

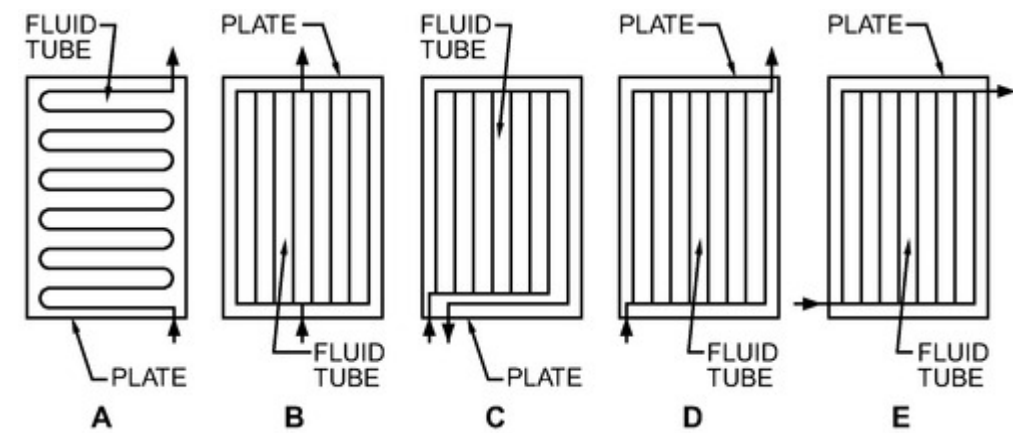


Figure 6. Plan View of Liquid Collector Absorber Plates

Figure 6 shows a plan view of typical absorber plates. The serpentine design (Figure 56) is used less frequently because it is difficult to drain and imposes a high pressure drop, but it can produce higher temperatures at lower flow rates. Most manufacturers use absorber plates similar to those shown in Figures 60 and 66. In liquid collectors, manifold selection is important because the design can restrict the array piping configuration. The manifold must be drainable and free-floating, with generous allowance for thermal expansion. Some manufacturers provide a choice of manifold connections to give designers flexibility in designing arrays. In air collectors, most manufacturers increase the heat transfer area by using fins, matrices, or corrugated surfaces (Figure 7). Many of these designs increase air turbulence, which improves collector efficiency (at the expense of increased fan power). Figure 7 also shows cross sections of typical solar water collectors. Water passages can be integrated with the absorber plate to ensure good thermal contact (Figure 7E), or they can be soldered, brazed, or otherwise fastened to the absorber plate (Figure 7F). Manufactured collectors are made in modular sizes, typically 4 by 8 ft, and these are connected to form a bank or row. Because most commercial-scale systems involve more than 1000 ft² of collector, there can be numerous piping connections, depending on design.

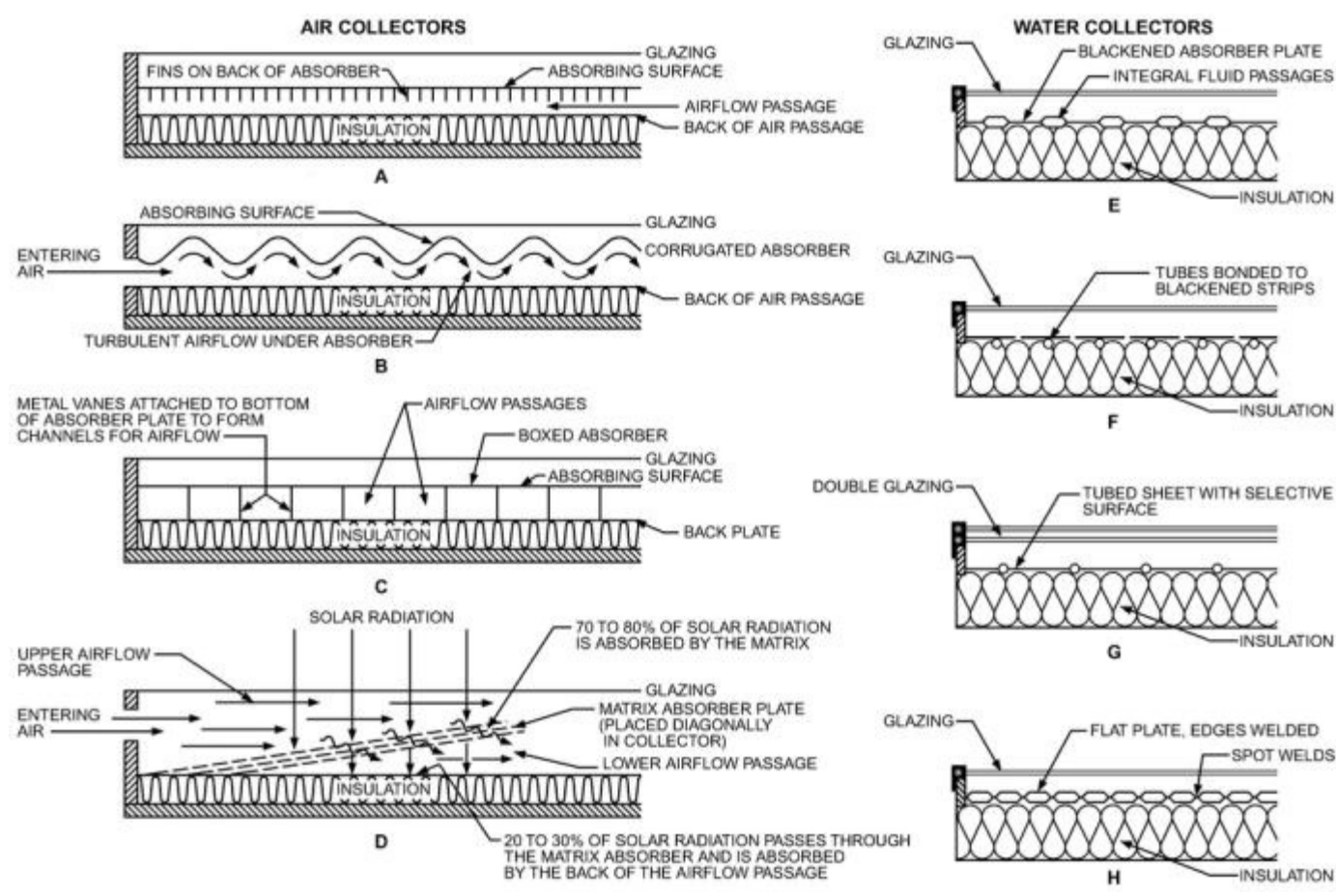


Figure 7. Cross Sections of Various Solar Air and Water Heater

Absorber plates may be coated with spectrally selective or nonselective materials. Selective coatings are more efficient, but cost more than nonselective, or flat black, coatings. However, their higher efficiencies may reduce the overall size of the array, resulting in lower total cost. The Argonne National Laboratory has detailed guidelines on various coating materials used in solar applications (ANL 1981). Solar collectors with colored absorbers (e.g., blue, red-brown, green) may be aesthetically attractive for building integration. Although colored collectors have lower efficiency than the typical black collectors, the difference in energy output depends on the absorber darkness. For a medium value of the coefficient of absorptance ($\alpha = 0.85$), the drop in amount of energy from colored collectors (about 7 to 18%, depending on location) is satisfactory compared to the performance of collectors with black absorbers ($\alpha = 0.95$). Colored collectors require a proportionately larger collector aperture area to have the same energy output as that of typical black collectors (Kalogirou et al. 2005). Selective colored absorbers are efficient in a wider range of operating temperatures than absorbers with high-emissivity paints, and selective absorbers stable to weather conditions are needed for unglazed collectors to avoid degradation by environmental effects. **Housing.** The collector housing is the container that provides structural integrity for the collector assembly. The housing must be structurally sound, weathertight, fire resistant, and mechanically connectable to a substructure to form an array. Collector housing materials include the following:

- Galvanized or painted steel
- Aluminum folded sheet stock or extruded wall materials
- Various plastics, either molded or extruded
- Composite wood products
- Standard elements of the building

Extruded anodized aluminum, including extruded channels, offers durability and ease of fabrication. Grooves are sometimes included in the extruded channels to accommodate proprietary mounting fixtures. The high temperatures of a solar collector deteriorate wood housings; consequently, wood is often forbidden by fire codes. **Glazing.** Solar collectors for domestic hot water are usually single-glazed to reduce absorber plate convective and radiative losses. Some collectors have double glazing to further reduce these losses; however, double glazing should be restricted to applications where the value of $(t_a - t_c)/L$ exceeds that of domestic hot-water applications (e.g., space heating or activating absorption refrigeration); the parameters for this ratio are defined in Equations (4) and (5). Glazing materials are plastic, plastic film, or glass. Glass can absorb the long-wave thermal radiation emitted by the absorber coating, but it is not affected by ultraviolet (UV) radiation. Because of their impact tolerance, only tempered, low-iron glass covers should be considered. These covers have a solar transmission rating of 84 to 91%.

If the probability of vandalism is high, polycarbonate, which has high impact resistance, should be considered. Unfortunately, its transmittance is not as high as that of low-iron glass, and it is susceptible to long-term UV degradation. **Insulation.** Collector enclosures must be well insulated to minimize heat losses. Insulation adjacent to the absorber plate must withstand temperatures up to 400°F and, most importantly, must not outgas volatile products within this range. Many insulation materials designed for construction applications are not suitable for solar collectors because the binders outgas volatiles at normal collector operating temperatures. Solar collector insulation is typically made of mineral fiber, ceramic fiber, glass foam, plastic foam, or fiberglass. Fiberglass is the least expensive insulation and is widely used in solar collectors. For high-temperature applications, rigid fiberglass board with a minimum of binder is recommended. Also, a layer of polyisocyanurate foam in collectors is often used because of its superior R-value. Because it can outgas at high temperatures, the foam must not be allowed to contact the collector plate.

Figure 8 illustrates the preferred method of combining fiberglass and foam insulations to obtain both high efficiency and durability. Note that the absorber plate should be free-floating to avoid thermal stresses. Despite attempts to make collectors watertight, moisture is always present in the interior. This moisture can physically degrade mineral wool and reduce the R-value of fiberglass, so drainage and venting are crucial.

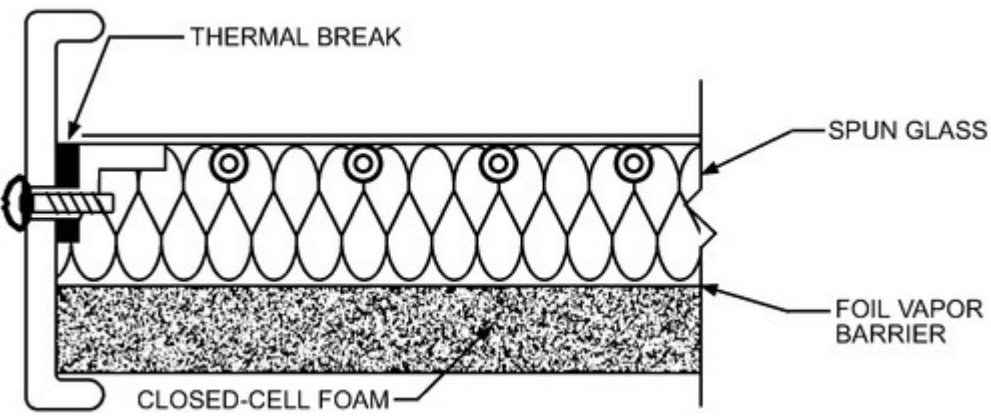


Figure 8. Cross Section of Suggested Insulation to Reduce Heat Loss from Back Surface of Absorber

1.4 ROW DESIGN

Piping Configuration Most commercial and industrial systems require a large number of collectors. Connecting the collectors with one set of manifolds makes it difficult to ensure drainability, balanced flow, and low pressure drop. An array usually includes many individual groups of collectors, called rows, to provide the necessary flow characteristics. Rows can be grouped into (1) parallel flow or (2) combined series-parallel flow. Parallel flow is the most frequently used because it is inherently balanced, has low pressure drop, and is drainable. Figure 9 illustrates various collector header designs for forming a parallel-flow row (the flow is parallel but the collectors are connected in series).

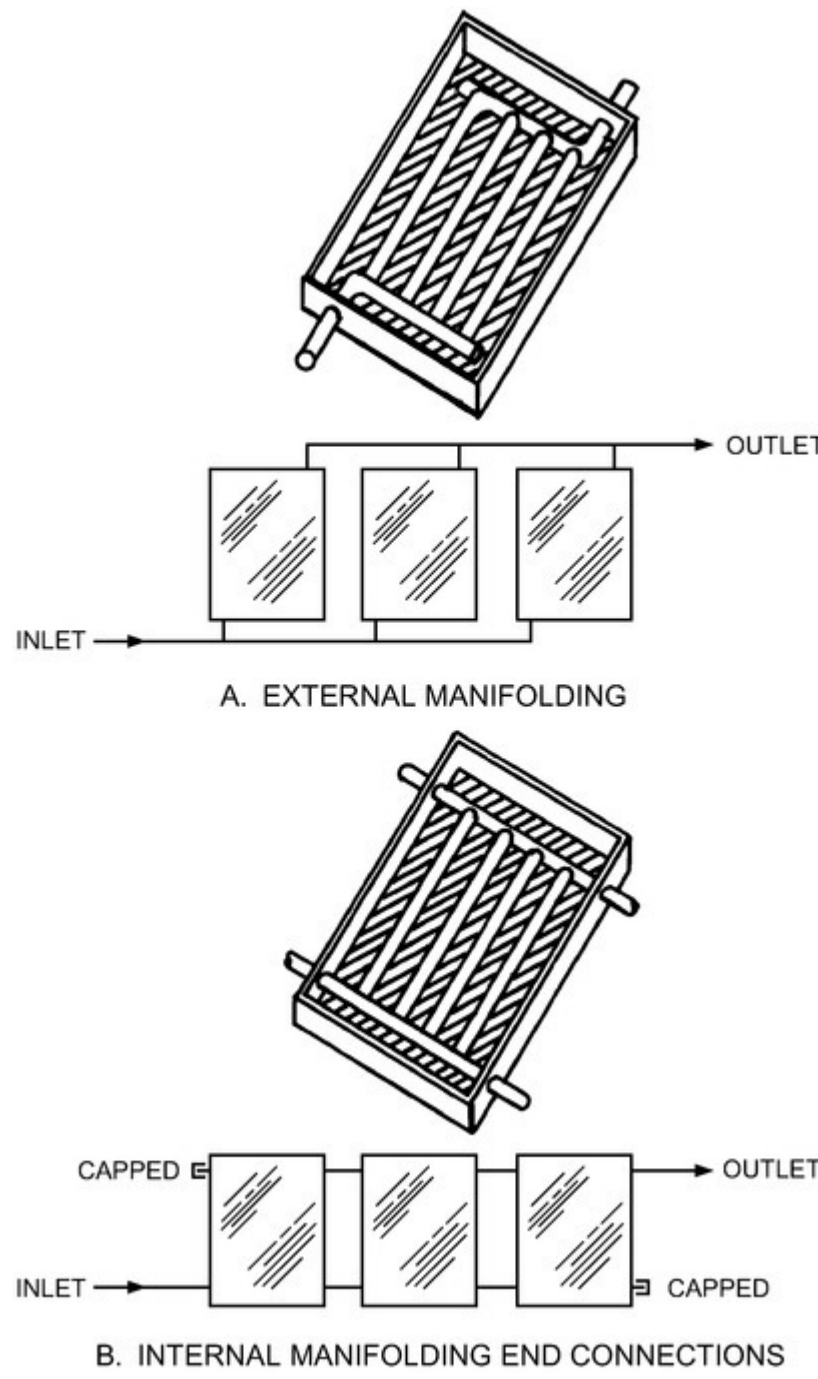


Figure 9. Collector Manifolding Arrangements for Parallel-Flow Row

Generally, flat-plate collectors connect to the main piping in one of the methods shown in Figure 9. The **external manifold** collector has a small-diameter connection meant to carry only the flow for one collector. It must be connected individually to the manifold piping, which is not part of the collector panel, as depicted in Figure 9A. **Internal manifold** collectors incorporate the manifold piping integral with each collector (Figure 9B). Headers at either end of the collector distribute flow to the risers. Several collectors with large headers can be placed side by side to form a continuous supply and return manifold. With 1 in. headers, four to six 40 ft² collectors can be placed side by side. Collectors with 1 in. headers can be mounted in a row without producing unbalanced flow. Most collectors have four plumbing connections, some of which may be capped if the collector is located on the end of the array. Internally manifolded collectors have the following advantages:

- Piping costs are lower because fewer fittings and less insulation are required.
- Heat loss is less because less piping is exposed.
- Installation is more attractive.

Some of their disadvantages are as follows:

- For drainback systems, the entire row must be pitched, thus complicating mounting.
- Flow may be imbalanced if too many collectors are connected in parallel.
- Removing the collector for servicing may be difficult.
- Stringent thermal expansion requirements must be met if too many collectors are combined in a row.

Velocity Limitations

Fluid velocity limits the number of internally manifolded collectors that can be contained in a row. For 1 in. headers, up to eight 40 ft² collectors can usually be connected for satisfactory performance. If too many are connected in parallel, the middle collectors will not receive enough flow, and performance will decrease. Connecting too many collectors also increases pressure drop. Figure 10 shows the effect of collector number on performance and pressure drop for one particular design. Newton and Gilman (1983) describe a general method to determine the number of internally manifolded collectors that can be connected.

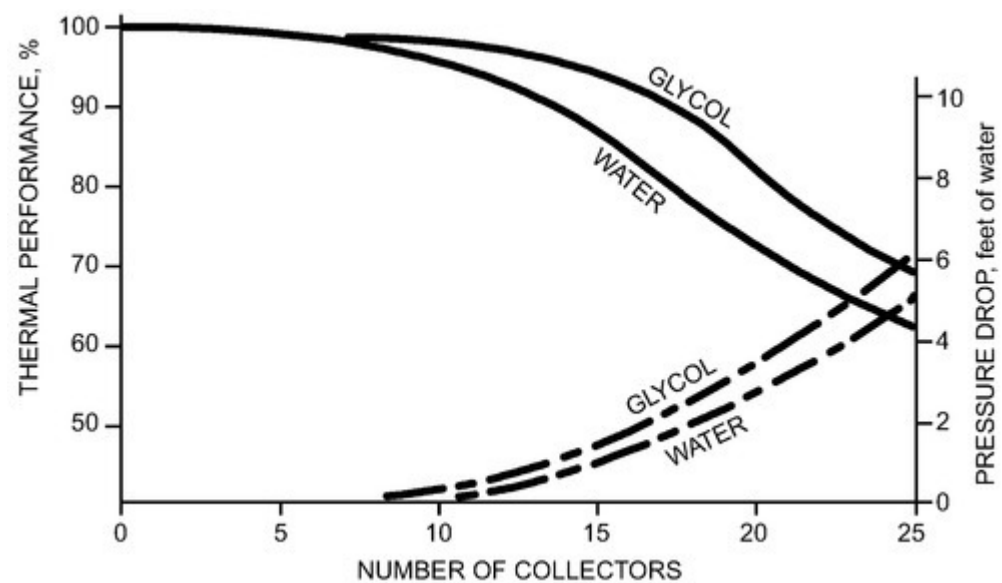


Figure 10. Pressure Drop and Thermal Performance of Collectors with Internal Manifolds Numbers

Flow restrictors can be used to accommodate a large number of collectors in a row. The flow distribution in the 12 collectors of Figure 11 would not be satisfactory without the flow restrictors shown at the interconnections. The flow restrictors are barriers with a drilled hole of the diameter indicated. Some manufacturers calculate the required hole diameters and provide predrilled restrictors. Chapter 22 of the 2013 ASHRAE Handbook—Fundamentals gives information on sizing piping. Knowles (1981) provides the following expression for the minimum acceptable header diameter:

$$D = 0.24(Q/\Delta p)^{0.45} N^{0.64} \tag{1}$$

where

- D = header diameter, in.
- N = number of collectors in module
- Q = recommended flow rate for collector, gpm
- Δp = pressure drop across collector at recommended flow rate, psi

Because pipe is available in a limited number of diameters, selection of the next larger size ensures balanced flow. Usually, the sizes of supply and return piping are graduated to maintain the same pressure drop while minimizing piping cost. Complicated configurations may require a hydraulic static regain calculation.

Thermal Expansion

Thermal expansion will affect the row shown in Figure 11. Thermal expansion (or contraction) of a module of collectors in parallel may be estimated by the following equation:

$$\Delta = 0.000335n(t_c - t_i) \tag{2}$$

where

- Δ = expansion or contraction of collector array, in.
- n = number of collectors in array
- t_c = collector temperature, °F (generally the maximum stagnation temperature)
- t_i = installation temperature of collector array, °F (generally the lowest possible air temperature)

Because absorbers are rigidly connected, the absorber must have sufficient clearance from the side frame to allow the expansion indicated in Equation (2). Information on dealing with expansion in piping may also be found in Chapter 46.

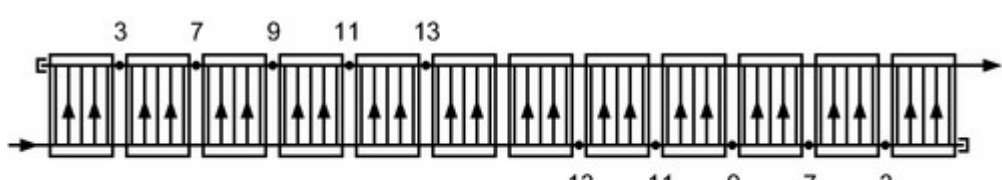
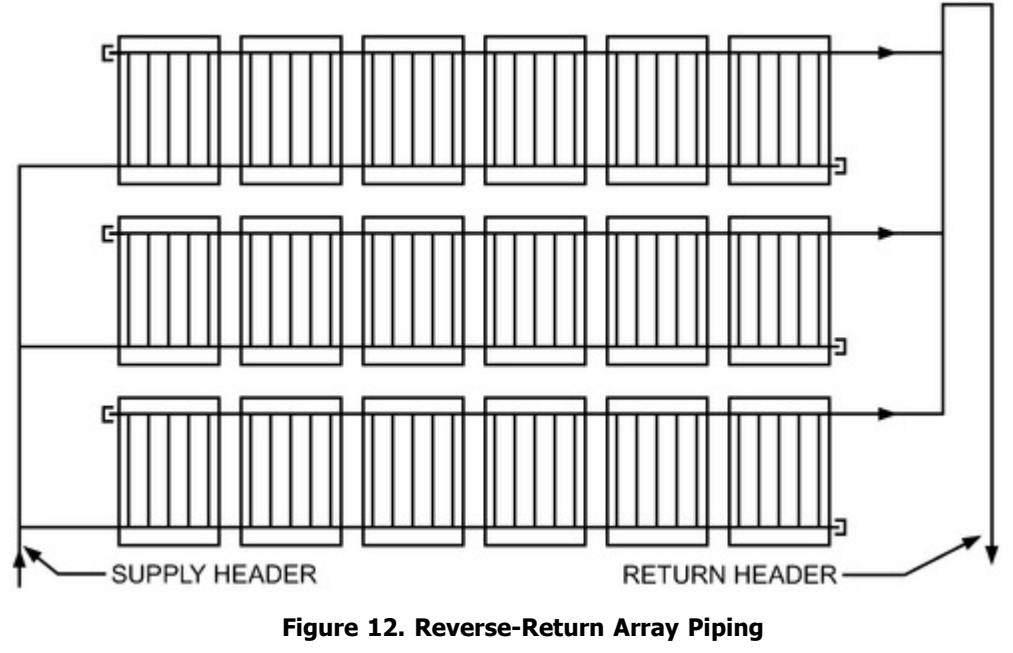


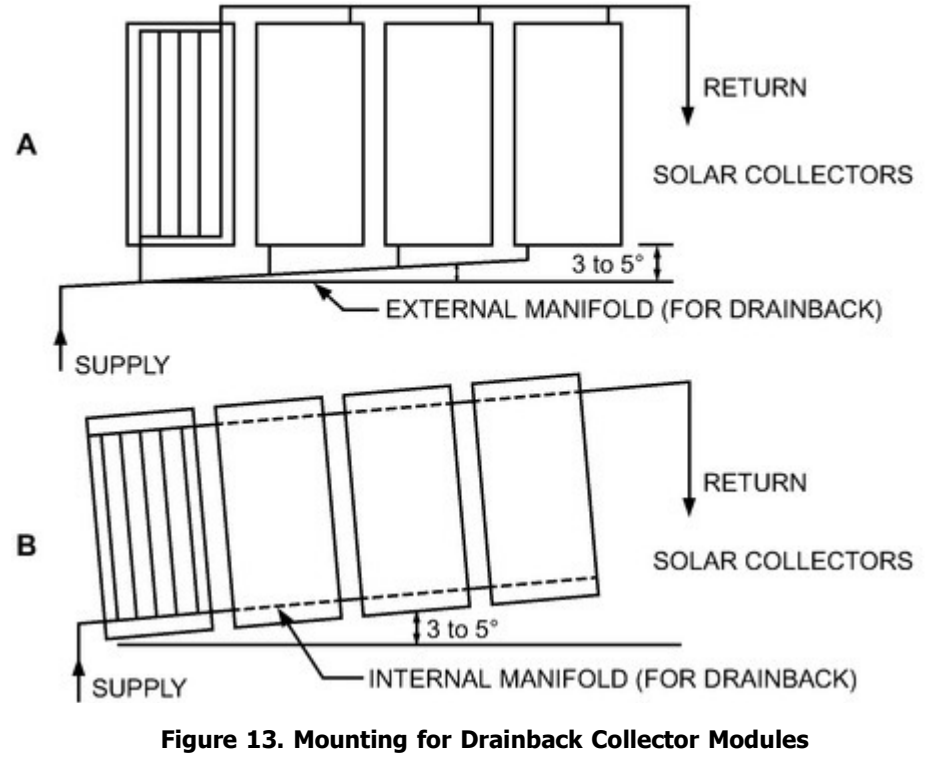
Figure 11. Flow Pattern in Long Collector Row with Restrictions

1.5 ARRAY DESIGN

Piping Configuration **Liquid Systems.** To maintain balanced flow, an array or field of collectors should be built from identical rows configured as described in previous sections. Whenever possible, rows must be connected in reverse-return fashion (Figure 12), which ensures that the array is self-balanced. With proper care, an array can drain, which is an essential requirement for drainback freeze protection. Piping to and from the collectors must be sloped properly in a drainback system. Typically, piping and collectors must slope to drain at 1/4 in. per linear foot. Elevations throughout the array, especially the highest and lowest point of the piping, should be noted on the drawings.

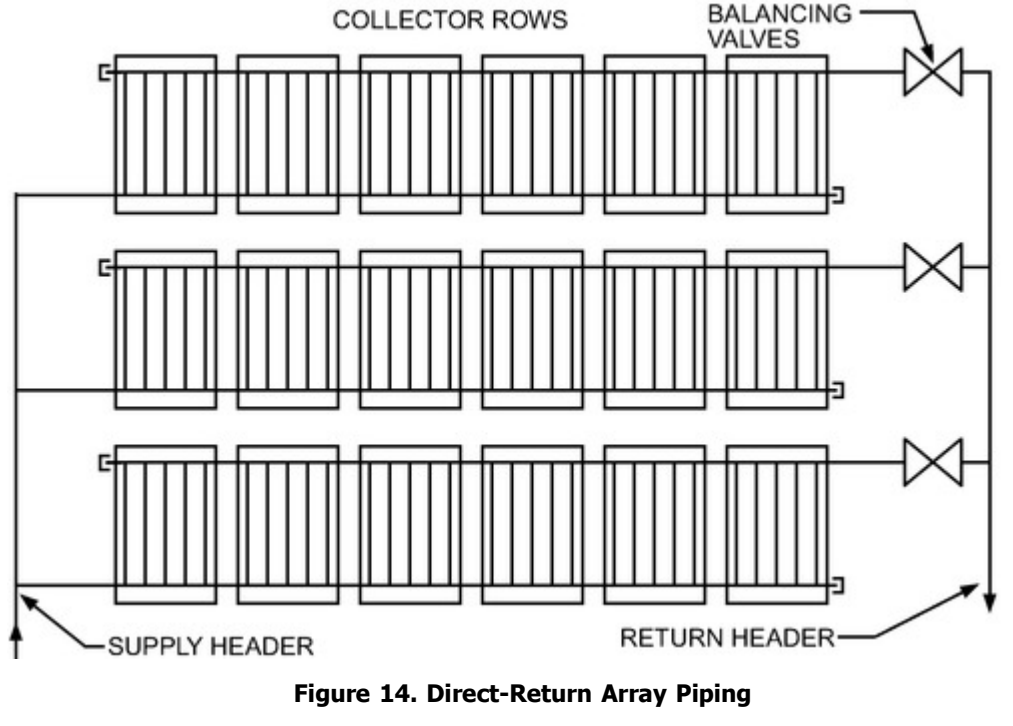


The external manifold collector has different mounting and plumbing considerations from the internal manifold collector (Figure 13). A row of externally manifolded collectors can be mounted horizontally, as shown in Figure 13a. The lower header must be pitched as shown. The pitch of the upper header can be either horizontal or pitched toward the collectors so it can drain back through the collectors. Arrays with internal manifolds pose a greater challenge in designing and installing the collector mounting system. For these collectors to drain, the entire bank must be tilted, as shown in Figure 13b. Reverse return always implies an extra pipe run. Sometimes, it is more convenient to use direct return (Figure 14). In this case, balancing valves are needed to ensure uniform flow through the rows. The balancing valves must be connected at the row outlet to provide the flow resistance necessary to ensure filling of all rows on pump start-up.



It is often impossible to configure parallel arrays because of the presence of rooftop equipment, roof penetrations, or other building-imposed constraints. Although the list is not complete, the following requirements should be considered when developing the array configuration:

- Strive for a self-balancing configuration.
- For drainback, design rows, subarrays, and arrays to be individually and collectively drainable.
- Always locate collectors or rows with high flow resistance at the outlet to improve flow balance and ensure the drainback system fills.
- Minimize flow and heat transfer losses.



In general, it is easier to configure complex array designs for nonfreezing fluid systems. Newton and Gilman (1983) provide some typical examples. However, with careful attention to these criteria, it is also possible to design successful large drainback arrays. **Air Systems.** Air distribution within the collector array is the most critical feature of an air system for effective operation. Proper airflow must take into account the overall pressure drop to allow efficient fan motor sizing. Balancing, automatic backdraft dampers, and fire dampers are usually needed. Air leaks (both into and out of the ducts and from component to component) must be minimized. For example, some air collector systems contain a water coil for preheating water. Despite the inclusion of automatic and backdraft dampers, leakage within the system can freeze coils. One possible solution is to position the coil near the warm end of the storage bin. Another is to circulate an antifreeze solution in the coil. Whatever the solution, the designer must remember that even the lowest-leakage dampers will leak if improperly installed or adjusted. As with liquid systems, the main supply and return ducts should be connected in a reverse-return configuration, with balancing dampers on each supply branch to the collector modules. If reverse return is not feasible, the main ducts should include balancing dampers at strategic locations. Here too, having fewer branch ducts reduces balancing needs and costs. Unlike liquid systems, air collectors can be built on site. Although material and cost savings can be substantial with site-built collectors, extreme care is necessary to ensure long life, low leakage, and proper air distribution. Quality control in the field can be a problem, so well-trained designers and installers are critical to the success of these systems. **Performance.** The effect of array and air distribution system designs on the overall system performance must be considered. Beckman et al. (1977) give standard procedures for estimating the impact of series connection and duct thermal losses. The effect on fan operation and fan power is more difficult to determine. For unique system designs and more detailed performance estimates, including fan power, an hourly simulation such as TRNSYS (Klein et al. 2010) can be used.

Shading When large collector arrays are mounted on flat roofs or level ground, multiple rows of collectors are usually installed in a sawtooth fashion. These multiple rows should be spaced so that they do not shade each other at low sun angles. However, it is usually not desirable to avoid mutual shading altogether. Additional rows can sometimes be added to a roof or other constrained area; this increases the solar fraction but sacrifices efficiency. Kutscher et al. (1982) presents a method of estimating the energy delivered annually when there is some row-to-row shading in the array. The most common practice is to base row spacing on the lowest sun angle during the most important time of the year. For example, water and pool heating systems that operate mostly in the summer can use the higher sun angles of summer or spring and fall. Space-heating systems may want to maximize exposure in the winter, so they should use the lowest sun angle at the winter solstice. A system that operates year round may use the spring equinox sun angle as a reasonable compromise to allow more rows.

Thermal Collector Performance Under steady-state flow conditions, the useful heat delivered by a solar collector is equal to the energy absorbed in the heat transfer fluid minus the direct and indirect heat losses from the surface to the surroundings. This principle can be stated in the following relationship:

$$q_u = A_c [I_r \tau \alpha - U_L (\bar{T}_p - t_a)] \tag{3}$$

where
 q_u = useful energy delivered by collector, Btu/h
 A_c = total aperture collector area, ft²
 I_r = irradiance, total (direct plus diffuse) solar energy incident on upper surface of sloping collector structure, Btu/h · ft²
 τ = transmittance (fraction of incoming solar radiation that reaches absorbing surface), dimensionless
 α = absorptance (fraction of solar energy reaching surface that is absorbed), dimensionless
 U_L = overall heat loss coefficient, Btu/h · ft² · °F
 $\bar{T}_p$ = average temperature of absorbing surface of absorber plate, °F
 t_a = atmospheric temperature, °F

Except for average plate temperature $\bar{T}_p$, these terms can be readily determined. For convenience, Equation (3) can be modified by substituting inlet fluid temperature for the average plate temperature, if a suitable correction factor is included. The resulting equation is

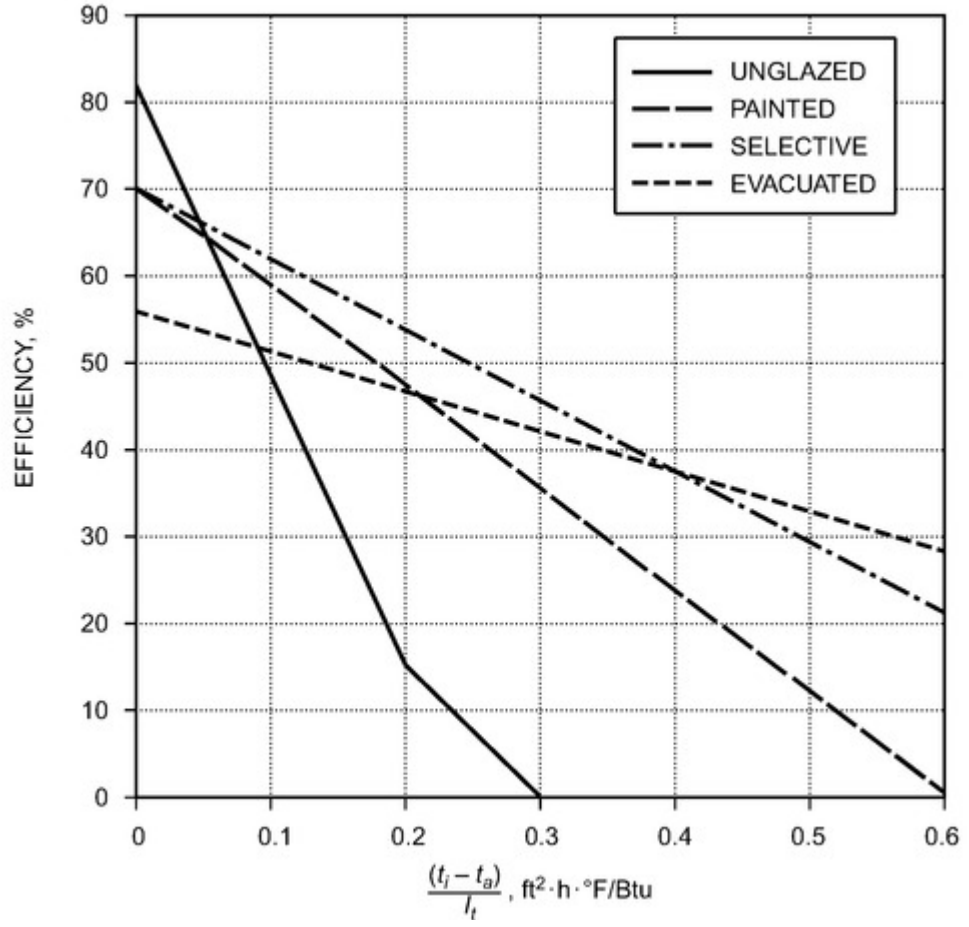
$$q_u = F_R A_c [I_r \tau \alpha - U_L (t_i - t_a)] \tag{4}$$

where
 F_R = correction factor, or collector heat removal efficiency factor, <1.0
 t_i = temperature of fluid entering collector, °F

The heat removal factor F_R can be considered the ratio of the heat actually delivered to that delivered if the collector plate were at a uniform temperature equal to that of the entering fluid. An F_R of 1.0 is theoretically possible if (1) the fluid is circulated at such a high rate that its temperature rise is negligible, and (2) the heat transfer coefficient and fin efficiency are so high that the temperature difference between the absorber surface and the fluid is negligible. In Equation (4), the temperature t_i of the inlet fluid depends on the characteristics of the complete solar heating system and the building's heat demand. However, F_R is affected only by the solar collector characteristics, fluid type, and fluid flow rate through the collector. Solar air heaters remove substantially less heat than liquid collectors. However, their lower collector inlet temperature makes their system efficiency comparable to that of liquid systems for space-heating applications. Equation (4) may be rewritten in terms of the instantaneous efficiency of total solar radiation collection by dividing both sides of the equation by $I_r A_c$. The result is

$$\eta = F_R \tau \alpha - F_R U_L \frac{(t_i - t_a)}{I_r} \tag{5}$$

where η = collector efficiency, dimensionless. Equation (5) plots as a straight line on a graph of efficiency versus the heat loss parameter $(t_i - t_a)/I_r$. Plots of Equation (5) for various liquid collectors are shown in Figure 15. The intercept (intersection of the line with the vertical efficiency axis) equals $F_R \tau \alpha$. The slope of the line (i.e., any efficiency difference divided by the corresponding horizontal scale difference) equals $-F_R U_L$. If experimental data on collector heat delivery at various temperatures and solar conditions are plotted, with efficiency as the vertical axis and $(t_i - t_a)/I_r$ as the horizontal axis, the best straight line through the data points correlates collector performance with solar and temperature conditions. The intersection of the line with the vertical axis is where the temperature of the fluid entering the collector equals the ambient temperature, and collector efficiency is at its maximum. At the intersection of the line with the horizontal axis, collection efficiency is zero. This condition corresponds to such a low radiation level, or to such a high temperature of the fluid into the collector, that heat losses equal solar absorption, and the collector delivers no useful heat. This condition, normally called **stagnation**, usually occurs when no coolant flows to a collector. Solving Equation (5) for t_i under the hottest conditions yields the stagnation temperature for the collectors at that site.



Equation (5) includes all important design and operational factors affecting steady-state performance except collector flow rate and solar incidence angle. Flow rate indirectly affects performance through the average plate temperature. If the heat removal rate is reduced, the average plate temperature increases, and more heat is lost. If the flow is increased, collector plate temperature and heat loss decrease. **Solar Incidence Angle.** These relationships assume that the sun is perpendicular to the plane of the collector, which rarely occurs. For glass cover plates, specular reflection of radiation occurs, thereby reducing the $\tau \alpha$ product. The incident angle modifier $K_{\tau \alpha}$, defined as the ratio of $\tau \alpha$ at some incidence angle θ to $\tau \alpha$ at normal radiation ($\tau \alpha_n$), is described by the following simple expression for specular reflection:

$$K_{\tau \alpha} = \frac{\tau \alpha}{(\tau \alpha)_n} = 1 + b_r \left(\frac{1}{\cos \theta} - 1 \right) \tag{6}$$

For a single glass cover, b_r is approximately -0.10. Many flat-plate collectors, particularly evacuated tubes, have some limited focusing capability. The incident angle modifiers for these collectors are not modeled well by Equation (6), which is a linear function of $(1/\cos \theta) - 1$. **Cellular Flow Rate.** Equation (5) is not convenient for air collectors when it is desirable to present data based on collector outlet temperature t_{out} rather than inlet temperature, which is the commonly measured variable for liquid systems. The relationship between the heat removal factors for these two cases is

$$F_R \tau \alpha = \frac{(F_R \tau \alpha)'}{1 + (F_R U_L)'/\dot{m} c_p} \tag{7a}$$

$$F_R U_L = \frac{(F_R U_L)'}{1 + (F_R U_L)'/\dot{m} c_p} \tag{7b}$$

where $\dot{m} c_p$ is mass flow times specific heat of air, $F_R U_L$ and $F_R \tau \alpha$ apply to $t_i - t_a$ in Equation (5), and $(F_R U_L)'$ and $(F_R \tau \alpha)'$ apply to $t_{out} - t_a$.

Testing Methods ASHRAE Standard 93 gives information on testing solar energy collectors using single-phase fluids and no significant internal storage. The data can be used to predict performance in any location and under any conditions where load, weather, and insulation are known. The standard presents efficiency in a modified form of Equation (5). It specifies that the efficiency be reported in terms of gross collector area A_g rather than aperture collector area A_c . The reported efficiency is lower than efficiency based on net area, but the total energy collected does not change by this simplification. Therefore, gross collector area must be used when analyzing performance of collectors based on experiments that determine $F_R \tau \alpha$ and $F_R U_L$ according to ASHRAE Standard 93. Standard 93 suggests that testing be done at 14.7 lbu/h per square foot of gross collector area for liquid systems, and that the test fluid be water. Although it is acceptable to use lower flow rates or a heat transfer fluid other than water, the designer must adjust the F_R for a different heat removal rate based on the product $\dot{m} c_p$ of mass flow and specific heat. The following approximate approach may be used to estimate small changes in $\dot{m} c_p$:

$$\frac{(F_R U_L)_2}{(F_R U_L)_1} = \frac{1 - \exp[-A_c (F_R U_L)' / (\dot{m} c_p)_2]}{1 - \exp[-A_c (F_R U_L)' / (\dot{m} c_p)_1]} \tag{8}$$

Air collectors are tested at a flow rate of 2 cfm per square foot, and the same relationship applies for adjusting to other flow rates. Annual compilations of collector test data that meet the criteria of ASHRAE Standards 93 (for glazed solar collectors) and 96 (for unglazed solar collectors) may be obtained from Solar Rating and Certification Corporation (www.solar-rating.org). Another source for this information is the collector manufacturer. However, a manufacturer sometimes publishes efficiency data at a much higher flow rate than the recommended design value, so collector data should be obtained from an independent laboratory qualified to conduct testing prescribed by ASHRAE Standard 93.

Collector Test Results and Initial Screening Methods Final collector selection should be made only after energy analyses of the complete system, including realistic weather conditions and loads, have been determined. Also, a preliminary screening of collectors with various performance parameters should be conducted to identify those that best match the load. One way to accomplish this is to identify the expected range of heat loss parameter $(t_i - t_a)/I_r$ for the load and climate on a plot of efficiency η as a function of heat loss parameter, as indicated in Figure 15. Ambient temperature during the swimming season may vary by 18°F above or below pool temperature. The corresponding parameter values range from 0.15 Btu/h · ft² · °F on cool overcast days (low I_r) to as low as -0.15 Btu/h · ft² · °F on hot overcast days. For most swimming pool heating, unglazed collectors offer the highest performance and are the least expensive. The heat loss parameter for service water heating can range from 0.05 to 0.35 Btu/h · ft² · °F, depending on the climate at the site and the desired hot-water delivery temperature. Convective space heating requires an even greater collector inlet temperature than water heating, and the primary load coincides with lower ambient temperature. In many areas of the United States, space heating coincides with low radiation values, which further increases the heat loss parameter. However, many space-heating systems are accompanied by water heating, and some space-heating systems (e.g., radiant panels) are available at a lower $(t_i - t_a)/I_r$ value. Air conditioning with solar-activated absorption equipment is economical only when the cooling season is long and solar radiation levels are high. These devices require at least 180°F water. Thus, on an 80°F day with radiation at 300 Btu/h · ft², the heat loss parameter is 0.3. Higher operating temperatures are desirable to prevent excessive derating of the air conditioner. Only the most efficient (low $F_R U_L$) collector is suitable. Derating may also be minimized by using convective-radiant hybrid air-conditioning systems. Collector efficiency curves may be used as an initial screening device. However, efficiency curves illustrate only instantaneous performance of a collector. They do not include incidence angle effects (which vary throughout the year); heat exchanger effects; probabilities of occurrence of t_i , t_a , and I_r ; system heat loss; or control strategies. Final selection requires determining a collector's long-term energy output and performing cost-effectiveness studies. Estimating a particular collector's and system's annual performance requires appropriate analysis tools such as fChart (Beckman et al. 1977) or TRNSYS (Klein et al. 2010). The Solar Rating and Certification Corporation (SRCC), International Organization for Standardization (ISO), European Committee for Standardization (CEN), and other rating agencies provide day-long collector outputs under various solar and operating conditions. These ratings can greatly simplify selection of the proper collector type, including cost considerations. Table 2 shows a comparison of generic types of liquid flat-plate collectors.

Generic Test Results The generic types of liquid flat-plate collectors are **unglazed**, **single-glazed painted-** and **selective-surface absorbers**, and **evacuated tubes**. Table 2 shows the average output, intercept, and slope for these. The normalized day-long energy output for a clear day in category C points out the difficulties in using only the instantaneous efficiency data. Category C holds the temperature difference constant at 36°F all day, and a clear day is defined as 2000 Btu/ft² · day, which is high for most of the United States. Note that evacuated-tube collectors do not perform very well under these circumstances. SRCC and other rating agencies publish a matrix of collector outputs for various temperature differences and solar days. Table 3 shows the SRCC matrix for each generic collector type. Further explanation of the categories is available on the SRCC website, www.solar-rating.org, or in their standards.

Table 2. Average Performance Parameters ^a for Generic Types of Liquid Flat-Plate Collectors			
Collector Type	Vertical Intercept	Slope, Btu/h · ft ² · °F	Clear-Day Category C Output, Btu/h · ft ² · °F
Unglazed	0.807	-3.289	420
Glazed painted absorber	0.701	-1.15	927
Selective-surface absorber	0.699	-0.814	1007
Evacuated tube	0.554	-0.443	840

^a Derived from data of Solar Rating and Certification Corporation (SRCC), www.solar-rating.org (Oct. 2006).

Table 3. Thermal Performance Ratings ^a for Generic Types of Liquid Flat-Plate Collectors			
Category ^a	Solar Day, Btu/ft ² · day		
	Clear Day 2000	Mildly Cloudy 1500	Cloudy Day 1000
Unglazed			
A	1600	1200	900
B	1000	700	400
C	400	200	—
D	—	—	—
E	—	—	—
Painted			
A	1285	971	658
B	1128	815	533

Category ¹	Solar Day, Btu/Rt ² · day		
	Clear Day 2000	Mildly Cloudy 1500	Cloudy Day 1000
C	908	595	282
D	407	157	—
E	—	—	—
Selective surface			
A	1316	971	658
B	1191	877	564
C	1003	689	376
D	595	313	63
E	219	31	—
Evacuated tube			
A	872	655	436
B	841	623	405
C	810	592	374
D	685	467	280
E	592	374	156
¹ Categories	$T_i - T_a$, °F	Application	
A	−9	Pool heating, warm climate	
B	9	Pool heating, cool climate	
C	36	Water heating, warm climate	
D	90	Water heating, cool climate	
E	144	Air conditioning	

¹ Derived from data of Solar Rating and Certification Corporation (SRCC), www.solar-rating.org (Oct. 2006).

The daily outputs in [Table 3](#) show a clear advantage for each type of collector starting from the upper left corner and proceeding diagonally down and across. At low temperature differences, for categories A and B under clear and mildly cloudy conditions, unglazed collectors provide a clear advantage. The middle of the matrix is a bit less definitive, but painted and selective-surface collectors tend to perform best in the range. At the lower right corner, evacuated tubes perform best under high temperature differences and low solar inputs. Determining where a project falls in this rating matrix is very important in selecting the proper collector; the designer should be familiar with the location's average daily solar input and the desired operating temperature. For example, selective-surface collectors might work well for a high-temperature application in Phoenix in the summer because the daytime temperature difference is low and the solar input is high, whereas evacuated tubes are a better choice in Atlanta.

1.6 THERMAL ENERGY STORAGE

Design and selection of thermal storage equipment is one of the most neglected elements of solar energy systems. In fact, the energy storage system has an enormous influence on overall system cost, performance, and reliability, and its design is highly interactive with that of other system elements (e.g., collector loop, thermal distribution system). Thus, it should be considered within the context of the total system. Energy can be stored in liquids, solids, or phase-change materials (PCMs). Water is the most frequently used liquid storage medium, although the collector loop may use water, oils, or aqueous glycol as a collection fluid. For service water heating and most building space heating, water is normally contained in some type of tank. Air systems typically store heat in rocks or pebbles, but sometimes use the building's structural mass. [Chapter 35 of the 2015 ASHRAE Handbook—HVAC Applications](#) and ASHRAE (1991) cover this topic in more detail.

Air System Thermal Storage

The most common storage media for air collectors have been rocks or a regenerator matrix made from concrete masonry units (CMUs). Gravel was widely used as a storage medium because it is plentiful and relatively inexpensive; however, the inherent cost and likelihood of mold growth in humid climates put an end to its use. Other possible media include PCMs, water, and the inherent building mass. Where large interior temperature swings are tolerable, the building's inherent mass may be sufficient for thermal storage. Designated storage may also be eliminated where the array output seldom exceeds the concurrent demand. Loads requiring no storage are usually the most cost-effective applications of air collectors, and heated air from the collectors can be distributed directly to the space. Water can be used as a storage medium for air collectors by using a conventional heating coil to transfer heat from the air to the water in the storage tank. Advantages of water storage include compatibility with hydronic heating systems and relative compactness (roughly one-third the volume of pebble beds).

Liquid System Thermal Storage

For units large enough for commercial liquid systems, the following factors should be considered:

- Pressurized versus unpressurized storage
- External versus internal heat exchanger
- Single versus multiple tanks
- Steel versus nonmetallic tank(s)
- Type of service [i.e., service hot water (SHW), building space heating (BSH), or a combination of the two]
- Location, space, and accessibility constraints imposed by architectural limitations
- Interconnect constraints imposed by existing mechanical systems
- Limitations imposed by equipment availability

The following sections present examples of the more common configurations. **Pressurized Storage.** Defined here as storage that is open to the city water supply, pressurized storage is preferred for small service water heating systems because it is convenient and provides an economical way of meeting ASME *Boiler and Pressure Vessel Code* requirements with off-the-shelf equipment. Typical storage size is about 1 to 2 gal per square foot of collector area. The largest off-the-shelf water heater is 120 gal; however, no more than three or four of these should be connected in parallel. Hence, the largest storage that can be considered with off-the-shelf water heater tanks is about 360 to 480 gal, and must be compared to a single tank with the additional labor and material to connect them, plus the amount of floor space required. For larger solar hot-water and combined systems, the following concerns are important when selecting storage:

- Higher cost per unit volume of ASME rated tanks in sizes greater than 120 gal
- Handling difficulties because of large weight
- Accessibility to locations suitable for storage
- Interfacing with existing SHW and BSH systems
- Corrosion protection for steel tanks

The choice of pressurized storage for medium-sized systems is based on the availability of suitable low-cost tanks near the site. Identifying a suitable supplier of low-cost tanks can extend the advantages of pressurized storage to larger SHW installations. Storage pressurized at city water supply pressure is not practical for building space heating, except for small applications such as residences, apartments, and small commercial buildings. With pressurized storage, the heat exchanger is always located on the collector side of the tank. Either the internal or external heat exchanger configuration can be used. [Figure 16](#) illustrates the three main types of internal heat exchanger concepts: immersed coil, wraparound jacket, and tube bundle. Small tanks (less than 120 gal) are available with either of the first two heat exchangers already installed. For larger tanks, a large assortment of tube bundle heat exchangers are available that can be incorporated into the tank design by the manufacturer.

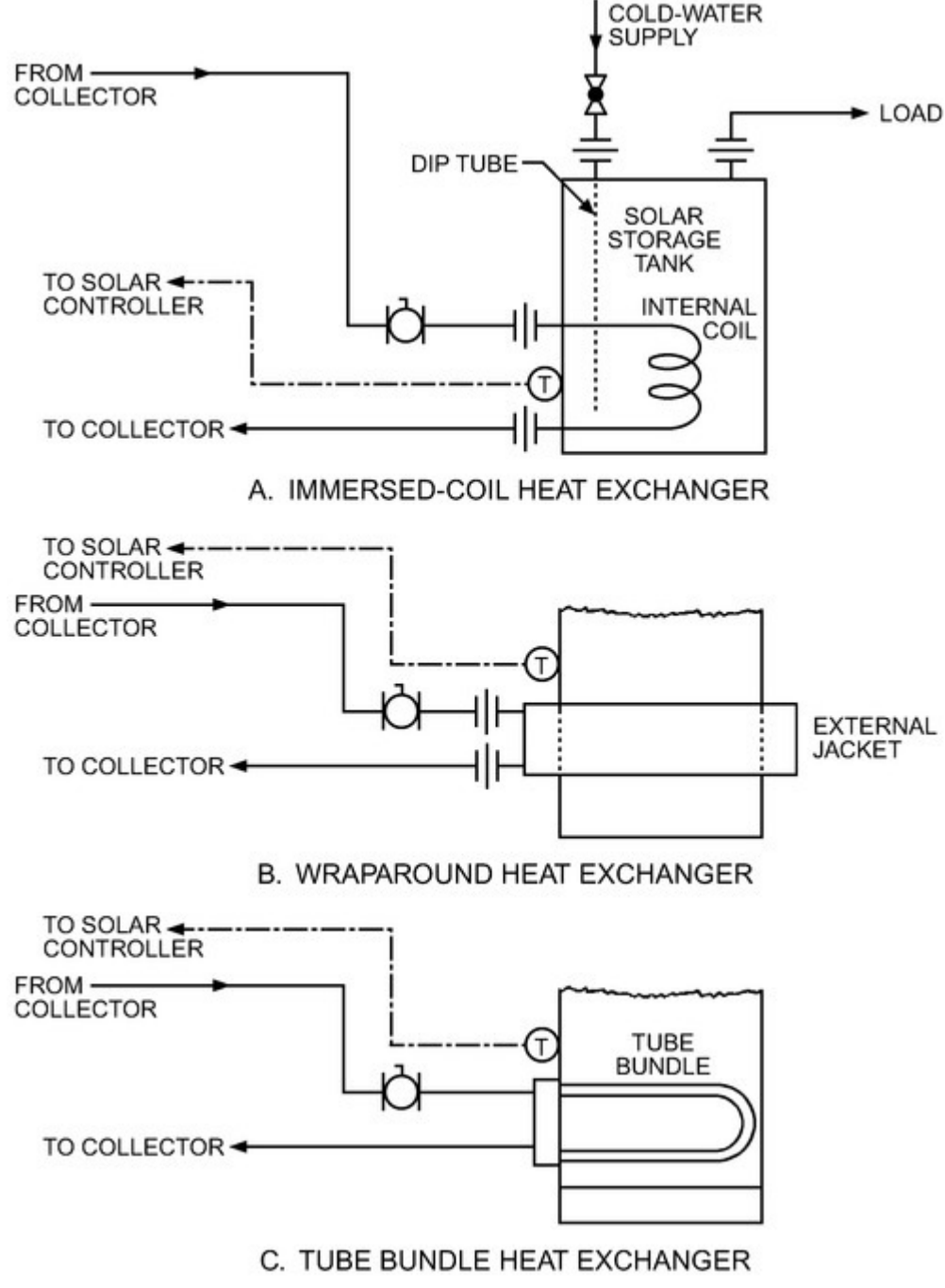


Figure 16. Pressurized Storage with Internal Heat Exchanger

Sometimes, more than one tank is needed to meet design requirements. Additional tanks offer the following benefits:

- Added storage volume
- Increased heat exchanger surface
- Reduced pressure drop in the collection loop
- Increased stratification for larger volumes

Disadvantages compared to a single tank of the same volume include the following:

- Higher installation costs
- Greater space requirements
- Higher heat losses (reduced performance)

[Figure 17](#) illustrates the multiple-tank configuration for pressurized storage. The exchangers are connected in reverse-return fashion to minimize flow imbalance. A third tank may be added. An external heat exchanger provides greater flexibility because the tank and heat exchanger can be selected independently of each other ([Figure 18](#)). A drawback to this method is that an additional pump, with its parasitic energy consumption, is required.

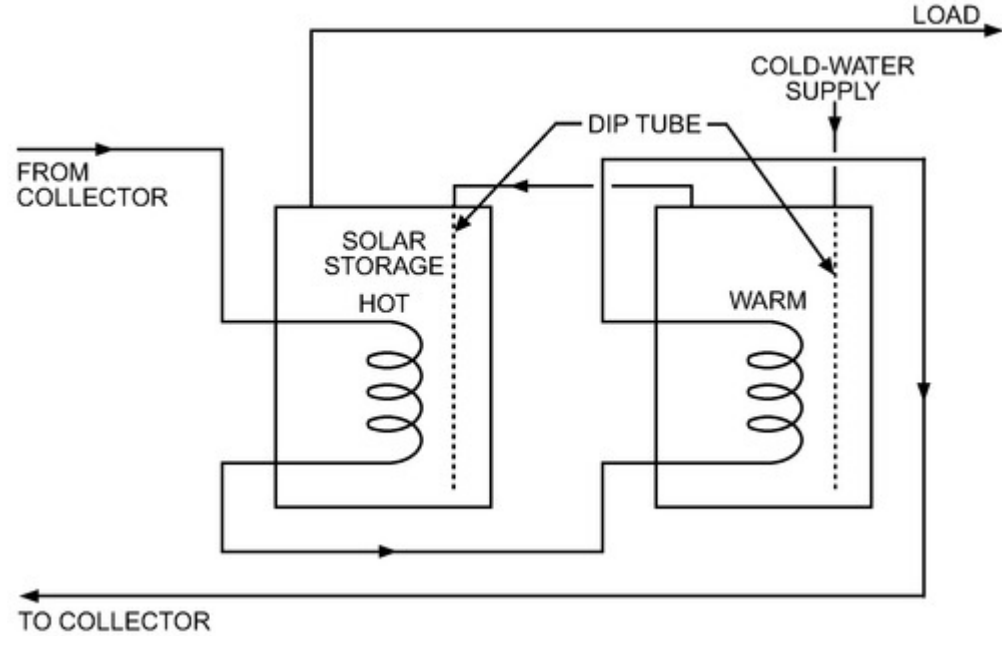


Figure 17. Multiple Storage Tank Arrangement with Internal Heat Exchangers

When selecting an external heat exchanger for a system protected by a nonfreezing liquid, consider the following factors related to start-up after at least one extremely cold night:

- Freeze-up of the water side of the heat exchanger
- Performance loss due to extraction of heat from storage

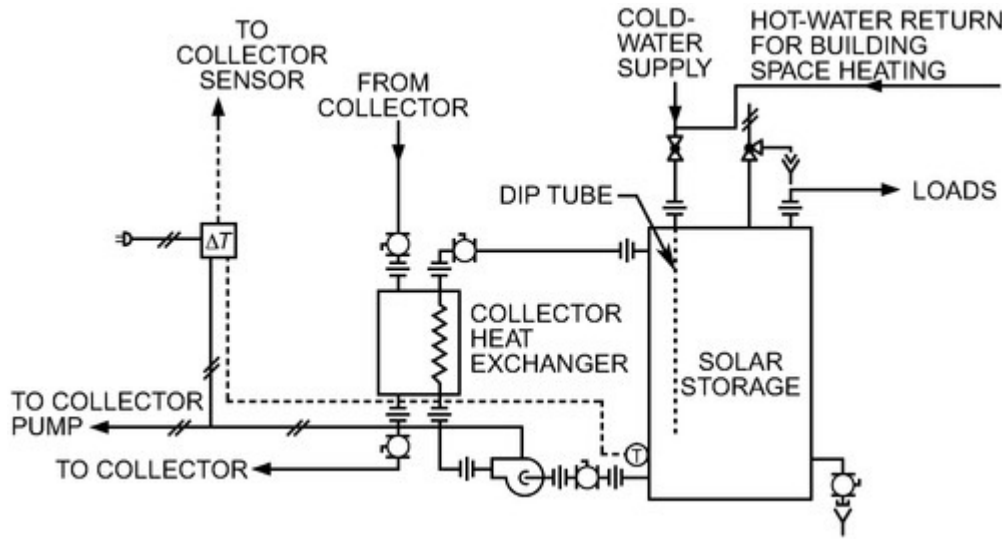


Figure 18. Pressurized Storage System with External Heat Exchanger

For small systems, an internal heat exchanger/tank arrangement prevents the water side of the heat exchanger from freezing. However, the energy required to maintain the water above freezing must be extracted from storage, thereby decreasing overall performance. With the external heat exchanger/tank combination, a bypass can be arranged to divert cold fluid around the heat exchanger until it has been heated to an acceptable level, such as 80°F. When the heat transfer fluid has warmed to this level, it can enter the heat exchanger without freezing or extracting heat from storage. If necessary, this arrangement can also be used with internal heat exchangers to improve performance.

Unpressurized Storage. For systems greater than about 1000 ft³ (1500 gal storage volume minimum), unpressurized storage is usually more cost-effective than pressurized. In this chapter, the term *unpressurized* means tanks at or below the pressure expected in an unvented drainback loop. The pH of the liquid and the electric potential of the metal are the primary governing factors of electrochemical corrosion. A sacrificial anode made from a metal more reactive than steel can provide protection. Magnesium is recommended for solar applications. Because protection ends when the anode has dissolved, the anode must be inspected annually. Oxygen can enter the tank through air dissolved in the water entering the tank or through an air vent. In pressurized storage, oxygen is continually replenished by the incoming water. Besides causing rust, oxygen catalyzes other types of corrosion. Unpressurized storage systems are less susceptible to corrosion caused by oxygen because they can be designed as unvented systems, so corrosion is limited to the small amount of oxygen contained in the initial fill water. The external heat exchanger in [Figure 19](#) provides good system flexibility and freedom in component selection. In some cases, system cost and parasitic power consumption may be reduced by an internal heat exchanger. Some field-fabricated heat exchangers use coiled soft copper tube. For larger systems, where custom fabrication is more feasible, a specified heat exchanger can be installed at the top of the tanks.

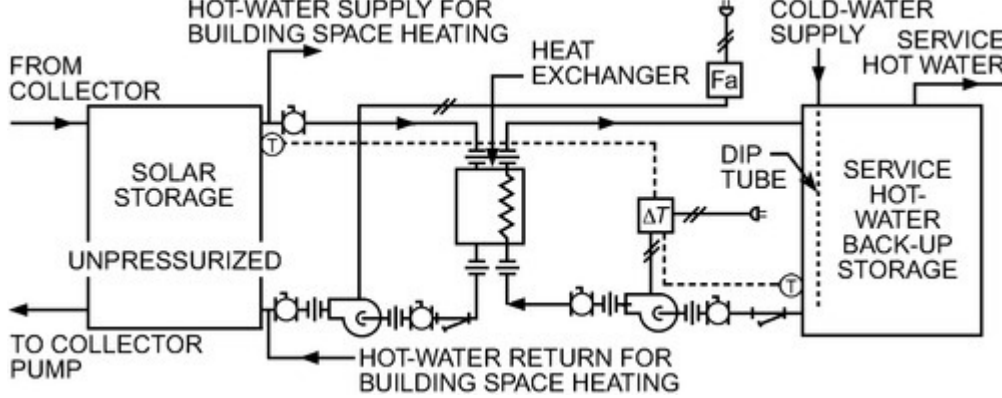


Figure 19. Unpressurized Storage System with External Heat Exchanger

Storage Tank Construction

Steel. For most liquid solar energy systems, steel is the preferred material for storage tank construction. Steel tanks are relatively easy to fabricate to ASME *Pressure Vessel Code* requirements, readily available, and easily attached to pipes and fittings. Steel tanks used for pressures of 30 psi and above must be ASME rated. Because water main pressure is usually above this level, open systems must use ASME code tanks. These pressure-rated storage tanks are more expensive than nonpressurized types. Costs can usually be significantly reduced if a nonpressurized tank can be used. Steel tanks are subject to corrosion, however. Because corrosion rates increase with temperature, the designer must be particularly aware of corrosion protection methods for solar energy applications. A steel tank must be protected against (1) electrochemical corrosion, (2) oxidation (rusting), and (3) galvanic corrosion (Cole et al. 1980). The pH of the liquid and the electric potential of the metal are the primary governing factors of electrochemical corrosion. A sacrificial anode made from a metal more reactive than steel can provide protection. Magnesium is recommended for solar applications. Because protection ends when the anode has dissolved, the anode must be inspected annually. Oxygen can enter the tank through air dissolved in the water entering the tank or through an air vent. In pressurized storage, oxygen is continually replenished by the incoming water. Besides causing rust, oxygen catalyzes other types of corrosion. Unpressurized storage systems are less susceptible to corrosion caused by oxygen because they can be designed as unvented systems, so corrosion is limited to the small amount of oxygen contained in the initial fill water. Coatings on the tank interior protect it from oxidation. Some of the most common coatings include the following:

- **Phenolic epoxy** should be applied in four coats.
- **Baked-on epoxy** is preferred over painted-on epoxy.
- **Glass lining** offers more protection than the epoxies and can be used under severe water conditions.
- **Hydraulic stone** provides the best protection against corrosion and increases the tank's heat retention capabilities. Its weight may cause handling problems in some installations.

The coating should either be flexible enough to withstand extreme thermal cycling or have the same coefficient of expansion as the steel tank. In the United States, all linings used for potable water tanks should be approved by the Food and Drug Administration (FDA) for the maximum temperature expected in the tank. Dissimilar materials in an electrolyte (water) are in electrical contact with each other and corrode by galvanic action. Copper fittings screwed into a steel tank corrode the steel, for example. Galvanic corrosion can be minimized by using dielectric bushings to connect pipes to tanks.

Plastic. Fiberglass-reinforced plastic (FRP) tanks offer the advantages of low weight, high corrosion resistance, and low cost. Premium-quality resins allow operating temperatures as high as 210°F, well above the temperature imposed by flat-plate solar collectors. Before delivery of an FRP tank is accepted, inspect the tank for damage that may have occurred during shipment. The gel coat on the inside must be intact, and no glass fibers should be exposed.

Concrete. Concrete vessels lined with a waterproofing membrane may also be used (in vented systems only) to contain a liquid thermal storage medium. Concrete storage tanks are inexpensive, and can be shaped to fit almost any retrofit application. Also, they have excellent resistance to the loading that occurs when they are placed below grade. Concrete tanks must have smooth corners and edges.

Concrete storage vessels have some disadvantages. Seepage often occurs unless a proper waterproofing surface is applied. Waterproofing paints are generally unsatisfactory because the concrete often cracks after setting. Other problems may occur because of poor workmanship, poor location, or poor design. Usually, tanks should stand alone and not be integrated with a building or other structure. Pay careful attention to expansion joints and seams because they are particularly difficult to seal. Sealing at pipe taps can be a problem. Penetrations should be above liquid level, if possible.

Finally, concrete tanks are heavy and may be more difficult to support in a proper location. Their weight may make insulating the tank bottom more expensive and difficult.

Storage Tank Insulation

Heat loss from storage tanks and appurtenances is one of the major causes of poor system performance. The average R-value of storage tanks in solar applications is about half the insulation design value because of poorly insulated supports. Different standards recommend various design criteria for tank insulation. Performance requirements for unfired storage tanks are included in the service-water-heating sections of ASHRAE *Standards* 90.1 and 90.2. A 12 h maximum capacity loss of 1 to 2% should be feasible. The following equation can be used to calculate the insulation R-value for this requirement:

$$\frac{1}{R} = \frac{fQ}{A\theta} \frac{1}{(t_{avg} - t_a)}$$

(9)

where

R = thermal resistivity of insulation, ft² · h · °F/Btu

f = specified fraction of stored energy that can be lost in time θ

Q = stored energy, Btu

A = exposed surface area of storage unit, ft²

θ = given time period, h

t_{avg} = average temperature in storage unit, °F

t_a = ambient temperature surrounding storage unit during heating season, °F

The insulation factor *RQ/Aθ* is found from [Table 4](#) for various tank shapes (Cole et al. 1980).

Most solar water-heating systems use large steel pressure vessels, which are usually shipped uninsulated. Materials suitable for field insulation include fiberglass, rigid foam, and flexible foam blankets. Fiberglass is easy to transport and make fire retardant, but it requires significant labor to apply and seal.

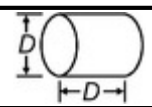
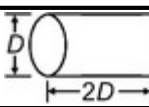
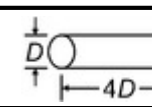
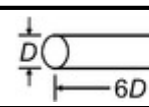
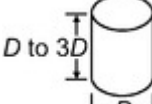
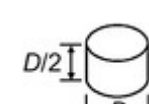
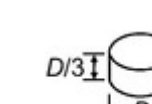
Another widely used insulation consists of rigid sheets of polyisocyanurate foam cut and taped around the tank. Material that is 3 to 4 in. thick can provide R-20 to R-30 insulation value. Rigid foam insulation is sprayed directly onto the tank from a foaming truck or in a shop. It bonds well to the tank surface (no air space between tank and insulation) and insulates better than an equivalent thickness of fiberglass. When most foams are exposed to flames, they ignite and/or produce toxic gases. When located in or adjacent to a living space, they often must be protected by a fire barrier and/or sprinkler system.

Some tank manufacturers and suppliers offer custom tank jackets of flexible foam with zipper-like connections that provide quick installation and neat appearance. Some of the fire considerations for rigid foam apply to these foam blankets.

Tank supports are a major source of heat loss. To provide suitable load-bearing capability, the supports must be in direct contact with the tank wall or attached to it. Thermal breaks must be provided between the supports and the tank ([Figure 20](#)). If insulating the tank from the support is impractical, the external surface of the supports must be insulated, and the supports must be placed on insulative material that can support the compressive load. Wood, foam glass, and closed-cell foam can be used, depending on the compressive load.

Stratification and Short Circuiting

Because hot water rises and cold water sinks in a vessel, the pipe to the collector inlet should always be connected to the bottom of the tank. The collector then operates at its best efficiency because it is always at the lowest possible temperature. The return from the collector should always run near the bottom of the tank to encourage stratification and avoid introducing cooler fluid to the top of the tank after a draw occurs. Techniques such as using diffusers and lower flow rates have been developed to reduce destratification of flow returning from the collector. Generally, the better the tank stratification, the better the solar system performs overall. Similarly, the load should be extracted from the top of the tank, and cold makeup water should be introduced at the bottom. Because of increased static pressure, vertical storage tanks enhance stratification better than horizontal tanks.

Table 4. Insulation Factor <i>RQ/Aθ</i> for Cylindrical Water Tanks				
Size, gal	Horizontal Tank Insulation Factor, Btu/h · ft ²			
				
250	3.63	3.46	3.05	2.77
500	4.57	4.36	4.84	3.49
750	5.24	4.99	4.40	3.99
1000	5.76	5.49	4.84	4.39
1500	6.60	6.28	5.54	5.03
2000	7.26	6.92	6.10	5.53
3000	8.31	7.92	6.98	6.33
4000	9.15	8.71	7.68	6.97
5000	9.86	9.39	8.28	7.51
Size, gal	Vertical Tank Insulation Factor, Btu/h · ft ²			
				

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