CHAPTER 14
GREENHOUSES

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14.1 INTRODUCTION

Greenhouse heating is one of the most common uses of geothermal resources. Because of the significant heating requirements of greenhouses and their ability to use very low- temperature fluids, they are a natural application. The evaluation of a particular greenhouse project involves consideration of the structure heating requirements, and the system to meet those requirements. This chapter is intended to provide information on each of these areas.

14.2 GREENHOUSE CONSTRUCTION

In order to make an evaluation of geothermal heating systems for greenhouses, it is first necessary to examine the different heating requirements imposed by various construction methods.

At one time, greenhouses were constructed exclusively of cypress wood frames and single glass lites. Recent years have seen substantial changes in construction techniques and materials. In general, construction may be considered to fall into one of the following four categories:

1. Glass
2. Plastic film
3. Fiberglass or similar rigid plastics
4. Combination of two and three.

All of the above are generally constructed of steel or aluminum frames.

Glass greenhouses are the most expensive to construct because of both the cost of the glazing material and the requirement for a stronger framework to support the glass. In many cases, fiberglass panels are employed on the side and end walls of the structure. The building profile is generally of peaked design, with 36 and 42 ft widths, and lengths in 20 ft increments most common. This type of greenhouse is preferred by growers whose plants require superior light transmission qualities. In addition to offering the highest light quality, the glass greenhouse also has the poorest energy efficiency. Heating costs are high because of the poor insulating quality of single glazing and the high infiltration of cold air through the many "cracks" in the construction. This issue of high transmission loss has been addressed in recent years through the introduction of new, double glazing panels for glass houses. However, because of the expense of these panels and their effect upon light transmission, most glass greenhouses remain single layer.

Plastic film greenhouses are the newest variation in greenhouse construction techniques. This type of structure is almost always of the arched roof or "quonset hut" design. The roof can come all the way down to the ground or can be fitted with side walls. The side walls, if employed, and end walls are generally of fiberglass construction. Maintenance requirements for the plastic film are high in that it generally requires replacement on 3-year intervals or less, depending on the quality of the material. Most plastic film houses employ a double layer of film separated by air space. The air space is maintained by a small blower that pressurizes the volume between the layers. This double poly design is a very energy efficient approach to greenhouse design. Double poly not only reduces transmission losses (losses through the walls and roof) by 30 to 40%, but also substantially reduces infiltration (in leakage of cold air). Although the plastic film tends to lose more heat than glass through radiation, the net effect is a reduction in heating requirements compared to glass construction. Infiltration is reduced because the "cracks" present in other types of construction are eliminated through the use of the continuous plastic film. As a result, there is less opportunity for the cold outside air to penetrate the structure. The superior energy efficiency of the film construction comes at the price of reduced light transmission, however. As a result, highly light sensitive crops cannot be grown in the double-poly greenhouse as successfully as in other constructions. These greenhouses are generally constructed in 30 ft width, and 100 and 150 ft lengths.

Fiberglass greenhouses are similar in construction to the glass houses described above. They are generally of peaked roof design, but require less structural support as a result of the lower weight of the fiber glass. Heat loss of the fiberglass house is about the same as the glass house. Although the fiberglass material has a lower conductivity than glass, when considered in the overall building heat loss, this has little effect.

14.3 HEATING REQUIREMENTS

In order to select a heating system for a greenhouse, the first step is to determine the peak heating requirement for the structure. Heat loss for a greenhouse is composed of two components: (a) transmission loss through the walls and roof, and (b) infiltration and ventilation losses caused by the heating of cold outside air.

To evaluate transmission loss, the first step is to calculate the surface area of the structure. This surface area should be subdivided into the various materials employed, i.e. square feet of double plastic, square feet of fiberglass, etc.

For example, consider a fiberglass wall, double-poly roof greenhouse 42 ft x 120 ft with 8 ft side walls (see Figure 14.1).

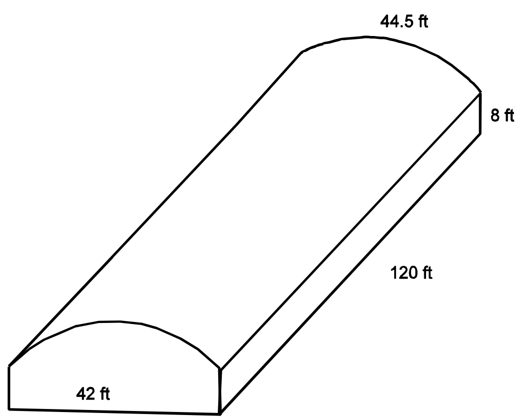


Figure 14.1 Example greenhouse.

Determine the double poly area (roof only):

A_1 = arch width x greenhouse length
A_1 = 44.5 ft x 120 ft
A_1 = 5,340 ft^2

Fiberglass area (side walls and end walls),
Side walls:

A_s = height x length x 2
A_s = 8 ft x 120 ft x 2
A_s = 1,920 ft^2

End walls:

A_e = 1,254 ft^2

Total fiberglass area:

$A_2 = A_s + A_e$
A_2 = 1,254 ft + 1,920 ft
A_2 = 3,174 ft^2.

After determining the total surface area (A) of the various construction materials, this value is then combined with a design temperature difference (ΔT) and a heat loss factor (U) for each component, to calculate the total transmission heat loss (q):

$$q = (A_1 \times \Delta T \times U_1) + (A_2 \times \Delta T \times U_2).$$

The design temperature difference is a function of two values: (a) design inside temperature, and (b) design outside temperature. The inside design value is simply the temperature to be maintained inside the space (typical values appear in Table 14.1 range). The design outdoor temperature is not the coldest outdoor temperature recorded at the site. It is generally considered to be a temperature that is valid for all but 22 h/y during the heating season. Acceptable values for various locations are generally available from state energy offices or organizations such as American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE, 1978).

Table 14.1 Temperature Requirements for Typical Greenhouse Crops

Vegetables	Day	Night
Peppers	65-85	60-65
Tomato	70-75	62-65
Cucumber	75-77	70
Lettuce (hydroponic)	75	65
(Reduce temp. 2° when picking)		
(During germination, humidity 30-70%)		
Flowers		
Roses	60-62	62
Poinsettias	70-80	64-72
Easter Lilies	60	
Carnations	75	50
Geraniums	70-80 (max)	
Fuchsia (min)	70 (min)	65

For this example, assume a design outdoor temperature of 0°F and a design indoor temperature of 60°F. This results in a design temperature difference (ΔT) of:

ΔT = 60°F - 0°F
ΔT = 60°F.

The final value in the transmission heat loss equation is the heat transfer coefficient (U). Acceptable values for various materials are shown in Table 14.2.

Table 14.2 Glazing Material U Values[a]

Material	Btu/h ft^2 °F
Glass	1.10
Fiberglass	1.00
Single poly	1.15
Double poly	0.70

a. Roberts, 1985

The U factor is also influenced by wind speed. The above values are based upon a wind speed of 15 mph. If other wind speeds are expected to occur at the design outside condition, then allowances should be made for this by adjusting the U factor are shown in Table 14.3.

Table 14.3 U Values at Various Wind Velocities

Material	Velocity (mph)					
	0	5	10	20	25	30
Glass	0.765	0.951	1.040	1.140	1.160	1.180
Fiberglass	0.695	0.865	0.949	1.034	1.058	1.078
Single poly	0.810	1.000	1.090	1.190	1.210	1.230
Double poly	0.535	0.631	0.675	0.716	0.728	0.736

For the example, the transmission heat loss (q_p) for the double poly roof area is:

$q_p = 5340\ ft^2 \times 60°F \times 0.70$ Btu/h ft^2 °F
$q_p = 224{,}280$ Btu/h

and for the fiberglass areas:

$q_F = 3{,}174\ ft^2 \times 60°F \times 1.00$ Btu/h ft^2 °F
$q_F = 190{,}440$ Btu/h

Total transmission heat loss (q_1) is then:

$q_1 = q_p + q_F$
$q_1 = 224{,}280$ Btu/h + 190,440 Btu/h
$q_1 = 414{,}720$ Btu/h

As mentioned previously, total heat loss is a function of two components: (a) transmission heat loss, and (b) infiltration. For greenhouse design, infiltration is generally analyzed via the air change method. This method is based upon the number of times per hour (ACH) that the air in the greenhouse is replaced by cold air leaking in from outside. The number of air changes which occur is a function of wind speed, greenhouse construction, and inside and outside temperatures. Table 14.4 outlines general values for different types of greenhouse construction.

Table 14.4 Air Change Data for Various Glazing Materials

Material	Air Changes/h
Single glass	2.5 to 3.5
Double glass	1.0 to 1.5
Fiberglass	2.0 to 3.0
Single poly	0.5 to 1.0
Double poly	0.0 to 0.5
Single poly w/low fiberglass sides	1.0 to 1.5
Double poly w/low fiberglass sides	0.5 to 1.0
Single poly w/high fiberglass sides	1.5 to 2.0
Double poly w/high fiberglass sides	1.0 to 1.5

a. Roberts, 1985, ASHRAE, 1978.

As the number of air changes is related to the volume of the greenhouse, after selecting the appropriate figure from above, it is necessary to calculate the volume of the structure. For the example structure, this is most easily accomplished in two steps. These figures do not include ventilation.

Volume (V_1) of the greenhouse:

V_1 = end wall area x greenhouse length
$V_1 = 627\ ft^{-2}\ ft \times 120\ ft$
$V_1 = 75{,}247\ ft^3$

From the Table 14.4, the number of air changes/h (ACH) would be 1.0 to 1.5--use 1.0 (double poly with high fiberglass sides).

Heat loss (q_2) caused by infiltration:

$q_2 = ACH \times V_T \times \Delta T \times 0.018$
$q_2 = 1.0 \times 75{,}247\ ft^3 \times 60°F \times 0.018$
$q_2 = 81{,}260$ Btu/h

Total greenhouse heating (q_T) requirement:

$q_T = q_1 + q_2$
$q_T = 414{,}720$ Btu/h + 81,260 Btu/h
$q_T = 495{,}980$ Btu/h (98.41 Btu/ft^2 of floor area)

This calculation assumes that infiltration will meet winter ventilation requirements. If artificial ventilation is required in excess of infiltration, this should be added to the peak load.

This is the peak or design heating load for the greenhouse. The heating equipment selected for the structure would have to be capable of meeting this requirement.

14.4 GREENHOUSE HEATING SYSTEMS

There are basically six different geothermal heating systems which are applied to greenhouses:

1. Finned pipe
2. Standard unit heaters
3. Low-temp. unit heaters
4. Fan coil units
5. Soil heating
6. Bare tube.

Often the choice of heating system type is not dictated by engineering considerations such as maximum use of the available geothermal resource or even the most economical system, but on grower preference. Grower preference may be based strictly on past experience and familiarity with growing crops with that system. It may also be influenced by factors such as the type of crop, or potential disease problems. Some crops, such as roses and mums, require closely controlled humidity and a considerable amount of air circulation to prevent leaf mildew. If a radiant floor system is used, auxiliary circulating fans will be required. Tropical and subtropical potted plants, on the other hand, may require high humidity and higher soil temperatures. In this case, a radiant, under the bench system will be preferred, perhaps combined with an overhead air system for snow melting, in order to get maximum sunlight during winter months in areas of high snow fall. Certain flowering plants may require shading to control blooming, thereby enabling the grower to market at the most opportune time. The type and location of the shading cover can affect the placement of heating and air handling equipment and, perhaps, the type of heating.

All these things should be taken into consideration and the heating system designer should maintain close communication with the grower in the selection of type and the placement of heating devices.

The following paragraphs outline the performance of the heating systems mentioned above.

14.4.1 Heat Exchangers

In most geothermal applications, a heat exchanger is required to separate actual heating equipment from the geothermal fluid. This is because of the scaling and corrosion associated with most geothermal fluids. Generally, the heat exchanger is placed between two circulating loops, the geothermal loop and the clean loop, as shown in Figure 14.2.

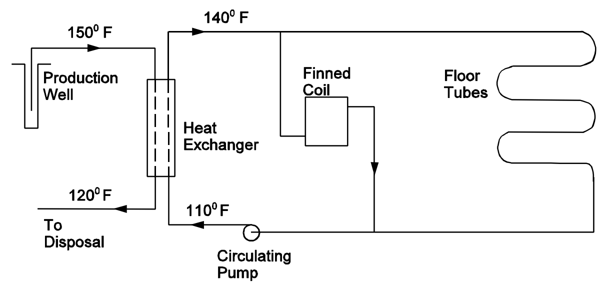


Figure 14.2 Heat exchanger schematic.

As a result of this heat exchanger, there is some loss in the temperature of the fluid available for use in the actual heating equipment. This temperature loss depends upon the type of heat exchanger used. For plate-type heat exchangers, a temperature of 5 to 10°F should be applied, for shell and tube heat exchangers 15 to 20°F, and for homemade configurations 20 to 40°F. For example, assuming a geothermal resource temperature of 150°F is available, use of a plate heat exchanger would result in 140°F supply water, as shown in Figure 14.2.

Now that the heating requirement and supply water temperature has been established, various heating systems can be evaluated with respect to their ability to meet this demand. For geothermal applications, the available geothermal resource temperature has a large impact upon the system chosen. This is a result of the fact that certain types of heating methods yield better results with low-temperature fluid than others.

Table 14.5 Steam and Extended Hot Water Ratings[a] (Bare Element)

Bare Heating Elements	Rows	Hot Water Ratings, Btu/h/lf Average Water Temperature 240°F	230°F	220°F	210°F	200°F	190°F	180°F	170°F
	1	1630	1480	1370	1240	1120	1010	900	790
33 fins/ft	2	2810	2570	2360	2140	1940	1760	1550	1370
	3	3660	3340	3080	2780	2520	2290	2020	1790
40 fins/ft	1	1750	1600	1470	1330	1220	1090	970	850
	2	2930	2670	2460	2220	2010	1830	1610	1430

a. Vulcan, 1976

Finned Pipe

As the name implies, finned pipe is usually constructed of steel or copper pipe with steel or aluminum fins attached to the outside. These fins can either be circular, square or rectangular in shape. In the size range employed in greenhouses, the steel pipe with steel fins is most common.

Since most finned-pipe heating equipment used in geothermal projects was originally designed for standard hot water use, heating capacity is generally based upon 200°F or higher average water temperature and 65°F entering air temperature. If the available supply temperature from the geothermal system is less than the 200°F value, the capacity of the heating equipment, in this case finned pipes, will be less than the rated value. In addition, heating capacity of finned pipe, usually expressed in Btu/h per lineal foot, is influenced by fin size, pipe size and flow velocity. Table 14.5 shows one manufacturer's rating for equipment.

Table 14.6 shows the appropriate de-rating factors to be applied for average water temperatures of <190°F.

Table 14.6 Derating Factors (Vulcan, 1976)

Average Water (°F)	Factor
180	0.80
160	0.62
140	0.47
120	0.30
100	0.17

It is important to note that the capacity of this equipment is indexed to <u>average</u> water temperature, not supply water temperature. In order to find average water temperature (AWT), it is first necessary to calculate the temperature drop (ΔT), which is found according to the following relationship:

$$\Delta T = q/(500 \times Q)$$

where

ΔT = temperature drop (°F)
q = heating requirement (Btu/h)
500 = constant, Btu/h gpm (°F)
Q = flow rate (gpm).

Using the greenhouse example from above, with a requirement of 495,980 Btu/h, assume a 150°F resource, a flow of 50 gpm, and the use of a plate-type heat exchanger.

$$\Delta T = (495{,}980 \text{ Btu/h})/(500 \text{ Btu/h gpm °F} \times 50 \text{ gpm})$$
$$\Delta T = 20°F$$

With a 150°F resource and a 10EF loss across the heat exchanger, this results in a 140°F supply temperature (T_s). Since a 20°F drop from supply to return was calculated, the average water temperature is then:

$$AWT = T_s - (\Delta T/2)$$
$$AWT = 140°F - (20°F/2)$$
$$AWT = 130°F.$$

This provides the information required to select the necessary length of finned-pipe heating element required. Using Table 14.5, for a 2-in. steel element having 4-1/4 in.(1 row) square fins spaced at 33/ft, output at 200°F AWT (factor of 1.00) is 1120 Btu/h lf. Using an interpolated correction factor of 0.385 from Table 14.6, actual capacity will be 0.385 x 1120 Btu/h lf = 431 Btu/h lf at the 130°F AWT.

With this value and the heating requirement of 495,980 Btu/h, calculate the length (l) of element required as:

$$l = (495{,}980 \text{ Btu/h})/(431 \text{ Btu/h lf})$$
$$l = 1{,}151 \text{ ft.}$$

This large length requirement points up the limitation of finned pipe with respect to low temperature. As average water temperature falls below about 150°F, large lengths of finned element are required to meet the heating load in colder regions. As a result, finned pipe is not a particularly good choice for low-temperature resources.

Finned elements are generally installed along the long dimension of the greenhouse adjacent to the outside wall. Improved heat distribution is achieved if about one-third of the total required length is installed in an evenly spaced pattern across the greenhouse floor (ASHRAE 1978). This system has the disadvantage of using precious floor space that would otherwise be available for plants. In addition, it is less capable of dealing effectively with ventilation if it is required. Maintenance requirements are low, particularly if a heat exchanger is used. In addition, the natural convection nature of the finned pipe system does not increase electrical costs as a result of fan operation.

The costs for finned pipe elements are a function of the type and size of piping (steel or copper), and fin spacing (fins/ft). It is not possible to present costs for all combinations of these characteristics; however, Table 14.7 should serve to illustrate cost trends in fin pipe equipment.

For labor cost estimating, a value of 0.25 to 0.35 man hours per lineal foot can be employed for finned pipe installation (Khashab, 1984).

Table 14.7 Comparative Costs of Finned Pipe Elements (Means, 1996)

Element	Cost/lf ($)
Copper/aluminum (3/4 in., 33 fin/ft)	5.40
Copper/aluminum (1 in., 33 fin/ft)	7.50
Steel/steel (1-1/4 in., 33 fin/ft)	11.00
Steel/steel (1-1/4 in., 40 fin/ft)	12.30
Steel/steel (2 in., 24 fin/ft)	10.80
Steel/steel (2 in., 33 fin/ft)	12.60

Standard Unit Heaters

Unit heaters consist of a finned coil and small propeller fan contained in a pre-designed unit. These units are available in either horizontal or vertical configurations and are generally hung from the greenhouse structure at roof level (see Chapter 12, Figures 12.24 and 12.25). Air is discharged either directly into the greenhouse or into a perforated plastic distribution tube ("poly tube").

As with the finned pipe equipment, unit heaters are generally rated at 200°F entering water temperature (EWT) and 60EF entering air temperature (EAT). Changes in either of these two parameters will affect unit capacity (usually expressed in Btu/h). Since most geothermal resources applied to greenhouses are <200°F, some adjustment of unit capacity is necessary. Table 14.8 shows a typical set of manufacturer's performance data for unit heaters at standard conditions (200°F EWT/60°F EAT). To adjust for other conditions, Table 14.9 values are employed. It is important that the gpm values shown in Table 14.8 are met. Providing a unit with a flow less than that shown will decrease capacity.

Table 14.8 Hot Water Unit Heater Ratings[a] (Modine, 1979)

Model	Btu/h	GPM	CFM	Final Air Temp.	HP
A	90,000	9.0	1775	110	1/6
B	133,000	13.4	3240	100	1/3
C	139,000	14.0	2900	107	1/3
D	198,000	20.0	4560	102	1/2
E	224,000	22.0	4590	108	1/2
F	273,000	27.0	5130	108	1/2

a. Standard Conditions, 200°F EWT/60°F EAT.

Because these units are generally constructed with copper tubes, even very small concentrations of dissolved hydrogen sulphide (H_2S) or ammonia (NH_3) will result in rapid failure. In addition, the long path through which the water must flow in the unit heater can result in scaling if the fluid has this tendency. As a result, a unit heater system should not be applied without an isolation heat exchanger.

Using information from the example greenhouse, unit heaters can be selected to meet the heating requirement. Example conditions are given in Table 14.10.

From Table 14.9, find a correction factor of 0.571. This factor is then applied to the capacity values shown in Table 14.8 to adjust them to the system conditions.

Table 14.9 Unit Heater Correction Factors[a] (Modine, 1979)

	EAT(°F)			
EWT (°F)	40	60	80	100
80	0.293	0.143	-0-	-0-
100	0.439	0.286	0.140	0.069
120	0.585	0.429	0.279	0.137
140	0.731	0.571	0.419	0.273
160	0.878	0.714	0.559	0.410
180	1.024	0.857	0.699	0.547
200	1.170	1.000	0.833	0.684

a. To be applied to standard ratings.

Table 14.10 Unit Heater Example Conditions

Condition	Value
Load	495,980 Btu/h
Resource temperature	150°F
Heat exchanger loss	10°F
Supply water temperature	140°F (150-10°F)
Greenhouse inside design temp.	60°F

For greenhouses over 50 ft in length, it is advisable to place unit heaters at each end to allow for better heat distribution. Assuming two units are used in this case, each would need a capacity (q) of:

$$q = (495{,}980 \text{ Btu/h})/2 = 247{,}990 \text{ Btu/h}.$$

To convert this to an equivalent in Table 14.8, dividing by the above correction factor of 0.571:

$$q = (247{,}990 \text{ Btu/h})/0.571 = 434{,}308 \text{ Btu/h}.$$

A two-unit system will not work because the largest unit capacity for a horizontal configuration is 273,000 Btu/h. The next step is to try a four-unit system--two-unit heaters at each end of the house. In this case, each unit would have a capacity of:

q = (434,308 Btu/h)/2 = 217,154 Btu/h.

This results in half the capacity calculated for the single unit above.

The proper selection would be the "E" unit at a capacity of 224,000 Btu/h. This is slightly more than the required 217,154 and will allow for a margin of safety in the design. As shown, the flow requirement (Q) for the four units will be:

Q = 22 gpm x 4 units = 88 gpm.

If the available flow rate is less than this value, unit capacity would have to be corrected for operation at this reduced flow, possibly resulting in the need for additional units.

Two types of hot-water unit heaters are commonly used in greenhouse applications: horizontal and vertical. Of these two configurations, the horizontal unit is the more common. Vertical unit heaters are generally available in larger capacities than the horizontal units. In addition to the unit heater itself, a "poly tube" adapter is frequently required to attach the distribution system to the front of the heating device. Prices for each of these items are shown in Table 14.11. Capacities for unit heaters are based on 200°F entering water temperature.

Table 14.11 Horizontal and Vertical Unit Heater Costs[a]

Horizontal Unit Heaters	
Capacity[b] (MBH)	Cost ($)
23	822
44	874
66	995
97	1210
133	1294
153	1294
198	1581
257	1811

a. Means, 1996.
b. 1000 Btu/h.

Poly tube adapter costs are given in Table 14.12

Table 14.12 Poly Tube Adapter Costs[a](1996)

Size (in.)	Cost ($)
12	100
18	115
24	175

a. Roper, undated.

Low-Temperature Unit Heaters

Low-temperature unit heaters are similar to standard unit heaters; but, their design is optimized for low-water temperature operation. These units incorporate a more effective water coil and a higher capacity fan. They are larger and heavier than standard unit heaters, and in some applications, may require additional support if suspended from the ceiling. These units are horizontal in configuration and use a propeller-type fan.

Performance of the low-temperature unit heaters falls between that of standard unit heaters and fan-coil units. Performance data for this equipment appear in Table 14.13. Costs appear in Table 14.14.

As indicated in the table, this equipment is rated in terms of its capacity per degree of entering temperature difference (ETD). Entering temperature difference is calculated by subtracting the space air temperature from the supply water temperature. For a greenhouse maintained at 60°F with a supply water temperature of 125 °F, an ETD value of 65°F would result.

Table 14.13 Low-Temperature Unit Heaters Performance Data (Modine, 1985)

Water	Btu/EF of Entering Temperature Difference	
Flow (gpm)	Single Fan (3850 cfm)	Two Fan (7700 cfm)
5	1500	2500
10	2200	3600
15	2500	4300
20	2750	4900
25	2850	5300
30	3000	5650
35	3100	5800
40	3100	6000

Based on the example, greenhouse heat loss of 495,980 Btu/hr, a 125°F supply water temperature, and a 30 °F ΔT, the following calculations can be made:

System flow rate = 495,980 Btu/hr ÷ (500 · 30)
=33.1 gpm

Using two units, the single fan rate would have a capacity of:

33.1 ÷ 2 = 16.6 gpm ea.

From Table 14.13:

Interpolate for capacity @ 16.6 gpm
= 2,580 Btu/hr °F ETD
Capacity = 2,580 · 65
= 167,700 Btu/hr

Number of units required:

= 495,980 Btu/hr ÷ 167,700
=2.96 or 3 units

Two-fan units:

Capacity @ 16.6 gpm = 4,492 Btu/hr °F ETD
@ 65°F ETD capacity
= 4,492 · 65
= 291,980 Btu/hr

Number of units required:

= 495,980 Btu/hr ÷ 291,980 Btu/hr
= 1.70 or 2 units.

Table 14.14 Cost Data for Low-Temperature Unit Heaters

Single fan unit	$2,800
Two-fan unit	$5,100

Fan Coil Units

These units are similar to the standard unit heater discussed previously. They consist of a finned coil and a centrifugal blower in a single cabinet. A few manufacturers offer units in an off-the-shelf line for low temperature greenhouse heating. It is much more common that they are custom selected. The difference between the fan coil unit and the hot-water unit heater is primarily in the coil itself. In the fan coil system, the coil is much thicker and usually has closer fin spacing than the coil in a unit heater. Unit heaters generally have only a one or two row coil. A custom designed coil can have as many as six or eight rows. The additional rows of tubes create more surface area. The added surface area allows for more effective heat transfer, resulting in the ability to extract more heat from the water. To illustrate this, consider the unit heater selected in the previous section. Conditions are given in Table 14.15.

Table 14.15 Unit Heater Example[a] (two row)

Condition	Value
Capacity	127,904 Btu/h (0.571 x 224,000)[a]
Air flow	4,590 cfm[b]
Water flow	22 gpm
Supply water temperature	140°F
Leaving water temperature	128.4°F
Leaving air temperature	85.8°F

a. Model E unit heater.
b. Cubic ft/min.

Supplying the same temperature water to a fan coil unit with a four-row coil would result in the values as shown in Table 14.16.

Table 14.16 Fan Coil Example[a] (four-row)

Condition	Value
Capacity	275,171 Btu/h
Air flow	4,590 cfm
Water flow	13.76 gpm
Supply water temp.	140°F
Leaving water temp.	100°F
Air in temp.	60°F
Air out temp.	115°F

a. Four-row coil with 11 fins/in., 2.5 ft x 3.67 ft.

Using only 60% of the water flow, the fan coil unit has the capability to more than double the heat output. In addition, the leaving air temperature is raised to 115°F from 85.8°F.

This benefit is not without cost, however. The fan coil units are generally larger and more bulky than the hot-water unit heater. As a result, they cost more. The larger coils discussed above generally require a larger fan motor to push the air through the added coil resistance. In this case, the unit heater would require a 0.5 horsepower (hp), motor and the fan coil unit would require a 1 hp motor. These factors may be compensated for by increased capacity, thus requiring fewer units.

The ability to extract more heat from each gallon of water pumped reduces well pumping requirements and allows the development of more greenhouse area, using the same resource. As a general rule of thumb, a well designed coil can cool water down to within about 15 to 25°F of the space temperature. For example, if a greenhouse is to be maintained at 60°F and the coils are supplied with water at 120°F, a system ΔT of 120°F - (60°F + 25°F), or 35°F could be achieved. If the well flow is known, then the total heat supplied (q) can be calculated as:

$$q = 500 \times gpm \times \Delta T = Btu/h.$$

This figure can then be compared to greenhouse heat loss to find the total area of greenhouse that can be developed.

The fan coil construction is very similar to that of the unit heater. For the same reasons, it is recommended that they be applied with an isolation heat exchanger. The fan-coil system is the most cost effective method for extracting large quantities of heat from very-low-temperature heating mediums.

Table 14.17 presents pricing information for fan coil equipment.

Table 14.17 Fan Coil Unit Prices (Means, 1996)

Unit (cfm)	Nominal Capacity[a] Btu/hr	Cost ($)
2000	120,000	1750
4000	240,000	2500
6000	360,000	3500
8000	480,000	4500

a. @ 115°F supply air temperature

As with the unit heater, a poly tube adapter would be required if this equipment is to be attached to such a distribution system. For prices, see Table 14.12.

Soil Heating

This system generally involves using the floor of the greenhouse as a large radiator. Tubes, through which warm water is circulated, are buried in the floor of the greenhouse. Heat from warm water is transferred through the tube to the soil and, eventually, to the air in the greenhouse.

In the past, tube materials were generally copper or steel. Because of corrosion and expansion problems with these materials, nonmetallic materials have seen increasing application in recent years. The most popular of these is polybutylene. This material is able to withstand relatively high temperatures (up to - 180°F) and is available in roll form for easy installation. PVC piping is only available in rigid form and is limited with respect to temperature. Polyethylene and similar materials are available in flexible roll form, but are (as PVC) generally limited in terms of temperature handling ability.

A soil heating system is preferred by many operators because it results in very even temperature distribution from floor to ceiling and does not obstruct floor space or cause shadows. However, its ability to supply 100% of the heating requirements of a greenhouse necessitates a rather mild climate and a low inside design temperature. This is caused by the nature of heat transfer in the system. As heating requirements are increased, the required heat output from the floor is increased. In order to produce more heat, the floor surface temperature must be increased. Very quickly a point is reached at which it is difficult to spend extended periods on such a hot floor. In addition, if plants are grown on or near the floor (including benches), heat transfer to the plants may be excessive with a radiant floor system. As a result, this system is generally employed in conjunction with another system such as unit heaters. The floor system supplies the base load for the greenhouse and the secondary system is used for occasional peaking purposes.

The procedure for designing a floor system consists of:

1. Determining the heat load for the greenhouse.
2. Calculating the required floor temperature to meet the load.
3. Calculating the required size, depth and spacing of the tubes.

The load analysis portion of the procedure has been covered. The next step is to determine the required floor surface temperature.

The heat output of the floor (usually expressed in Btu/h ft^2) is a function of the floor surface temperature, greenhouse air temperature and average temperature of unheated surfaces in the room (AUST). Heat output from the floor occurs by two mechanisms: convection and radiation.

After the heat loss of the greenhouse has been calculated, it is divided by the area of the floor which will be used for heating purposes (usually about 10% less than the actual floor area). Using the previous greenhouse example, 42 ft x 120 ft, with a total heat loss of 495,980 Btu/h, the value for heat loss (q/A) is:

$q/A = (495{,}980\ \text{Btu/h})/(42\ \text{ft} \times 120\ \text{ft} \times 0.90)$
$q/A = 109.4\ \text{Btu/h ft}^2$.

This value is then used in the following equation to solve for the required floor surface temperature (ASHRAE, 1984):

$$q/A = 0.15x\left[\left(\frac{T_f+460}{100}\right)^4-\left(\frac{AUST+460}{100}\right)^4\right]+0.32(T_f-T_a)^{1.32}$$

where

T_f = floor surface temperature
T_a = indoor air temperature.

Before the above can be solved for T_f, a value for AUST must first be calculated. As mentioned earlier, AUST is the area weighted average temperature of unheated surfaces in the room. For a greenhouse, these surfaces are the walls and roof.

Inside surface temperature can be calculated according to the formula below. Referring back to the heat loss example, the greenhouse is constructed of both double poly (roof) and single fiberglass (walls). The calculation for AUST is:

$IST = IDT - ((0.595/(1/U)) \times \Delta T)$

where

IST = inside surface temperature (°F)
IDT = inside design temperature (°F)
U = glazing material U factor, Btu/h ft^2 (°F)
ΔT = design temperature difference (°F).

For the example greenhouse, the inside surface temperature of the double poly roof area is:

IST = 60°F - ((0.595/(1/0.70)) x 60°F)
IST = 35.0°F.

The inside surface temperature for the single fiberglass area is:

IST = 60°F - ((0.595/(1/1.0)) x 60°F)
IST = 24.3°F

$AUST = (A_1 \times IST_1 + A_2 \times IST_2)/(A_1 + A_2)$

$$AUST = \frac{(5{,}340 ft^2\ x\ 35^oF) + (3{,}174\ ft^2\ x\ 24.3^oF)}{(5{,}340\ ft^2 + 3{,}174\ ft^2)}$$

AUST = 31.0°F

This value can now be inserted into the equation for floor temperature developed by ASHRAE as:

$$q/A = 0.15((T_f + 460/100)^4 - (31.0 + 460/100)^4) + (0.32(T_f - 60)^{1.32}) = 109.4\ \text{Btu/h ft}^2$$

Solving for T_f:

T_f = 103°F.

This means that in order to meet the peak demand, a floor surface temperature of 103°F would be required. Plants could not be grown on or near such warm soil. In addition, the amount of time that workers could be exposed would be limited. As a result, it would be advisable to supply a portion of the design capacity with this system and the rest with a secondary system. If the system is designed for only 60% of peak requirements (65.5 Btu/h ft^2), a floor temperature of only 84 °F would be required. This figure is close to the maximum recommended floor surface temperature of 85°F for occupied areas. If the greenhouse is occupied only for brief periods, this value can be exceeded somewhat. A secondary system would be used for peaking.

The next step is to determine the depth and spacing of the tubes supplying the heat. Tube spacing and size is dependent upon the available water temperature. Generally, depth is more a function of protecting the tubes from surface activity than system design, and a figure of 2 to 6 in. below the surface is common.

Since it is the purpose of the floor panel system to use the floor as a large radiator, it follows that the installation of the tubing should result in as uniform a floor surface temperature as possible. This is accomplished by two general approaches: (a) placing smaller diameter tubes at close spacing near the surface of the floor, or (b) placing larger tubes spaced further apart at a greater burial depth. The theory behind this approach is to reduce the difference between the distance heat must travel vertically (from the tube to the surface directly above it) and laterally (from each tube to the surface between the tubes)(Adlam, 1947).

The depth at which the tubes are to be buried is often a function of protecting them from surface activity. For burial in the soil floor of a greenhouse, a depth of at least 2 to 3 in. should be employed. If crops are to be grown directly in the soil, depth requirements are such that this type of system becomes impractical.

Tubing size is a function of heating requirements. Common sizes are ½ in., 3/4 in. and 1 in. with the smaller sizes used generally in the 2 to 4 in. depth and the larger lines for depths of 5 in. and greater.

The final determination of the size and spacing is a function of heat output (Btu/ft^2) required, mean water temperature, soil conductivity, and burial depth.

The required heat loss is fixed by the type of greenhouse construction used. Soil conductivity is also fixed by site characteristics. As mentioned earlier, the minimum burial depth is fixed by surface activity. As a result, the choice of size and spacing is balanced against mean water temperature, the single parameter over which the designer has some control. Table 14.18 lists some maximum mean water temperatures for various situations. Employing mean water temperatures above these values will result in floor surface temperatures greater than 90^{0}F. If workers are to spend extended periods in the greenhouse, floor surface temperatures above this value would be unacceptable.

Table 14.18 Maximum Recommended Mean Water Temperatures (ºF)

Burial Depth (in.)	Steel Pipe k = 0.5	Steel Pipe k = 0.75	Polybutylene Tube k = 0.5	Polybutylene Tube k = 0.75
1	111	105	124	112
2	116	110	131	120
3	122	115	139	128
4	125	117	144	131
5	128	120	148	135
6	134	125	156	142

a. k = soil conductivity in Btu/hr ft ºF

In addition to the maximum mean water temperature, it is also important when making this calculation to be aware of system ΔT (supply temperature minus return water temperature) and its impact upon system design. Temperature drops above approximately 15ºF should employ a double serpentine to balance the circuit output. For ΔT below 15ºF, a single serpentine can be used as shown in Figure 14.3.

Using the heating requirement and floor surface temperature calculated above, some combinations of tubing size and spacing can be determined. It will be assumed that, because of surface activity, the tubes would have to be buried a minimum of 3 in. below the surface. Soil conductivity is 0.75 Btu/h ft^2 ºF. Resource temperature is 140ºF and a flow of 60 gpm is available. Polybutylene tubing will be employed. Plate heat exchanger loss is 7ºF.

As a result of the heat exchanger loss, 133ºF fluid will be available for supply. If the entire flow is used, the system ΔT would be:

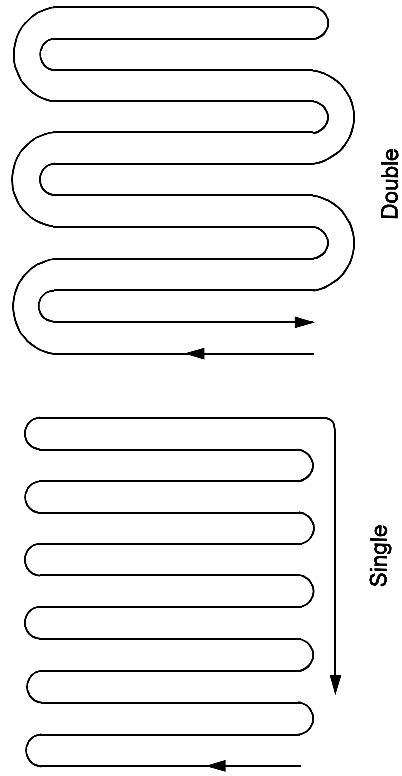


Figure 14.3 Single- and double-serpentine piping layout.

ΔT = (297,108 Btu/h)/(500 Btu/h gpm ºF x 60 gpm) = 9.9ºF

The resulting mean water temperature (Tw) would be:

Tw = 133ºF - (9.9ºF/2) = 128ºF

This value is equal to the recommended maximum mean water temperature found in Table 14.18, so design can proceed. If this value had been above the recommended temperature, either the tubes would have to be buried deeper or the radiant floor system operated at a lower supply-water temperature.

Subtracting the required floor surface temperature from the mean water temperature results in the tube-to-surface temperature difference. Using this and the value from Figure 14.4, the heat out-put per lineal foot (lf) of tube can be determined. From Figure 14.4, for a burial depth of 3 in., a value of 1.60 Btu/h lf ºF for 3/4 in. tubing results. For 1 in. tubing due to greater surface area, the value would be (1.60 x 1.00/0.75) = 2.13 Btu/h lf ºF.

The heat output per lf for each of these tubes would be arrived at by multiplying the Btu/hr lf ºF value times the tube-to-surface temperature difference.

For 3/4 in. tube: 1.60 x (128ºF - 84ºF) = 70.4 Btu/h lf

For 1 in. tube: 2.13 x (128ºF - 84ºF) = 93.7 Btu/h lf

The tube spacing is determined by dividing the tube output per lineal foot into the heating requirement (per square foot).

For 3/4 in. tube: (65.5 Btu/ft^2 h)/(70.4 Btu/h lf)
= 0.93 lf/ft^2

For 1 in. tube: (65.5 Btu/ft^2 h)/(93.7 Btu/h lf)
= 0.70 lf/ft^2.

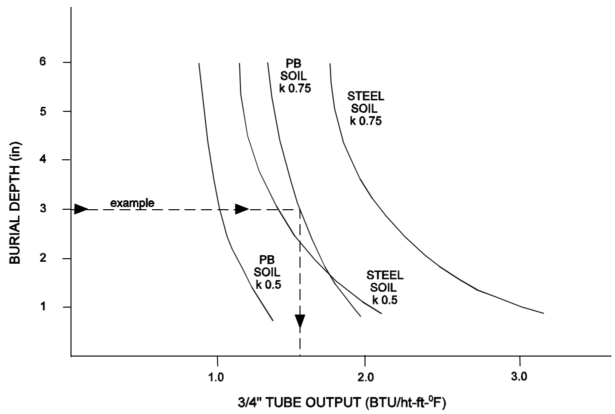


Figure 14.4 Heat output for radiant floor system.

Taking the inverse of the above results and multiplying by 12 in./ft yields tube spacing:

For 3/4 in. tube: (1/0.93) x 12 = 12.9 in.

For 1 in. tube: (1/0.70) x 12 = 17.1 in.

In most cases, because of losses downward and at the edges, a safety factor of 10 to 15% is added to the tube requirements. This is most conveniently accomplished by reducing the tube spacing by 10 to 15%.

In order to demonstrate the sensitivity of the system to other parameters, Table 14.19 shows some additional tube spacing calculations that are made:

Table 14.19 Tube Spacing (in.)

				Tubing	
	Soil k	Depth	MWT	3/4 in.	1 in.
Base case	0.75	3	128	12.9	17.1
	0.5	3	128	8.10	10.8
	0.75	6	128	10.23	13.6
	0.5	3	118	8.94	11.9
	0.5	6	118	5.43	7.2

Using the base case tube spacing and 3/4 in. tubes, a total of 4,218 ft of tubing will be required. In order that a reasonable pressure drop will be attained, the total 60 gpm flow would be divided among a number of individual circuits. At a velocity of approximately 3 ft/s, each circuit would carry 5 gpm. This would require 12 circuits for the total flow. If the 1 in. tubing is used, a smaller number of higher flow circuits could be employed.

As suggested above, a heat exchanger is used in this case. This is for two reasons: protection from scaling and control of temperature.

Control of temperature is the most critical. The only method of controlling the output of a floor system is by controlling the water temperature in the tubes. The use of a heat exchanger allows this control to be carried out more easily. The flow of geothermal fluid to the exchanger is regulated to maintain a given supply temperature to the heating loop as shown in Figure 14.2.

As suggested in the example, a great deal of piping material is required to supply just 60% of the peak requirement of a greenhouse in a cold location. In addition, the inability to grow directly in or on the soil surface also restricts the wide acceptance of this type of system.

The cost of both polybutylene and polyethylene piping is a function of pipe size and the standard dimension ratio (SDR). The SDR is related to the nominal pipe size divided by the wall thickness, or as the SDR increases, the wall thickness decreases. Material costs shown in Table 14.20 are for SDR 11. This material is rated at 100 psi at 180ºF (polybutylene) and 160 psi at 70ºF (polyethylene).

Table 14.20 Polyethylene and Polybutylene Pipe Costs (Means, 1996)

Size (in.)	Polybutylene ($/lf)	Polyethylene ($/lf)
1/4	0.32	-
3/8	0.40	-
1/2	0.40	0.20
3/4	0.74	0.29
1	1.25	0.44

Bare Tube System

This system involves the use of bare tubing, usually small diameter polybutylene or similar material. The tubing is installed either on the floor or suspended under benches. It is preferable for the tubing to be located low in the greenhouse, although a portion may be located overhead. Regardless of the installation location, it is very important that the tubing be arranged such that each tube is separated from the others. If the tubes are bunched together, the effective surface area of each is reduced, thus lowering heating capacity.

In colder regions, this system encounters the same problem as the floor panel system in that large quantities of tubing are required to meet the design requirement.

Control of the system in many cases has been manual by way of gate valves. However, as with the floor panel system, the use of a heat exchanger can allow accurate control of temperature and, hence, output. Design of a system is based upon the average water temperature of the heating loop. For a system using a heat exchanger:

1. Determine the flow of geothermal fluid available. We will assume 80 gpm at 150°F for the example case.

2. Calculate the greenhouse heat loss; i.e., 495,980 Btu/h for the example.

3. Determine the temperature drop in the available water flow:

   $\Delta T = q/(500 \text{ x gpm})$
   $\Delta T = (495{,}980 \text{ Btu/h})/(500 \text{ Btu/h gpm }^{o}\text{F x } 80 \text{ gpm})$
   $\Delta T = 12.4^{o}\text{F}$.

4. Determine heating loop average water temperature (AWT) using:

   Ts = Tg - 10°F

   where

   Ts = supply temperature (°F)
   Tg = geothermal resource temp. (°F)
   Ts = 150°F - 10°F
   Ts = 140F

   $\text{AWT} = \text{Ts} - (\Delta T/2)$
   AWT = 140°F - (12.4°F/2)
   AWT = 134°F

5. Calculate heat output per foot of tubing based on the average water temperature (AWT) using:

   $$q/l = ((1.016 \text{ x } (1/D)^{0.2} \text{ x } (1/T_{avg})^{0.181} \text{ x } (?\ T^{1.266})) + ((15.7 \text{ x } 10^{-10}) \text{ x } (T_1^{\ 4} - T_2^{\ 4}))) \text{ x ft}^2/\text{lf pipe}$$

   where

   D = tube outside diameter (in.)
   $T_{ave} = 460 + (\text{AWT} + T_{air})/2$
   $\Delta T = \text{AWT} - (T_{air} + 3^{o}\text{F})$
   $T_1 = 460 + \text{AWT}$
   $T_2 = 460 + T_3$
   $T_3 = (\text{AUST} + T_{air})/2$

Using a 3/4 in. tube, 60°F air temperature and 134°F AWT, Btu/h lf for the example case:

$$((1.016 \text{ x } (1/1.05)^{0.2} \text{ x } (1/557)^{0.181} \text{ x } (71)^{1.266}) + ((15.7 \text{ x } 10^{-10}) \text{ x } ((594)^4 - (505)^4))) \text{ x } (0.275)$$

q/l = 45.1 Btu/h lf

The total length (l) required to meet the design load becomes:

l = (495,980 Btu/h)/(45.1 Btu/h lf)
l = 10,997 lf

This length requirement can then be compared to requirements for other tubing sizes and water temperatures to determine the most economical system.

Costs for polybutylene and polyethylene piping used in the bare tube system are shown under the previous section.

The procedures presented in this chapter are intended to familiarize the reader with some of the considerations appropriate to greenhouse heating systems. It is strongly recommended that the services of a consulting engineer be retained for final design purposes.

14.5 PEAKING WITH FOSSIL FUEL

To this point, design methods in this chapter have been based upon meeting 100% of the peak load with the geothermal heating equipment. Under some circumstances, a strategy in which the geothermal system is designed for less than 100% of the peak may be worthwhile.

A situation where this may be considered is one in which a grower wishes to expand an existing operation, but is faced with limited resource flow. Using low-temperature effluent from the existing facility, it may be difficult to configure a system which will meet the peak load, particularly with bare tube-type terminal equipment. In this case, designing the geothermal system for 50 to 70% of the peak and meeting the remaining load with a conven-tional system may have some merit. In most climates, this design will still allow the geothermal to meet 95% or more of the annual heating energy requirement.

14.5.1 <u>Climate Considerations</u>

The rationale behind using different base load and peak load heating systems lies in the annual temperature profile. Figure 14.5 presents a comparison of the number of hours per year at various temperatures. It is apparent that the annual number of hours at very low outside temperatures is quite low compared to the number of hours at more moderate tempeatures.

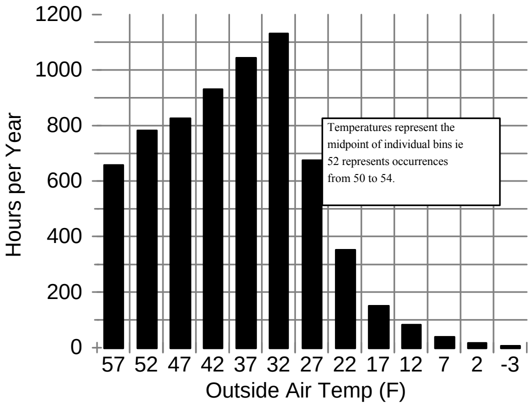


Figure 14.5 Temperature occurrences, Klamath Falls, Oregon.

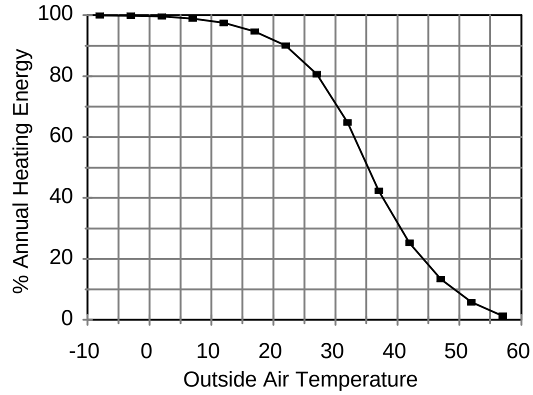


Figure 14.6 Annual heating energy requirement.

This data is arranged in 5°F increments (i.e., 70E to 74 °F). These 5°F increments are known as temperature "bins" and data from which it comes is referred to as bin data. Bin data for many locations in the U.S. is published by the Defense Department in Engineering Weather Data, AFM 88-29, 1978.

It is apparent from Figure 14.5, that a system designed for 100% of the peak load actually operates at those conditions for only a very few hours per year. As a result, a system designed for 100% of the peak load is grossly underutilized.

The amount of energy required to heat a building (on an annual basis) is determined by the number of hours occurring at outside temperatures less than the temperature maintained in the structure. The quantity of annual energy required at a particular temperature bin is determined by the number of hours at that bin and the temperature difference between it and the inside temperature of the structure. Summing the number of hours at various outside temperatures permits the development of a cumulative heating requirement curve similar to that in Figure 14.6 . This particular plot was developed for an inside temperature of 60°F using the weather data from Figure 14.6. The plot indicates the percentage of annual heating requirements occurring above (or below) a particular outside air temperature. For example, reading vertically from 30°F to the intersection with the curve and then horizontally to the axis, yields a figure of approximately 71%. That is, 71% of the annual heating requirement occurs at this design temperature.

This is significant since the normal design temperature in the Klamath Falls area is 0°F. A system designed for 30°F would be only 50% the size of a system designed for 100% of the load (IDT 60°F). Despite this, it could capture 71% of the annual heating requirements. In addition to this, the down-sized system would capture most of the remaining 29% of heating energy requirement by operating in parallel with a peaking system.

Figure 14.7 presents a plot of the annual energy requirements which could be met by a base load system designed for various percentages of the peak load. This plot assumes that the base load system continues to operate (at its maximum capacity) in parallel with the peak load system below the balance point. The 50% (of peak load) system described above would capture approximately 93% of the annual heating requirements of the structure (assuming a 60°F IDT, 0°F ODT and Figure 14.5 weather data).

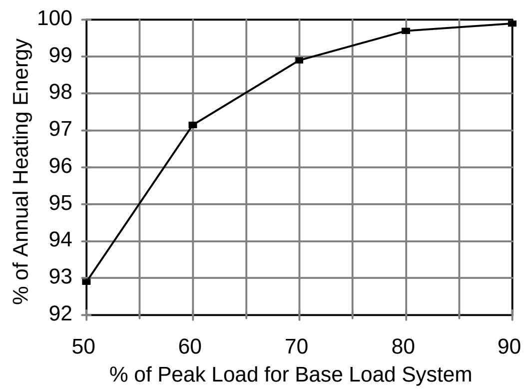


Figure 14.7 Annual heating energy capture, 60°F inside temperature, Klamath Falls, Oregon.

It is clear that due to the nature of temperature occurrences, the base load heating system is capable of meeting only half the peak heating requirement and still meets more than 90% of the annual heating energy needs of a structure.

14.5.2 <u>Peaking Equipment Capital Costs</u>

Two broad approaches are available for the use of conventionally-fired peak heating equipment in a hot-water greenhouse heating system: individual unit heaters and central peaking boiler.

Individual unit heaters offer the advantage of zero floor space requirements (since they can be hung from the ceiling). Because each unit requires accessory equipment (flue pipe, thermostat, distribution "poly tube", fuel line, electrical connection, etc.), the cost of a given amount of heating capacity is relatively high in comparison to the boiler approach. This affect is compounded by the need to use a large number of units to assure adequate air distribution. For example, consider a 1-acre greenhouse for which a peaking system capacity of 1,300,000 Btu/hr is required. Although it is possible to supply this capacity with just three or four large units, to assure adequate air distribution, a minimum of 8 or 10 units should be employed. Costs for unit heater capacity assuming 10 units per acre appear in Figure 14.8.

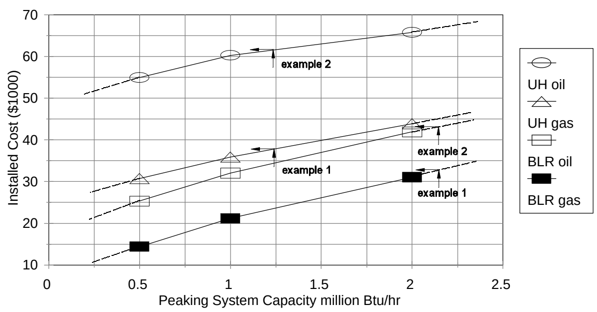


Figure 14.8 Peaking equipment costs.

The costs shown include, for the propane- (or natural gas), fired unit heaters (UH gas): unit heater (blower type), installation, flue pipe and cap, thermostat and wire, fuel distribution pipe (inside greenhouse), and electrical connection (120 v). Costs for the oil unit heater (UH oil) equipment reflects the much higher cost for this type of unit and includes the cost of a double-wall oil storage tank (2500 gal). Oil-fired unit heaters are much more expensive (50 - 80% depending upon size) than equivalent capacity gas-fired units. This fact along with the cost of the oil tank tends to push the cost of the oil-fired unit heater system far above the other alternatives. All unit heater equipment costs assume the use of blower-type units.

The central boiler (BLR) approach involves the installation of a peaking boiler downstream of the geothermal heat exchanger. The boiler's function is to boost the supply water temperature to the heating equipment during the peak load period. The higher water temperature allows a down-sized tubing system to provide the required capacity to meet the space heating requirement. Because only a single piece of equipment (along with its accessory equipment) is required, the cost of a given heat output is much lower than for the unit heater equipment cited above. Figuer 14.8 presents costs for both propane- (BLR gas) and oil-fired (BLR oil) cast iron boiler equipment. These costs include boiler, stack, electrical connection, fuel lines, controls, 3-way valve, circulating pump, installation, and for the oil system, a double-wall storage tank of 2500 gal.

14.5.3 Controls and Operational Considerations

The object of the peaking equipment is to provide the capacity difference between the structure's requirement and the capacity of the base load (geothermal) system. This task must be accomplished in such a way as to produce even heat output without compromising the performance of the base load system.

Peaking with individual unit heaters is a simple process with regard to controls. Each individual unit is equipped with a thermostat which initiates operation of the unit when additional capacity is required in the zone that it serves. To eliminate unnecessary operation, it is useful to incoprorate an outside temperature driven lockout to prevent use of the peaking unit above the balance point temperature.

For the boiler design, the situation is somewhat more complex. This results from the boiler being incorporated into the heating loop. Because the boiler changes the temperature of the supply water, it not only influences the output of the terminal equipment, but also the capacity of the geothermal heat exchanger.

Figure 14.9 presents a common design for installing a boiler on a circulating water loop. Located downstream of the heat exchanger, the boiler's function is to raise the supply water temperature to the terminal equipment during the peak heat load period. This is accomplished by resetting the supply water upward as the outside air temperature decreases. Table 14.21 presents a typical temperature reset schedule. In this case, the boiler begins operation between 30 and 25°F outside air temperature. Actual temperatures will vary with system design.

As the supply water temperature rises, the output of the terminal equipment rises. At the same time, the temperature of the return water rises as well.

The rise in return temperature occurs at a rate less than the supply water increase due to the higher output of the terminal equipment (which results in an increasing system ΔT). However, the rising return water temperature erodes the capacity of the geothermal heat exchanger to the extent that its capacity at the peak condition (0°F outside) is approximately 50% of its capacity prior to the initiation of boiler operation.

The impact of this decreased geothermal heat exchanger capacity is illustrated in Table 14.22 which compares the performance of unit heaters and boiler peaking strategies for the same example case.

As indicated for this example, the boiler design requires approximately 78% more peaking fuel than the unit heater design. At the peak condition (0°F), the unit heater supplies 58% of the heating energy needs of the structure compared to the boiler's 27%.

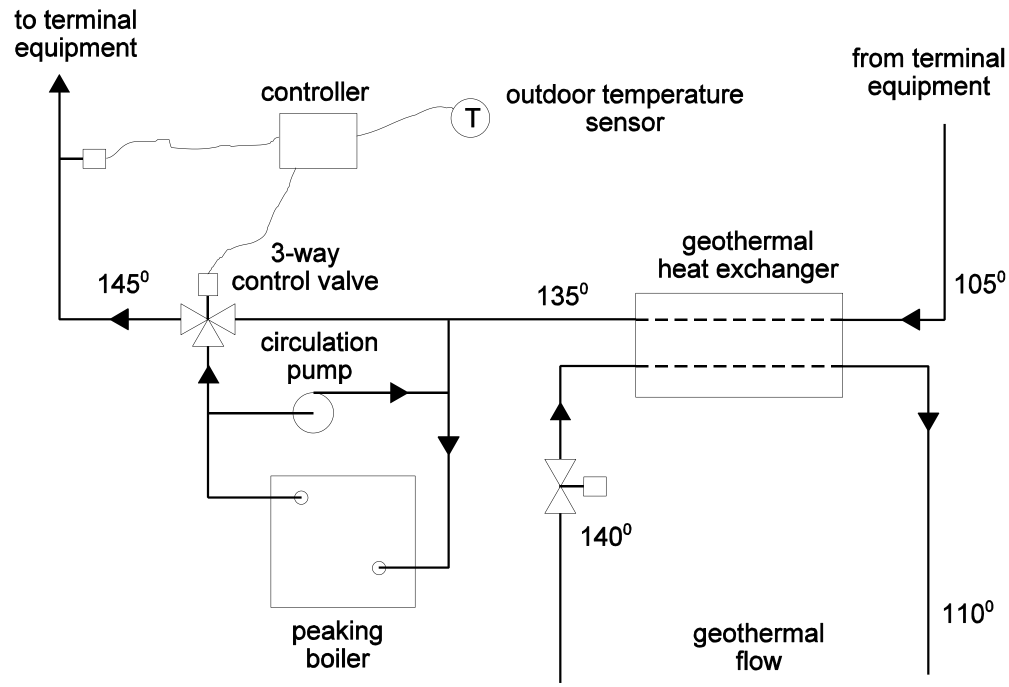


Figure 14.9 Heating system flow diagram.

Table 14.21 Typical Supply Water Temperature Reset Schedule and System Performance

Outside Air Temp (ºF)	Supply Water Temp (ºF)	Return Temp	Geothermal Heat Exchanger Capacity	Greenhouse Load	Required Boiler Output	% Geothermal
25	140	105.0	2,116,000	2,116,000	0	100
20	149	109.6	1,866,000	2,418,000	552,000	77
15	159	114.1	1,627,000	2,721,000	1,092,000	60
10	168	118.3	1,407,000	3,023,000	1,616,000	47
5	177	122.3	1,197,000	3,325,000	2,128,000	36
0	186	126.3	989,000	3,627,000	2,638,000	27

Table 14.22 Comparison of Boiler and Unit Heater Peaking Strategies

Outside Air Temp (ºF)	Hrs/Yr	Boiler Fuel (gal Propane)	% Geothermal	Unit Heater Fuel (gal Propane)	% Geothermal
20	352	3,107	77	1,687	88
15	150	2,591	66	1,440	78
10	82	2,085	47	1,180	70
5	39	1,317	36	748	64
0	17	617	27	407	58
		9,717 gal		5,462 gal	

Table 14.23 Peaking System Sizing Requirements (60ºF Inside, 0ºF Outside)

Base Load System Capacity (% of Peak)	Unit Heater System Peaking Capacity (% of Peak)	Boiler Peaking Capacity (% of Peak)
40	60	93
60	40	73
80	20	27

This means that the required capacity of the peaking boiler is larger than that of the unit heater equipment for the same application. This disparity in required capacity at the peak load becomes more pronounced as the percentage of peak load carried by the base load system decreases. For example, a system in which the base load capacity is 40% of the peak would suggest a peaking boiler sized for 60% of the load. In fact, due to issues discussed above, the boiler would have to be sized for 93% of the peak. Table 14.23 provides a summary of the peaking boiler and unit heater sizing requirements for selected base load system capacities.

Figures 14.11, 14.12 and 14.13 present heating energy displaced for unit heater type peaking systems in three different climates for a variety of inside temperatures set points. Figures 14.14, 14.15 and 14.16 present the same information for boiler peaking system. In each case in these figures, the results are strongly influenced by day setpoint temperature (the first value as indicated in the key of each figure). Although the percentages of displaced energy appear to be quite similar to the unit heater values for boiler system, because the heating energy requirement for greenhouses are so high, small percentage differences translate into substantial fuel cost differences.

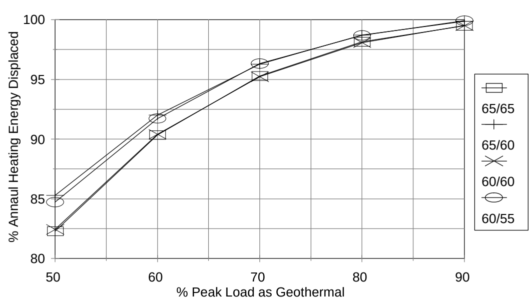


Figure 14.10 Unit heater annual energy displaced, Helena, MT.

Table 14.24 presents the fuel consumption for 1-acre greenhouse in the three climates for the same temperature set points as in Figures 14.10 through 14.15. Using the Klamath Falls climate data as an example, for a system with a base load capacity of 60% of the peak and a 60º day/60ºF night set point, the boiler system would displace 94.8% of the annual heating requirements compared to 97.2% for the unit heater design.

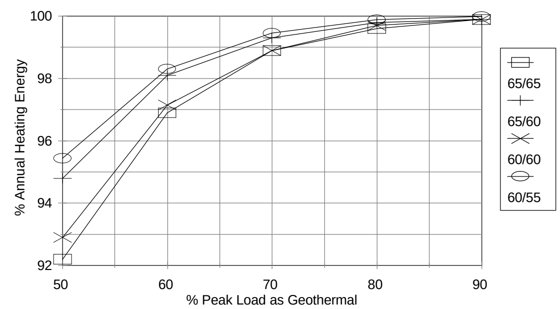


Figure 14.11. Unit heater annual energy displaced, Klamath Falls, OR

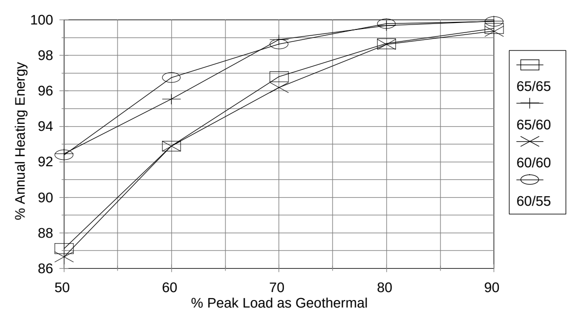
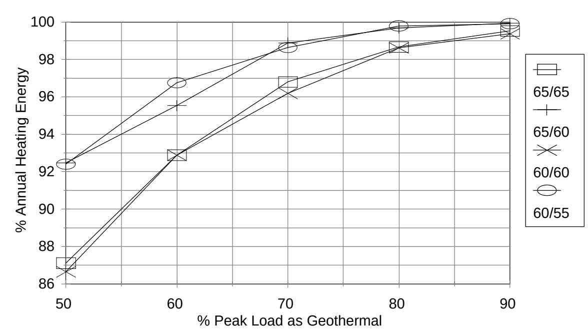


Figure 14.12. Unit heater annual energy displaced,



San Bernardino, CA.

Figure 14.13. Boiler annual energy displaced, Helena, MT.

Table 14.24 Fuel Consumption for 1-Acre Greenhouse - Btu x 10^9

	Helena, MT	Klamath Falls, OR	San Bernardino, CA
60º/60º	7.36	5.59	1.78
60º/55º	6.37	4.52	1.09
65º/60º	7.59	5.81	1.88
65º/65º	8.69	6.96	2.77

Notes: Double poly roof, single fiberglass sides, 1 ACH.
To convert to gallons of propane per year, divide by 63,000.
To convert to gallon of fuel oil per year, divide by 93,000.
To convert to therms of natural gas, divide by 70,000.
Conversions assume 70% efficiency.
At \$1.00/gal and 70% efficiency, fuel oil cost \$10.20/10^9 Btu and propane \$15.87/10^9 Btu. At the same efficiency at \$0.50 per therm, gas cost \$7.14/10^9 Btu.

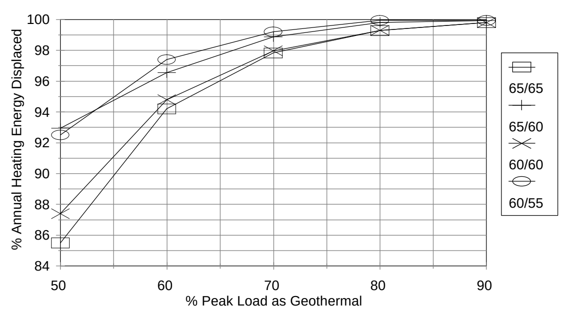


Figure 14.14 Boiler annual energy displaced, Klamath Falls, OR.

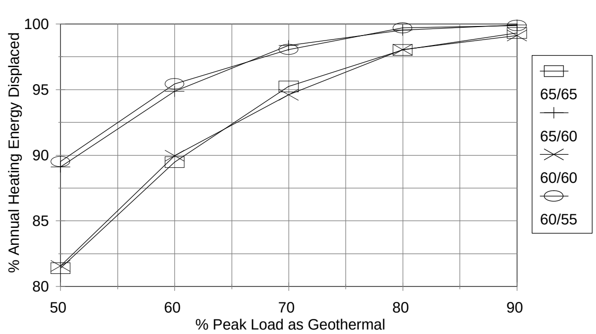


Figure 14.15 Boiler annual energy displaced, San Bernardino, CA.

Although these figures seem comparable, attaching fuel consumption values to them clearly indicates the difference. Using data from Table 14.24, assuming the use of propane as the fuel, the boiler would require 4,613 gal/yr and the unit heater system 2,484 gal/yr.

14.5.4 Cost of Implementation

Using Figures 14.10 through 14.15 along with Table 14.24, the capital cost for equipment and the annual fuel cost can be calculated for any application (based on the three climates for which data is provided). As discussed above, the boiler approach is characterized by lower equipment cost than the unit heater approach, but higher fuel consumption in a given application. As a result of this, for a given set of conditions, there will be an optimum system from a total cost standpoint.

Calculation of the lowest cost system for a particular application involves consideration of equipment ownership cost (capital cost and financing), fuel costs, equipment maintenance and fan energy (unit heater system).

This is best illustrated with an example. Consider a 1-acre greenhouse to be built in a moderate climate (Klamath Falls) in which effluent from an existing facility will be used as the supply for the new construction. Using the effluent will permit the heating system to meet 55% of the peak load. Propane will be employed for the peaking fuel and inside temperature set point will be 60ºF day and night.

Assuming a double poly roof/single fiberglass construction, the peak heating load for the structure is determined to be 2.77 x 10^6 Btu/hr. As a result, the unit heater peaking equipment would be sized for 0.45 C 2,770,000 = 1,247,000 Btu/hr. The boiler would be sized (interpolating from Table 14.23) for 0.78 C 2,770,000 = 2.16 x 10^6 Btu/hr. From Figure 14.8, the capital cost for the peaking system would be \$38,000 for the unit heaters and \$32,500 for the boiler. Based on 15 years at 8% financing, the annual cost of the unit heater equipment would be \$4,440 and \$3,797 for the boiler system.

Table 14.25. Summary of Peaking System Costs - Propane Example

	Unit Heaters $	Unit Heaters $/ft^2	Boiler $	Boiler $/ft^2
Equipment (15 yrs at 8%)	4,440	0.102	3,797	0.087
Maintenance (2% of capital)	760	0.017	650	0.015
Electricity ($0.07/kWh)	269	0.006	0	0
Fuel ($1.00/gal)	4,436	0.102	7,986	0.183
Total	9,905	0.227	12,433	0.285

Table 14.26. Summary of Peaking System Annual Costs - Fuel Oil Example

	Unit Heaters $	Unit Heaters $/ft^2	Boiler $	Boiler $/ft^2
Equipment (15 yrs at 8%)	7,243	0.166	4,965	0.114
Maintenance (2% of capital)	1,240	0.029	850	0.020
Electricity ($0.07/kWh)	269	0.006	0	0.000
Fuel ($1.00/gal)	3,005	0.069	5,410	0.124
Total	11,757	0.270	11,225	0.258

Using Figures 14.11 and 14.14, along with Table 14.24, the annual propane consumption for the unit heater system would be 4,436 gallons ((1 - 0.95) C 5.59 x 10^9 ÷ 63,000) and 7,986 gallons ((1 - 0.91) C 5.59 x 10^9 ÷ 63,000) for the boiler system.

Assuming a value of 2% of capital cost for equipment maintenance, the cost for the boiler system would be $650/yr and for the unit heater system $760/yr. Fan energy consumption is a function of the size and number of unit heaters installed. Assuming 10 units at 125,000 Btu/hr each, the fan motor in each unit would be 1/3 hp. For 10 units, 3.3 hp or approximately 2.9 kW at 85% efficiency. For 1325 hours per year operation, the electric consumption would amount to 3842 kWh or about $269 at $0.07/kWh.

Table 14.25 presents a summary of the costs for the two peaking systems in both $ and $/ft^2 of greenhouse.

In this case, the unit heater design is the clear choice due to its lower equipment and fuel costs. If fuel oil was to be the peaking fuel in the same situation, the results are quite different. Table 14.26 presents the results for the oil case.

In the case of fuel oil, the much higher cost of oil-fired unit heater equipment tends to be the pivotal cost item. Despite the lower fuel costs for the unit heater system, the boiler design is the most economic choice.

Figures 14.16, 14.17 and 14.18 summarize the cost data discussed in the previous section and present the total costs associated with the peaking system for the three climates discussed in this report. In each case, the costs are presented in $/ft^2 of greenhouse, a value commonly used in the greenhouse industry.

Figures 14.16, 14.17 and 14.18 are based on a constant 60^o set point (night and day) in the greenhouse. Because the set point temperature, and whether or not set back is used, has a substantial impact upon energy usage, the above conclusions are valid for the 60^o set point only. For other temperatures calculations, using Figures 14.10 through 14.15 and Table 14.24 should be done.

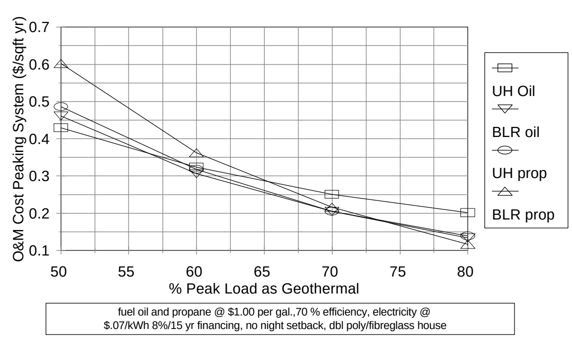


Figure 14.16 Peaking system cost, Helena, MT.

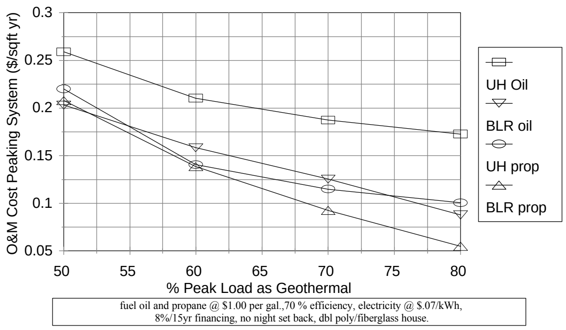


Figure 14.17 Peaking system cost, Klamath Falls, OR.

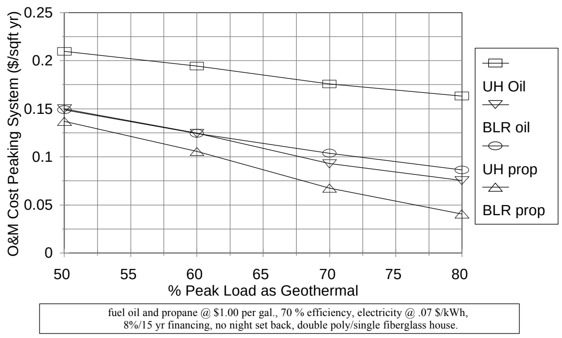


Figure 14.18. Peaking system cost, San Bernardino, CA.

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