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EVALUATING THE HEAT PUMP ALTERNATIVE FOR HEATING ENCLOSED WASTE--ETC(U)
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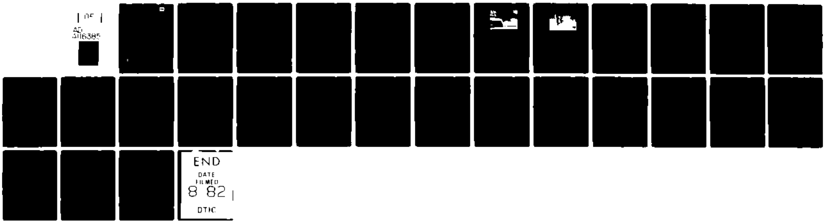

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Evaluating the heat pump alternative for heating enclosed wastewater treatment facilities in cold regions

C.J. Martel and Gary E. Phetteplace

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This report presents a five-step procedure for evaluating the technical and economic feasibility of using heat pumps to recover heat from treatment plant effluent. The procedure is meant to be used at the facility planning level by engineers who are unfamiliar with this technology. An example of the use of the procedure and general design information are provided. Also, the report reviews the operational experience with heat pumps at wastewater plants located in Fairbanks, Alaska, Madison, Wisconsin, and Wilton, Maine.

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PREFACE

This report was prepared by C. James Martel, Environmental Engineer, of the Civil Engineering Research Branch, and Gary E. Phetteplace, Mechanical Engineer, of the Applied Research Branch, Experimental Engineering Division, U.S. Army Cold Regions Research nd Engineering Laboratory. Funding for this research was provided by DA Project 4A762730AT42, Design, Construction and Operations Technology for Cold Regions, Task C, Cold Regions Base Support: Maintenance and Operations, Work Unit 001, Operations and Maintenance of Cold Regions Sanitary Engineering Facilities.

Sherwood Reed and Dr. Virgil Lunardini of CRREL technically reviewed the manuscript of this report.

A

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CONVERSION FACTORS: U.S. CUSTOMARY TO METRIC (SI) UNITS OF MEASUREMENT

These conversion factors include all the significant digits given in the conversion tables in the ASTM Metric Practice Guide (E 380), which has been approved for use by the Department of Defense. Converted values should be rounded to have the same precision as the original (see E 380).

Multiply	By	To obtain
inch	25.4*	millimeter
$foot^2$	0.09290304*	$meter^2$
$foot^3$	0.02831685	$meter^3$
gallon	3.785412*	liter
$pound/foot^3$	16.01846	$kilogram/meter^3$
ton	907.1847	kilogram
Btu	1055.056	joule
Btu/lb °F	4186.800	joule/kilogram kelvin
degrees Fahrenheit	$t_{°F} = (t_{°C} - 32)/1.8$	degrees Celsius

*Exact.

EVALUATING THE HEAT PUMP ALTERNATIVE FOR HEATING ENCLOSED WASTEWATER TREATMENT FACILITIES IN COLD REGIONS

by

C. James Martel and Gary Phetteplace

INTRODUCTION

Wastewater treatment facilities located in cold regions are often enclosed to facilitate operation and maintenance. Because of today's high cost of energy, heating these large enclosures with a conventional oil- or gas-fired boiler can be expensive. A potentially less costly method is to use a heat pump. This device extracts heat from the treatment plant effluent and reuses it for space heating. Usually there is enough heat energy in the effluent to supply the entire heating requirements of the facility, including office and laboratory areas. A heat pump is especially attractive in cold regions where the wastewater is kept warm in insulated pipelines and heated utilidors. Also, the water supply is often heated in order to avoid freezing in the distribution system. An added environmental benefit of installing a heat pump is the reduction of thermal discharges to the receiving stream.

The purpose of this report is to present a simple procedure that can be used to evaluate the technical and economic feasibility of installing a heat pump at new or existing wastewater treatment facilities. The intended users of this procedure are environmental engineers, who generally are not familiar with this technology. Information used to develop this procedure was obtained from site visits, technical reports and papers, and heating/ventilation and air conditioning (HVAC) manuals. It should be noted that this procedure is intended for feasibility studies at the facility planning level only. The actual unit specifications should be determined by a qualified HVAC specialist.

EXISTING HEAT PUMP INSTALLATIONS

One of the first sewage treatment plants to include a heat pump in its initial design is located in Fairbanks, Alaska. This plant, completed in the summer of 1976, has an average design flow of 8.0 million gallons per

Figure 1. Wastewater treatment plant in Wilton, Maine.

day (mgd) and an average wastewater temperature of 45°F. The wastewater treatment processes include an aerated grit chamber, several pure-oxygen-activated sludge units and a chlorine contact chamber. Approximately 12.5% of the chlorinated effluent is diverted through two heat transfer coils and a heat pump. The heat transfer coils supply the base heat load of the facility while the heat pump is used as a booster unit when more heat is needed. The heat transfer coils and heat pump are designed to heat and ventilate a 38,880-ft^2 enclosure at 40°F when the outside air temperature is as low as -60°F (Crews 1977). The total estimated heat load, including ventilation, is 3.728×10^9 Btu/yr. The maximum rate of heat loss through the enclosure was calculated to be 743,400 Btu/hr and the ventilation requirement is 10,000 ft^3/min. A backup heating system, consisting of electrical resistance coils, is provided for supplemental heating.

The Wilton, Maine, Wastewater Treatment Plant (Fig. 1), also successfully uses a heat pump to heat their enclosed treatment area. This 0.45-mgd plant was constructed using the latest energy-saving technologies, including a heat pump, active and passive solar heating systems, digester gas recovery and air-to-air heat recovery (Wilke and Fuller 1976). A study conducted by Wright-Pierce, Architects and Engineers (1980) indicated that the heat pump unit provided 60% of the total facility heating requirements. The unit was designed for a total heat output of 320,000 Btu/hr at

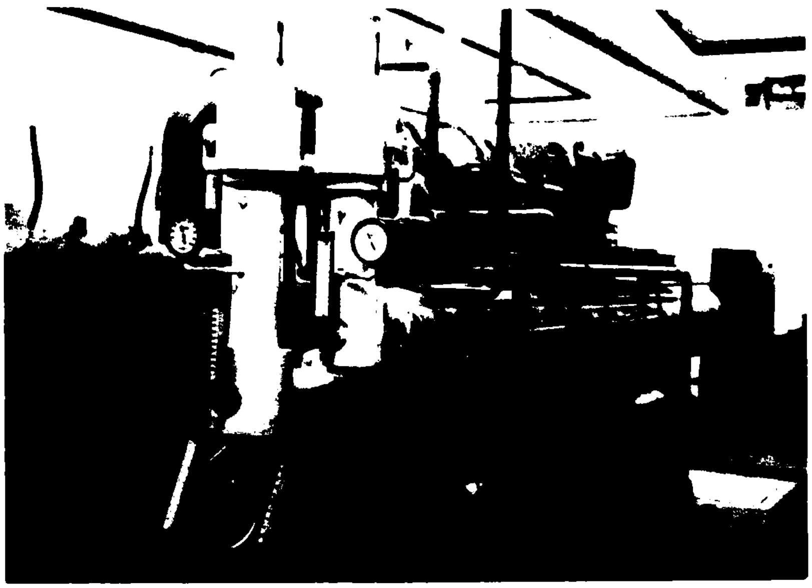

Figure 2. Heat pump unit at the Delafield-Hartland plant.

an effluent flow rate through the heat pump of 60 gal./min. The initial cost of this unit was $11,400 in 1978.

After initial problems with the control circuitry were solved, a heat pump unit performed satisfactorily during its first year of operation at the Delafield-Hartland plant near Madison, Wisconsin[1]. This new plant has a design flow of 2.2 mgd, and the treatment scheme includes primary sedimentation, rotating biological contactors and rapid sand filtration. The total heating load of this facility was estimated to be 1.0×10^6 Btu/hr (Budde et al. 1979). The heat pump (Fig. 2) was designed to supply 120°F water to the hot water heating system rather than the conventional 180°F. Because of this lower temperature, the size of the building heating elements had to be increased by approximately 20%. The hot water produced by the heat pump is stored in a large tank which can also be heated by stand-by electric immersion heaters.

Heat pumps have been used for many years at numerous commercial and industrial installations, but their use with sewage effluent is a relatively new application of this technology. Some problems have been reported due to the solids content and corrosive nature of wastewater.

[1] R. Hyde, Superintendent, Delafield-Hartland Wastewater Treatment Plant, pers. comm., 1981.

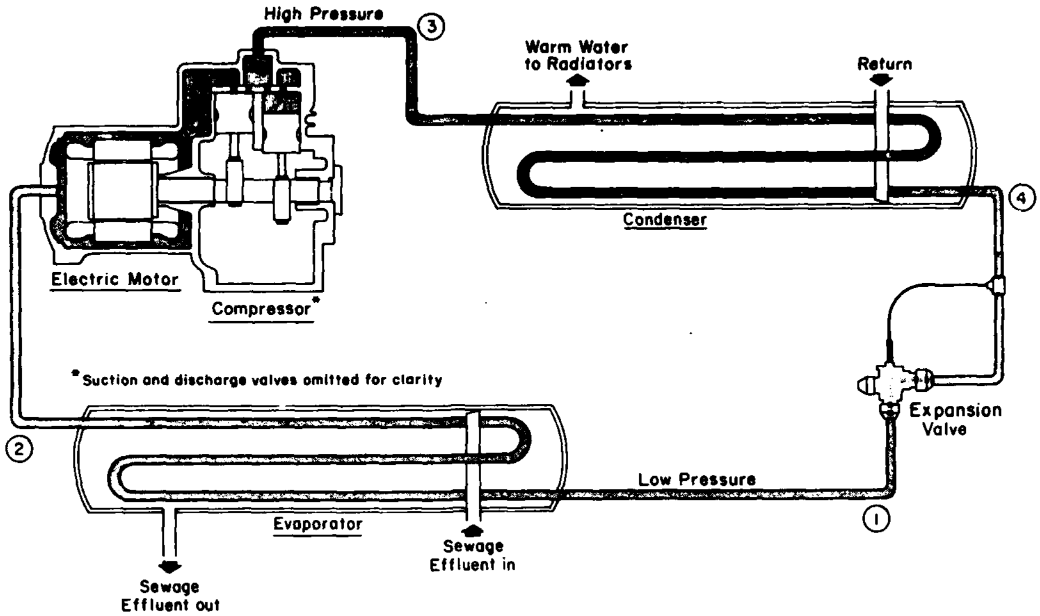

Figure 3. The principal components of a water-source heat pump (Phetteplace 1979).

Recently, the Fairbanks plant had to shut down its heat pump unit because of corrosion in the evaporator caused by chlorine and sulfides in the effluent[2] (Miko 1981). At the Wilton plant, the operator reported that he had to frequently clean the effluent strainers which are located in the supply line to the evaporator[3]. He indicated that this problem could have been avoided by installing larger strainers.

The heat pump unit in each of these plants is supplied with wastewater of secondary effluent quality or better. A higher strength wastewater is not used because of possible solids accumulatiom and fouling problems on the interior surface of the tubes in the evaporator unit of the heat pump (see Fig. 3). Even with secondary effluent, a fine-screening device is sometimes placed in the effluent supply line to trap solids escaping from the clarifier.

The preferred location for the intake to the evaporator supply line is just upstream of the overflow weir in the chlorine contact chamber. As noted by Budde et al. (1979), this location is preferred because the

[2] J. Miko, Superintendent, Fairbanks Municipal Wastewater Treatment Plant, pers. comm., 1981.
[3] J. Thornton, Superintendent, Wilton, Maine, Wastewater Treatment Plant, pers. comm., 1981.

chlorine should help to maintain clean evaporator tubes. However, chlorine residual concentration should be carefully controlled in order to prevent corrosion. The return line should be located downstream of the chlorine contact chamber just before the effluent metering device. This location avoids the potential adverse effects of low temperature return water on disinfection.

THEORY OF OPERATION

A heat pump is a cyclic device that transfers heat from a lower temperature source (e.g. the effluent) to a higher temperature sink (e.g. the hot water heating systm). The most common example of a heat pump is a refrigerator. It removes heat from the low temperature of its interior and rejects that heat to the higher temperature on its exterior. Thus, it "pumps" heat back into its surroundings, where the heat originated. A heat pump allows heat from a low temperature source such as treatment plant effluent to be used at a higher temperature, such as is required for space heating. The price paid for recovery of this waste heat is the cost of the electrical energy used by the compressor.

With the aid of Figure 3 it is possible to trace the thermodynamic cycle of a heat pump. Since this is a cyclic device, there is no true starting or finishing point for the cycle; arbitrarily, the cycle will be assumed to begin where the refrigerant is mostly liquid and at low temperature and pressure. This is the state of the refrigerant after it leaves the expansion valve as shown in Figure 3 (state 1). The refrigerant then enters the evaporator where it takes on sufficient heat from the effluent to be evaporated from the liquid to the gaseous state at nearly constant temperature and pressure (state 2). Next, the refrigerant enters the compressor where the pressure is increased, a process that also increases the temperature of the refrigerant (state 3). Finally, the gaseous refrigerant flows into the condenser where heat is given up to the hot water heating system. This heat is the latent heat of vaporization of the refrigerant as it condenses from a gas to a liquid (state 4). The liquid refrigerant now enters the expansion valve and has its pressure reduced. Upon exit from the expansion valve, the refrigerant is now ready again to repeat the cycle just described. The advantage of this cycle is that most of the heat input occurs in the evaporator. This energy does not have an economic cost, although some electrical energy is needed to run the compressor.

COEFFICIENT OF PERFORMANCE (COP)

The ratio of energy delivered to the building heating system vs the energy required by the compressor is a measure of the efficiency of the heat pump. This ratio, called the coefficient of performance (COP), is the most important factor in determining the economic advantage of installing a heat pump. A COP higher than 1.0 indicates that the energy output by the heat pump is greater than the energy input so that an energy savings would be realized. However, the operation and maintenance cost of the heat pump must also be recovered. Typically, COP's of heat pumps used in wastewater treatment plants range from 2.5 to 4.5.

The COP depends mainly on the temperature difference between the heat source (wastewater effluent) and the heat delivery system, which in most cases will be a hot water heating system. The greater the temperature difference, the lower the COP. For this reason it is desirable to use the highest temperature heat source and the lowest practical delivery temperature. The relationship between effluent temperature, delivered temperatures and COP is shown in Figure 4. This figure assumes a temperature drop of 5°F across the evaporator, an approach temperature of 15°F (temperature difference between the effluent, or hot water, and refrigerant as they approach the heat exchangers), and a combined motor/compressor efficiency of 65%. Although these factors could vary for each heat pump installation, they should be accurate enough for estimation purposes.

To maintain a high COP it is better to provide a greater flow through the evaporator and drop the temperature of the effluent only a few degrees than to provide a low flow and a large temperature drop. Usually, the increased COP compensates for the higher pumping costs. The allowable temperature drop will depend on the minimum temperature of the effluent during the winter. In the case of an existing plant, the minimum effluent temperature can be determined from past data. For new plants the minimum temperature must be estimated.

Another way to improve the COP is to design the hot water heating system for temperatures in the 120° to 140°F range rather than th standard 180°F. As mentioned previously, the COP improves as the temperature difference between the effluent and the hot water heating system decreases. The cost of the larger radiators needed for a 120° to 140°F heating system is small compared to the reduced operational cost at the higher COP (Niess 1981). A heat pump can also be used at an existing plant equipped with a

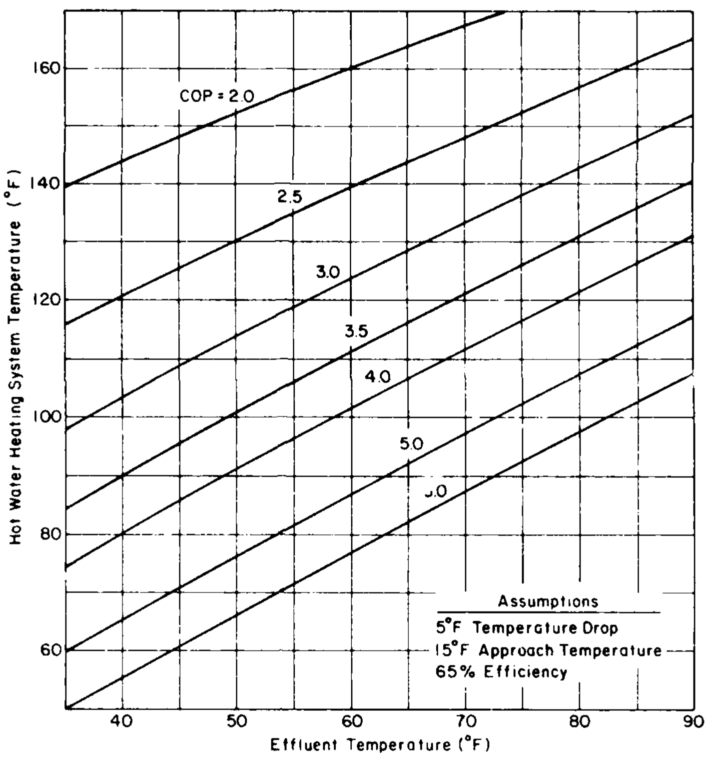

Figure 4. Approximate COP values for given effluent and heating system temperatures.

standard heating system. However, supplemental heating may be required on occasion. This supplemental heat could be supplied by the old boiler.

The heat pump system controls should be designed to automatically lower the hot water supply temperature as the outdoor temperature increases. This can be achieved with a control device which senses the temperature of the return water from the heating system. As the temperature of the return water increases, the control system reduces the compressor energy input and causes the temperature of the supply water to decrease so that the heat pump system operates at near the optimum COP.

Another way to increase the COP is to reduce on-off cycling by closely matching the capacity of the heat pump with the load requirement. One effective method of load control is through the use of a multi-speed or variable speed compressor. This type of compressor reduces the energy input by idling one or more cylinders as the load decreases. The improved efficiency easily compensates for the additional cost of the special controls.

SIZING THE HEAT PUMP AND ESTIMATING COSTS

Building Heat Load

The most important step in sizing a heat pump is to estimate the building heat load. The heating load results from basic conduction losses through walls, windows, ceiling and floor. Also air infiltration and ventilation cause sizable heat losses, especially in enclosed sewage treatment facilities where air exchange is necessary to control odors. These losses vary from structure to structure depending on type and quality of construction, ventilation requirements, and climatic conditions. The heating load will also depend on the desired internal temperature of the building. At cold regions facilities, the administration and laboratory areas are usually maintained at 68°F while the enclosed treatment area may be maintained at only 40 to 45°F.

Conduction heat losses per 1000 ft^2 of floor area can be calculated as follows:

$$E_c = 24\ H\ D \tag{1}$$

where E_c = annual building conduction losses, Btu/1000 ft^2 of floor area

H = hourly heat loss, Btu/1000 ft^2-hr-°F

D = degree days, °F day/yr.

The hourly heat loss value (H) can be estimated from the building construction criteria in Table 1. Degree day (D) information for 65°, 55° and 45°F base temperature can be obtained from Appendix A or local sources such as a heating oil distributors. The designer should use the base temperature closest to the desired internal temperature of the building.

Determining the total heating load will also require calculation of the amount of infiltration and ventilation. Infiltration is the uncontrolled leakage of air into a building through cracks in the walls and ceilings, and around doors and windows (including the result of opening and closing doorways). Ventilation is the intentional movement of air into and out of a building through specific openings. In general it is more energy efficient to produce air exchange by installing a ventilation system, which can be controlled, than by relying on infiltration, which cannot be controlled.

Table 1. Typical hourly heat loss for three types of buildings (Wesner et al. 1978).

Type	Construction	Hourly heat loss (H), Btu/1000 ft^2-hr-°F
A	Uninsulated	820
B	3.5-in. wall insulation, 6.0-in. ceiling insulation, storm windows	450
C	Same insulation as Type B, double glazed windows, floor insulation	325

The energy required to heat the infiltration and ventilation air, per 1000 ft^2 of floor area, can be calculated as follows:

$$E_v = 466 \text{ haD} \qquad (2)$$

where E_v = annual ventilation heat losses, Btu/1000 ft^2-yr

h = height of ceiling, ft

a = number of air changes/hr.

The constant 466 includes the specific heat of air (0.24 Btu/lb-°F), the density of air (0.0809 lb/ft^3), the unit area of the building (1000 ft^2) and the number of hours per day. Accepted values of "a" for both administrative and enclosed treatment areas range between four to six air changes per hour (Wesner et al. 1978). The term D is degree days as defined in eq 1.

The combined annual heating loss for each 1000 ft^2 is the sum of E_c and E_v. This sum, when multiplied by the heated floor area of the facility (expressed in 1000 ft^2), will give the total heating load of the facility. This is a conservative estimate because no credit is taken for heat produced by internal sources such as lights and machinery. Solar heat gain during the day is also not taken as a credit because the heating equipment should be sized to meet the most adverse conditions, which generally occur at night.

Heat Pump Capacity

Once the building heating load is calculated, the required heat pump capacity (Q, in Btu/hr) can be determined by dividing the total annual heating load by the number of hours per year of equivalent full load operation. The yearly hours of equivalent full load operation (N), i.e. the hours of use resulting if the total heating load for the year were supplied at full capacity, can be calculated from climatic information as follows:

$$N = \frac{24\ D}{B - T} \qquad (3)$$

where B is the base temperature of degree days (D) used in heat load calculations (°F), and T the winter design temperature (°F).

The winter design temperature T is based on a probability distribution of average hourly temperatures during the months of December, January and February. This temperature can be found in the ASHRAE Cooling and Heating Load Calculation Manual (1979) for many U.S., Canadian and foreign cities. Appendix A lists winter design temperatures for a few selected cities in the United States.

Since winter design temperatures were selected at the 97 1/2% and 99% levels, there could be 54 and 22 hours, respectively, during an average winter when the outdoor temperature is below the winter design temperature. During this period, the heat pump would be inadequate to supply the heating load and freezing problems might occur, especially if the internal design temperature of the building is already near freezing, as in the case of an enclosed treatment area. Typically this area is only heated to 40° to 45°F. Therefore, for enclosed treatment areas it would be safer to use the winter design temperature at the 99% level. The winter design temperature at the 97-1/2% level could be used for other areas where the internal design temperature will be higher, say 68°F, and the risk of freezing is smaller.

Estimation of COP

As mentioned eariler, the COP can be estimated from Figure 4, but the average effluent temperature during the winter must first be determined. For existing plants this is not a problem since the effluent temperatures can be measured directly. For new plants, however, the temperature must be estimated from data obtained at other northern facilities. Data from eight such facilities are shown in Table 2 for the months of October through March. The average effluent temperature for these facilities during this time period was approximately 50°F. Although these temperatures are for raw sewage, they should not be significantly different from effluent temperatures if the facility is enclosed. In evaluating the temperature drop through an enclosed pure-oxygen-activated sludge system, Boyle (1976) con-

Table 2. Monthly average temperature (°F) of raw sewage (Alter 1969).

Place	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.
Aurora, Ill.	49	49	50	48	47	46
Canton, Ohio	56	50	47	42	40	42
Flint, Mich.	63	58	53	50	48	45
Schenectady, N.Y.	64	60	56	48	43	36
Traverse City, Mich.	60	55	52	48	46	49
Fairbanks, Alaska	53	52	52	52	51	51
Juneau, Alaska	46	46	44	38	36	37
Wilton, Me.*	62	57	50	46	44	43
Average	57	53	51	47	44	43

* Temperature of secondary effluent (Wright-Pierce 1980)

cluded that the amount of heat loss was negligible and the temperature drop through the system should not be greater than 1°F.

The data shown in Table 2 also indicate that effluent temperatures will gradually decrease over the winter heating season. October usually has the highest effluent temperatures and the March lowest. Thus, there is a significant delay in the impact of cold ambient air on the temperature of the sewage effluent. This delay is caused by the insulating effect of the cover material above the collection system. An important benefit of this delay is that higher effluent temperatures are available during the coldest months of December and January. By the time the effluent temperature reaches its lowest point in March the demand for heat has lessened considerably.

Effluent Flow rate to Heat Pump

The effluent flow rate needed to supply the evaporator will depend on the heat pump capacity, COP, and the temperature drop across the evaporator. The heat pump capacity and COP can be determined as indicated earlier. The temperature drop across the evaporator should only be 3° to 5°F according to Niess (1981). Budde et al. (1979) report flow rates ranging between 12 to 17% of the average wastewater flow. This percentage will be higher for facilities where the entire treatment area is enclosed and heated. Also, the percentage of effluent used by the heat pump will be higher during the first few years because the initial flow will be lower than the 20-year design value.

Once the required heat pump capacity, COP, and temperature drop across

Table 3. Average heating value of some fuels.

Fuel	Heating value (10^6 Btu)
#2 fuel oil, 1 US gallon	0.1385
#4 fuel oil, 1 US gallon	0.145
Natural gas, 1000 ft^3	1.092
Propane, 1000 ft^3	2.560
Bituminous steam coal, short ton	24.580
Electricity, 1000 kilowatt hours	3.413

the evaporator are established, the volumetric flow rate can be calculated as follows:

$$V = \frac{2000\ Q\ (1 - \frac{\eta_m}{100\ COP})}{\Delta T} \tag{4}$$

where V = volumetric flow rate, gpm

Q = heat pump capacity, million Btu/hr

η_m = compressor motor efficiency, %

ΔT = temperature drop across the evaporator, °F.

Since the motor efficiency is usually around 90%, eq 4 can be simplified to

$$V = \frac{2000\ Q\ (1 - \frac{0.9}{COP})}{\Delta T}\ . \tag{5}$$

Estimation of Cost and Energy Savings

The construction, O&M labor, and maintenance material costs as a function of heat pump capacity can be estimated from Figures 5, 6 and 7. These costs are based on an Engineering News Record (ENR) Construction Cost Index of 2499 (Jan. 1977).

The energy cost savings will depend on the cost of heating with an alternative fuel source. The heating values of some alternative fuels are shown in Table 3. The cost of these fuels will vary regionally and can be determined by contacting local fuel suppliers. Boiler efficiency will also affect the cost of an alternative heating systems. Oil- and coal-fired boiler efficiences are usually about 70% while that of a gas-fired boiler is about 80%. Electrical resistance heating is normally figured at 100% efficiency. The heat pump efficiency is already included in the assumptions for determining the COP from Figure 4.

EXAMPLE

The following is an example of how to use the previous information to properly size a heat pump unit and estimate energy savings. As mentioned earlier, this procedure is meant for the feasibility planning stage only. Nevertheless, this procedure should provide a relatively accurate estimate of the energy savings, if any, resulting from a heat pump installation. A competent HVAC specialist should be contacted for more detail in regard to design specifications and cost information.

As part of a facility plan, the project engineer wishes to evaluate the cost-effectiveness of installing a heat pump to supply the heating needs for a 5-mgd plant located near Minneapolis, Minnesota. The total floor area of this facility will be 27,000 ft^2 of which 25,000 ft^2 is for an enclosed treatment area and the remaining 2000 ft^2 is for the administrative and laboratory functions.

Step 1. Determine the heating load.

The building heat loss can be determined from eq 1. Type B construction is planned for the enclosed treatment area which results in an hourly heat loss value (H) of 450 Btu/1000 ft^2-hr-°F. This area will only be heated to 40°F which should be sufficient to prevent freeze-ups and facilitate operator maintenance. With the 45°F base, the average degree day reading for the Minneapolis area is 3309°F-day (see App. A). Type C construction is proposed for the office area which has a H value of 325 Btu/1000 ft^2-hr-°F. This area will be heated to the standard indoor design temperature of 68°F. Therefore, the 65°F base will be used for this area which results in an average degree day reading of 8382°F-day. From eq 1, the building heat losses for the treatment and office areas are

Conduction heat loss for enclosed treatment area $= 24 \times 450 \times 3309 = 35.7 \times 10^6$ Btu/1000 ft^2-yr

Conduction heat loss for office area $= 24 \times 325 \times 8382 = 65.4 \times 10^6$ Btu/1000 ft^2-yr.

The heating requirements due to ventilation can be calculated from eq 2. The ceiling heights of the treatment and office areas will be 12 and 8 ft respectively. A ventilation requirement of four changes per hour was assumed for both areas. From equation 2, the ventilation losses are

Ventilation heat loss, enclosed treatment area $= 466 \times 12 \times 4 \times 3309 = 74.0 \times 10^6$ Btu/1000 ft^2-yr

Table 4. Summary of heating requirements for hypothetical treatment facility in Minneapolis, Minnesota.

Service area	Conduction loss, E_c (Btu/1000 ft^2-yr)	Ventilation heat loss, E_v (Btu/1000 ft^2-yr)	Combined heat loss, $E_c + E_v$ (Btu/1000 ft^2-yr)	Area heat load (Btu/yr)
Enclosed treatment area (25,000 ft^2)	35.7×10^6	74.0×10^6	109.7×10^6	2472×10^6
Offices (2000 ft^2)	65.4×10^6	125.0×10^6	190.4×10^6	381×10^6

Ventilation heat loss, office area $= 466 \times 8 \times 4 \times 8382 = 125.0 \times 10^6$ Btu/1000 ft^2-yr

The combined heat losses can be determined by adding building and ventilation heat losses for each area. To convert the combined heat losses to a heat load, multiply the heat losses by the floor area expressed in 1000 ft^2. The total facility heat load is then the sum of each area heat load. A summary of heat losses and heat loads in each area is shown in Table 4. The total facility heat load is estimated to be 3123×10^6 Btu/yr.

Step 2. Determine the capacity of the heat pump.

The capacity or size of the heat pump can be determined by dividing the heating load by the number of equivalent full load hours. From eq 3, the number of equivalent full load hours (N) for the treatment and office areas are 1302 and 2613 hours respectively. The winter design temperatures (T) used in eq 3 were -16°F for the treatment area and -12°F for the office area. For reasons discussed earlier, a lower winter design temperature (-16°F) was used for the treatment area to allow a greater margin of safety.

Since the heating requirements of the two areas are quite different, separate heat pump units would be more efficient. Based on a heat load of 2742×10^6 Btu/yr (see Table 4) and 1302 equivalent full load hours, the required capacity of unit 1, the heat pump unit for the enclosed treatment area, is 2.1×10^6 Btu/hr (2742×10^6 Btu/yr ÷ 1302 hr/yr). Likewise, the required capacity of a smaller heat pump unit (unit 2) for the office area is 0.15×10^6 Btu/hr.

Step 3. Determine the COP.

The COP can be estimated from Figure 4 once the effluent and hot water heating system temperatures are established. Since this is a new plant, an effluent temperature of 50°F will be assumed. The heating system temperature for unit 1 will be set at 90°F while unit 2 will be set at 120°F. These temperatures should provide adequate heat transfer and allow each heat pump to operate at a high COP. From Figure 4 the COP's are 4.1 and 2.8 for units 1 and 2 respectively.

Step 4. Determine the evaporator flow rate.

By assuming a 5°F drop across the evaporator, the flow rate can be determined from eq 5.

$$\text{Effluent flow rate through evaporator of unit 1} = \frac{2000 \times 2.1 \left(1 - \frac{0.9}{4.1}\right)}{5} = 657 \text{ gal./min}$$

$$\text{Effluent flow rate through evaporator of unit 2} = \frac{2000 \times 0.15 \left(1 - \frac{0.9}{2.8}\right)}{5} = 41 \text{ gal./min}$$

The total flow of effluent through both evaporators is then almost 700 gpm at peak load, which is equivalent to 1.0 mgd or approximately 20% of the design flow.

Step 5. Estimate costs and energy savings.

The estimated construction O&M labor and maintenance material costs for each heat pump unit were calculated from Figures 5, 6 and 7 for an Engineering News-Record Construction Cost Index of 3610 (3 Sept 1981). These costs (see Table 5) may be higher than for comparable oil- or gas-fired systems; however, the energy saved by the heat pump should more than compensate. In a comparison of oil and heat pump heating systems, Budde et al. (1979) determined that, although the heat pump system had higher capital, O&M labor, and maintenance material costs, the present worth cost was lower because of the energy savings. As energy costs escalate, this differential will undoubtedly increase. Also, for municipal treatment plants, the initial cost of the heat pump could be offset by federal and state grants.

The energy costs of heating with the heat pumps and various other fuels are shown in Table 6. These costs were based on an annual heating

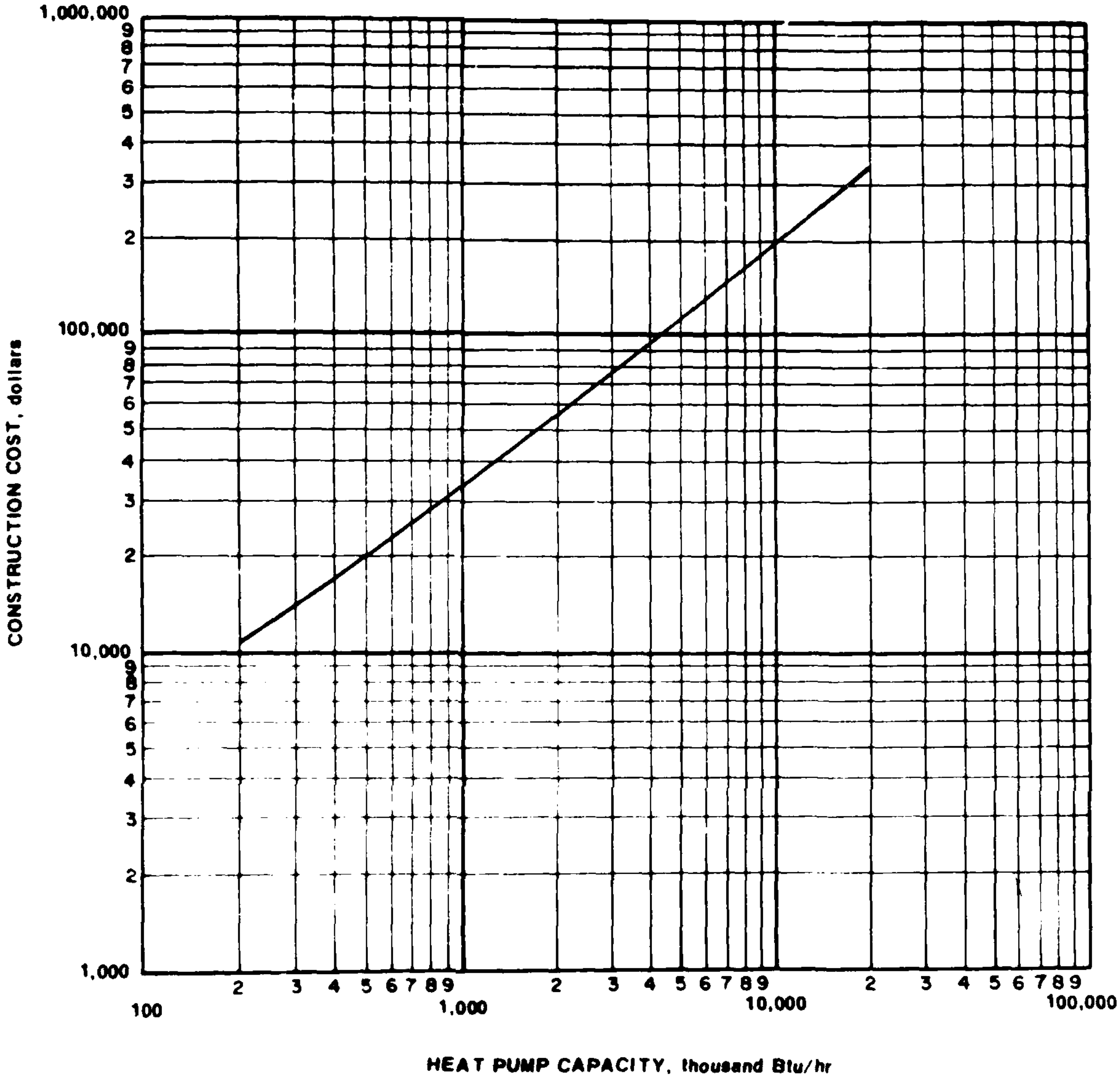

Figure 5. Heat pump construction cost (Wesner et al. 1978).

Table 5. Estimated costs of heat pump units.

Heat pump	Service area	Capacity (1000 Btu/hr)	Construction* costs ($)	O&M labor and maintenance material† costs ($/yr)
1	Enclosed Treatment Area	2100	86,676	4,400
2	Office area	150	13,000	530

* Based on ENR Construction Cost Index of 3610 (3 Sept. 1981).

† Includes labor costs @ $8/hr and maintenance material costs (see Fig. 6 and 7).

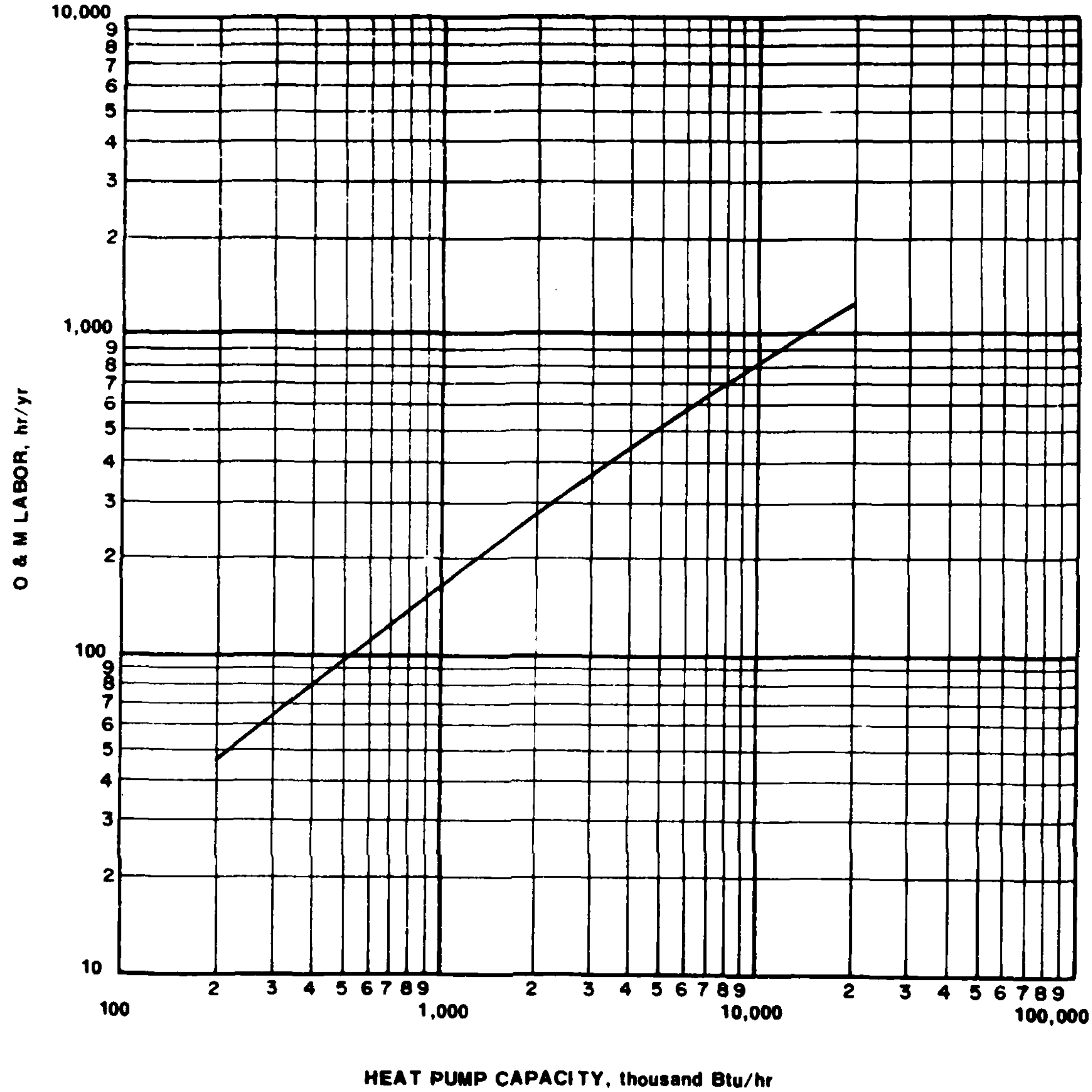

Figure 6. Heat pump O&M labor requirements (Wesner et al. 1978).

load of 3123 x 10^6 Btu and unit costs for the Minneapolis area of 0.04/kWh for electricity, $1.19/gal. for fuel oil, $97/ton for coal and $4.06/1000 ft^3 of natural gas[4]. Also shown in Table 6 are the potential annual savings resulting from a heat pump installation vs other types of heating equipment. The minimum annual savings is $5,082 compared to an equivalent gas boiler. The annual savings compared to a typical oil-fired boiler is $28,905. This cost differential in favor of a heat pump installation will probably increase in the future if fuel prices continue to rise.

[4] A. Duval, St. Paul District, U.S. Army Corps of Engineers, St. Paul Minnesota, pers. comm., 1981.

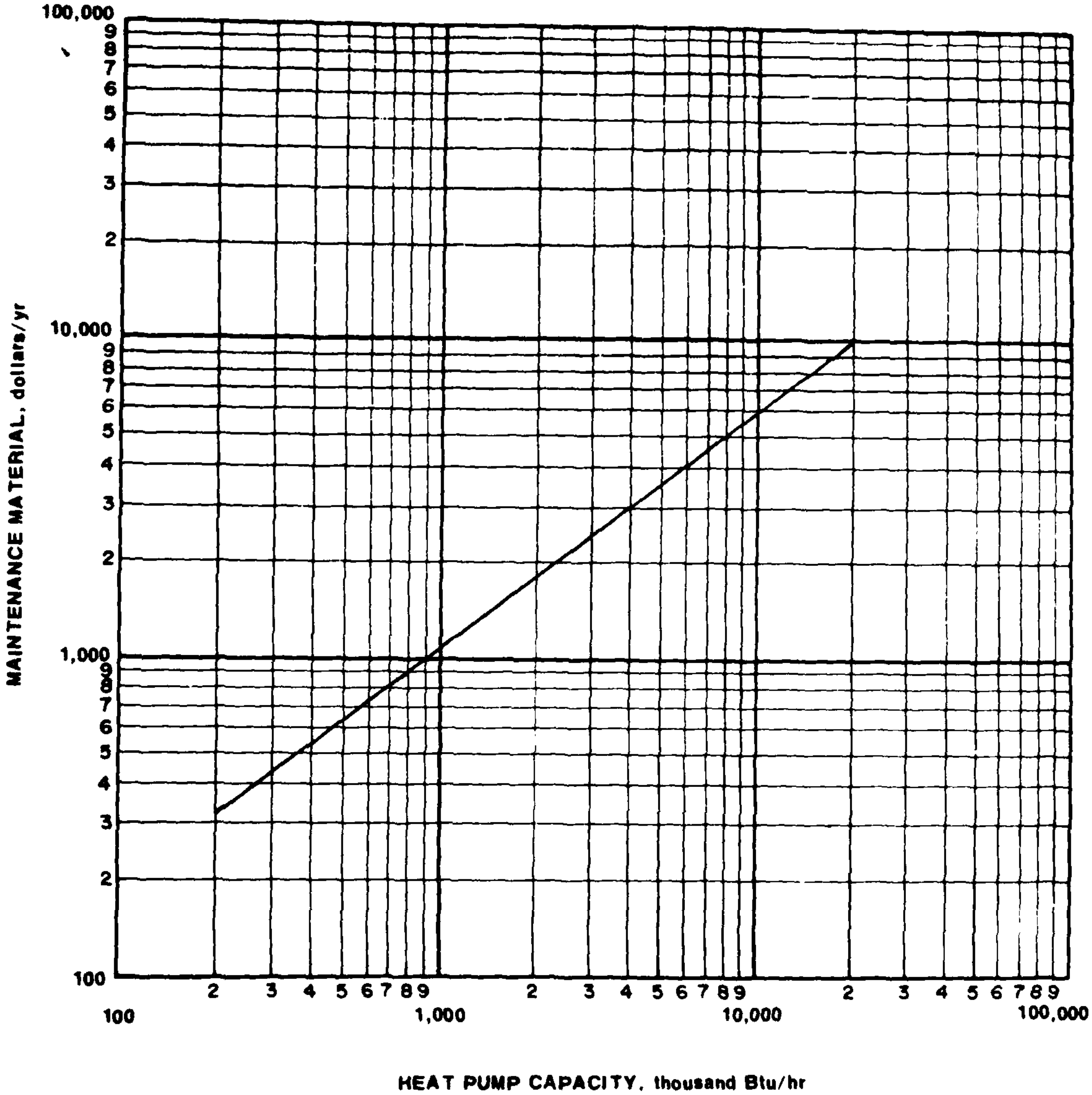

Figure 7. Heat pump maintenance material costs (Wesner et al. 1978).

Table 6. Comparison of energy costs.

Type of heating	Annual* energy cost ($)	Extra energy cost compared to heat pump ($/yr)
Heat pumps	9,433	--
Elec. resistance	52,294	42,861
Oil-fired boiler	38,338	28,905
Coal fired boiler	17,608	8,175
Natural gas boiler	14,515	5,082

* Electrical resistance cost figures at 100% efficiency, oil- and coal-fired boilers at 70% efficiency, and natural gas at 80% efficiency. The heat pump energy cost was calculated as follows:

$$\frac{2742\times10^{6}\ \text{Btu/yr}}{4.1} + \frac{381\times10^{6}\ \text{Btu/yr}}{2.8}\ \frac{\$0.04/\text{kWh}}{3413\ \text{Btu/kWh}} = \$9433.00$$

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APPENDIX A. DEGREE-DAYS AND WINTER DESIGN TEMPERATURES FOR SELECTED U.S. CITIES

State and City	Degree days (°F-day) 45°F base[1]	55°F base[1]	65°F base[2]	Winter design temp[2] (°F) 99%	97.5%
Alaska					
Anchorage			10864	-23	-18
Fairbanks			14279	-51	-47
Juneau			9075	- 4	- 1
Nome			14171	-31	-27
Colorado					
Denver	1548	3440	6283	- 5	1
Grand Junc.	1757	3433	5641	2	7
Pueblo	1499	3261	5462	- 7	0
Connecticut					
Meriden	--	734	--	--	--
New Haven	1769	3237	5897	3	7
District of Columbia	1041	2487	4224	14	17
Idaho					
Boise	1045	2814	5809	3	10
Lewiston	1034	2688	5542	- 1	6
Pocatello	2161	4140	7033	- 8	- 1
Illinois					
Cairo	749	2119	3821	--	--
Chicago	1969	3743	5882	- 3	2
Springfield	1677	3289	5429	- 3	2
Indiana					
Evansville	799	2335	4435	4	9
Indianapolis	1397	2829	5699	- 2	2
Iowa					
Davenport	2296	4142	--	--	--
Des Moines	2440	4180	6588	-10	- 5
Kansas					
Dodge City	1385	2962	4986	0	5
Topeka	1518	1811	5182	0	4
Wichita	1152	2587	4620	3	7
Kentucky					
Lexington	--	2557	4683	3	8
Louisville	1073	2294	4660	5	10

[1] Strock and Koral (1956) Handbook of Air Cond., Heating and Ventilation, 2nd ed.
[2] ASHRAE (1979) Cooling and Heating Load Calculation Manual.

State and City	Degree days (°F-day) 45°F base	55°F base	65°F base	Winter design temp (°F) 99%	97.5%
Maine					
East Port	2956	5236	--	--	--
Portland	2530	4572	7511	- 6	- 1
Maryland					
Baltimore	986	2491	4654	14	17
Massachusetts					
Boston	1787	3603	5634	6	9
Nantucket	1514	3419	5891	--	--
Michigan					
Alpena	3131	5499	8506	-11	- 6
Detroit	2240	4089	6232	3	6
Escanaba	3699	5918	8481	-11	- 7
Grand Haven	2405	3435	--	--	--
Grand Rapids	2332	4177	6894	1	5
Houghton	4029	6112	--	--	--
Lansing	2537	4444	6909	- 3	1
Sault Ste. Marie	4049	6575	9048	-12	- 8
Minnesota					
Duluth	4419	6774	1000	-21	-16
Minneapolis	3309	5417	8382	-16	-12
Moorhead	4796	6572	--	--	--
St. Paul	2497	5497	--	-16	-12
Missouri					
Kansas City	1463	2980	4711	2	6
Saint Louis	1186	2745	4484	2	6
Springfield	982	2423	4900	3	9
Montana					
Havre	3736	5874	8182	-18	-11
Helena	2843	5071	8129	-21	-16
Kalispell	2874	5131	8191	-14	- 7
Nebraska					
Lincoln	3023	3850	5864	- 5	- 2
North Platte	2291	4152	6684	- 8	- 4
Omaha	2284	3982	6612	- 8	- 3
Valentine	2833	4801	7425	--	--
Nevada					
Winnemucca	1670	3468	6761	- 1	3

State and City	Degree days (°F-day) 45°F base	55°F base	65°F base	Winter design temp (°F) 99%	97.5%
New Hampshire					
Concord	2646	4640	7383	- 8	- 3
New Jersey					
Atlantic City	1123	2904	4812	10	13
New York					
Albany	2018	4302	6875	- 4	1
Binghamton	2073	4296	7286	- 2	1
Buffalo	2359	4316	7062	2	6
Ithaca	2412	4023	--	- 5	0
New York	1412	3089	4871	11	15
Oswego	2274	4363	--	1	7
Rochester	2341	4231	6748	1	5
North Dakota					
Bismarck	3831	6468	8851	-23	-19
Williston	4616	6399	9243	-25	-21
Ohio					
Cincinnati	1376	3003	4410	1	6
Cleveland	1525	3795	6351	1	5
Columbus	1600	3255	5660	0	5
Dayton	1487	3147	5622	- 1	4
Sandusky	1949	3425	5796	1	6
Toledo	1990	3757	6494	- 1	3
Oklahoma					
Oklahoma City	600	1835	3725	9	13
Oregon					
Baker	2321	4307	--	- 1	6
Portland	373	1911	4109	17	23
Roseburg	272	1868	4491	18	23
Pennsylvania					
Erie	2337	3837	6451	4	9
Harrisburg	1565	3236	5251	7	11
Philadelphia	1122	2695	4486	10	14
Pittsburgh	1377	3088	5053	3	7
Scranton	1938	3755	6254	1	5
Rhode Island					
Block Island	871	3388	5804	--	--
South Dakota					
Yankton	2898	6045	--	-13	- 7

State and City	Degree days (°F-day) 45°F base	55°F base	65°F base	Winter design temp (°F) 99%	97.5%
Tennessee					
Chattanooga	242	1398	3254	13	18
Knoxville	431	1741	3494	13	19
Memphis	166	1284	3015	13	18
Nashville	419	1678	3578	9	14
Utah					
Modena	1978	3981	--	--	--
Salt Lake City	1475	3202	6052	3	8
Vermont					
Burlington	3014	4984	8269	-12	- 7
Virginia					
Lynchburg	554	1928	4166	12	16
Norfolk	260	1496	3421	20	22
Richmond	549	1895	3865	14	17
Washington					
North Head	184	2062	—	--	--
Seattle	408	2185	4424	22	27
Spokane	1741	3672	6655	- 6	2
West Virginia					
Elkins	1506	3327	5675	1	6
Parkersburg	1147	2784	4754	7	11
Wisconsin					
Greenbelt	3318	5331	8029	-13	- 9
LaCrosse	3034	3992	7589	-13	- 9
Madison	3067	4850	7863	-11	- 7
Milwaukee	2657	4617	7635	- 8	- 4
Wyoming					
Cheyenne	2500	4700	7381	- 9	- 1
Lander	3208	5450	7870	-16	-11

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