu/h)

<u>Card 5: 8 F 10.4</u>

Cols 1-10	B1(23)	=	Cooling power required during defrost cycle (kW)
Cols 11-20	B1(15)	=	Solar collector area (ft^2)
Cols 21-30	B1(17)	=	Solar collector efficiency
Cols 31-40	B1(14)	=	Solar-water-air initial storage water temperature (°F)
Cols 41-50	B1(18)	=	Solar-water-air lower limit for storage water temperature (°F)
Cols 51-60	B1(16)	=	MC_p for storage water (Btu/°F)

Cols	61-70	B1(24)	=	Solar-water-air upper limit for storage water temperature (°F)
Cols	71-80	B1(25)	=	Water source temperature for water-to-air heat pumps (°F)

Cards 6 (must be at least two): 2 F 15.1

Cols	1-15	T1(I)	=	outdoor dry-bulb temperature for heating
Cols	16-30	C1(I)	=	Corresponding heat pump heating capacity

Cards 7 (must be at least two): 2 F 15.1

Cols	1-15	T1(I)	=	Outdoor dry-bulb temperature for heating
Cols	16-30	P1(I)	=	Corresponding heat pump heating power requirement

Cards 8 (must be at least two): 2 F 15.1

Cols	1-15	T2(I)	=	Outdoor dry-bulb temperature for cooling
Cols	16-30	C2(I)	=	Corresponding heat pump cooling capacity

Cards 9 (must be at least two): 2 F 15.1

Cols	1-15	T2(I)	=	Outdoor dry-bulb temperature for cooling
Cols	15-30	P2(I)	=	Corresponding heat pump cooling power requirement.

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