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R. L. D. Cane*
S. B. Clemes*
A. Morrison*
P. J. Hughes

*Caneta Research , Inc.
Mississauga, Ontario, Canada

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R. L. Douglas Cane, S. Blair Clemes, and Andrew Morrison
Caneta Research, Inc.
Mississauga, Ontario
Canada

Patrick J. Hughes
Oak Ridge National Laboratory
Oak Ridge, Tennessee

ABSTRACT

A building energy simulation program has been used in conjunction with a ground heat exchanger sizing algorithm to develop general guidelines on how to size vertical ground heat exchangers for closed-loop ground-source heat pump systems in large buildings. The analysis considered three commercial building types of varying size with different internal loads and heat pump efficiencies. Each building variation was simulated in seven cities, three in the United States and four in Canada. The ground heat exchanger sizing algorithm has been previously validated against actual system data.

The analysis results showed a strong correlation between heat exchanger length required and annual energy rejected to the ground, if the building was cooling-dominated, or annual energy extracted from the ground, if the building was heating-dominated.

The resulting sizing guidelines recommend hour-by-hour energy analysis to determine the energy extracted from and rejected to the building water loop. Using this information the designer will have available easy-to-use, accurate sizing guidelines that should result in more economical installations than those based on previous "rule of thumb" guidelines.

INTRODUCTION

The decentralized ground-source heat pump system for commercial buildings is an evolution of the traditional water loop heat pump (WLHP) system used in North American buildings for over 30 years. The traditional WLHP system consists of water-source heat pumps which provide space conditioning, a water-circulating pump, supply and return piping to all heat pumps from the circulating pump, a cooling tower, and a boiler. A ground-source WLHP, simply called a ground-source heat pump system, replaces the cooling tower and boiler with a heat exchanger buried in the ground. The heat exchanger can be either piping located in horizontal trenches or piping installed in vertically drilled boreholes. Fluid circulating from the heat pumps on the building water loop will extract energy from the ground if the heat pumps are in a net heating mode, or will reject energy to the ground if the heat pumps are in a net cooling mode. Because of smaller land use in commercial buildings, vertical heat exchangers have become more popular than horizontal heat exchangers. Owner benefits of a ground-source heat pump system include design flexibility, lower operating costs (both purchased energy and demand), less mechanical room space, less outdoor equipment, no exterior noise, and excellent building aesthetics.

Most ground-source heat pump systems installed in North America to date have been for residential applications. Experience has been gained on the sizing of vertical heat exchangers based on these residential applications. This wealth of experience has been transferred to the design of commercial vertical ground-source heat pump systems. The use of residential "rule of thumb" sizing guidelines has resulted in commercial heat exchangers that are oversized and therefore not as cost-effective as they could be.

A need existed to replace the conventional residential sizing guidelines with more accurate and appropriate commercial sizing guidelines. To gain acceptance by designers it was necessary to maintain the simplicity of the earlier guidelines. This paper presents the development of sizing guidelines for the most popular commercial ground-source heat exchanger, the closed-loop vertical heat exchanger.

METHODOLOGY

The size of a ground heat exchanger is dependent upon the building loads that it serves. These loads are a function of building type, size, location, internal loads (such as equipment usage, lighting, people), infiltration, solar gain, envelope heat

TABLE 1
KEY CHARACTERISTICS OF MODELLED COMMERCIAL INSTITUTIONAL BUILDINGS

Building Type	Building Floor Area (m^2)			Occupant Density (m^2/person)		Lighting & Equipment (W/m^2)		Heat Pump Performance (COP/EER)*	
	Small	Medium	Large	Low	High	Low	High	Low	High
Multi-unit Residential	3,900	9,290	13,940	46.5	35.8	8.6	14.0	3.7/12.0	4.5/18.9
Office	2,790	5,570	10,030	20.0	7.0	24.8	53.8	3.8/12.0	4.3/16.1
School	4,650	11,150	16,720	7.0	4.6	22.6	43.1	3.8/12.0	4.3/16.1

* Coefficient of performance (COP) based on 50°F (10°C) entering water temperature and 70°F (21°C) dry bulb.
Energy Efficiency Ratio (EER) based on 70°F (21°C) entering water temperature, and 80°F (27°C) dry bulb and 67°F (19°C) wet bulb air temperature.

conduction, and ventilation levels. Other factors which influence the amount of heat exchange between the building and ground include ground temperature, ground soil or rock type, and heat pump efficiency.

To account for all of these factors we used two simulation tools. An hour-by-hour building energy analysis program was used to generate the loads on the ground heat exchanger, while a computerized ground heat exchanger algorithm was used to model the ground-coupling. This two-step approach was used because as of 1994 no commercially available building energy analysis program included a ground heat exchanger sizing algorithm.

Building Simulation

A large number of energy analysis computer simulations were performed for a variety of commercial and institutional buildings. The purpose of the simulations was to determine

- building peak block heating and cooling loads (i.e., peak diversified load of the building);
- maximum heat added to and extracted from the water loop;
- total energy added to and extracted from the building loop for each month over an entire year.

Table 1 summarizes the key characteristics of the commercial/institutional buildings used in the simulations. Three building types were selected: office, school, and multi-unit residential. Within each building type three building sizes were simulated. Prescriptive requirements from ASHRAE 90.1-1989 were used for envelope insulation levels, infiltration rates, occupancy, lighting, equipment, ventilation, and service hot water schedules. Ventilation rates were based on ASHRAE 62-1989. Window-to-wall ratios were based on ASHRAE 90.1-1989 maximum fenestration values for the office and multi-unit residential buildings, but were halved for the schools.

In the simulations all ventilation loads were met by the space-conditioning heat pumps. Ventilation was scheduled off when there was no occupancy in the office and school buildings. The heating and cooling setpoints were 70°F (21°C) and 75°F (24°C), respectively.

To generate a variation in building use and associated loads, internal loads (lights, equipment, and people) were varied from low values based on ASHRAE 90.1-1989 criteria to high values based on the Ontario Hydro Applications Handbook (Ontario Hydro, 1987), as shown in Table 1.

To assess the impact of heat pump performance, both high and low values for heat pump coefficient of performance (COP) and energy efficiency ratio (EER) were used and were obtained from one manufacturer's product lines. The values used are shown in Table 1.

To account for the impact of location (which includes weather and ground temperature effects) the buildings were modelled in three cities in the United States (Albany, Atlanta, and Philadelphia) and four cities in Canada (Halifax, Ottawa, Toronto, and Winnipeg).

The range in parameters resulted in the performance of 252 building energy analysis simulations (3 building types * 3 building sizes * 2 internal load settings * 2 levels of heat pump efficiency * 7 locations). Modifications were also made to the buildings located in the Canadian cities, resulting in additional simulations. The multi-unit residential buildings were modelled with the service hot water loads met by a dedicated water-to-water loop-connected heat pump; the offices were modelled with unoccupied temperature setback to 55°F (13°C) and set up to 99°F (37°C); the schools were modelled with both the service hot water heat pumps and temperature scheduling as described above. This brought the total number of simulations to 396.

Modelling of Ground Heat Exchanger

The modelling of heat extraction from or heat rejection to the ground was based on a procedure developed by Hart and Couvillion (1986). Their algorithm was implemented on a spreadsheet program. This implementation has been validated against actual system data collected from a large secondary school installation and several residential installations (Caneta Research Inc., 1992, 1993).

The method employs a line-source theory and the superposition principle, where warranted, to solve the ground heat exchange problem. It accounts for effects such as on/off cycling, U-tube pipe-to-pipe interference, semi-infinite assumptions, and earth temperature variation with both depth and time of year.

Hourly values for the required heat extraction from or heat rejection to the ground were calculated by the building energy analysis program and summed monthly. These values represent the net load on the ground by the loop-connected heat pumps (i.e., energy extracted from the building space by heat pumps in cooling mode plus their compressor energy equals the energy rejected to the ground, while the energy rejected to the building space by heat pumps in heating mode less the compressor energy equals the energy extracted from the ground). Results from each of the 396 building simulations were input into the heat exchanger model. In addition to the building simulation results described above, inputs to the heat exchanger model were the following:

- The minimum ground loop temperature limit was set to approximately 18°F (10°C) less than the mean ground temperature for that location.
- The maximum ground loop temperature limit was set to approximately 36°F (20°C) above the mean ground temperature for that location.
- The flow rate through the ground loop was set equal to 3 gallons per minute per ton (0.054 liters per second per kilowatt), where the tonnage was the greater of the building peak block cooling load or peak block heating load.

The entering water temperature used to estimate the EER and COP, and hence the energy consumed by the heat pumps on the water loop, was determined after one iteration through the building energy analysis program and the ground heat exchanger model. One entering water temperature was chosen as representative of heating operation while another was representative of cooling operation.

Mean ground temperatures were obtained from either Environment Canada (1994) or Oklahoma State University (1988). All buildings were simulated with 1¼-in. (31.75 mm) vertical U-tube heat exchangers made of high density polyethylene and filled with a 20% ethanol solution. The boreholes were assumed to be spaced such that there was no thermal interference. The overburden depth was assumed to be 30 ft (9.1 m). The soil was assumed to be heavy, damp soil (conductivity of 1.30 W/mK) in winter and heavy, dry soil (conductivity of 0.87 W/mK) in summer. Below the overburden was assumed to be dense rock or granite (conductivity of 3.46 W/mK).

With the above information, the ground heat exchanger model was used to calculate the design vertical heat exchanger length. This length was the minimum allowed while still maintaining the minimum or maximum ground loop temperature limits. One of these temperature limits would define the heat exchanger length. If the minimum ground loop temperature limit was the defining limit, the heat exchanger length was governed by the heating load. Similarly, if the maximum ground loop temperature limit was the defining limit, the heat exchanger length was governed by the cooling load. In some cases a building could have cooling dominant loads, yet the heat exchanger was sized to the heating load. This occurred because a smaller temperature difference was allowed between the minimum ground loop temperature limit and the mean ground temperature [i.e., 18°F (10°C)] than between the maximum ground loop temperature limit and the mean ground temperature [i.e., 36°F (20°C)].

Where the maximum ground loop temperature limit determined the length, a shorter heat exchanger length, based on the minimum ground loop temperature limit, was also determined. This represented the case where the designer used the option of a supplementary heat rejector or cooling tower in addition to the ground heat exchanger to meet the cooling loads.

RESULTS

The heat exchanger lengths predicted from the sizing model were plotted against either the net energy extracted from the ground or the net energy rejected to the ground, depending on which governed the heat exchanger sizing. Figures 1 through 3 present the heat exchanger length results in this way.

Figure 1 shows the predicted heat exchanger length when the cooling load determined the length. None of the schools or multi-unit residential buildings modelled in Canada with a loop-connected heat pump for service water heating had their heat exchanger length determined by the cooling load. Because of the additional heating load due to supplying heat for hot water, all these heat exchanger lengths were limited by the lower ground loop temperature limit.

Figure 2 shows the results for the heat exchanger length as determined by the heating load for all simulations that did not have a heat pump for service water heating. Figure 3 shows the results for all 396 simulations, including those that had a heat pump for water heating simulated. Both of these plots show a high degree of scatter. The results from Winnipeg, the coldest climate of the seven locations modelled, showed the greatest deviation from the group.

After examining the heating length plots, we realized that a parameter was not being accounted for, resulting in the considerable scatter. We hypothesized that the cooling or heating heat exchanger length would be a function of the mean ground temperature, T_{mean}, in each locale and the maximum and minimum ground loop temperature (T_{max} and T_{min}, respectively) used to size the heat exchangers. That is:

$$\text{Cooling Heat Exchanger Length} \propto \frac{\text{Energy Rejected to Ground}}{(T_{max} - T_{mean})}$$

$$\text{Heating Heat Exchanger Length} \propto \frac{\text{Energy Extracted from Ground}}{(T_{mean} - T_{min})}$$

Dividing the energy rejected or extracted from the ground by the appropriate temperature difference resulted in the data collapsing to a more linear relationship, as shown in Figures 4 to 6. This was especially true for the heating lengths (Figures 5 and 6), but had little impact on the cooling length (Figure 4) which initially had less scatter. The linear fit has been forced through the origin in these figures because of the physical limit of zero heat exchanger length required when the heating/cooling energy was zero.

The results in Figures 1 to 6 were based on the assumption of 30 ft (9.1 m) of overburden on top of granite rock. The overburden was assumed to be heavy, damp soil in winter and heavy, dry soil in summer. The sensitivity of the heat exchanger length to these assumptions for ground thermal conductivity was assessed by additional simulations of the large office building in Albany. The predicted heat exchanger length using the original assumptions was 23,100 ft (7030 m). If the entire borehole were drilled in granite, a near-ideal situation for improved heat transfer, the predicted heat exchanger length was 22,300 ft (6790 m), or 3.5% less than the original assumption. If the depth of overburden were dramatically increased to 150 ft (46 m), the predicted heat exchanger length would be 28,100 ft (8560 m), or 22% greater than that predicted under the original assumption.

HEAT EXCHANGER SIZING

The following are the proposed guidelines for the sizing of vertical closed-loop heat exchangers.

Calculation of the vertical heat exchanger length requires that the design engineer undertake an hour-by-hour computer energy simulation of the design building to account for the annual net energy added to or extracted from the closed water loop by the heat pumps in the zones. In each hour there will either be a net energy extraction from or a net energy rejected to the water loop by the heat pumps, or there will be no energy exchange because the loop is perfectly balanced (an unlikely event) or because the system is idle due to temperature scheduling. Each of the hourly net energy rejected and the net energy extracted values should be individually summed for all 8760 hours of a year.

The mean annual ground temperature, T_{mean}, for the locale should be determined either from Oklahoma State University (1988) or other source. As a first approximation, the minimum temperature limit (T_{min}) should be set equal to T_{mean} - 18°F (or T_{mean} - 10°C), while the maximum temperature limit (T_{max}) should be set equal to T_{mean} + 36°F (or T_{mean} + 20°C). Temperature limits should not violate the heat pump manufacturer's recommended entering water temperature limits, typically 20°F (-7°C) and 110°F (43°C) for extended range units. For the energy simulations of the building, the recommended average temperature of the fluid in the loop is the midpoint of the applicable ΔT [i.e., $(T_{mean} + T_{min})/2$ for energy extracted from the water loop by heat pumps in heating mode and $(T_{max} + T_{mean})/2$ for energy added to the water loop by heat pumps in cooling mode].

For building energy analysis computer programs that do not report the energy extracted from or rejected to the water loop, the thermal capacity of the loop should be set to zero. The above energy extracted and energy added values will be equal to the energy added by a boiler or the energy extracted by a cooling tower of a conventional water loop system. The COP and EER of the heat pumps should be calculated on the basis of recommended average temperatures described above. The temperature of the water loop should be set at the standard rating value.

The annual energy rejected by the heat pumps and the annual energy extracted by the heat pumps, calculated by the hour-by-hour energy simulation, should be divided by (T_{max} - T_{mean}) and (T_{mean} - T_{min}), respectively. Entering Figure 4, 5 or 6 as appropriate will allow determination of the cooling and heating heat exchanger lengths (note that borehole length will be one-half the heat exchanger length). The longer of the heating and cooling lengths will determine the design length for the building. If the cooling length is longer than the heating length, the engineer can consider using a supplemental heat rejector to augment a ground heat exchanger sized to the shorter heating length. The design heat rejection rate of the supplemental heat rejector, Q_{Rej}, should be sized to

$$Q_{Rej} = (Q_{Tot.Rej} - Q_{Loop\ Rej}) / 2 / \text{hours}$$

where $Q_{Tot.Rej}$ is the total amount of heat to be rejected during the design cooling month, $Q_{Loop\ Rej}$ is the heat that can be rejected to the earth over the design cooling month by the heat exchanger, and hours is the number of hours in the design cooling month.

The design outdoor condition for the heat rejector sizing is the average design month dry bulb for a dry closed-circuit liquid cooler or the average design month wet bulb for an evaporative closed-circuit liquid cooler or cooling tower. This sizing approach assumes that 50% of the hours in the design month have less extreme conditions and therefore the heat rejector has greater heat rejection capacity.

Example

As an example of the difference in heat exchanger sizing as given by the proposed guidelines versus conventional rule of thumb guidelines, consider the large office building in Philadelphia. When the simulation was performed with low internal loads and low-efficiency heat pumps, the peak building cooling load was 143 tons (504 kW) and the annual heat rejected was 3044 MBTU (892 MWh). The conventional guideline would call for a heat exchanger length of 300 feet per ton of installed cooling capacity (i.e., double the bore length). The conventional design length based on peak building cooling load, a value less than the installed heat pump cooling capacity, in this example would be 43,000 ft (13,100 m). The proposed sizing guidelines, using Figure 4 with a (T_{max} - T_{mean}) difference of 20°C, predict a required length of 26,900 ft (8200 m), about 60% of the conventional length. The difference between the conventional and guideline lengths increases with higher-efficiency heat pumps (because the annual heat rejected decreases) and with higher internal loads.

CONCLUSIONS AND RECOMMENDATIONS

The preparation of the closed-loop vertical heat exchanger sizing guidelines has involved 396 energy analysis and heat exchanger sizing runs. The guidelines can result in accurate sizing of vertical heat exchanger systems, but require that the designer have an energy analysis computer program that can model water loop heat pumps to provide the energy rejected and extracted from the loop over a 12-month period. This is a degree of sophistication not previously required with simple rules of thumb for sizing heat exchanger systems, but it is necessary if more accurate sizing is desired.

There appears to be a considerable difference between conventional heat exchanger design lengths and those based on these guidelines for certain building types. To date, only one validation attempt has been made (Caneta Research Inc., 1993), and this was a school in Toronto, Canada. Additional validation, particularly in hot climates, is needed. With installed systems, limited ranges in entering water temperatures are symptomatic of oversizing and wide ranges of undersizing. The ground-source industry should monitor its installations to determine if its sizing methods are accurate, conservative, or optimistic.

The guidelines will yield conservative design lengths when the borehole is contained in solid granite and will undersize the heat exchanger where there is large overburden (e.g., greater than 100 ft). In most locations where closed-loop ground-source heat pumps are installed, the sizing guidelines should give accurate required heat exchanger lengths.

The required heat exchanger lengths for a given annual energy extraction were found to be a strong function of the ΔT between the mean ground temperature and the T_{min} value. By accounting for this effect, better correlation was obtained in the final analysis.

One area that has not been thoroughly investigated but warrants investigation is the provision of hot water heating by a dedicated water-to-water heat pump operating from the loop. This, of course, is an additional load on the loop and the ground. The building energy analysis program we used does not model such a heat pump; therefore, it was necessary to do the calculation outside the program each month, rather than on an hour-by-hour basis. Hour-by-hour calculations would more accurately account for the coincidence of the water heating, space heating, and cooling loads on the loop. Further work should be performed to account for varying effects of ground properties, pipe size, and heat exchange fluid.

ACKNOWLEDGMENTS

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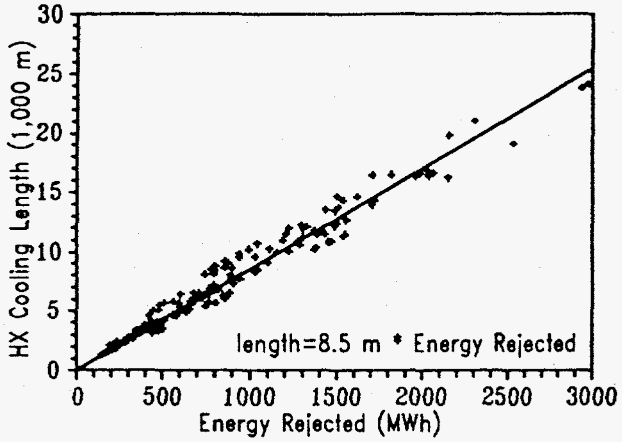

Figure 1 Heat Exchanger Cooling Length versus Annual Energy Rejected (SHW Loads Not Included)

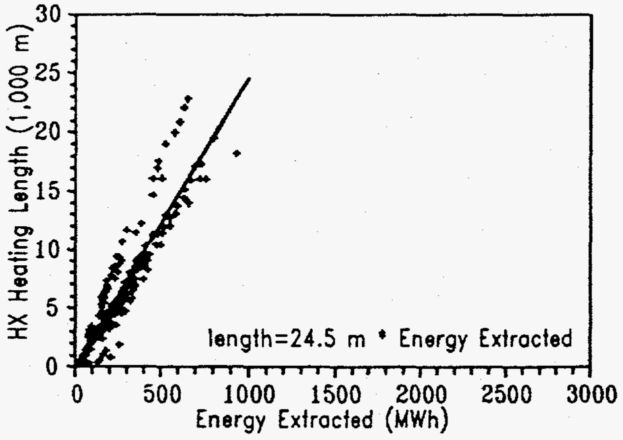

Figure 2 Heat Exchanger Heating Length versus Annual Energy Rejected (SHW Loads Not Included)

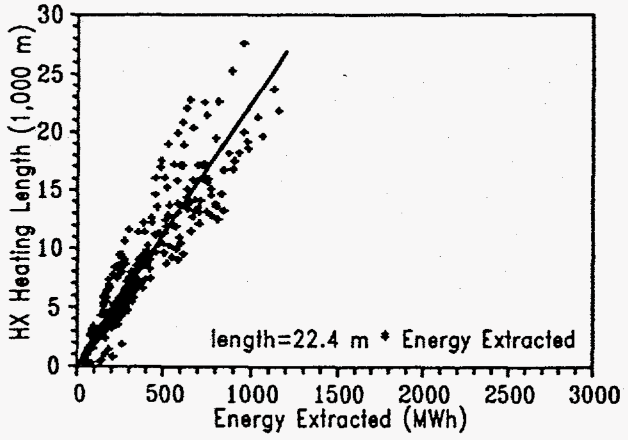

Figure 3 Heat Exchanger Heating Length versus Annual Energy Rejected (SHW Loads Included)

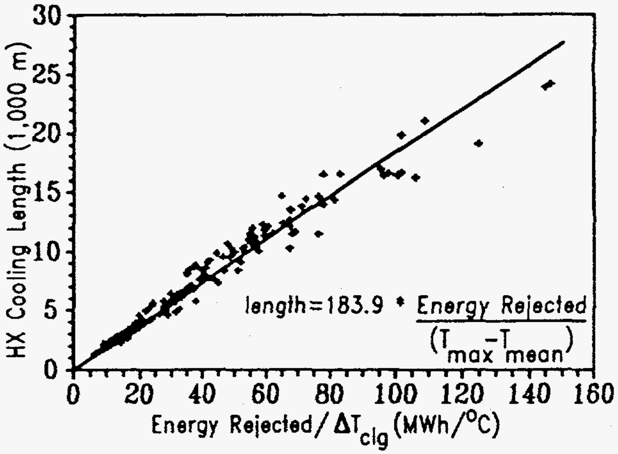

Figure 4 Heat Exchanger Cooling Length versus Annual Energy Rejected (SHW Loads Not Included)

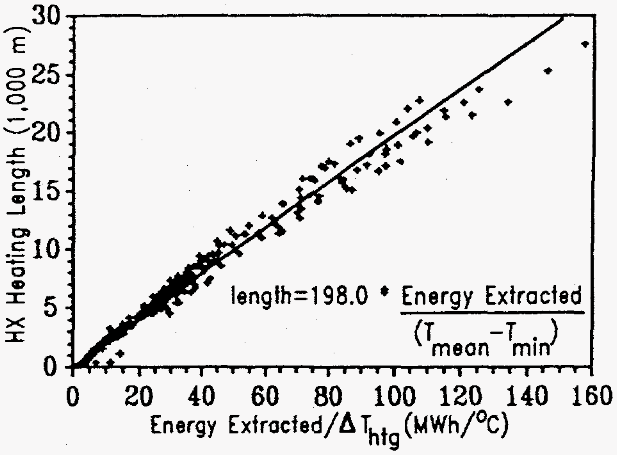

Figure 5 Heat Exchanger Heating Length versus Annual Energy Rejected (SHW Loads Not Included)

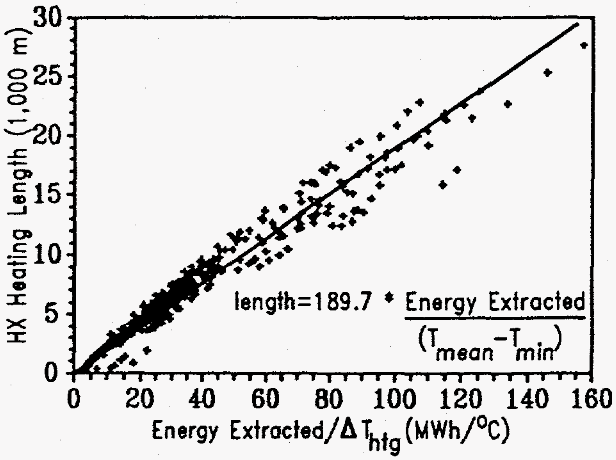

Figure 6 Heat Exchanger Heating Length versus Annual Energy Rejected (SHW Loads Included)