

SOLAR ENERGY USE

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THE major obstacle encountered in solar heating and cooling is economic—the equipment needed to collect and store solar energy is expensive. In some cases, the cost of the solar equipment is greater than the resulting savings in fuel costs. Some of the challenges inherent in using solar radiation are that it is

- Relatively low in intensity, rarely exceeding 300 Btu/h·ft². Consequently, when large amounts of energy are needed, large collectors must be used.
- Intermittent because solar radiation intensity varies from zero at sunrise to a maximum at noon and back to zero at sunset. Some means of energy storage must be provided at night and during periods of low solar radiation.
- Subject to unpredictable interruptions by clouds, rain, snow, hail, or dust.

Systems should make maximum use of the solar energy input by effectively using the energy at the lowest temperatures possible.

QUALITY AND QUANTITY OF SOLAR ENERGY

Solar Constant

Solar energy approaches the earth as electromagnetic radiation, with wavelengths ranging from 0.1 μm (x-rays) to 100 m (radio waves). The earth maintains a thermal equilibrium between the annual input of shortwave radiation (0.3 to 2.0 μm) from the sun and the outward flux of longwave radiation (3.0 to 30 μm). Only a limited band need be considered in terrestrial applications, because 99% of the sun's radiant energy has wavelengths between 0.28 and 4.96 μm. The current value of the solar constant (which is defined as the intensity of solar radiation on a surface normal to the sun's rays, just beyond the earth's atmosphere at the average earth-sun distance) is 433 Btu/h·ft². The section on Determining Incident Solar Flux in Chapter 30 of the 2001 *ASHRAE Handbook—Fundamentals* has further information on this topic.

Solar Angles

The axis about which the earth rotates is tilted at an angle of 23.45° to the plane of the earth's orbital plane and the sun's equator. The earth's tilted axis results in a day-by-day variation of the angle between the earth-sun line and the earth's equatorial plane, called the **solar declination** δ . This angle varies with the date, as shown in [Table 1](#) for the year 1964 and in [Table 2](#) for 1977. For other dates, the declination may be estimated by the following equation:

$$\delta = 23.45 \sin \left[360^\circ \frac{284 + N}{365} \right] \quad (1)$$

The preparation of this chapter is assigned to TC 6.7, Solar Energy Utilization.

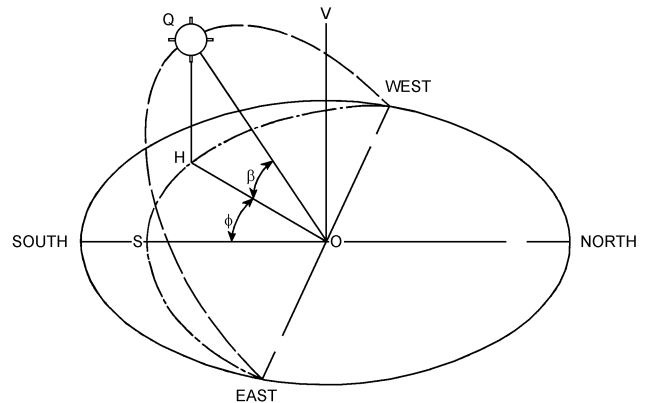


Fig. 1 Apparent Daily Path of the Sun Showing Solar Altitude (β) and Solar Azimuth (ϕ)

where N = year day, with January 1 = 1. For values of N , see [Tables 1 and 2](#).

The relationship between δ and the date from year to year varies to an insignificant degree. The daily change in the declination is the primary reason for the changing seasons, with their variation in the distribution of solar radiation over the earth's surface and the varying number of hours of daylight and darkness.

The earth's rotation causes the sun's apparent motion ([Figure 1](#)). The position of the sun can be defined in terms of its altitude β above the horizon (angle HOQ) and its azimuth ϕ , measured as angle HOS in the horizontal plane.

At solar noon, the sun is exactly on the meridian, which contains the south-north line. Consequently, the solar azimuth ϕ is 0°. The noon altitude β_N is given by the following equation as

$$\beta_N = 90^\circ - \text{LAT} + \delta \quad (2)$$

where LAT = latitude.

Because the earth's daily rotation and its annual orbit around the sun are regular and predictable, the solar altitude and azimuth may be readily calculated for any desired time of day when the latitude, longitude, and date (declination) are specified. Apparent solar time (AST) must be used, expressed in terms of the hour angle H , where

$$\begin{aligned} H &= (\text{number of hours from solar noon}) \times 15^\circ \\ &= \frac{\text{number of minutes from solar noon}}{4} \end{aligned} \quad (3)$$

Solar Time

Apparent solar time (AST) generally differs from local standard time (LST) or daylight saving time (DST), and the difference can be significant, particularly when DST is in effect. Because the sun

Table 1 Date, Declination, and Equation of Time for the 21st Day of Each Month of 1964, with Data (A, B, C)
Used to Calculate Direct Normal Radiation Intensity at the Earth's Surface

	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec
Year Day	21	52	80	111	141	172	202	233	264	294	325	355
Declination δ , degrees	-19.9	-10.6	0.0	+11.9	+20.3	+23.45	+20.5	+12.1	0.0	-10.7	-19.9	-23.45
Equation of time, minutes	-11.2	-13.9	-7.5	+1.1	+3.3	-1.4	-6.2	-2.4	+7.5	+15.4	+13.8	+1.6
Solar noon	late		early		late		early					
A, Btu/h·ft ²	390	385	376	360	350	345	344	351	365	378	387	391
B, dimensionless	0.142	0.144	0.156	0.180	0.196	0.205	0.207	0.201	0.177	0.160	0.149	0.142
C, dimensionless	0.058	0.060	0.071	0.097	0.121	0.134	0.136	0.122	0.092	0.073	0.063	0.057

A = apparent solar irradiation at air mass zero for each month.

B = atmospheric extinction coefficient.

C = ratio of the diffuse radiation on a horizontal surface to the direct normal irradiation.

Table 2 Solar Position Data for 1977

Date	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec
1 Year Day	1	32	60	91	121	152	182	213	244	274	305	335
Declination δ	-23.0	-17.0	-7.4	+4.7	+15.2	+22.1	+23.1	+17.9	+8.2	-3.3	-14.6	-21.9
Eq of Time	-3.6	-13.7	-12.5	-4.0	+2.9	+2.4	-3.6	-6.2	+0.0	+10.2	+16.3	+11.0
6 Year Day	6	37	65	96	126	157	187	218	249	279	310	340
Declination δ	-22.4	-15.5	-5.5	+6.6	+16.6	+22.7	+22.7	+16.6	+6.7	-5.3	-16.1	-22.5
Eq of Time	-5.9	-14.2	-11.4	-2.5	+3.5	+1.6	-4.5	-5.8	+1.6	+11.8	+16.3	+9.0
11 Year Day	11	42	70	101	131	162	192	223	254	284	315	345
Declination δ	-21.7	-13.9	-3.5	-8.5	+17.9	+23.1	+22.1	+15.2	+4.4	-7.2	-17.5	-23.0
Eq of Time	-8.0	-14.4	-10.2	-1.1	+3.7	+0.6	-5.3	-5.1	+3.3	+13.1	+15.9	+6.8
16 Year Day	16	47	75	106	136	167	197	228	259	289	320	350
Declination δ	-20.8	-12.2	-1.6	+10.3	+19.2	+23.3	+21.3	+13.6	+2.5	-8.7	-18.8	-23.3
Eq of Time	-9.8	-14.2	-8.8	+0.1	+3.8	-0.4	-5.9	-4.3	+5.0	+14.3	+15.2	+4.4
21 Year Day	21	52	80	111	141	172	202	233	264	294	325	355
Declination δ	-19.6	-10.4	+0.4	+12.0	+20.3	+23.4	+20.6	+12.0	+0.5	-10.8	-20.0	-23.4
Eq of Time	-11.4	-13.8	-7.4	+1.2	+3.6	-1.5	-6.2	-3.1	+6.8	+15.3	+14.1	+2.0
26 Year Day	26	57	85	116	146	177	207	238	269	299	330	360
Declination δ	-18.6	-8.6	+2.4	+13.6	+21.2	+23.3	+19.3	+10.3	-1.4	-12.6	-21.0	-23.4
Eq of Time	-12.6	-13.1	-5.8	+2.2	+3.2	-2.6	-6.4	-1.8	+8.6	+15.9	+12.7	-0.5

Source: ASHRAE Standard 93-1986 (Reaffirmed 1991).

Notes: Units for declination are angular degrees; units for equation of time are minutes. Values of declination and equation of time vary slightly for specific dates in other years.

appears to move at the rate of 360° in 24 h, its apparent rate of motion is 4 min per degree of longitude. The AST can be determined from the following equation:

$$\text{AST} = \text{LST} + \text{Equation of Time} + (4 \text{ min})(\text{LST Meridian} - \text{Local Longitude}) \quad (4)$$

The longitudes of the seven standard time meridians that affect North America are Atlantic ST, 60°; Eastern ST, 75°; Central ST, 90°; Mountain ST, 105°; Pacific ST, 120°; Yukon ST, 135°; and Alaska-Hawaii ST, 150°.

The equation of time is the measure, in minutes, of the extent by which solar time, as determined by a sundial, runs faster or slower than local standard time (LST), as determined by a clock that runs at a uniform rate. Table 1 gives values of the declination of the sun and the equation of time for the 21st day of each month for the year 1964 (when the ASHRAE solar radiation tables were first calculated), and Table 2 gives values of δ and the equation of time for six days each month for the year 1977.

Example 1. Find AST at noon DST on July 21 for Washington, D.C., longitude = 77°, and for Chicago, longitude = 87.6°.

Solution: Noon DST is actually 11:00 A.M. LST. Washington is in the eastern time zone, and the LST meridian is 75°. From Table 1, the equation of time for July 21 is -6.2 min. Thus, from Equation (4), noon DST for Washington is actually

$$\text{AST} = 11:00 - 6.2 + 4(75 - 77) = 10:45.8 \text{ AST} = 10.76 \text{ h}$$

Chicago is in the central time zone, and the LST meridian is 90°. Thus, from Equation (4), noon central DST is

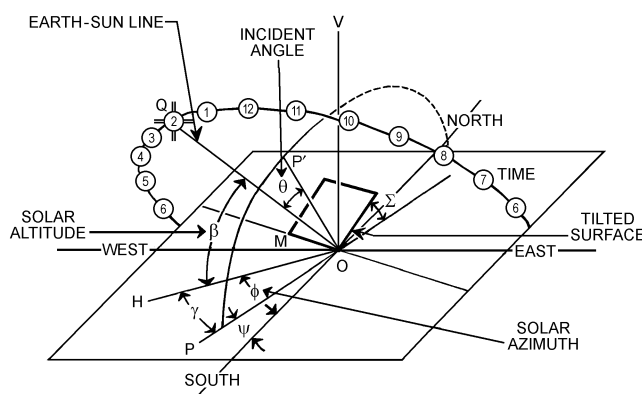


Fig. 2 Solar Angles with Respect to a Tilted Surface

$$\text{AST} = 11:00 - 6.2 + 4(90 - 87.6) = 11:03.4 \text{ AST} = 11.06 \text{ h}$$

The hour angles H for these two examples (see Figure 2) are

$$\text{for Washington, } H = (12.00 - 10.76)15^\circ = 18.6^\circ \text{ east}$$

$$\text{for Chicago, } H = (12.00 - 11.06)15^\circ = 14.10^\circ \text{ east}$$

To find the solar altitude β and the azimuth ϕ when the hour angle H , the latitude LAT , and the declination δ are known, the following equations may be used:

$$\sin \beta = \cos(\text{LAT}) \cos \delta \cos H + \sin(\text{LAT}) \sin \delta \quad (5)$$

$$\sin \phi = \cos \delta \sin H / \cos \beta \quad (6)$$

$$\text{or} \quad \cos \phi = \frac{\sin \beta \sin(\text{LAT}) - \sin \delta}{\cos \beta \cos(\text{LAT})} \quad (7)$$

Tables 15 through 21 in Chapter 29 of the 1997 *ASHRAE Handbook—Fundamentals* give values for latitudes from 16 to 64° north. For any other date or latitude, interpolation between the tabulated values will give sufficiently accurate results. More precise values, with azimuths measured from the north, are given in the U.S. Hydrographic Office Bulletin No. 214 (1958).

Incident Angle

The angle between the line normal to the irradiated surface (OP' in Figure 2) and the earth-sun line OQ is called the incident angle θ . It is important in solar technology because it affects the intensity of the direct component of the solar radiation striking the surface and the ability of the surface to absorb, transmit, or reflect the sun's rays.

To determine θ , the surface azimuth ψ and the surface-solar azimuth γ must be known. The surface azimuth (angle POS in Figure 2) is the angle between the south-north line SO and the normal PO to the intersection of the irradiated surface with the horizontal plane, shown as line OM. The surface-solar azimuth, angle HOP, is designated by γ and is the angular difference between the solar azimuth ϕ and the surface azimuth ψ . For surfaces facing east of south, $\gamma = \phi - \psi$ in the morning and $\gamma = \phi + \psi$ in the afternoon. For surfaces facing west of south, $\gamma = \phi + \psi$ in the morning and $\gamma = \phi - \psi$ in the afternoon. For south-facing surfaces, $\psi = 0^\circ$, so $\psi = \phi$ for all conditions. The angles δ , β , and ϕ are always positive.

For a surface with a tilt angle Σ (measured from the horizontal), the angle of incidence θ between the direct solar beam and the normal to the surface (angle QOP' in Figure 2) is given by:

$$\cos \theta = \cos \beta \cos \gamma \sin \Sigma + \sin \beta \cos \Sigma \quad (8)$$

For vertical surfaces, $\Sigma = 90^\circ$, $\cos \Sigma = 0$, and $\sin \Sigma = 1.0$, so Equation (8) becomes

$$\cos \theta = \cos \beta \cos \gamma \quad (9)$$

For horizontal surfaces, $\Sigma = 0^\circ$, $\sin \Sigma = 0$, and $\cos \Sigma = 1.0$, so Equation (8) leads to

$$\theta_H = 90^\circ - \beta \quad (10)$$

Example 2. Find θ for a south-facing surface tilted upward 30° from the horizontal at 40° north latitude at 4:00 P.M., AST, on August 21.

Solution: From Equation (3), at 4:00 P.M. on August 21,

$$H = 4 \times 15^\circ = 60^\circ$$

From Table 1,

$$\delta = 12.1^\circ$$

From Equation (5),

$$\begin{aligned} \sin \beta &= \cos 40^\circ \cos 12.1^\circ \cos 60^\circ + \sin 40^\circ \sin 12.1^\circ \\ \beta &= 30.6^\circ \end{aligned}$$

From Equation (6),

$$\begin{aligned} \sin \phi &= \cos 12.1^\circ \sin 60^\circ / \cos 30.6^\circ \\ \phi &= 79.7^\circ \end{aligned}$$

The surface faces south, so $\phi = \gamma$. From Equation (8),

$$\begin{aligned} \cos \theta &= \cos 30.6^\circ \cos 79.7^\circ \sin 30^\circ + \sin 30.6^\circ \cos 30^\circ \\ \theta &= 58.8^\circ \end{aligned}$$

ASHRAE *Standard* 93, Methods of Testing to Determine the Thermal Performance of Solar Collectors, provides tabulated values of θ for horizontal and vertical surfaces and for south-facing surfaces tilted upward at angles equal to the latitude minus 10° , the latitude, the latitude plus 10° , and the latitude plus 20° . These tables cover the latitudes from 24° to 64° north, in 8° intervals.

Solar Spectrum

Beyond the earth's atmosphere, the effective black body temperature of the sun is $10,370^\circ\text{R}$. The maximum spectral intensity occurs at $0.48 \mu\text{m}$ in the green portion of the visible spectrum. Thekaekara (1973) presents tables and charts of the sun's extraterrestrial spectral irradiance from 0.120 to $100 \mu\text{m}$, the range in which most of the sun's radiant energy is contained. The ultraviolet portion of the spectrum below $0.40 \mu\text{m}$ contains 8.73% of the total, another 38.15% is contained in the visible region between 0.40 and $0.70 \mu\text{m}$, and the infrared region contains the remaining 53.12%.

Solar Radiation at the Earth's Surface

In passing through the earth's atmosphere, some of the sun's direct radiation I_D is scattered by nitrogen, oxygen, and other molecules, which are small compared to the wavelengths of the radiation; and by aerosols, water droplets, dust, and other particles with diameters comparable to the wavelengths (Gates 1966). This scattered radiation causes the sky to appear blue on clear days, and some of it reaches the earth as diffuse radiation I .

Attenuation of the solar rays is also caused by absorption, first by the ozone in the outer atmosphere, which causes a sharp cutoff at $0.29 \mu\text{m}$ of the ultraviolet radiation reaching the earth's surface (Figure 3). In the longer wavelengths, there are series of absorption bands caused by water vapor, carbon dioxide, and ozone. The total amount of attenuation at any given location is determined by (1) the length of the atmospheric path through which the rays travel and (2) the composition of the atmosphere. The path length is expressed in terms of the air mass m , which is the ratio of the mass of atmosphere in the actual earth-sun path to the mass that would exist if the sun were directly overhead at sea level ($m = 1.0$). For all practical purposes, at sea level, $m = 1.0/\sin \beta$. Beyond the earth's atmosphere, $m = 0$.

Before 1967, solar radiation data were based on an assumed solar constant of $419.7 \text{ Btu/h} \cdot \text{ft}^2$ and on a standard sea level atmosphere containing the equivalent depth of 2.8 mm of ozone, 20 mm of precipitable moisture, and 300 dust particles per cubic centimeter. Threlkeld and Jordan (1958) considered the wide variation of water vapor in the atmosphere above the United States at any given time, and particularly the seasonal variation, which finds three times as much moisture in the atmosphere in midsummer as in December, January, and February. The basic atmosphere was assumed to be at sea level barometric pressure, with 2.5 mm of ozone, 200 dust particles per cm^3 , and an actual precipitable moisture content that varied throughout the year from 8 mm in midwinter to 28 mm in mid-July. Figure 4 shows the variation of the direct normal irradiation with solar altitude, as estimated for clear atmospheres and for an atmosphere with variable moisture content.

Stephenson (1967) showed that the intensity of the direct normal irradiation I_{DN} at the earth's surface on a clear day can be estimated by the following equation:

$$I_{DN} = A e^{-B/\sin \beta} \quad (11)$$

where A , the apparent extraterrestrial irradiation at $m = 0$, and B , the atmospheric extinction coefficient, are functions of the date and take into account the seasonal variation of the earth-sun distance and the air's water vapor content.

The values of the parameters A and B given in Table 1 were selected so that the resulting value of I_{DN} would be in close agreement

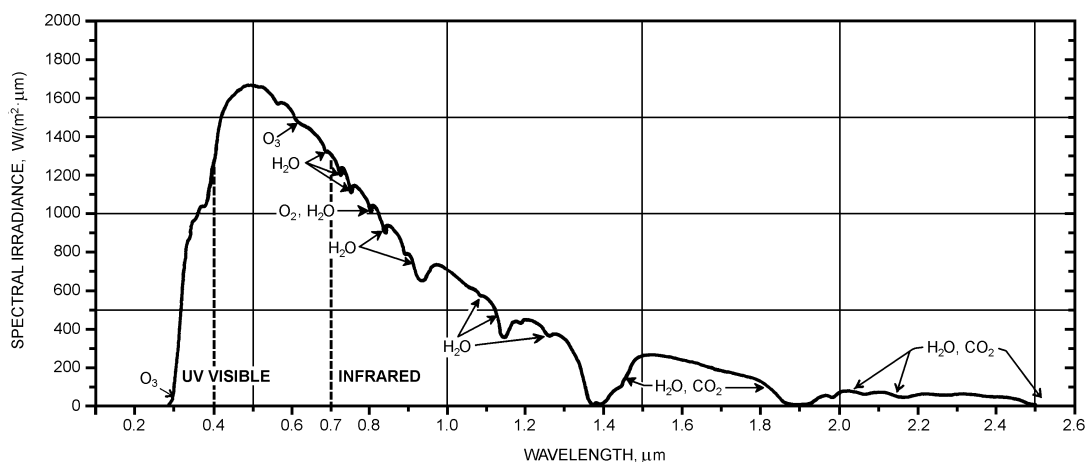


Fig. 3 Spectral Solar Irradiation at Sea Level for Air Mass = 1.0

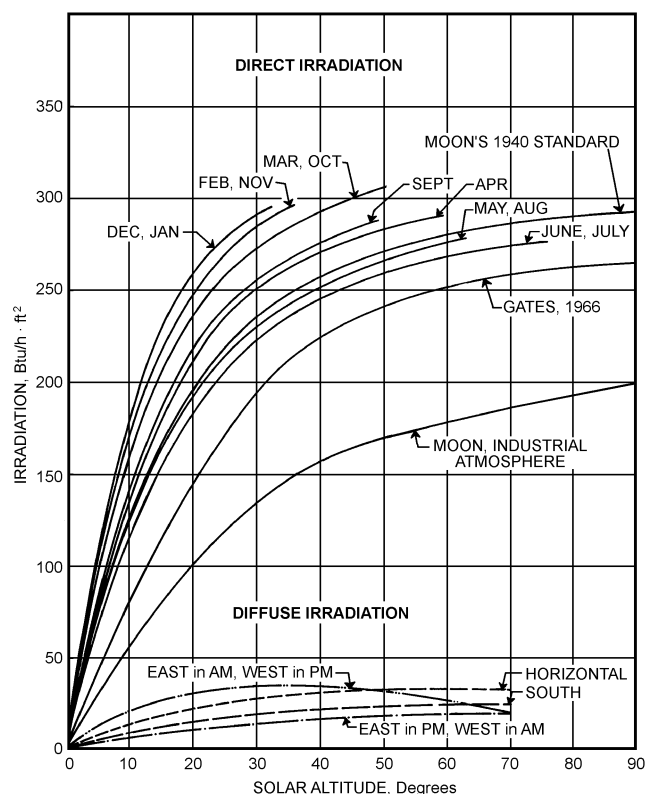


Fig. 4 Variation with Solar Altitude and Time of Year for Direct Normal Irradiation

with the Threlkeld and Jordan (1958) values on average cloudless days. The values of I_{DN} given in Tables 15 through 21 in Chapter 29 of the 1997 *ASHRAE Handbook—Fundamentals*, were obtained by using Equation (11) and data from Table 1. The values of the solar altitude β and the solar azimuth ϕ may be obtained from Equations (5) and (6).

Because local values of atmospheric water content and elevation can vary markedly from the sea level average, the concept of **clearness number** was introduced to express the ratio between the actual clear-day direct irradiation intensity at a specific location and the intensity calculated for the standard atmosphere for the same location and date.

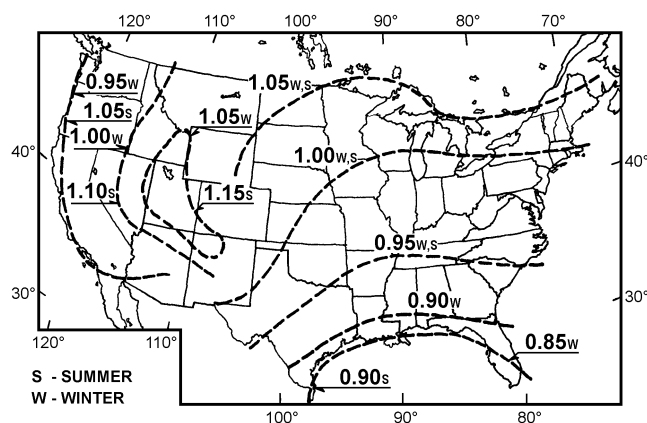


Fig. 5 Clearness Numbers for the United States

Figure 5 shows the Threlkeld-Jordan map of winter and summer clearness numbers for the continental United States. Irradiation values should be adjusted by the clearness numbers applicable to each particular location.

Design Values of Total Solar Irradiation

The total solar irradiation $I_{t\theta}$ of a terrestrial surface of any orientation and tilt with an incident angle θ is the sum of the direct component $I_{DN} \cos \theta$ plus the diffuse component $I_{d\theta}$ coming from the sky plus whatever amount of reflected shortwave radiation I_r may reach the surface from the earth or from adjacent surfaces:

$$I_{t\theta} = I_{DN} \cos \theta + I_{d\theta} + I_r \quad (12)$$

The diffuse component is difficult to estimate because of its non-directional nature and its wide variations. Figure 4 shows typical values of diffuse irradiation of horizontal and vertical surfaces. For clear days, Threlkeld (1963) has derived a dimensionless parameter (designated as C in Table 1), which depends on the dust and moisture content of the atmosphere and thus varies throughout the year:

$$C = \frac{I_{dH}}{I_{DN}} \quad (13)$$

where I_{dH} is the diffuse radiation falling on a horizontal surface under a cloudless sky.

The following equation may be used to estimate the amount of diffuse radiation $I_{d\theta}$ that reaches a tilted or vertical surface:

$$I_{d\theta} = CI_{DN}F_{ss} \quad (14)$$

where

$$F_{ss} = \frac{1 + \cos \Sigma}{2} \quad (15)$$

= angle factor between the surface and the sky

The reflected radiation I_r from the foreground is given by the following equation:

$$I_r = I_{tH} \rho_g F_{sg} \quad (16)$$

where

ρ_g = reflectance of the foreground
 I_{tH} = total horizontal irradiation

$$F_{sg} = \frac{1 + \cos \Sigma}{2} \quad (17)$$

= angle factor between the surface and the earth

The intensity of the reflected radiation that reaches any surface depends on the nature of the reflecting surface and on the incident angle between the sun's direct beam and the reflecting surface. Many measurements made of the reflection (albedo) of the earth under varying conditions show that clean, fresh snow has the highest reflectance (0.87) of any natural surface.

Threlkeld (1963) gives values of reflectance for commonly encountered surfaces at solar incident angles from 0 to 70°. Bituminous paving generally reflects less than 10% of the total incident solar irradiation; bituminous and gravel roofs reflect from 12 to 15%; concrete, depending on its age, reflects from 21 to 33%. Bright green grass reflects 20% at $\theta = 30^\circ$ and 30% at $\theta = 65^\circ$.

The maximum daily amount of solar irradiation that can be received at any given location is that which falls on a flat plate with its surface kept normal to the sun's rays so it receives both direct and diffuse radiation. For fixed flat-plate collectors, the total amount of clear day irradiation depends on the orientation and slope. As shown by Figure 6 for 40° north latitude, the total irradiation of horizontal surfaces reaches its maximum in midsummer, while vertical south-facing surfaces experience their maximum irradiation during the winter. These curves show the combined effects of the varying length of days and changing solar altitudes.

In general, flat-plate collectors are mounted at a fixed tilt angle Σ (above the horizontal) to give the optimum amount of irradiation for each purpose. Collectors intended for winter heating benefit from higher tilt angles than those used to operate cooling systems in summer. Solar water heaters, which should operate satisfactorily throughout the year, require an angle that is a compromise between the optimal values for summer and winter. Figure 6 shows the monthly variation of total day-long irradiation on the 21st day of each month at 40° north latitude for flat surfaces with various tilt angles.

Tables in ASHRAE Standard 93 give the total solar irradiation for the 21st day of each month at latitudes 24 to 64° north on surfaces with the following orientations: normal to the sun's rays (direct normal data do not include diffuse irradiation); horizontal; south-facing, tilted at (LAT-10), LAT, (LAT+10), (LAT+20), and 90° from the horizontal. The day-long total irradiation for fixed surfaces is highest for those that face south, but a deviation in azimuth of 15 to 20° causes only a small reduction.

Solar Energy for Flat-Plate Collectors

The preceding data apply to clear days. The irradiation for average days may be estimated for any specific location by referring to publications of the U.S. Weather Service. The *Climatic Atlas of*

the United States (U.S. GPO 1968) gives maps of monthly and annual values of percentage of possible sunshine, total hours of sunshine, mean solar radiation, mean sky cover, wind speed, and wind direction.

The total daily horizontal irradiation data reported by the U.S. Weather Bureau for approximately 100 stations prior to 1964 show that the percentage of total clear-day irradiation is approximately a linear function of the percentage of possible sunshine. The irradiation is not zero for days when the percentage of possible sunshine is reported as zero, because substantial amounts of energy reach the earth in the form of diffuse radiation. Instead, the following relationship exists for the percentage of possible sunshine:

$$\frac{\text{Day-long actual } I_{tH}}{\text{Clear day } I_{tH}} 100 = a + b \quad (18)$$

where a and b are constants for any specified month at any given location. See also Duffie and Beckman (1974) and Jordan and Liu (1977).

Longwave Atmospheric Radiation

In addition to the shortwave (0.3 to 2.0 μm) radiation it receives from the sun, the earth receives longwave radiation (4 to 100 μm , with maximum intensity near 10 μm) from the atmosphere. In turn, a surface on the earth emits longwave radiation q_{Rs} in accordance with the Stefan-Boltzmann law:

$$q_{Rs} = e_s \sigma T_s^4 \quad (19)$$

where

e_s = surface emittance
 σ = Stefan-Boltzmann constant, $0.1712 \times 10^{-8} \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{R}^4$
 T_s = absolute temperature of the surface, $^\circ\text{R}$

For most nonmetallic surfaces, the longwave hemispheric emittance is high, ranging from 0.84 for glass and dry sand to 0.95 for black built-up roofing. For highly polished metals and certain selective surfaces, e_s may be as low as 0.05 to 0.20.

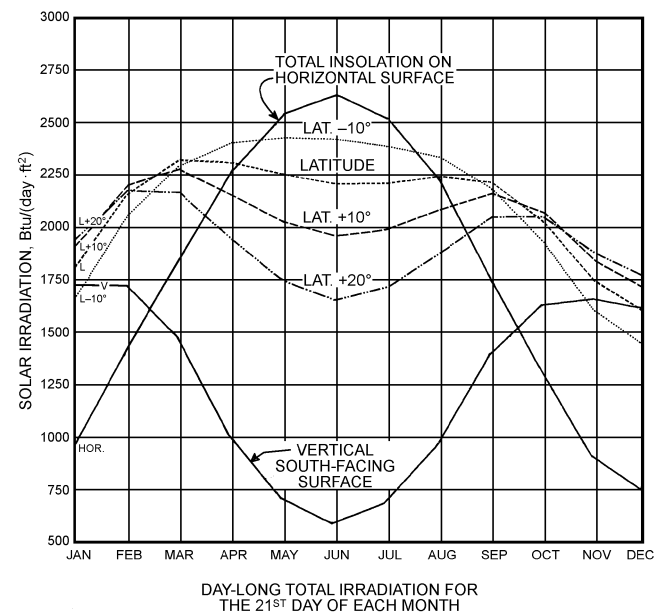


Fig. 6 Total Daily Irradiation for Horizontal, Tilted, and Vertical Surfaces at 40° North Latitude

Table 3 Sky Emittance and Amount of Precipitable Moisture Versus Dew-Point Temperature

Dew Point, °F	Sky Emittance, e_{at}	Precipitable Water, in.
-20	0.68	0.12
-10	0.71	0.16
0	0.73	0.18
10	0.76	0.22
20	0.77	0.29
30	0.79	0.41
40	0.82	0.57
50	0.84	0.81
60	0.86	1.14
70	0.88	1.61

Atmospheric radiation comes primarily from water vapor, carbon dioxide, and ozone (Bliss 1961); very little comes from oxygen and nitrogen, although they make up 99% of the air.

Approximately 90% of the incoming atmospheric radiation comes from the lowest 300 ft. Thus, the air conditions at ground level largely determine the magnitude of the incoming radiation. The downward radiation from the atmosphere q_{Rad} may be expressed as

$$q_{Rad} = e_{at} \sigma T_{at}^4 \quad (20)$$

The emittance of the atmosphere is a complex function of air temperature and moisture content. The dew point of the atmosphere near the ground determines the total amount of moisture in the atmosphere above the place where the dry-bulb and dew-point temperatures of the atmosphere are determined (Reitan 1963). Bliss (1961) found that the emittance of the atmosphere is related to the dew-point temperature, as shown by Table 3.

The apparent sky temperature is defined as the temperature at which the sky (as a blackbody) emits radiation at the rate actually emitted by the atmosphere at ground level temperature with its actual emittance e_{at} . Then,

$$\sigma T_{sky}^4 = e_{at} \sigma T_{at}^4 \quad (21)$$

or
$$T_{sky}^4 = e_{at} T_{at}^4 \quad (22)$$

Example 3. Consider a summer night condition when the ground level temperatures are 65°F dew point and 85°F dry bulb. From Table 3, e_{at} at 65°F dew point is 0.87, and the apparent sky temperature is

$$T_{sky} = 0.87^{0.25} (85 + 459.6) = 526.0^\circ\text{R}$$

Thus, $T_{sky} = 526.0 - 459.6 = 66.4^\circ\text{F}$, which is 18.6°F below the ground level dry-bulb temperature.

For a winter night in Arizona, when the temperatures at ground level are 60°F dry bulb and 25°F dew point, from Table 3, the emittance of the atmosphere is 0.78, and the apparent sky temperature is 488.3°R or 28.7°F.

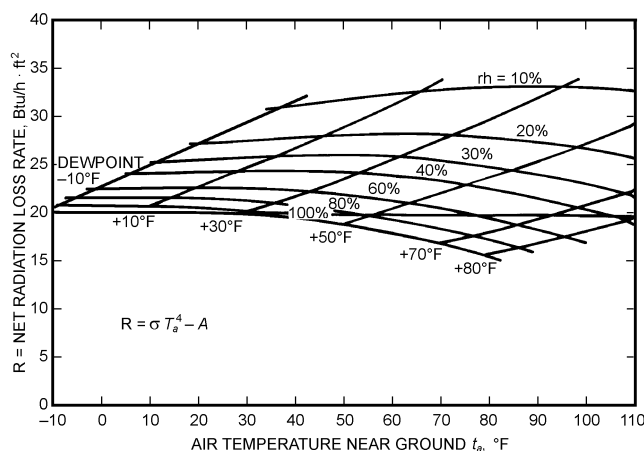
A simple relationship, which ignores vapor pressure of the atmosphere, may also be used to estimate the apparent sky temperature:

$$T_{sky} = 0.0411 T_{at}^{1.5} \quad (23)$$

where T is in degrees Rankine.

If the temperature of the radiating surface is assumed to equal the atmospheric temperature, the loss of heat from a black surface ($e_s = 1.0$) may be found from Figure 7.

Example 4. For the conditions in the previous example for summer, 85°F dry bulb and 65°F dew point, the rate of radiative heat loss is about 23

**Fig. 7 Radiation Heat Loss to Sky from Horizontal Blackbody**

Btu/h·ft². For winter, 60°F dry bulb and 25°F dew point, the heat loss is about 27 Btu/h·ft².

Where a rough, unpainted roof is used as a heat dissipater, the rate of heat loss rises rapidly as the surface temperature goes up. For the summer example, a painted metallic roof, $e_s = 0.96$, at 100°F (559.6°R) will have a heat loss rate of

$$\begin{aligned} q_{RAD} &= 0.96 \times 0.1713 \times 10^{-8} [559.6^4 - 526.0^4] \\ &= 35.4 \text{ Btu/h} \cdot \text{ft}^2 \end{aligned}$$

This analysis shows that radiation alone is not an effective means of dissipating heat under summer conditions of high dew-point and high ambient temperature. In spring and fall, when both the dew-point and dry-bulb temperatures are relatively low, radiation becomes much more effective.

On overcast nights, when the cloud cover is low, the clouds act much like blackbodies at ground level temperature, and virtually no heat can be lost by radiation. The exchange of longwave radiation between the sky and terrestrial surfaces occurs in the daytime as well as at night, but the much greater magnitude of the solar irradiation masks the longwave effects.

SOLAR ENERGY COLLECTION

Solar energy can be converted to (1) chemical, (2) electrical, and (3) thermal processes. Photosynthesis is a chemical process that produces food and converts CO₂ to O₂. Photovoltaic cells convert solar energy to electricity. The section on Photovoltaic Applications discusses some of the applications for these devices. The thermal conversion process, the primary subject of this chapter, provides thermal energy for space heating and cooling, domestic water heating, power generation, distillation, and process heating.

Solar Heat Collection by Flat-Plate Collectors

The solar irradiation data presented in the foregoing sections may be used to estimate how much energy is likely to be available at any specific location, date, and time of day for collection by either a concentrating device, which uses only the direct rays of the sun, or by a flat-plate collector, which can use both direct and diffuse irradiation. The temperatures needed for space heating and cooling do not exceed 200°F, even for absorption refrigeration, and they can be attained with carefully designed flat-plate collectors. Depending on the load and ambient temperatures, single-effect absorption systems can use energizing temperatures of 110 to 230°F.

A flat-plate collector generally consists of the following components (see Figure 8):

- **Glazing.** One or more sheets of glass or other diathermanous (radiation-transmitting) material.
- **Tubes, fins, or passages.** To conduct or direct the heat transfer fluid from the inlet to the outlet.
- **Absorber plates.** Flat, corrugated, or grooved plates, to which the tubes, fins, or passages are attached. The plate may be integral with the tubes.
- **Headers or manifolds.** To admit and discharge the fluid.
- **Insulation.** To minimize heat loss from the back and sides of the collector.
- **Container or casing.** To surround the other components and keep them free from dust, moisture, etc.

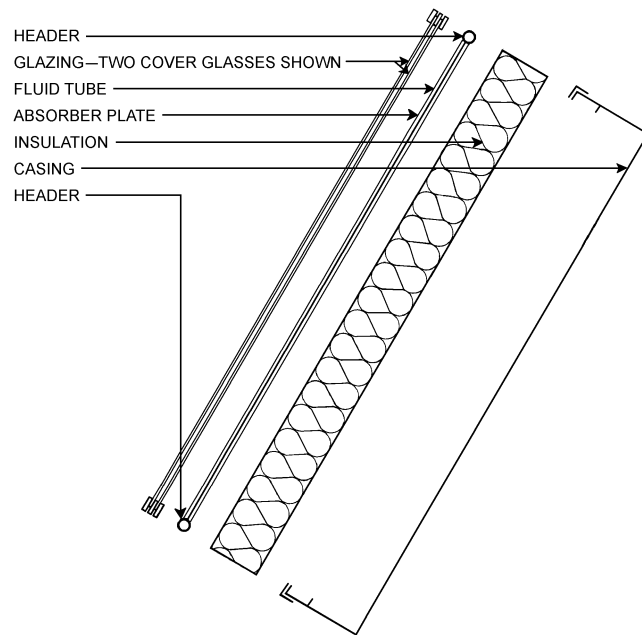


Fig. 8 Exploded Cross Section Through Double-Glazed Solar Water Heater

Flat-plate collectors have been built in a wide variety of designs from many different materials ([Figure 9](#)). They have been used to heat fluids such as water, water plus an antifreeze additive, or air. Their major purpose is to collect as much solar energy as possible at the lowest possible total cost. The collector should also have a long effective life, despite the adverse effects of the sun's ultraviolet radiation; corrosion or clogging because of acidity, alkalinity, or hardness of the heat transfer fluid; freezing or air-binding in the case of water, or deposition of dust or moisture in the case of air; and breakage of the glazing because of thermal expansion, hail, vandalism, or other causes. These problems can be minimized by using tempered glass.

Glazing Materials

Glass has been widely used to glaze flat-plate solar collectors because it can transmit as much as 90% of the incoming shortwave solar irradiation while transmitting virtually none of the longwave radiation emitted outward by the absorber plate. Glass with low iron content has a relatively high transmittance for solar radiation (approximately 0.85 to 0.90 at normal incidence), but its transmittance is essentially zero for the longwave thermal radiation (5.0 to 50 μm) emitted by sun-heated surfaces.

Plastic films and sheets also possess high shortwave transmittance, but because most usable varieties also have transmission bands in the middle of the thermal radiation spectrum, they may have longwave transmittances as high as 0.40.

Plastics are also generally limited in the temperatures they can sustain without deteriorating or undergoing dimensional changes. Only a few kinds of plastics can withstand the sun's ultraviolet radiation for long periods. However, they are not broken by hail and other stones and, in the form of thin films, they are completely flexible and have low mass.

The glass generally used in solar collectors may be either single-strength (0.085 to 0.100 in. thick) or double-strength (0.115 to 0.133 in. thick). The commercially available grades of window and greenhouse glass have normal incidence transmittances of about 0.87 and 0.85, respectively. For direct radiation, the transmittance varies markedly with the angle of incidence, as shown in [Table 4](#), which gives transmittances for single- and double-glazing using double-strength clear window glass.

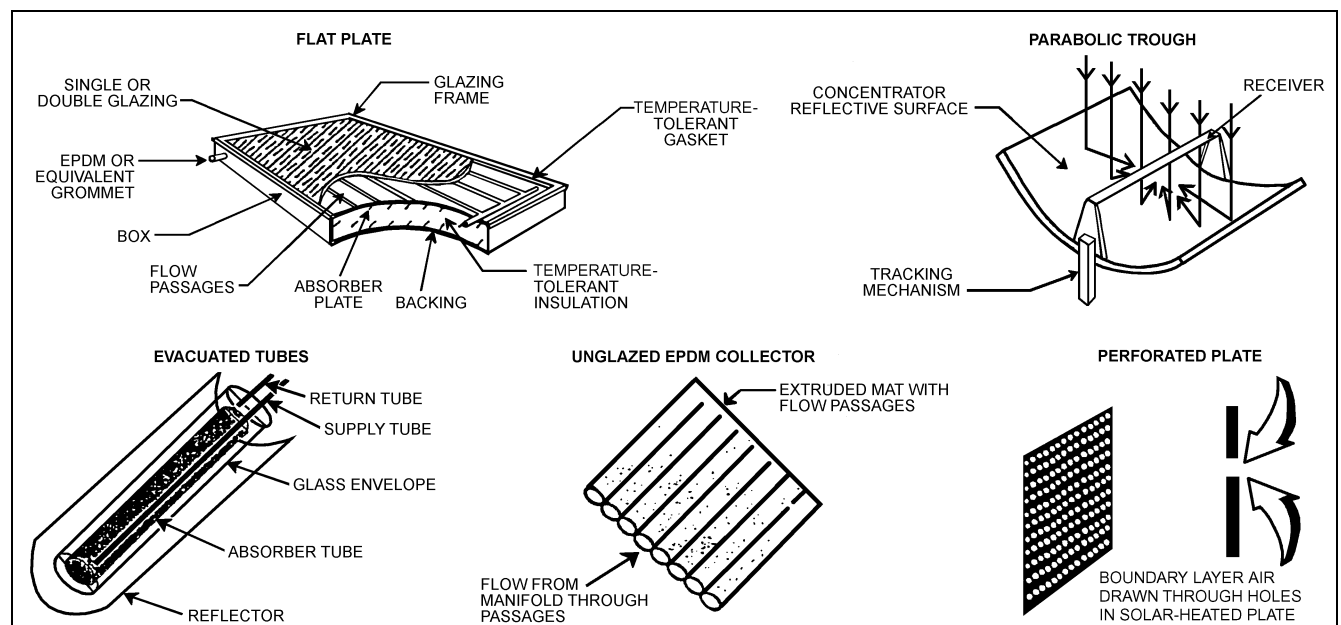


Fig. 9 Various Types of Solar Collectors

The 4% reflectance from each glass-air interface is the most important factor in reducing transmission, although a gain of about 3% in transmittance can be obtained by using water-white glass. Antireflective coatings and surface texture can also improve transmission significantly. The effect of dirt and dust on collector glazing may be quite small, and the cleansing effect of an occasional rainfall is usually adequate to maintain the transmittance within 2 to 4% of its maximum.

The glazing should admit as much solar irradiation as possible and reduce the upward loss of heat as much as possible. Although glass is virtually opaque to the longwave radiation emitted by collector plates, absorption of that radiation causes an increase in the glass temperature and a loss of heat to the surrounding atmosphere by radiation and convection. This type of heat loss can be reduced by using an infrared reflective coating on the underside of the glass; however, such coatings are expensive and reduce the effective solar transmittance of the glass by as much as 10%.

In addition to serving as a heat trap by admitting shortwave solar radiation and retaining longwave thermal radiation, the glazing also reduces heat loss by convection. The insulating effect of the glazing is enhanced by the use of several sheets of glass, or glass plus plastic. The loss from the back of the plate rarely exceeds 10% of the upward loss.

Collector Plates

The collector plate absorbs as much of the irradiation as possible through the glazing, while losing as little heat as possible upward to the atmosphere and downward through the back of the casing. The collector plates transfer the retained heat to the transport fluid. The absorbance of the collector surface for shortwave solar radiation depends on the nature and color of the coating and on the incident angle, as shown in Table 4 for a typical flat black paint.

By suitable electrolytic or chemical treatments, selective surfaces can be produced with high values of solar radiation absorptance a and low values of longwave emittance e_s . Essentially, typical selective surfaces consist of a thin upper layer, which is highly absorbent to shortwave solar radiation but relatively transparent to longwave thermal radiation, deposited on a substrate that has a high reflectance and a low emittance for longwave radiation. Selective surfaces are particularly important when the collector surface temperature is much higher than the ambient air temperature.

For fluid-heating collectors, passages must be integral with or firmly bonded to the absorber plate. A major problem is obtaining a good thermal bond between tubes and absorber plates without incurring excessive costs for labor or materials. Materials most frequently used for collector plates are copper, aluminum, and steel. UV-resistant plastic extrusions are used for low-temperature application. If the entire collector area is in contact with the heat transfer fluid, the thermal conductance of the material is not important.

**Table 4 Variation with Incident Angle
of Transmittance for Single and Double Glazing and
Absorbance for Flat Black Paint**

Incident Angle, Deg	Transmittance		Absorbance for Flat Black Paint
	Single Glazing	Double Glazing	
0	0.87	0.77	0.96
10	0.87	0.77	0.96
20	0.87	0.77	0.96
30	0.87	0.76	0.95
40	0.86	0.75	0.94
50	0.84	0.73	0.92
60	0.79	0.67	0.88
70	0.68	0.53	0.82
80	0.42	0.25	0.67
90	0.00	0.00	0.00

Whillier (1964) concluded that steel tubes are as effective as copper if the bond conductance between tube and plate is good. Potential corrosion problems should be considered for any metals. Bond conductance can range from a high of 1000 Btu/h·ft²·°F for a securely soldered or brazed tube to a low of 3 Btu/h·ft²·°F for a poorly clamped or badly soldered tube. Plates of copper, aluminum, or stainless steel with integral tubes are among the most effective types available. Figure 9 shows a few of the solar water and air heaters that have been used with varying degrees of success.

Concentrating Collectors

Temperatures far above those attainable by flat-plate collectors can be reached if a large amount of solar radiation is concentrated on a relatively small collection area. Simple reflectors can markedly increase the amount of direct radiation reaching a collector, as shown in Figure 10A.

Because of the apparent movement of the sun across the sky, conventional concentrating collectors must follow the sun's daily motion. There are two methods by which the sun's motion can be readily tracked. The altazimuth method requires the tracking device to turn in both altitude and azimuth; when performed properly, this method enables the concentrator to follow the sun exactly. Paraboloidal solar furnaces, Figure 10B, generally use this system. The polar, or equatorial, mounting points the axis of rotation at the North Star, tilted upward at the angle of the local latitude. By rotating the collector 15° per hour, it follows the sun perfectly (on March 21 and September 21). If the collector surface or aperture must be kept normal to the solar rays, a second motion is needed to correct for the change in the solar declination. This motion is not essential for most solar collectors.

The maximum variation in the angle of incidence for a collector on a polar mount will be ±23.5° on June 21 and December 21; the incident angle correction would then be $\cos 23.5^\circ = 0.917$.

Horizontal reflective parabolic troughs, oriented east and west, as shown in Figure 10C, require continuous adjustment to compensate for the changes in the sun's declination. There is inevitably some morning and afternoon shading of the reflecting surface if the concentrator has opaque end panels. The necessity of moving the concentrator to accommodate the changing solar declination can be reduced by moving the absorber or by using a trough with two sections of a parabola facing each other, as shown in Figure 10D. Known as a compound parabolic concentrator (CPC), this design can accept incoming radiation over a relatively wide range of angles. By using multiple internal reflections, any radiation that is accepted finds its way to the absorber surface located at the bottom of the apparatus. By filling the collector shape with a highly transparent material having an index of refraction greater than 1.4, the acceptance angle can be increased. By shaping the surfaces of the array properly, total internal reflection is made to occur at the medium-air interfaces, which results in a high concentration efficiency. Known as a dielectric compound parabolic concentrator (DCPC), this device has been applied to the photovoltaic generation of electricity (Cole et al. 1977).

The parabolic trough of Figure 10E can be simulated by many flat strips, each adjusted at the proper angle so that all reflect onto a common target. By supporting the strips on ribs with parabolic contours, a relatively efficient concentrator can be produced with less tooling than the complete reflective trough.

Another concept applied this segmental idea to flat and cylindrical lenses. A modification is shown in Figure 10F, in which a linear Fresnel lens, curved to shorten its focal distance, can concentrate a relatively large area of radiation onto an elongated receiver. Using the equatorial sun-following mounting, this type of concentrator has been used as a means of attaining temperatures well above those that can be reached with flat-plate collectors.

One disadvantage of concentrating collectors is that, except at low concentration ratios, they can use only the direct component of

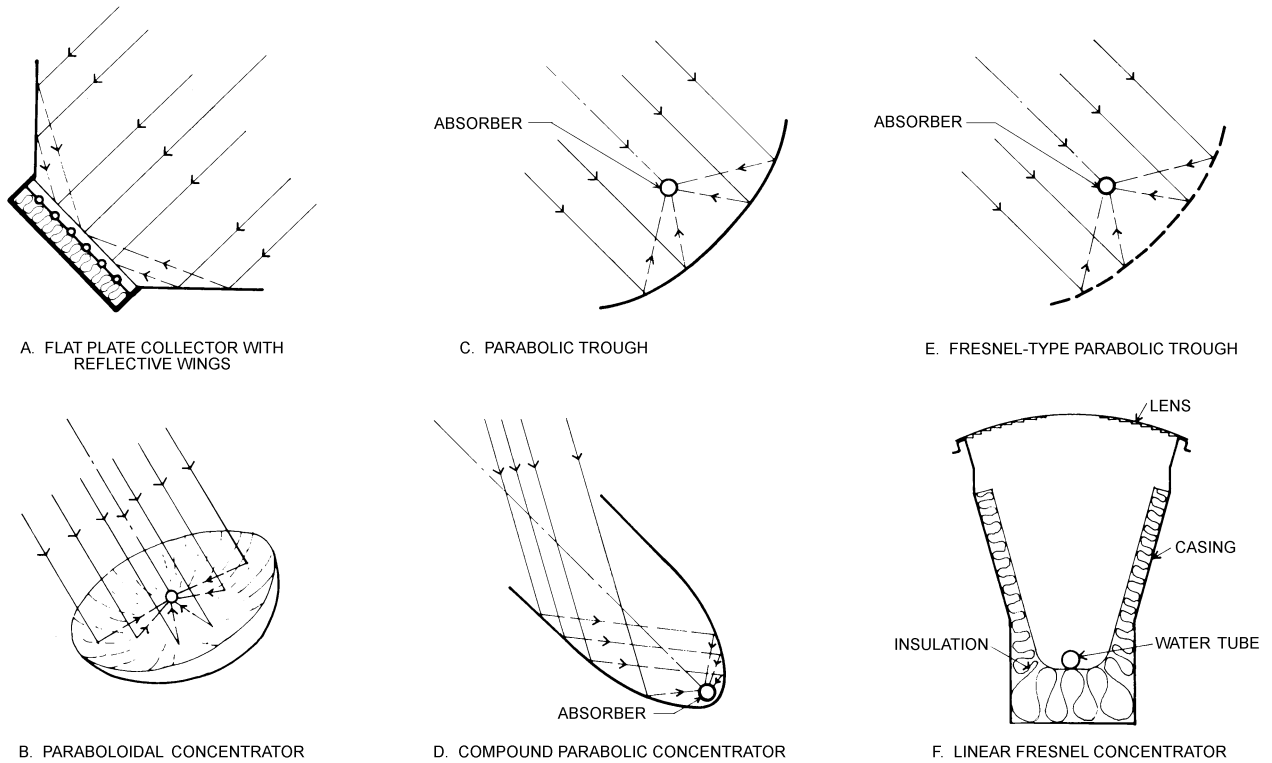


Fig. 10 Types of Concentrating Collectors

solar radiation, because the diffuse component cannot be concentrated by most types. However, an advantage of concentrating collectors is that, in summer, when the sun rises and sets well to the north of the east-west line, the sun-follower, with its axis oriented north-south, can begin to accept radiation directly from the sun long before a fixed, south-facing flat plate can receive anything other than diffuse radiation from the portion of the sky that it faces. At 40° north latitude, for example, the cumulative direct radiation available to a sun-follower on a clear day is 3180 Btu/ft², while the total radiation falling on the flat plate tilted upward at an angle equal to the latitude is only 2220 Btu/ft² each day. Thus, in relatively cloudless areas, the concentrating collector may capture more radiation per unit of aperture area than a flat-plate collector.

To get extremely high inputs of radiant energy, many flat mirrors, or heliostats, using altazimuth mounts, can be used to reflect their incident direct solar radiation onto a common target. Using slightly concave mirror segments on the heliostats, large amounts of thermal energy can be directed into the cavity of a steam generator to produce steam at high temperature and pressure.

Collector Performance

The performance of collectors may be analyzed by a procedure originated by Hottel and Woertz (1942) and extended by Whillier (ASHRAE 1977). The basic equation is

$$q_u = I_{\theta}(\tau\alpha)_{\theta} - U_L(t_p - t_{at}) = \dot{m}c_p(t_{fe} - t_{fi})/A_{ap} \quad (24)$$

Equation (24) also may be adapted for use with concentrating collectors:

$$q_u = I_{DN}(\tau\alpha)_{\theta}(\rho\Gamma) - U_L(A_{abs}/A_{ap})(t_{abs} - t_a) \quad (25)$$

where

$$q_u = \text{useful heat gained by collector per unit of aperture area, Btu/h} \cdot \text{ft}^2$$

- I_{θ} = total irradiation of collector, Btu/h · ft²
- I_{DN} = direct normal irradiation, Btu/h · ft²
- $(\tau\alpha)_{\theta}$ = transmittance τ of cover times absorptance α of plate at prevailing incident angle θ
- U_L = upward heat loss coefficient, Btu/h · ft² · °F
- t_p = temperature of the absorber plate, °F
- t_a = temperature of the atmosphere, °F
- t_{abs} = temperature of the absorber, °F
- \dot{m} = fluid flow rate, lb/h
- c_p = specific heat of fluid, Btu/lb · °F
- t_{fe}, t_{fi} = temperatures of the fluid leaving and entering the collector, °F
- $\rho\Gamma$ = reflectance of the concentrator surface times fraction of reflected or refracted radiation that reaches the absorber
- A_{abs}/A_{ap} = areas of absorber surface and of aperture that admit or receive radiation, ft²

The total irradiation and the direct normal irradiation for clear days may be found in ASHRAE *Standard* 93. The transmittance for single and double glazing and the absorptance for flat black paint may be found in Table 4 for incident angles from 0 to 90°. These values, and the products of τ and α , are also shown in Figure 11. The solar-optical properties of the glazing and absorber plate change little until θ exceeds 30°, but, because all values reach zero when $\theta = 90^\circ$, they drop off rapidly for values of θ beyond 40°.

For nonselective absorber plates, U_L varies with the temperature of the plate and the ambient air, as shown in Figure 12. For selective surfaces, which effect major reductions in the emittance of the absorber plate, U_L will be much lower than the values shown in Figure 12. Manufacturers of such surfaces should be asked for values applicable to their products, or test results that give the necessary information should be consulted.

Example 5. A flat-plate collector is operating in Denver, latitude = 40° north, on July 21 at noon solar time. The atmospheric temperature is assumed to be 85°F, and the average temperature of the absorber plate is 140°F. The collector is single-glazed with flat black paint on the absorber. The collector faces south, and the tilt angle is 30° from the

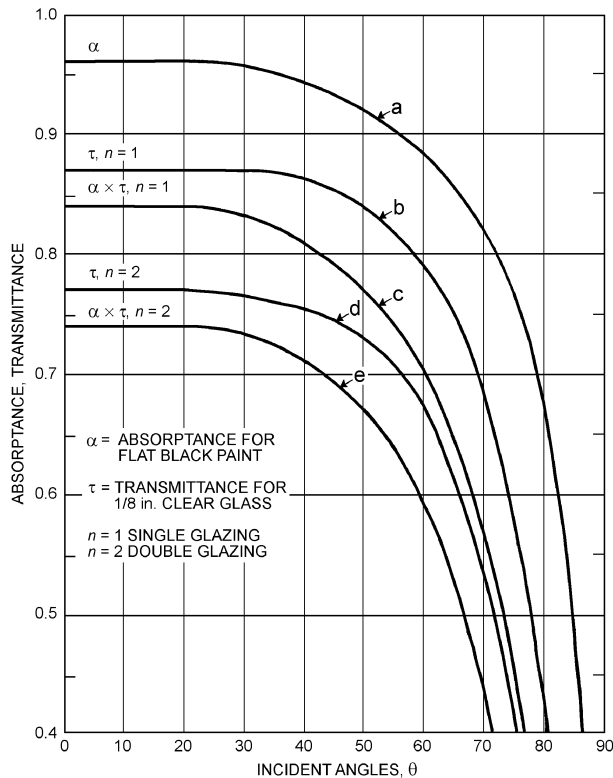
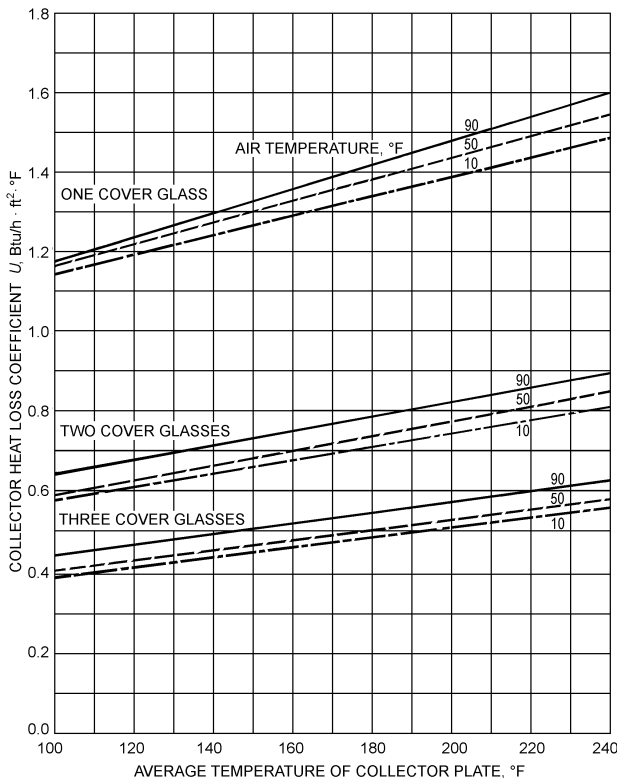


Fig. 11 Variation of Absorptance and Transmittance with Incident Angle



NOTE: ABSORBER PLATE IS COATED WITH A NONSELECTIVE FLAT BLACK PAINT.

Fig. 12 Variation of Upward Heat Loss Coefficient U_L with Collector Plate Temperature and Ambient Air Temperatures for Single-, Double-, and Triple-Glazed Collectors

horizontal. Find the rate of heat collection and the collector efficiency. Neglect the losses from the back and sides of the collector.

Solution: From Table 2, $\delta = 20.6^\circ$.

From Equation (2),

$$\beta_N = 90^\circ - 40^\circ + 20.6^\circ = 70.6^\circ$$

From Equation (3), $H = 0$; therefore from Equation (6), $\sin \phi = 0$ and thus, $\phi = 0^\circ$. Because the collector faces south, $\psi = 0^\circ$, and $\gamma = \phi$. Thus $\gamma = 0^\circ$. Then Equation (8) gives

$$\begin{aligned} \cos \theta &= \cos 70.6^\circ \cos 0^\circ \sin 30^\circ + \sin 70.6^\circ \cos 30^\circ \\ &= (0.332)(1)(0.5) + (0.943)(0.866) \\ &= 0.983 \\ \theta &= 10.6^\circ \end{aligned}$$

From Table 1, $A = 344 \text{ Btu/h} \cdot \text{ft}^2$, $B = 0.207$, and $C = 0.136$. Using Equation (11),

$$I_{DN} = 344 e^{-0.207/\sin 70.6^\circ} = 276 \text{ Btu/h} \cdot \text{ft}^2$$

Combining Equations (14) and (15) gives

$$I_{d\theta} = 0.136 \times 276(1 + \cos 30^\circ)/2 = 35 \text{ Btu/h} \cdot \text{ft}^2$$

Assuming $I_r = 0$, Equation (12) gives a total solar irradiation on the collector of

$$I_{t\theta} = 276 \cos 10.6^\circ + 35 + 0 = 306 \text{ Btu/h} \cdot \text{ft}^2$$

From Figure 11, for $n = 1$, $\tau = 0.87$ and $\alpha = 0.96$.

From Figure 12, for an absorber plate temperature of 140°F and an air temperature of 85°F , $U_L = 1.3 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F}$.

Then from Equation (24),

$$q_u = 306(0.87 \times 0.96) - 1.3(140 - 85) = 184 \text{ Btu/h} \cdot \text{ft}^2$$

The collector efficiency η is

$$184/306 = 0.60$$

The general expression for collector efficiency is

$$\eta = (\tau\alpha)_\theta - U_L(t_p - t_{at})/I_{t\theta} \quad (26)$$

For incident angles below about 35° , the product τ times α is essentially constant and Equation (26) is linear with respect to the parameter $(t_p - t_{at})/I_{t\theta}$, as long as U_L remains constant.

ASHRAE (1977) suggested that an additional term, the **collector heat removal factor** F_R , be introduced to permit the use of the fluid inlet temperature in Equations (24) and (26):

$$q_u = F_R [I_{t\theta}(\tau\alpha)_\theta - U_L(t_{fi} - t_{at})] \quad (27)$$

$$\eta = F_R(\tau\alpha)_\theta - F_R U_L(t_{fi} - t_{at})/I_{t\theta} \quad (28)$$

where F_R equals the ratio of the heat actually delivered by the collector to the heat that would be delivered if the absorber were at t_{fi} . F_R is found from the results of a test performed in accordance with ASHRAE Standard 93.

The results of such a test are plotted in Figure 13. When the parameter is zero, because there is no temperature difference between the fluid entering the collector and the atmosphere, the value of the y-intercept equals $F_R(\tau\alpha)$. The slope of the efficiency line equals the heat loss factor U_L multiplied by F_R . For the single-glazed, nonselective collector with the test results shown in Figure 13, the y-intercept is 0.82, and the x-intercept is $0.69^\circ\text{F} \cdot \text{ft}^2 \cdot \text{h/Btu}$. This collector used high-transmittance single glazing, $\tau = 0.91$, and

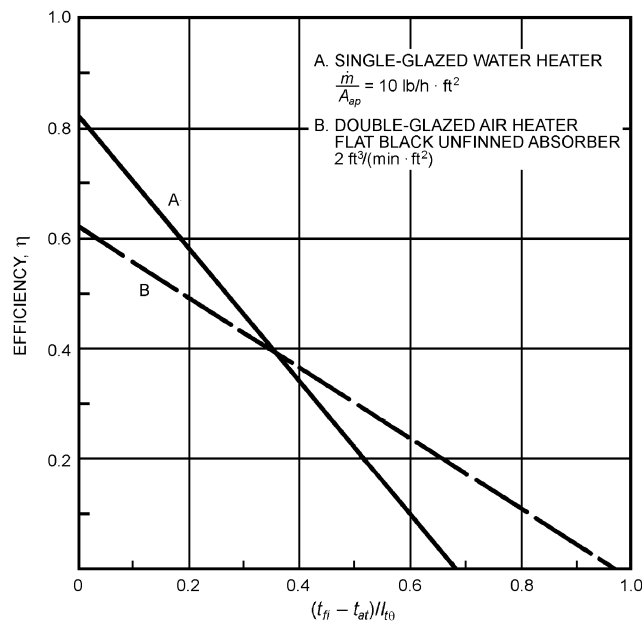


Fig. 13 Efficiency Versus $(t_{fi} - t_{at})/I_0$ for Single-Glazed Solar Water Heater and Double-Glazed Solar Air Heater

black paint with an absorptance of 0.97, so Equation (28) gives $F_R = 0.82/(0.91 \times 0.97) = 0.93$.

Assuming that the relationship between η and the parameter is actually linear, as shown, then the slope is $-0.82/0.69 = -1.19$; thus $U_L = 1.19/F_R = 1.19/0.93 = 1.28 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F}$. The tests on which Figure 13 is based were run indoors. Factors that affect the measured efficiency are wind speed and fluid velocity.

Figure 13 also shows the efficiency of a double-glazed air heater with an unfinned absorber coated with flat black paint. The y-intercept for the air heater B is considerably less than it is for water heater A because (1) transmittance of the double glazing used in B is lower than the transmittance of the single glazing used in A and (2) F_R is lower for B than for A because of the lower heat transfer coefficient between air and the unfinned metal absorber.

The x-intercept for air heater B is greater than it is for the water heater A because the upward loss coefficient U_L is much lower for the double-glazed air heater than for the single-glazed water heater. The data for both A and B were taken at near-normal incidence with high values of I_0 . For Example 5, using a single-glazed water heater, the value of the parameter would be close to $(140 - 85)/306 = 0.18^\circ\text{F} \cdot \text{ft}^2 \cdot \text{h/Btu}$, and the expected efficiency, 0.60, agrees closely with the test results.

As ASHRAE Standard 93 shows, the incident angles encountered with south-facing tilted collectors vary widely throughout the year. Considering a surface located at 40° north latitude with a tilt angle $\Sigma = 40^\circ$, the incident angle θ will depend on the time of day and the declination δ . On December 21, $\delta = 23.456^\circ$; at 4 h before and after solar noon, the incident angle is 62.7° , and it remains close to this value for the same solar time throughout the year. The total irradiation at these conditions varies from a low of $45 \text{ Btu/h} \cdot \text{ft}^2$ on December 21 to approximately $140 \text{ Btu/h} \cdot \text{ft}^2$ throughout most of the other months.

When the irradiation is below about $100 \text{ Btu/h} \cdot \text{ft}^2$, the losses from the collector may exceed the heat that can be absorbed. This situation varies with the temperature difference between the collector inlet temperature and the ambient air, as suggested by Equation (27).

When the incident angle rises above 30° , the product of the transmittance of the glazing and the absorptance of the collector plate

begins to diminish; thus, the heat absorbed also drops. The losses from the collector are generally higher as the time moves farther from solar noon, and consequently the efficiency also drops. Thus, the daylong efficiency is lower than the near-noon performance. During the early afternoon, the efficiency is slightly higher than at the comparable morning time, because the ambient air temperature is lower in the morning than in the afternoon.

ASHRAE Standard 93 describes the **incident angle modifier**, which may be found by tests run when the incident angle is set at 30° , 45° , and 60° . Simon (1976) showed that for many flat-plate collectors, the incident angle modifier is a linear function of the quantity $(1/\cos \theta - 1)$. For evacuated tubular collectors, the incident angle modifier may grow with rising values of θ .

ASHRAE Standard 93 specifies that the efficiency be reported in terms of the gross collector area A_g rather than the aperture area A_{ap} . The reported efficiency will be lower than the efficiency given by Equation (28), but the total energy collected is not changed by this simplification:

$$\eta_g = \frac{\eta_{ap} A_{ap}}{A_g} \quad (29)$$

COMPONENTS

This section describes the major components involved in the collection, storage, transportation, control, and distribution of solar heat for a domestic hot-water system.

Collectors. Flat-plate collectors are most commonly used for water heating because of the year-round load requiring temperatures of 80 to 180°F . For discussions of other collectors and applications, see ASHRAE Standard 93, Chapter 33 of the 2000 *ASHRAE Handbook—HVAC Systems and Equipment*, and previous sections of this chapter. Collectors must withstand extreme weather (such as freezing, stagnation, and high winds), as well as system pressures.

Heat Transfer Fluids. Heat transfer fluids transport heat from the solar collectors to the domestic water. The potential safety problems that exist in this transfer are both chemical and mechanical and apply primarily to systems in which a heat exchanger interface exists with the potable water supply. Both the chemical compositions of the heat transfer fluids (pH, toxicity, and chemical durability), as well as their mechanical properties (specific heat and viscosity) must be considered.

Except in unusual cases, or when potable water is being circulated, the energy transport fluid is nonpotable and has the potential for contaminating potable water. Even potable or nontoxic fluids in closed circuits are likely to become nonpotable because of contamination from metal piping, solder joints, and packing, or by the inadvertent installation of a toxic fluid at a later date.

Thermal Energy Storage. Heat collected by solar domestic and service water heaters is virtually always stored as a liquid in tanks. Storage tanks and bins should be well insulated. In domestic hot-water systems, heat is usually stored in one or two tanks. The hot-water outlet is at the top of the tank, and cold water enters the tank through a dip tube that extends down to within 4 to 6 in. of the tank bottom. The outlet on the tank to the collector loop should be approximately 4 in. above the tank bottom to prevent scale deposits from being drawn into the collectors. Water from the collector array returns to the upper portion of the storage tank. This plumbing arrangement may take advantage of thermal stratification, depending on the delivery temperature from the collectors and the flow rate through the storage tank.

Single-tank electric auxiliary systems often incorporate storage and auxiliary heating in the same vessel. Conventional electric water heaters commonly have two heating elements: one near the top and one near the bottom. If a dual-element tank is used in a solar energy system, the bottom element should be disconnected and the

top left functional to take advantage of fluid stratification. Standard gas- and oil-fired water heaters should not be used in single-tank arrangements. In gas and oil water heaters, heat is added to the bottom of the tanks, which reduces both stratification and collection efficiency in single-tank systems.

Dual-tank systems often use the solar domestic hot-water storage tank as a preheat tank. The second tank is normally a conventional domestic hot-water tank containing the auxiliary heat source. Multiple tanks are sometimes used in large institutions, where they operate similarly to dual-tank heaters. Although the use of two tanks may increase collector efficiency and the solar fraction, it increases tank heat losses. The water inlet is usually a dip tube that extends near the bottom of the tank.

Estimates for sizing storage tanks usually range from 1 to 2.5 gal per square foot of solar collector area. The estimate used most often is 1.8 gal per square foot of collector area, which usually provides enough heat for a sunless period of about a day. Storage volume should be analyzed and sized according to the project water requirements and draw schedule; however, solar applications typically require larger-than-normal tanks.

Heat Exchangers. Indirect solar water heaters require one or more heat exchangers. The potential exists for contamination in the transfer of heat energy from solar collectors to potable hot water. Heat exchangers influence the effectiveness of energy collected to heat domestic water. They also separate and protect the potable water supply from contamination when nonpotable heat transfer fluids are used. For this reason various codes regulate the need for and design of heat exchangers.

Heat exchanger selection should consider the following:

- Heat exchange effectiveness
- Pressure drop, operating power, and flow rate
- Design pressure, configuration, size, materials, and location
- Cost and availability
- Reliable protection of the potable water supply from contamination by the heat transfer fluid
- Leak detection, inspection, and maintainability
- Material compatibility with other elements (e.g., metals and fluids)
- Thermal compatibility with design parameters such as operating temperature, and fluid thermal properties

Heat exchanger selection depends on the characteristics of the fluids that pass through the heat exchanger and the properties of the exchanger itself. Fluid characteristics to consider are fluid type, specific heat, mass flow rate, and hot and cold fluid inlet and outlet temperatures. Physical properties of the heat exchanger to consider are the overall heat transfer coefficient of the heat exchanger and the heat transfer surface area.

For most solar domestic hot-water designs, only the hot and cold inlet temperatures are known; the other temperatures must be calculated using the physical properties of the heat exchanger. Two quantities that are useful in determining the heat transfer in a heat exchanger and the performance characteristics of a collector when it is combined with a given heat exchanger are (1) the fluid capacitance rate, which is the product of the mass flow rate and the specific heat of the fluid passing through the heat exchanger, and (2) the heat exchanger effectiveness, which relates the capacitance rate of the two fluids to the fluid inlet and outlet temperatures. The effectiveness is equal to the ratio of the actual heat transfer rate to the maximum heat transfer rate theoretically possible. Generally, a heat exchanger effectiveness of 0.4 or greater is desired.

Expansion Tanks. An indirect solar water heater operating in a closed collector loop requires an expansion tank to prevent excessive pressure. Fluid in solar collectors under stagnation conditions can boil, causing excessive pressure to develop in the collector loop, and expansion tanks must be sized for this condition. Expansion tank sizing formulas for closed-loop hydronic systems, found in

Chapter 12 of the 2000 *ASHRAE Handbook—HVAC Systems and Equipment*, may be used for solar heater expansion tank sizing, but the expression for volume change due to temperature increase should be replaced with the total volume of fluid in the solar collectors and of any piping located above the collectors, if significant. This sizing method provides a passive means for eliminating fluid loss due to overtemperature or stagnation, common problems in closed-loop solar systems. This results in a larger expansion tank than typically found in hydronic systems, but the increase in cost is small compared to the savings in fluid replacement and maintenance costs (Lister and Newell 1989).

Pumps. Pumps circulate heat transfer liquid through collectors and heat exchangers. In solar domestic hot water heaters, the pump is usually a centrifugal circulator driven by a motor of less than 300 W. The flow rate for collectors generally ranges from 0.015 to 0.04 gpm/ft². Pumps used in drainback systems must provide pressure to overcome friction and to lift the fluid to the collectors.

Piping. Piping can be plastic, copper, galvanized steel, or stainless steel. The most widely used is nonlead, sweat-soldered L-type copper tubing. M-type copper is also acceptable if permitted by local building codes. If water/glycol is the heat transfer fluid, galvanized pipes or tanks must not be used because unfavorable chemical reactions will occur; copper piping is recommended instead. Also, if glycol solutions or silicone fluids are used, they may leak through joints where water would not. Piping should be compatible with the collector fluid passage material; for example, copper or plastic piping should be used with collectors having copper fluid passages.

Piping that carries potable water can be plastic, copper, galvanized steel, or stainless steel. In indirect systems, corrosion inhibitors must be checked and adjusted routinely, preferably every three months. Inhibitors should also be checked if the system overheats during stagnation conditions. If dissimilar metals are joined, dielectric or nonmetallic couplings should be used. The best protection is sacrificial anodes or getters in the fluid stream. Their location depends on the material to be protected, the anode material, and the electrical conductivity of the heat transfer fluid. Sacrificial anodes consisting of magnesium, zinc, or aluminum are often used to reduce corrosion in storage tanks. Because many possibilities exist, each combination must be evaluated. A copper-aluminum or copper-galvanized steel joint is unacceptable because of severe galvanic corrosion. Aluminum, copper, and iron have a greater potential for corrosion.

Elimination of air, pipe expansion, and piping slope must be considered to avoid possible failures. Collector pipes (particularly manifolds) should be designed to allow expansion from stagnation temperature to extreme cold weather temperature. Expansion control can be achieved with offset elbows in piping, hoses, or expansion couplings. Expansion loops should be avoided unless they are installed horizontally, particularly in systems that must drain for freeze protection. The collector array piping should slope 0.06 in. per foot for drainage (DOE 1978a).

Air can be eliminated by placing air vents at all piping high points and by air purging during filling. Flow control, isolation, and other valves in the collector piping must be chosen carefully so that these components do not restrict drainage significantly or back up water behind them. The collectors must drain completely.

Valves and Gages. Valves in solar domestic hot water systems must be located to ensure system efficiency, satisfactory performance, and the safety of equipment and personnel. Drain valves must be ball-type; gate valves may be used if the stem is installed horizontally. Check valves or other valves used for freeze protection or for reverse thermosiphoning must be reliable to avoid significant damage.

Auxiliary Heat Sources. On sunny days, a typical solar energy system should supply water at a predetermined temperature, and the solar storage tank should be large enough to hold sufficient water for a day or two. Because of the intermittent nature of solar radiation, an

auxiliary heater must be installed to handle hot-water requirements. If a utility is the source of auxiliary energy, operation of the auxiliary heater can be timed to take advantage of off-peak utility rates. The auxiliary heater should be carefully integrated with the solar energy heater to obtain maximum solar energy use. For example, the auxiliary heater should not destroy any stratification that may exist in the solar-heated storage tank, which would reduce collector efficiency.

Ductwork, particularly in systems with air-type collectors, must be sealed carefully to avoid leakage in duct seams, damper shafts, collectors, and heat exchangers. Ducts should be sized using conventional air duct design methods.

Control. Controls regulate solar energy collection by controlling fluid circulation, activate system protection against freezing and overheating, and initiate auxiliary heating when it is required. The three major control components are sensors, controllers, and actuators. Sensors detect conditions or measure quantities, such as temperature. Controllers receive output from the sensors, select a course of action, and signal a component to adjust the condition. Actuators, such as pumps, valves, dampers, and fans, execute controller commands and regulate the system.

Temperature sensors measure the temperature of the absorber plate near the collector outlet and near the bottom of the storage tank. The sensors send signals to a controller, such as a differential temperature thermostat, for interpretation.

The differential thermostat compares the signals from the sensors with adjustable set points for high and low temperature differentials. The controller performs different functions, depending on which set points are met. In liquid systems, when the temperature difference between the collector and storage reaches a high set point, usually 20°F, the pump starts, automatic valves are activated, and circulation begins. When the temperature difference reaches a low set point, usually 4°F, the pump is shut off and the valves are deenergized and returned to their normal positions. To restart the system, the high-temperature set point must again be met. If the system has either freeze or overheat protection, the controller opens or closes valves or dampers and starts or stops pumps or fans to protect the system when its sensors detect conditions indicating that either freezing or overheating is about to occur.

Sensors must be selected to withstand high temperature, such as may occur during collector stagnation. Collector loop sensors can be located on the absorber plate, in a pipe above the collector, on a pipe near the collector, or in the collector outlet passage. Although any of these locations may be acceptable, attaching the sensor on the collector absorber plate is recommended. When attached properly, the sensor gives accurate readings, can be installed easily, and is basically unaffected by ambient temperature, unlike sensors mounted on exterior piping.

A sensor installed on an absorber plate reads temperatures 3 to 4°F higher than the temperature of the fluid leaving the collector. However, such temperature discrepancies can be compensated for in the differential thermostat settings.

The sensor must be attached to the absorber plate with good thermal contact. If a sensor access cover is provided on the enclosure, it must be gasketed for a watertight fit. Adhesives and adhesive tapes should not be used to attach the sensor to the absorber plate.

The storage temperature sensor should be near the bottom of the storage tank to detect the temperature of the fluid before it is pumped to the collector or heat exchanger. The storage fluid is usually coldest at that location because of thermal stratification and the location of the makeup water supply. The sensor should be either securely attached to the tank and well insulated, or immersed inside the tank near the collector supply.

The freeze protection sensor, if required, should be located so that it will detect the coldest liquid temperature when the collector is shut down. Common locations are the back of the absorber plate at the bottom of the collector, the collector intake or return manifolds, or the center of the absorber plate. The center absorber plate

location is recommended because reradiation to the night sky will freeze the collector heat transfer fluid, even though the ambient temperature is above freezing. Some systems, such as the recirculation system, have two sensors for freeze protection; others, such as the draindown, use only one.

Control of on-off temperature differentials affects system efficiency. If the differential is too high, the collector starts later than it should; if it is too low, the collector starts too soon. The turn-on differential for liquid systems usually ranges from 15 to 30°F and is most commonly 20°F. For air systems, the range is usually 25 to 45°F.

The turn-off temperature differential is more difficult to estimate. Selection depends on a comparison between the value of the energy collected and the cost of collecting it. It varies with individual systems, but a value of 4°F is typical.

Water temperature in the collector loop depends on ambient temperature, solar radiation, radiation from the collector to the night sky, and collector loop insulation. Freeze protection sensors should be set to detect 40°F.

Sensors are important but often overlooked control components. They must be selected and installed properly because no control can produce accurate outputs from unreliable sensor inputs. Sensors are used in conjunction with a differential temperature controller and are usually supplied by the controller manufacturer. Sensors must survive the anticipated operating conditions without physical damage or loss of accuracy. Low-voltage sensor circuits must be located away from high-voltage lines to avoid electromagnetic interference. Sensors attached to collectors should be able to withstand the stagnation temperature.

Sensor calibration, which is often overlooked by installers and maintenance personnel, is critical to system performance; a routine calibration maintenance schedule is essential.

Another control is for the photovoltaic (PV) panel that powers a pump. A PV panel converts sunlight into electricity to run a small circulating pump. No additional sensing is required because the PV panel and pump output increase with sunlight intensity and stop when no sunlight (collector energy) is available. Cromer (1984) has shown that with proper matching of pump and PV electrical characteristics, PV panel sizes as low as 5 W per 40 ft² of thermal panel may be used successfully. Difficulty with late starting and running too long in the afternoon can be alleviated by tilting the PV panel slightly to the east.

WATER HEATING

A solar water heater includes a solar collector that absorbs solar radiation and converts it to heat, which is then absorbed by a heat transfer fluid (water, a nonfreezing liquid, or air) that passes through the collector. The heat transfer fluid's heat is stored or used directly.

Portions of the solar energy system are exposed to the weather, so they must be protected from freezing. The system must also be protected from overheating caused by high insolation levels during periods of low energy demand.

In solar water heating, water is heated directly in the collector or indirectly by a heat transfer fluid that is heated in the collector, passes through a heat exchanger, and transfers its heat to the domestic or service water. The heat transfer fluid is transported by either natural or forced circulation. Natural circulation occurs by natural convection (thermosiphoning), whereas forced circulation uses pumps or fans. Except for thermosiphon systems, which need no control, solar domestic and service water heaters are controlled by differential thermostats.

Five types of solar energy systems are used to heat domestic and service hot water: thermosiphon, direct circulation, indirect, integral collector storage, and site built. Recirculation and draindown are two methods used to protect direct solar water heaters from freezing.

Thermosiphon Systems

Thermosiphon systems (Figure 14) heat potable water or a heat transfer fluid and rely on natural convection to transport it from the collector to storage. For direct systems, pressure-reducing valves are required when the city water pressure is greater than the working pressure of the collectors. In a thermosiphon system, the storage tank must be elevated above the collectors, which sometimes requires designing the upper level floor and ceiling joists to bear this additional load. Extremely hard or acidic water can cause scale deposits that clog or corrode the absorber fluid passages. Thermosiphon flow is induced whenever there is sufficient sunshine, so these systems do not need pumps.

Direct-Circulation Systems

A direct-circulation system (Figure 15) pumps potable water from storage to the collectors when there is enough solar energy available to warm it. It then returns the heated water to the storage tank until it is needed. Collectors can be mounted either above or below the storage tank. Direct-circulation systems are only feasible in areas where freezing is infrequent. Freeze protection is provided either by recirculating warm water from the storage tank or by flushing the collectors with cold water. Direct water-heating systems

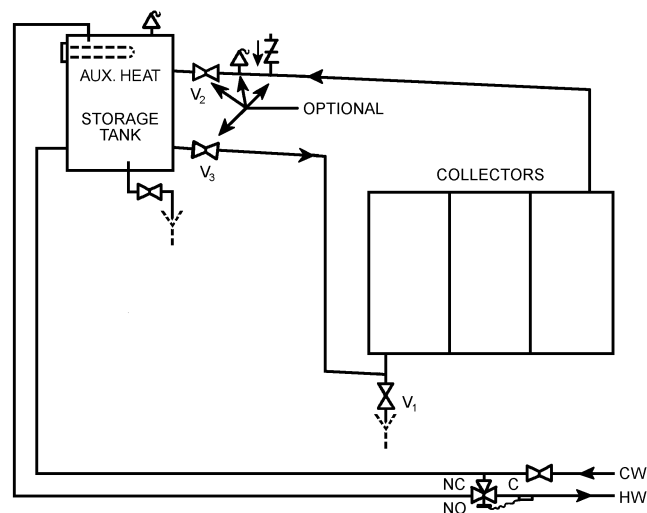


Fig. 14 Thermosiphon System

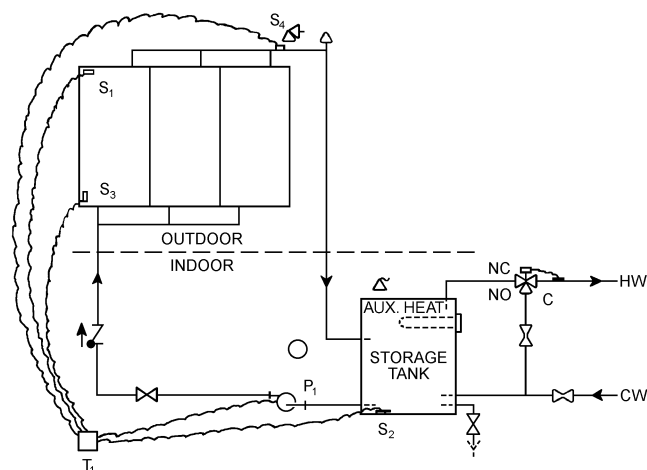


Fig. 15 Direct Circulation System

should not be used in areas where the water is extremely hard or acidic because scale deposits may clog or corrode the absorber fluid passages, rendering the system inoperable.

Direct-circulation systems are exposed to city water line pressures and must withstand pressures as required by local codes. Pressure-reducing valves and pressure-relief valves are required when the city water pressure is greater than the working pressure of the collectors. Direct-circulation systems often use a single storage tank for both solar energy storage and the auxiliary water heater, but two-tank storage systems can be used.

Draindown Systems. Draindown systems (Figure 16) are direct-circulation water-heating systems in which potable water is pumped from storage to the collector array where it is heated. Circulation continues until usable solar heat is no longer available. When a freezing condition is anticipated or a power outage occurs, the system drains automatically by isolating the collector array and exterior piping from the city water pressure and using one or more valves for draining. Solar collectors and associated piping must be carefully sloped to drain the collector's exterior piping.

Indirect Water-Heating Systems

Indirect water-heating systems (Figure 17) circulate a freeze-protected heat transfer fluid through the closed collector loop to a heat exchanger, where its heat is transferred to the potable water. The most commonly used heat transfer fluids are water/ethylene glycol and water/propylene glycol solutions, although other heat transfer fluids such as silicone oils, hydrocarbons, and refrigerants can also be used (ASHRAE 1983). These fluids are nonpotable, sometimes toxic, and normally require double-wall heat exchangers. The double-wall heat exchanger can be located inside the storage tank, or an external heat exchanger can be used. The collector loop is closed and therefore requires an expansion tank and a pressure-relief valve. A one- or two-tank storage can be used. Additional overtemperature protection may be needed to prevent the collector fluid from decomposing or becoming corrosive.

Designers should avoid automatic water makeup in systems using water/antifreeze solutions because a significant leak may raise the freezing temperature of the solution above the ambient temperature, causing the collector array and exterior piping to freeze. Also, antifreeze systems with large collector arrays and long pipe runs may need a time-delayed bypass loop around the heat exchanger to avoid freezing the heat exchanger on startup.

Drainback Systems. Drainback systems are generally indirect water-heating systems that circulate treated or untreated water through the closed collector loop to a heat exchanger, where its heat

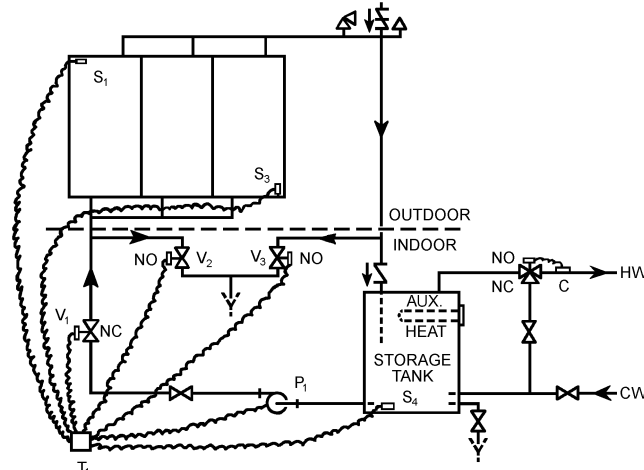


Fig. 16 Draindown System

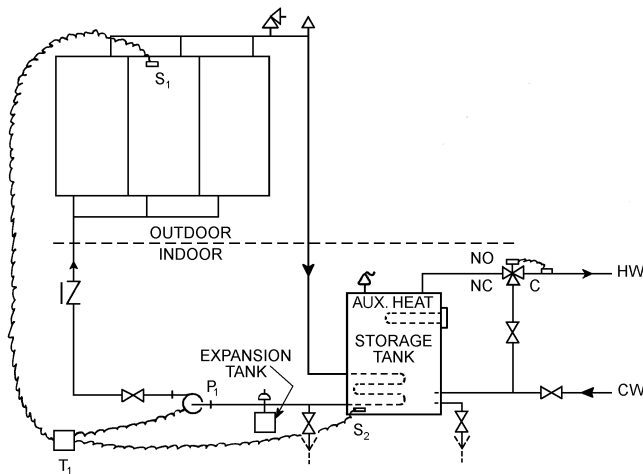


Fig. 17 Indirect Water Heating

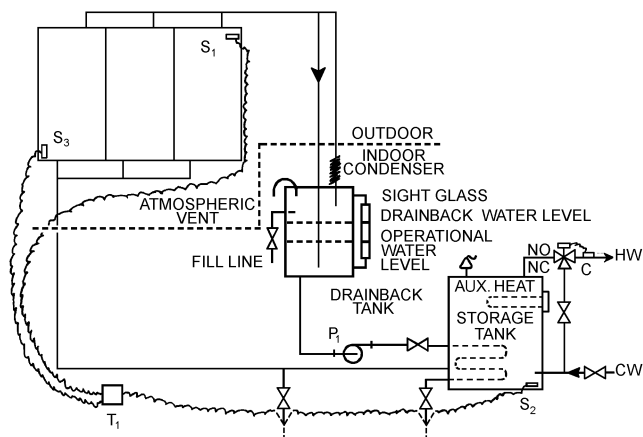


Fig. 18 Drainback System

is transferred to the potable water. Circulation continues until usable energy is no longer available. When the pump stops, the collector fluid drains by gravity to a storage or tank. In a pressurized system, the tank also serves as an expansion tank, so it must have a temperature- and pressure-relief valve to protect against excessive pressure. In an unpressurized system (Figure 18), the tank is open and vented to the atmosphere.

The collector loop is isolated from the potable water, so valves are not needed to actuate draining, and scaling is not a problem. The collector array and exterior piping must be sloped to drain completely, and the pumping pressure must be sufficient to lift water to the top of the collector array.

Integral Collector Storage Systems

Integral collector storage (ICS) systems use hot-water storage as part of the collector. Some types use the surface of a single tank as the absorber, and others use multiple long, thin tanks placed side by side horizontally to form the absorber surface. In this type of ICS, hot water is drawn from the top tank, and cold replacement water enters the bottom tank. Because of the greater nighttime heat loss from ICS systems, they are typically less efficient than pumped systems, and selective surfaces are recommended. ICS systems are normally installed as a solar preheater without pumps or controllers. Flow through the ICS system occurs on demand, as hot water flows from the collector to a hot-water auxiliary tank in the structure.

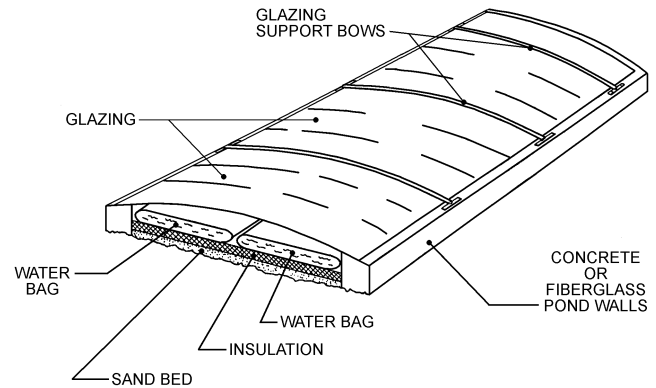


Fig. 19 Shallow Solar Pond

Site-Built Systems

Site-built, large-volume solar, water, air, or water-heating equipment is used in commercial and industrial applications. These site-built systems are based on a transpired solar collector for air heating and shallow solar pond technologies.

Transpired Solar Collector. This collector preheats outdoor air by drawing it through small holes in a metal panel. It is typically installed on south-facing walls and is designed to heat outdoor air for building ventilation or process applications (Kutscher 1996). The prefabricated panel, made of dark metal with thousands of small holes, efficiently heats and captures fresh air by drawing it through a perforated adsorber, eliminating the cost and the reflection losses associated with a glazing. The sun heats the metal panel, which in turn heats a boundary layer of air on its surface. Air is heated as it is drawn through the small holes into a ventilation system for delivery as ventilation air, crop drying, or other process applications.

Shallow Solar Pond. The shallow solar pond (SSP) is a large-scale ICS solar water heater (Figure 19) capable of providing more than 5000 gal of hot water per day for commercial and industrial use. These ponds are built in standard modules and tied together to supply the required load. The SSP module can be ground mounted or installed on a roof. It is typically 16 ft wide and up to 200 ft long. The module contains one or two flat water bags similar to a water bed. The bags rest on a layer of insulation inside concrete or fiberglass curbs. The bag is protected against damage and heat loss by greenhouse glazing. A typical pond filled to a 4 in. depth holds approximately 6000 gal of water.

Pool Heaters

Solar pool heaters do not require a separate storage tank, because the pool itself serves as storage. In most cases, the pool's filtration pump forces the water through the solar panels or plastic pipes. In some retrofit applications, a larger pump may be required to handle the needs of the solar heater, or a small pump may be added to boost the pool water to the solar collectors.

Automatic control may be used to direct the flow of filtered water to the collectors when solar heat is available; this may also be accomplished manually. Normally, solar heaters are designed to drain down into the pool when the pump is turned off; this provides the collectors with freeze protection.

Four primary types of collector designs are used for swimming pool heat: (1) rigid black plastic panels (polypropylene), usually 4 by 10 ft or 4 by 8 ft; (2) tube-on-sheet panels, which usually have a metal deck (copper or aluminum) with copper water tubes; (3) an EPDM rubber mat, extruded with the water passages running its length; and (4) arrays of black plastic pipe, usually 1.5 in. diameter ABS plastic (Root et al. 1985).

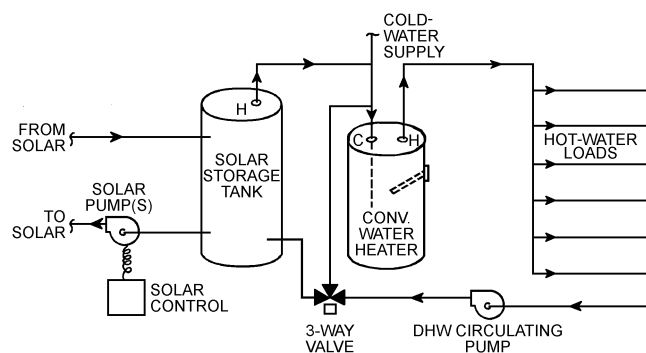


Fig. 20 DHW Recirculation System

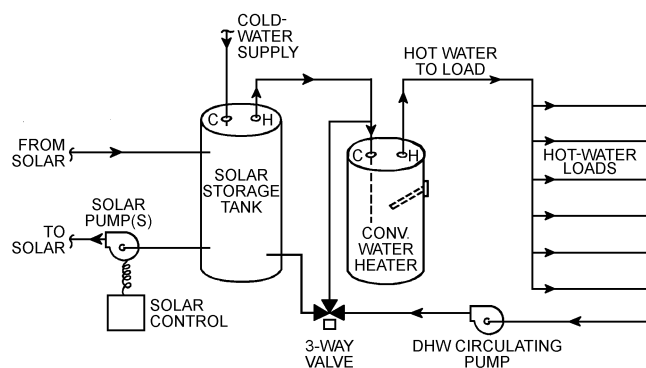


Fig. 21 DHW Recirculation System with Makeup Preheat

Hot-Water Recirculation

Domestic hot-water (DHW) recirculation systems (Figures 20 and 21), which continuously circulate domestic hot water throughout a building, are found in motels, hotels, hospitals, dormitories, office buildings, and other commercial buildings. The recirculation heat losses in these systems are usually a significant part of the total water heating load. A properly integrated solar heater can make up much of this loss.

SOLAR HEATING AND COOLING SYSTEMS

The components and subsystems discussed previously may be combined to create a wide variety of solar heating and cooling systems. These systems fall into two principal categories: passive and active.

Passive solar systems require little, if any, nonrenewable energy to make them function (Yellott 1977; Yellott et al. 1976). Every building is passive in the sense that the sun tends to warm it by day, and it loses heat at night. Passive systems incorporate solar collection, storage, and distribution into the architectural design of the building and make minimal or no use of fans to deliver the collected energy to the structure. Passive solar heating, cooling, and lighting design must consider the building envelope and its orientation, the thermal storage mass, and window configuration and design. ASHRAE (1984), DOE (1980/1982), LBL (1981), and Mazria (1979) give estimates of energy savings resulting from the application of passive solar design concepts.

Active solar systems use either liquid or air as the collector fluid. Active systems must have a continuous availability of nonrenewable energy, generally in the form of electricity, to operate pumps and fans. A complete system includes solar collectors, energy storage devices, and pumps or fans for transferring energy to storage or to the

load. The load can be space cooling, heating, or hot water. Although it is technically possible to construct a solar heating and cooling system to supply 100% of the design load, such a system would be uneconomical and oversized. The size of the solar system, and thus its ability to meet the load, is determined by life-cycle cost analysis that weighs the cost of energy saved against the amortized solar cost.

Active solar energy systems have been combined with heat pumps for water and/or space heating. The most economical arrangement in residential heating is a solar system in parallel with a heat pump, which supplies auxiliary energy when the solar source is not available. For domestic water systems requiring high water temperatures, a heat pump placed in series with the solar storage tank may be advantageous. Freeman et al. (1979) and Morehouse and Hughes (1979) present information on performance and estimated energy savings for solar-heat pumps.

Hybrid systems combine elements of both active and passive systems. Hybrid systems require some nonrenewable energy, but the amount is so small that they can maintain a coefficient of performance of about 50.

Passive Systems

Passive systems may be divided into several categories. The first residence to which the name **solar house** was applied used a large expanse of south-facing glass to admit solar radiation; this is known as a **direct-gain** passive system.

Indirect-gain solar houses use the south-facing wall surface or the roof of the structure to absorb solar radiation, which causes a rise in temperature that, in turn, conveys heat into the building in several ways. This principle was applied to the pueblos and cliff dwellings of the southwestern United States. Glass has led to modern adaptations of the indirect-gain principle (Balcomb et al. 1977; Trombe et al. 1977).

By glazing a large south-facing, massive masonry wall, solar energy can be absorbed during the day, and conduction of heat to the inner surface provides radiant heating at night. The mass of the wall and its relatively low thermal diffusivity delays the heat's arrival at the indoor surface until it is needed. The glazing reduces the loss of heat from the wall back to the atmosphere and increases the collection efficiency of the system.

Openings in the wall near the floor and ceiling allow convection to transfer heat to the room. The air in the space between the glass and the wall warms as soon as the sun heats the outer surface of the wall. The heated air rises and enters the building through the upper openings. Cool air flows through the lower openings, and convective heat gain can be established as long as the sun is shining.

In another indirect-gain passive system, a metal roof-ceiling supports transparent plastic bags filled with water (Hay and Yellott 1969). Movable insulation above these water-filled bags is rolled away during the winter day to allow the sun to warm the stored water. The water then transmits heat indoors by convection and radiation. The insulation remains over the water bags at night or during overcast days. During the summer, the water bags are exposed at night for cooling by (1) convection, (2) radiation, and (3) evaporation of water on the water bags. The insulation covers the water bags during the day to protect them from unwanted irradiation. Pittenger et al. (1978) tested a building for which water rather than insulation was moved to provide summer cooling and winter heating.

Attached greenhouses (sunspaces) can be used as solar attachments when the orientation and other local conditions are suitable. The greenhouse can provide a buffer between the exterior wall of the building and the outdoors. During daylight, warm air from the greenhouse can be introduced into the house by natural convection or a small fan.

In most passive systems, control is accomplished by moving a component that regulates the amount of solar radiation admitted into the structure. Manually operated window shades or venetian blinds are the most widely used and simplest controls.

Passive heating and cooling systems have been effective in field demonstrations (Howard and Pollock 1982; Howard and Saunders 1989).

COOLING BY NOCTURNAL RADIATION AND EVAPORATION

Radiative cooling is a natural heat loss that causes the formation of dew, frost, and ground fog. Because its effects are the most obvious at night, it is sometimes termed **nocturnal radiation**, although the process continues throughout the day. Thermal infrared radiation, which affects the surface temperature of a building wall or roof, may be estimated by using the sol-air temperature concept. Radiative cooling of window and skylight surfaces can be significant, especially under winter conditions when the dew-point temperature is low.

The most useful parameter for characterizing the radiative heat transfer between horizontal nonspectral emitting surfaces and the sky is the **sky temperature** T_{sky} . If S designates the total downcoming radiant heat flux emitted by the atmosphere, then T_{sky} is defined as

$$T_{sky}^4 = S/\sigma \quad (30)$$

where $\sigma = 0.1713 \times 10^{-8} \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{R}^4$.

The sky radiance is treated as if it originates from a blackbody emitter of temperature T_{sky} . The **net radiative cooling rate** R_{net} of a horizontal surface with absolute temperature T_{rad} and a nonspectral emittance ε is then

$$R_{net} = \varepsilon\sigma(T_{rad}^4 - T_{sky}^4) \quad (31)$$

Values of ε for most nonmetallic construction materials are about 0.9.

Radiative building cooling has not been fully developed. Design methods and performance data compiled by Hay and Yellott (1969) and Marlatt et al. (1984) are available for residential roof ponds that use a sealed volume of water covered by sliding insulation panels as the combined rooftop radiator and thermal storage. Other conceptual radiative cooling designs have been proposed, but more developmental work is required (Givoni 1981; Mitchell and Biggs 1979).

The sky temperature is a function of atmospheric water vapor, the amount of cloud cover, and air temperature; the lowest sky temperatures occur under an arid, cloudless sky. The monthly average sky temperature depression, which is the average of the difference between the ambient air temperature and the sky temperature, typically lies between 9 and 43°F throughout the continental United States. Martin and Berdahl (1984) have calculated this quantity using hourly weather data from 193 sites, as shown in the contour map for the month of July (Figure 22).

The sky temperature may be too high at night to effectively cool the structure. Martin and Berdahl (1984) suggest that the sky temperature should be less than 61°F to achieve reasonable cooling in July (Figure 23). In regions where sky temperatures fall below 61°F 40% or more of the month, all nighttime hours are effectively available for radiative cooling.

Clark (1981) modeled a horizontal radiator at various surface temperatures in convective contact with outdoor air for 77 U.S. locations. The average monthly cooling rates for a surface temperature of 76°F are plotted in Figure 24. If effective steps are taken to reduce the surface convection coefficient by modifying the radiator geometry or using an infrared-transparent glazing, it may be possible to improve performance beyond these values.

Active Systems

Active systems absorb solar radiation with collectors and convey it to storage using a suitable fluid. As heat is needed, it is obtained

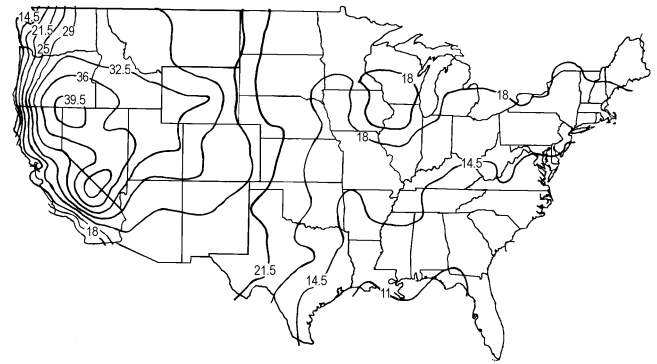


Fig. 22 Average Monthly Sky Temperature Depression ($T_{air} - T_{sky}$) for July, °F
(Adapted from Martin and Berdahl 1984)

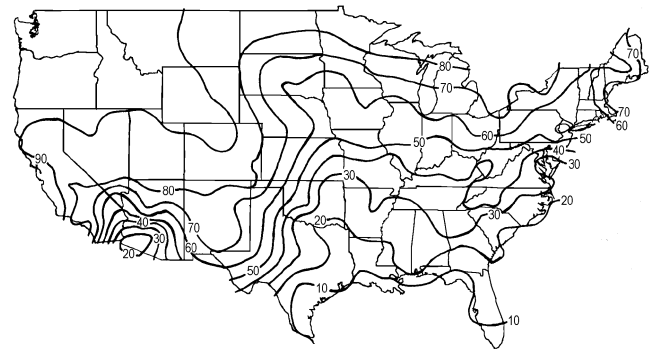


Fig. 23 Percentage of Monthly Hours when Sky Temperature Falls below 61°F
(Adapted from Martin and Berdahl 1984)

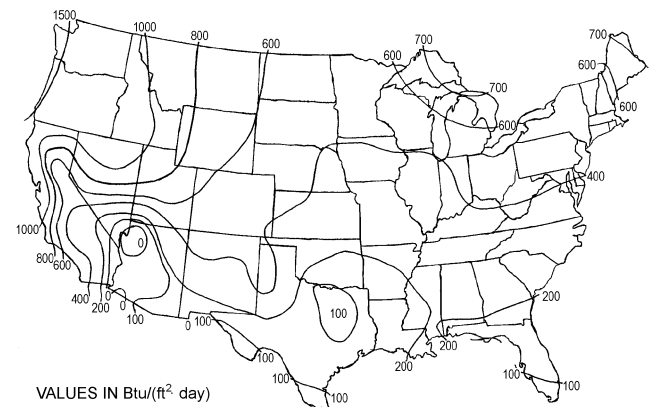


Fig. 24 July Nocturnal Net Radiative Cooling Rate from Horizontal Dry Surface at 76°F
(Adapted from Clark 1981)

from storage via heated air or water. Control is exercised by several types of thermostats, the first being a differential device that starts the flow of fluid through the collectors when they have been sufficiently warmed by the sun. It also stops the fluid flow when the collectors no longer gain heat. In locations where freezing occurs only rarely, a low-temperature sensor on the collector controls a circulating pump when freezing is impending. This process wastes some stored heat, but it prevents costly damage to the collector panels.

This system is not suitable for regions where freezing temperatures persist for long periods.

The space heating thermostat is generally the conventional double-contact type that calls for heat when the temperature in the controlled space falls to a predetermined level. If the temperature in storage is adequate to meet the heating requirement, a pump or fan is started to circulate the warm fluid. If the temperature in the storage subsystem is inadequate, the thermostat calls on the auxiliary or standby heat source.

Space Heating and Service Hot Water

Figure 25 shows one of the many systems for service hot water and space heating. In this case, a large, atmospheric pressure storage tank is used, from which water is pumped to the collectors by pump P_1 in response to the differential thermostat T_1 . Drainback is used to prevent freezing, because the amount of antifreeze required would be prohibitively expensive. Service hot water is obtained by placing a heat exchanger coil in the tank near the top, where, even if stratification occurs, the hottest water will be found.

An auxiliary water heater boosts the temperature of the sun-heated water when required. Thermostat T_2 senses the indoor temperature and starts pump P_2 when heat is needed. If the water in the storage tank becomes too cool to provide enough heat, the second contact on the thermostat calls for heat from the auxiliary heater.

Standby heat becomes increasingly important as heating requirements increase. The heating load, winter availability of solar radiation, and cost and availability of the auxiliary energy must be determined. It is rarely cost effective to do the entire heating job for either space or service hot water by using the solar heat collection and storage system alone.

Electric resistance heaters have the lowest first cost, but often have high operating costs. Water-to-air heat pumps, which use sun-heated water from the storage tank as the evaporator energy source, are an alternative auxiliary heat source. The heat pump's COP is 10 to 14 Btu of heat for each watt-hour of energy supplied to the compressor. When summer cooling as well as winter heating are needed, the heat pump becomes a logical solution, particularly in large systems where a cooling tower is used to dissipate the heat withdrawn from the system.

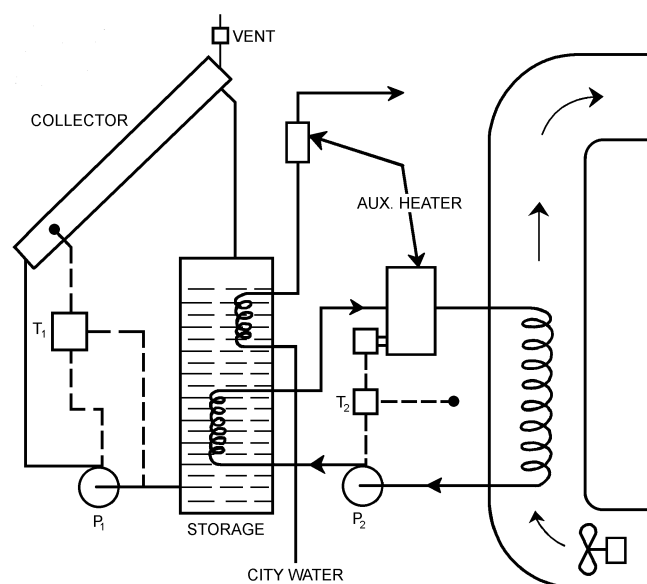


Fig. 25 Solar Collection, Storage, and Distribution System for Domestic Hot Water and Space Heating

The system shown in Figure 25 may be retrofitted into a warm-air furnace. In such systems, the primary heater is deleted from the space heating circuit, and the coil is located in the return duct of the existing furnace. Full backup is thus obtained, and the auxiliary heater provides only the heat not available at the storage temperature.

COOLING BY SOLAR ENERGY

Swartman et al. (1974) emphasize various absorption systems. Newton (in Jordan and Liu 1977) discusses commercially available water vapor/lithium bromide absorption refrigeration systems. Standard absorption chillers are generally designed to give rated capacity for activating fluid temperatures well above 200°F at full load and design condenser water temperature. Few flat-plate collectors can operate efficiently in this range; therefore, a lower hot-fluid temperature is used when solar energy provides the heat. Both the temperature of the condenser water and the percentage of design load are determinants of the optimum energizing temperature, which can be quite low, sometimes below 120°F. Proper control can raise the COP at these part-load conditions.

Many large commercial or institutional cooling installations must operate year-round, and Newton (Jordan and Liu 1977) showed that the low-temperature cooling water available in winter enables the LiBr/H₂O to function well with a hot-fluid inlet temperature below 190°F. Residential chillers in sizes as low as 1.5 tons, with an inlet temperature in the range of 175°F, have been developed.

Solar Cooling with Absorption Refrigeration

When solar energy is used for cooling as well as for heating, the absorption system shown in Figure 26, or one of its many modifications, may be used. The collector and storage must operate at a temperature approaching 200°F on hot summer days when the water from the cooling tower exceeds 80°F, but considerably lower operating water temperatures may be used when cooler water is available from the tower. The controls for collection, cooling, and distribution are generally separated, with the circulating pump P_1 operating in response to the collector thermostat T_1 , which is located within the air-conditioned space. When T_2 calls for heating, valves V_1 and V_2 direct the water flow from the storage tank through the unactivated auxiliary heater to the fan coil in the air distribution system. The fan F_1 in this unit may respond to the thermostat also, or it may have its own control circuit so that it can bring in outdoor air when a suitable temperature condition is present.

When thermostat T_2 calls for cooling, the valves direct the hot water into the absorption unit's generator, and pumps P_3 and P_4 are activated to pump the cooling tower water through the absorber and condenser circuits and the chilled water through the cooling coil in the air distribution system. A relatively large hot-water storage tank allows the unit to operate when no sunshine is available. A chilled-water storage tank (not shown) may be added so that the absorption unit can operate during the day whenever water is available at a sufficiently high temperature to make the unit function properly. The COP of a typical lithium bromide-water absorption unit may be as high as 0.75 under favorable conditions, but frequent on-off cycling of the unit to meet a high variable cooling load may cause significant loss in performance because the unit must be heated to operating temperature after each shutdown. Modulating systems are analyzed differently than on-off systems.

Water-cooled condensers are required with the absorption cycles, because the lithium bromide-water cycle operates with a relatively delicate balance among the temperatures of the three fluid circuits—cooling tower water, chilled water, and activating water. The steam-operated absorption systems, from which solar cooling systems are derived, customarily operate at energizing temperatures of 230 to 240°F, but these are above the capability of most flat-plate

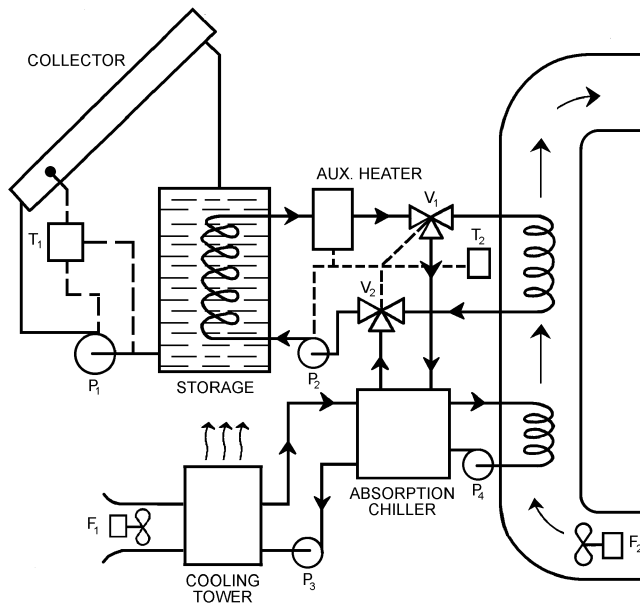


Fig. 26 Space Heating and Cooling System Using Lithium Bromide Absorption Chiller

collectors. The solar cooling units are designed to operate at considerably lower temperature, but unit ratings are also lowered.

Smaller domestic units may operate with natural circulation, or **percolation**, which carries the lithium bromide-water solution from the generator (to which the activating heat is supplied) to the separator and condenser; there, the re-concentrated LiBr is returned to the absorber while the water vapor goes to the condenser before being returned to the evaporator where cooling takes place. Larger units use a centrifugal pump to transfer the fluid.

-sizing solar heating and cooling systems—energy requirements

Methods used to determine solar heating and/or cooling energy requirements for both active and passive/hybrid systems are described by Feldman and Merriam (1979) and Hunn et al. (1987). Descriptions of public- and private-domain methods are included. An overview of simulation techniques suitable for active heating and cooling systems analysis, and for passive/hybrid heating, cooling, and lighting analysis follows.

Performance Evaluation Methods

The performance of any solar energy system is directly related to the (1) heating load, (2) amount of solar radiation available, and (3) solar energy system characteristics. Various calculation methods use different procedures and data when considering the available solar radiation. Some simplified methods consider only average annual incident solar radiation; complex methods may use hourly data.

Solar energy system characteristics, as well as individual component characteristics, are required to evaluate performance. The degree of complexity with which these systems and components are described varies from system to system.

The cost effectiveness of a solar domestic and service hot-water heating system depends on the initial cost and energy cost savings. A major task is to determine how much energy is saved. The **annual solar fraction**—the annual solar contribution to the water heating load divided by the total water heating load—can be used to estimate these savings. It is expressed as a decimal or percentage and generally ranges from 0.3 to 0.8 (30 to 80%), although more extreme values are possible.

Simplified Analysis Methods

Simplified analysis methods have the advantages of computational speed, low cost, rapid turnaround (especially important during iterative design phases), and ease of use by persons with little technical experience. Disadvantages include limited flexibility for design optimization, lack of control over assumptions, and a limited selection of systems that can be analyzed. Thus, if the application, configuration, or load characteristics under consideration are significantly nonstandard, a detailed computer simulation may be required to achieve accurate results. This section describes the *f*-Chart method for active solar heating and the solar load ratio method for passive solar heating (Dickinson and Cheremisinoff 1980; Klein and Beckman 1979; Lunde 1980).

Water Heating Load

The amount of hot water required must be estimated accurately because it affects component selection. Oversized storage may result in low-temperature water that requires auxiliary heating to reach the desired supply temperature. Undersizing can prevent the collection and use of available solar energy. [Chapter 49, Service Water Heating](#), gives methods to determine the load.²⁰

Active Heating/Cooling

Beckman et al. (1977) developed the *f*-Chart method using an hourly simulation program (Klein et al. 1976) to evaluate space heating and service water heating in many climates and conditions. The results of these analyses correlate the fraction *f* of the heat load met by solar energy. The correlations give the fraction *f* of the monthly heating load (for space heating and hot water) supplied by solar energy as a function of collector characteristics, heating loads, and weather. The standard error of the differences between detailed simulations in 14 locations in the United States and the *f*-Chart predictions was about 2.5%. Correlations also agree within the accuracy of measurements of long-term performance data. Beckman et al. (1977, 1981) and Duffie and Beckman (1980) discuss the method in detail.

The *f*-Chart method requires the following data:

- Monthly average daily radiation on a horizontal surface
- Monthly average ambient temperatures
- Collector thermal performance curve slope and intercept from standard collector tests; that is, $F_R U_L$ and $F_R (\tau\alpha)_n$ (see ASHRAE Standard 93 and Chapter 33 of the 2000 *ASHRAE Handbook—HVAC Systems and Equipment*)
- Monthly space- and water-heating loads

Standard Systems

The *f*-Chart assumes several standard systems and applies only to these liquid configurations. The standard **liquid heater** uses water, an antifreeze solution, or air as the heat transfer fluid in the collector loop and water as the storage medium ([Figure 27](#)). Energy is stored in the form of sensible heat in a water tank. A water-to-air heat exchanger transfers heat from the storage tank to the building. A liquid-to-liquid heat exchanger transfers energy from the main storage tank to a domestic hot-water preheat tank, which in turn supplies solar-heated water to a conventional water heater. A conventional furnace or heat pump is used to meet the space heating load when the energy in the storage tank is depleted.

[Figure 28](#) shows the assumed configuration for a **solar air heater** with a pebble-bed storage unit. Energy for domestic hot water is provided by heat exchange from the air leaving the collector to a domestic water preheat tank as in the liquid system. The hot water is further heated, if necessary, by a conventional water heater. During summer operation, a seasonal, manually operated storage bypass damper is used to avoid heat loss from the hot bed into the building.

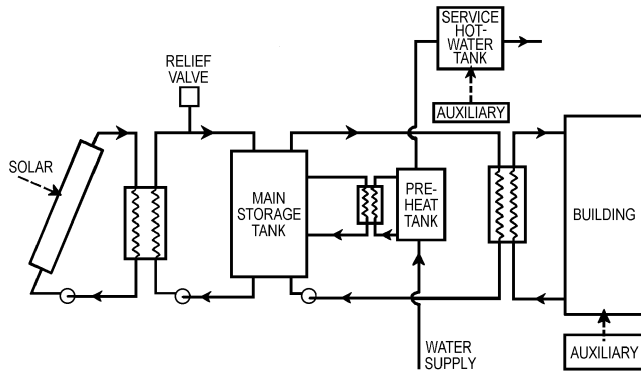


Fig. 27 Liquid-Based Solar Heating System
(Adapted from Beckman et al. 1977)

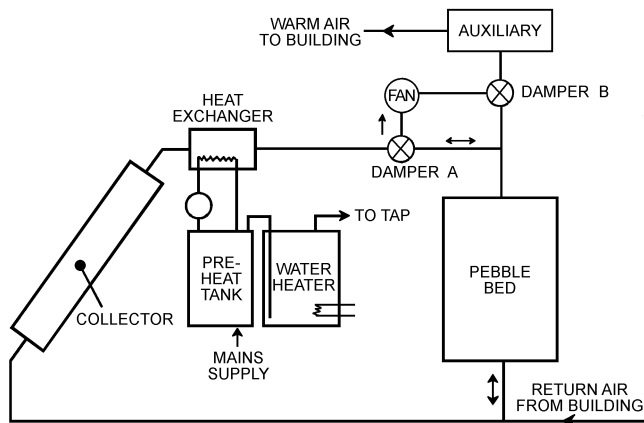


Fig. 28 Solar Air Heating System
(Adapted from Beckman et al. 1977)

The standard **solar domestic water heater** collector heats either air or liquid. Collected energy is transferred by a heat exchanger to a domestic water preheat tank that supplies solar-heated water to a conventional water heater. The water is further heated to the desired temperature by conventional fuel if necessary.

***f*-Chart Method**

Computer simulations correlate dimensionless variables and the long-term performance of the systems. The fraction f of the monthly space- and water-heating loads supplied by solar energy is empirically related to two dimensionless groups. The first dimensionless group X is collector loss; the second Y is collector gain:

$$X = \frac{F_R U_L A_c \Delta \theta}{L} \left(\frac{F_r}{F_R} \right) (t_{ref} - \bar{t}_a) \quad (32)$$

$$Y = \frac{F_R (\tau \alpha)_n \bar{H}_T N A_c \left(\frac{F_r}{F_R} \right) \left[\frac{(\tau \alpha)_n}{(\tau \alpha)_n} \right]}{L} \quad (33)$$

where

- A_c = area of solar collector, ft²
- F_r = collector-heat exchanger efficiency factor
- F_R = collector efficiency factor
- U_L = collector overall energy loss coefficient, Btu/h·ft²·°F
- $\Delta \theta$ = total number of hours in month

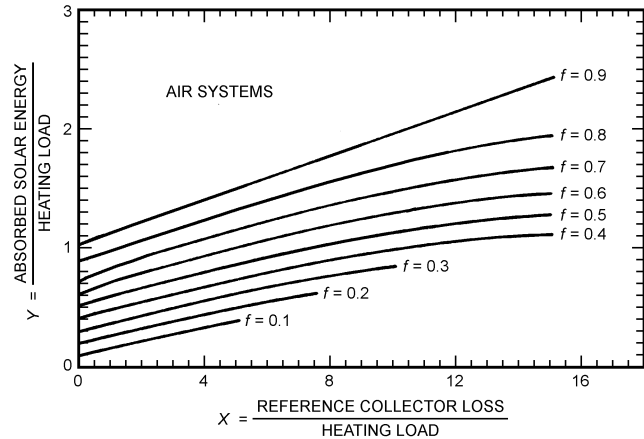


Fig. 29 Chart for Air System
(Adapted from Beckman et al. 1977)

\bar{t}_a = monthly average ambient temperature, °F

L = monthly total heating load for space heating and hot water, Btu

\bar{H}_T = monthly averaged, daily radiation incident on collector surface per unit area, Btu/day·ft²

N = number of days in month

$(\tau \alpha)_n$ = monthly average transmittance-absorptance product

$(\tau \alpha)_n$ = normal transmittance-absorptance product

t_{ref} = reference temperature, 212°F

$F_R U_L$ and $F_R (\tau \alpha)_n$ are obtained from collector test results. The ratios F_r/F_R and $(\tau \alpha)_n/(\tau \alpha)_n$ are calculated using methods given by Beckman et al. (1977). The value of \bar{t}_a is obtained from meteorological records for the month and location desired. \bar{H}_T is calculated from the monthly averaged, daily radiation on a horizontal surface by the methods in Chapter 33 of the 2000 *ASHRAE Handbook—HVAC Systems and Equipment* or in Duffie and Beckman (1980). The monthly load L can be determined by any appropriate load estimating method, including analytical techniques or measurements. Values of the collector area A_c are selected for the calculations. Thus, all the terms in these equations can be determined from available information.

Transmittance τ of the transparent collector cover and the absorptance α of the collector plate depend on the angle at which solar radiation is incident on the collector surface. Collector tests are usually run with the radiation incident on the collector in a nearly perpendicular direction. Thus, the value of $F_R (\tau \alpha)_n$ determined from these tests ordinarily corresponds to the transmittance and absorptance values for radiation at normal incidence. Depending on collector orientation and time of year, the monthly average values of the transmittance and absorptance can be significantly lower. The f -Chart method requires a knowledge of the ratio of the monthly average to normal incidence transmittance-absorptance.

The f -Chart method for liquid systems is similar to that for air systems. The fraction of the monthly total heating load supplied by the solar air heating system is correlated with the dimensionless groups X and Y , as shown in Figure 29. To determine the fraction of the heating load supplied by solar energy for a month, values of X and Y are calculated for the collector and heating load in question. The value of f is determined at the intersection of X and Y on the f -Chart, or from the following equivalent equations.

Air system:

$$f = 1.04Y - 0.065X - 0.159Y^2 + 0.00187X^2 - 0.0095Y^3 \quad (34)$$

Liquid system:

$$f = 1.029Y - 0.065X - 0.245Y^2 + 0.0018X^2 + 0.025Y^3 \quad (35)$$

This is done for each month of the year. The solar energy contribution for the month is the product of f and the total heating load L for the month. Finally, the fraction f of the annual heating load supplied by solar energy is the sum of the monthly solar energy contributions divided by the annual load:

$$F = \sum fL / \sum L$$

Example 6. Calculating the heating performance of a residence, assume that a solar heating system is to be designed for use in Madison, WI, with two-cover collectors facing south, inclined 58° with respect to the horizontal. The air heating collectors have the characteristics $F_R U_L = 0.50 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F}$ and $F_R(\tau\alpha)_n = 0.49$. The T_a is 19.4°F , the total space and water heating load for January is calculated to be $34.1 \times 10^6 \text{ Btu}$, and the solar radiation incident on the plane of the collector is calculated to be $1.16 \times 10^3 \text{ Btu/day} \cdot \text{ft}^2$. Determine the fraction of the load supplied by solar energy with a system having a collector area of 538.2 ft^2 .

Solution: For air systems, there is no heat exchanger penalty factor and $F_p/F_R = 1$. The value of $(\bar{\tau\alpha})/(\tau\alpha)_n$ is 0.94 for a two-cover collector in January. Therefore, the values of X and Y are

$$X = 0.50(1)(212 - 19.4)(31)(24)(538.2)/(34.1 \times 10^6) = 1.13$$

$$Y = (0.49)(1)(0.94)(1.16 \times 10^3)(31)(538.2)/(34.1 \times 10^6) = 0.26$$

Then the fraction f of the energy supplied for January is 0.19. The total solar energy supplied by this system in January is

$$fL = (0.19)(34.1 \times 10^6) = 6.4 \times 10^6 \text{ Btu}$$

The annual system performance is obtained by summing the energy quantities for all months. The result is that 37% of the annual load is supplied by solar energy.

The collector heat removal factor F_R that appears in X and Y is a function of the collector fluid flow rate. Because of the higher cost of power for moving fluid through air collectors than through liquid collectors, the capacitance rate used in air heaters is ordinarily much lower than that in liquid heaters. As a result, air heaters generally have a lower value of F_R . Values of F_R corresponding to the expected airflow in the collector must be used to calculate X and Y .

An increase in airflow rate tends to improve collector performance by increasing F_R , but it tends to decrease performance by reducing the degree of thermal stratification in the pebble bed (or water storage tank). The f -Chart for air systems is based on a collector airflow rate of 2 scfm per square foot of collector area. The performance with different collector airflow rates can be estimated by using the appropriate values of F_R in both X and Y . A further modification to the value of X is required to account for the change in degree of stratification in the pebble bed.

Air system performance is less sensitive to storage capacity than that of liquid systems for two reasons: (1) air systems can operate with air delivered directly to the building in which storage is not used, and (2) pebble beds are highly stratified and additional capacity is effectively added to the cold end of the bed, which is seldom heated and cooled to the same extent as the hot end. The f -Chart for air systems is for a nominal storage capacity. Performance of systems with other storage capacities can be determined by modifying the dimensionless group X as described in Beckman et al. (1977).

With modification, f -Charts can be used to estimate the performance of solar water heating operating in the range of 120 to 160°F . The main water supply temperature and the minimum acceptable

hot-water temperature (i.e., the desired delivery temperature) both affect the performance of solar water heating. The dimensionless group X , which is related to collector energy loss, can be redefined to include these effects. If monthly values of X are multiplied by a correction factor, the f -Chart for liquid-based solar space and water heating systems can be used to estimate monthly values of f for water heating. Experiments and analysis show that the load profile for a well-designed heater has little effect on long-term performance. Although the f -Chart was originally developed for two-tank systems, it may be applied to single- and double-tank domestic hot water systems with and without collector tank heat exchangers.

For industrial process heating, absorption air conditioning, or other processes for which the delivery temperature is outside the normal f -Chart range, modified f -Charts are applicable (Klein et al. 1976). The concept underlying these charts is that of solar usability, which is the fraction of the total solar energy that is useful in the given process. This fraction depends on the required delivery temperature as well as collector characteristics and solar radiation. The procedure allows the energy delivered to be calculated in a manner similar to that for f -Charts. An example of the application of this method to solar-assisted heat pumps is presented in Svard et al. (1981).

Other Active Collector Methods

The **relative areas method**, based on correlations of the f -Chart method, predicts annual rather than monthly active heating performance (Barley and Winn 1978). An hourly simulation program has been used to develop the **monthly solar-load ratio (SLR) method**, another simplified procedure for residential systems (Dickinson and Cheremisinoff 1980). Based on hour-by-hour simulations, a method was devised to estimate performance based on monthly values of horizontal solar radiation and heating degree-days. This SLR method has also been extended to nonresidential buildings for a range of design water temperatures (Dickinson and Cheremisinoff 1980; Schnurr et al. 1981).

Passive Heating

A widely accepted simplified passive space heating design tool is the solar-load ratio method (ASHRAE 1984; DOE 1980, 1982). It can be applied manually, although like the f -Chart, software is also available. The SLR method for passive systems is based on correlating results of multiple hour-by-hour computer simulations, the algorithms of which have been validated against test cell data for the following generic passive heating types: direct gain, thermal storage wall, and attached sunspace. Monthly and annual performance, as expressed by the auxiliary heating requirement, is predicted by this method. The method applies to single-zone, envelope-dominated buildings. A simplified, annual-basis distillation of SLR results, the **load collector ratio (LCR) method**, and several simple-to-use rules have grown out of the SLR method. Several hand-held calculator and microcomputer programs have been written using the method (Nordham 1981).

The SLR method uses a single dimensionless correlating parameter (SLR), which Balcomb et al. (1982) define as a particular ratio of solar energy gains to building heating load:

$$\text{SLR} = \frac{\text{Solar energy absorbed}}{\text{Building heating load}} \quad (36)$$

A correlation period of 1 month is used; thus the quantities in the SLR are calculated for a 1 month period.

The parameter that is correlated to the SLR, the **solar savings fraction (SSF)**, is defined as

$$\text{SSF} = 1 - \frac{\text{Auxiliary heat}}{\text{Net reference load}} \quad (37)$$

The SSF measures the energy saving expected from the passive solar building, relative to a reference nonpassive solar building.

In Equation (37), the net reference load is equal to the degree-day load DD of the nonsolar elements of the building:

$$\text{Net reference load} = (\text{NLC})(\text{DD}) \quad (38)$$

where NLC is the net load coefficient, which is a modified UA coefficient computed by leaving out the solar elements of the building. The nominal units are Btu/°F·day. The term DD is the temperature departure in degree-days computed for an appropriate base temperature. A building energy analysis based on the SLR correlations begins with a calculation of the monthly SSF values. The monthly auxiliary heating requirement is then calculated by

$$\text{Auxiliary heat} = (\text{NLC})(\text{DD})(1 - \text{SSF}) \quad (39)$$

Annual auxiliary heat is calculated by summing the monthly values.

By definition, SSF is the fraction of the heat load of the nonsolar portions of the building met by the solar element. If the solar elements of the building (south-facing walls and window in the northern hemisphere) were replaced by other elements so that the net annual flow of heat through these elements was zero, the annual heat consumption of the building would be the net reference load. The savings achieved by the solar elements would therefore be the net reference load in Equation (38) minus the auxiliary heat in Equation (39), which gives

$$\text{Solar saving} = (\text{NLC})(\text{DD})(\text{SSF}) \quad (40)$$

Although simple, in many situations and climates, Equation (40) is only approximately true because a normal solar-facing wall with a normal complement of opaque walls and windows has a near-zero effect over the entire heating season. In any case, the auxiliary heat estimate is the primary result and does not depend on this assumption.

The hour-by-hour simulations used as the basis for the SLR correlations are done with a detailed model of the building in which all the design parameters are specified. The only parameter that remains a variable is the solar collector area, which can be expressed in terms of the load collector ratio (LCR):

$$\text{LCR} = \frac{\text{NLC}}{A_p} = \frac{\text{Net load coefficient}}{\text{Projected collector area}} \quad (41)$$

Performance variations are estimated from the correlations, which allow the user to account directly for thermostat set point, internal heat generation, glazing orientation, and configuration, shading, and other solar radiation modifiers. Major solar system characteristics are accounted for by selecting one of 94 reference designs. Other design parameters, such as thermal storage thickness and conductivity, and the spacing between glazings, are included in a series of sensitivity calculations obtained using hour-by-hour simulations. The results are generally presented in graphic form so that the designer can see the effect of changing a particular parameter.

Solar radiation correlations for the collector area have been determined using hour-by-hour simulation and typical meteorological year (TMY) weather data. These correlations are expressed as ratios of incident-to-horizontal radiation, transmitted-to-incident radiation, and absorbed-to-transmitted radiation as a function of the latitude minus mid-month solar declination and the atmospheric clearness index K_T .

The performance predictions of the SLR method have been compared to predictions made by the detailed hour-by-hour simulations for a variety of climates in the United States. The standard error in the prediction of the annual SSF, compared to the hour-by-hour simulation, is typically 2 to 4%.

The annual solar savings fraction calculation involves summing the results of 12 monthly calculations. For a particular city, the resulting SSF depends only on the LCR of Equation (41), the type of system, and the temperature base used in calculating the temperature departure. Thus, tables that relate SSF to LCR for the various systems and for various degree-day base temperatures may be generated for a particular city. Such tables are easier for hand analysis than are the SLR correlations.

Annual SSF versus LCR tables have been developed for 209 locations in the United States and 14 cities in southern Canada for 94 reference designs and 12 base temperatures (ASHRAE 1984).

Example 7. Consider a small office building located in Denver, Colorado, with 3000 ft² of usable space and a sunspace entry foyer that faces due south; the projected collector area A_p is 420 ft². A sketch and preliminary plan are shown in Figure 30. Distribution of solar heat to the offices is primarily by convection through the doorways from the sunspace. The principle thermal mass is in the common wall that separates the sunspace from the offices and in the sunspace floor. This example is abstracted from the detailed version given in Balcomb et al. (1982). Even though lighting and cooling are likely to have the greatest energy costs for this building, heating is a significant energy item and should be addressed by a design that integrates passive solar heating, cooling, and lighting.

Solution: Table 5 shows calculations of the net load coefficient. Then,

$$\text{NLC} = 24 \times 525 = 12,600 \text{ Btu/°F} \cdot \text{day}$$

and the total load coefficient includes the solar aperture

$$\text{TLC} = 24 \times 676 = 16,224 \text{ Btu/°F} \cdot \text{day}$$

From Equation (43), the load collector ratio is

$$\text{LCR} = 12,600/420 = 30 \text{ Btu/ft}^2 \cdot \text{°F} \cdot \text{day}$$

Table 5 Calculations for Example 7

	Area A , ft ²	U-Factor, Btu/h·ft ² ·°F	UA, Btu/h·°F
Opaque wall	2000	0.04	80
Ceiling	3000	0.03	90
Floor (over crawl space)	3000	0.04	120
Windows (E, W, N)	100	0.55	55
		Subtotal	345
Infiltration			180
		Subtotal	525
Sunspace (treated as unheated space)			151
		Total	676

ASHRAE procedures are approximated with V = volume, ft³; c = heat capacity of Denver air, Btu/ft³·°F; ACH = air changes/h; equivalent UA for infiltration = $Vc\text{ACH}$. In this case, $V = 24,000$ ft³, $c = 0.015$ Btu/ft³·°F, and ACH = 0.5.

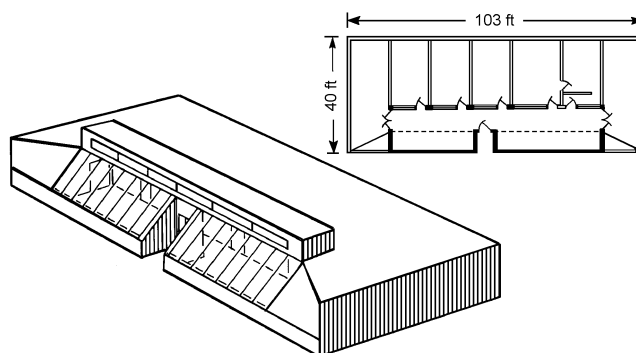


Fig. 30 Commercial Building in Example 7

If the daily internal heat is 130,000 Btu/day, and the average thermostat setting is 68°F, then

$$T_{base} = 68 - 130,000 / 16,224 = 60^\circ\text{F}$$

(Note that solar gains to the space are not included in the internal gain term as they customarily are for nonsolar buildings.)

In this example, the solar system is type SSD1, defined in ASHRAE (1984) and Balcomb et al. (1982). It is a semiclosed sunspace with a 12 in. masonry common wall between it and the heated space (offices). The aperture is double-glazed, with a 50° tilt and no night insulation. To achieve a projected area of 420 ft², a sloped glazed area of $420/\sin 50^\circ = 548 \text{ ft}^2$ is required.

The SLR correlation for solar system type SSD1 is shown in Figure 31. Values of the absorbed solar energy S and heating degree-days DD to base temperature 60°F are determined monthly. For S , the solar radiation correlation presented in ASHRAE (1984) for tabulated Denver, CO, weather data is used. For January, the horizontal surface incident radiation is 840 Btu/ft²·day. The 60°F base degree-days are 933. Using the tabulated incident-to-absorbed coefficients found in ASHRAE (1984), $S = 43,681 \text{ Btu/ft}^2$. Thus $S/DD = 46.8 \text{ Btu/ft}^2 \cdot ^\circ\text{F} \cdot \text{day}$. From Figure 31, at an LCR = 30 Btu/ft²·°F·day, the SSF = 0.51. Therefore, for January,

$$\text{Net reference load} = (933)(12,600) = 11.76 \times 10^6 \text{ Btu}$$

$$\text{Solar savings} = (11.76 \times 10^6)(0.51) = 6.00 \times 10^6 \text{ Btu}$$

$$\text{Auxiliary heat} = (11.76 \times 10^6) - (6.00 \times 10^6) = 5.76 \times 10^6 \text{ Btu}$$

Repeating this calculation for each month and adding the results for the year yields an annual auxiliary heat of $19.94 \times 10^6 \text{ Btu}$.

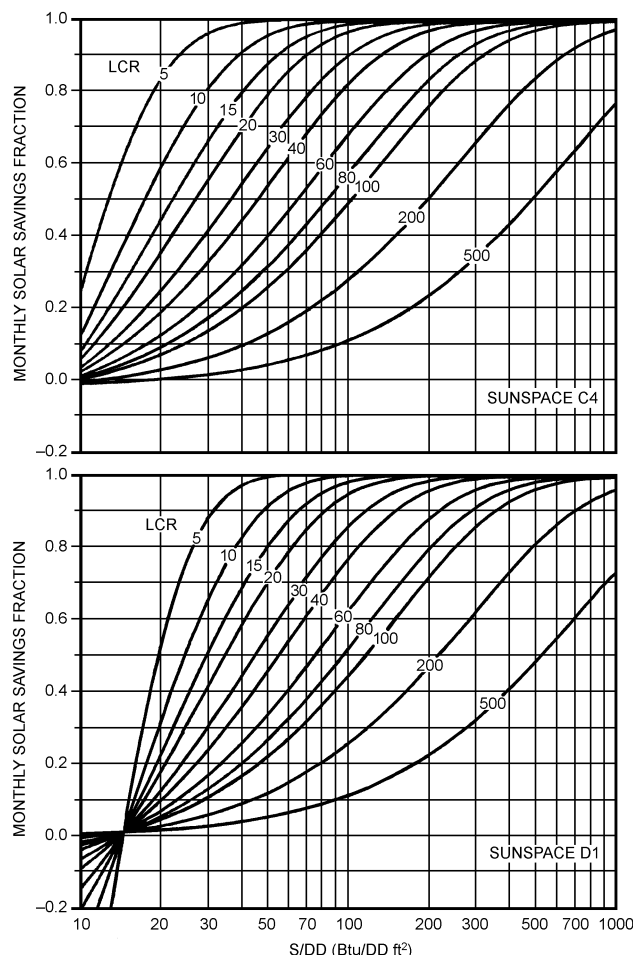


Fig. 31 Monthly SSF Versus Monthly S/DD for Various LCR Values

Other Passive Heating Methods

The concept of usability has been applied to passive buildings. In this approach, the energy requirements of zero- and infinite-capacity buildings are calculated. The amount of solar energy that enters the building and exceeds the instantaneous load of the zero-capacity building is then calculated. This excess energy must be dumped in the zero-capacity building, but it can be stored to offset heating loads in a finite-capacity building. Methods are provided to interpolate between the zero- and infinite-capacity limits for finite-capacity buildings. Equations and graphs for direct gain and collector-storage wall systems are given in Monsen et al. (1981, 1982).

INSTALLATION GUIDELINES

Most solar components are the same as those in HVAC and hot-water systems (pumps, piping, valves, and controls), and their installation is not much different from a conventional installation. Solar collectors are the most unfamiliar component in a solar heater. They are located outdoors, which requires penetration of the building envelope. They also require a structural element to support them at the proper tilt and orientation toward the sun.

The site must taken into account. Collectors should be (1) located so that shading is minimized and (2) installed so that they are attractive both on and off site. They should also be located to minimize vandalism and to avoid a safety hazard.

Collectors should be placed near the storage tank to reduce piping cost and heat loss. The collector and piping must be installed so that they can be drained without trapping fluid in the system.

For best annual performance, collectors should be installed at a tilt angle above the horizontal that is appropriate for the local latitude. In the northern hemisphere they should be oriented toward true south, not magnetic south. Small variations in tilt ($\pm 10^\circ$) and orientation ($\pm 20^\circ$) do not reduce performance significantly.

Collector Mounting

Solar collectors are usually mounted on the ground or on flat or pitched roofs. A roof location necessitates penetration of the building envelope by mounting hardware, piping, and control wiring. Ground or flat-roof-mounted collectors are generally rack mounted.

Pitched-roof mounting can be done several ways. Collectors can be mounted on structural **standoffs**, which support them at an angle other than that of the roof to optimize solar tilt. In another pitched-roof-mounting technique known as **direct mounting**, collectors are placed on a waterproof membrane on top of the roof sheeting. The finished roof surface together with the necessary collector structural attachments and flashing are then built up around the collector. A weatherproof seal between the collector and the roof must be maintained to prevent leakage, mildew, and rotting.

Integral mounting can be done for new pitched-roof construction. The collector is attached to and supported by the structural framing members. The top of the collector then serves as the finished roof surface. Weathertightness is crucial to avoid damage and mildew.

Collectors should support snow loads that occur on the roof area they cover. The collector tilt usually expedites snow sliding with only a small loss in efficiency. The roof structure should be free of objects that could impede snow sliding, and the collectors should be raised high enough to prevent snow buildup over them.

The mounting structure should be built to withstand winds of at least 100 mph, which impose a wind load of 40 lb/ft² on a vertical surface or an average of 25 lb/ft² on a tilted roof (HUD 1977). Wind load requirements may be higher, depending on local building codes. Flat-plate collectors mounted flush with the roof surface should be constructed to withstand the same wind loads. See Chapter 54 for additional information.

The collector array becomes more vulnerable to wind gusts as the angle of the mount increases. This wind load, in addition to the

equivalent roof area wind loads, should be determined according to accepted engineering procedures (ASCE *Standards* 7, 8, and 9).

Expansion and contraction of system components, material compatibility, and the use of dissimilar metals must be considered. Collector arrays and mounting hardware (bolts, screws, washers, and angles) must be well protected from corrosion. Steel-mounting hardware in contact with aluminum, and copper piping in contact with aluminum hardware are both examples of metal combinations that have a high potential for corrosion. Dissimilar metals can be separated by washers made of fluorocarbon polymer, phenolic, or neoprene rubber.

Freeze Protection

Freeze protection is extremely important and is often the determining factor when selecting a system in the United States. Freezing can occur at ambient temperatures as high as 42°F because of radiation to the night sky. Manual freeze protection should not be used for commercial installations.

One simple way of protecting against freezing is to drain the fluid from the collector array and interior piping when potential freezing conditions exist. The drainage may be automatic, as in draindown and drainback systems, or manual, as in direct thermosiphon systems. Automatic systems should be capable of fail-safe drainage operation—even in the event of pump failure or power outage. In some cases water may be designed to drain back through the pump, so the design must allow refilling without causing cavitation.

In areas where freezing is infrequent, recirculating water from storage to the collector array can be used as freeze protection. Freeze protection can also be provided by using fluids that resist freezing. Fluids such as water/glycol solutions, silicone oils, and hydrocarbon oils are circulated by pumps through the collector array and double wall heat exchanger. Draining the collector fluid is not required, because these fluids have freezing points well below the coldest anticipated outdoor temperature.

In mild climates where recirculation freeze protection is used, a second level of freeze protection should be provided by flushing the collector with cold supply water when the collector approaches near-freezing temperatures. This can be accomplished with a temperature-controlled valve that will automatically open a small port at a near-freezing temperature of about 40°F and then close at a slightly higher temperature.

Overheat Protection

During periods of high insolation and low hot-water demand, overheating can occur in the collectors or storage tanks. Protection against overheating must be considered for all portions of the solar hot-water system. Liquid expansion or excessive pressure can burst piping or storage tanks. Steam or other gases within a system can restrict liquid flow, making the system inoperable.

The most common methods of overheat protection stop circulation in the collection loop until the storage temperature decreases, discharge the overheated water and replace it with cold makeup water, or use a heat exchanger as a means of heat rejection. Some freeze protection methods can also provide overheat protection.

For nonfreezing fluids such as glycol antifreezes, overheat protection is needed to limit fluid degradation at high temperatures during collector stagnation.

Safety

Safety precautions required for installing, operating, and servicing a solar domestic hot-water heater are essentially the same as those required for a conventional domestic hot-water heater. One major exception is that some solar systems use nonpotable heat transfer fluids. Local codes may require a double wall heat exchanger for potable water installations.

Pressure relief must be provided in all parts of the collector array that can be isolated by valves. The outlet of these relief valves should be piped to a container or drain, and not where people could be affected.

Start-Up Procedure

After completing the installation, certain tests must be performed before charging or filling the system. The system must be checked for leakage, and pumps, fans, valves, and sensors must be checked to see that they function. Testing procedures vary with system type.

Closed-loop systems should be hydrostatically tested. The system is filled and pressurized to 1.5 times the operating pressure for one hour and inspected for leaks and any appreciable pressure drop.

Draindown systems should be tested to be sure that all water drains from the collectors and piping located outdoors. All lines should be checked for proper pitch so that gravity drains them completely. All valves should be verified to be in working order.

Drainback systems should be tested to ensure that the collector fluid is draining back to the reservoir tank when circulation stops and that the system refills properly.

Air systems should be tested for leaks before insulation is applied by starting the fans and checking the ductwork for leaks.

Pumps and sensors should be inspected to verify that they are in proper working order. Proper cycling of the pumps can be checked by a running time meter. A sensor that is suspected of being faulty can be dipped alternately in hot and cold water to see if the pump starts or stops.

After system testing and before filling or charging it with heat transfer fluid, the system should be flushed to remove debris.

Maintenance

All systems should be checked at least once a year in addition to any periodic maintenance that may be required for specific components. A log of all maintenance performed should be kept, along with an owner's manual that describes system operational characteristics and maintenance requirements.

The collectors' outer glazing should be hosed down periodically. Leaves, seeds, dirt, and other debris should be carefully swept from the collectors. Care should be taken not to damage plastic covers.

Without opening up a sealed collector panel, the absorber plate should be checked for surface coating damage caused by peeling, crazing, or scratching. Also, the collector tubing should be inspected to ensure that it contacts the absorber. If the tubing is loose, the manufacturer should be consulted for repair instructions.

Heat transfer fluids should be tested and replaced at intervals suggested by the manufacturer. Also, the solar energy storage tank should be drained about every six months to remove sediment.

Performance Monitoring/Minimum Instrumentation

Temperature sensors and temperature differential controllers are required to operate most solar systems. However, additional instruments should be installed for monitoring, checking, and troubleshooting.

Thermometers should be located on the collector supply and return lines so that the temperature difference in the lines can be determined visually.

A pressure gage should be inserted on the discharge side of the pump. The gage can be used to monitor the pressure that the pump must work against and to indicate if the flow passages are blocked.

Running time meters on pumps and fans may be installed to determine if the system is cycling properly.

DESIGN, INSTALLATION, AND OPERATION CHECKLIST

The following checklist is for designers of solar heating and cooling systems. Specific values have not been included because these vary for each application. The designer must decide whether design figures are within acceptable limits for any particular project (see DOE 1978b for further information). The review order listed does not reflect their precedence or importance during design.

Collectors

- Check flow rate for compliance with manufacturer's recommendation.
- Check that collector area matches the application and claimed solar fraction of the load.
- Review collector instantaneous efficiency curve and check match between collector and system requirements.
- Relate collector construction to end use; two cover plates are not required for low-temperature collection in warm climates and may, in fact, be detrimental. Two cover plates are more efficient when the temperature difference between the absorber plate and outdoor air is high, such as in severe winter climates or when collecting at high temperatures for cooling. Radiation loss only become significant at relatively high absorber plate temperatures. Selective surfaces should be used in these cases. Flat-black surfaces are acceptable and sometimes more desirable for low collection temperatures.
- Check match between collector tilt angle, latitude, and collector end use.
- Check collector azimuth.
- Check collector location for potential shading and exposure to vandalism or accidental damage.
- Review provisions made for high stagnation temperature. If not used, are liquid collectors drained or left filled in the summer?
- Check for snow hang-up and ice formation. Will casing vents become blocked?
- Review precautions, if any, against outgassing.
- Check access for cleaning covers.
- Check mounting for stability in high winds.
- Check for architectural integration. Do collectors on roof present rainwater drainage or condensation problems? Do roof penetrations present potential leak problems?
- Check collector construction for structural integrity and durability. Will materials deteriorate under operating conditions? Will any pieces fall off?
- Are liquid collector passages organized in such a way as to allow natural fill and drain? Does mounting configuration affect this?
- Does air collector duct connection promote balanced airflow and an even heat transfer? Are connections potentially leaky?

Hydraulics

- Check that the flow rate through the collector array matches system parameters.
- If antifreeze is used, check that the flow rate has been modified to allow for the viscosity and specific heat.
- Review properties of proposed antifreeze. Some fluids are highly flammable. Check toxicity, vapor pressure, flash point, and boiling and freezing temperatures at atmospheric pressure.
- Check means of makeup into antifreeze system. An automatic water makeup system can result in freezing.
- Check that provisions are made for draining and filling the system. (Air vents at high points, drains at low points, pipes correctly graded in-between, drainback vented to storage or expansion tank.)
- If system uses drainback freeze protection, check that
 - Provision is made for drainback volume and back venting
 - Pipes are graded for drainback

- Solar primary pump is sized for lift head
- Pump is self-priming if tank is below pump

- Check that collector pressure drop for drainback is slightly higher than static pressure between the supply and return headers.
- Optimum pipe arrangement is reverse return with collectors in parallel. Series collectors reduce flow rate and increase head. A combination of parallel/series can sometimes be beneficial, but check that equipment has been sized and selected properly.
- Cross-connections under different operating modes sometimes result in pumps operating in opposition or tandem, causing severe hydraulic problems.
- If heat exchangers are used, check that approach temperature differential has been recognized in the calculations.
- Check that adequate provisions are made for water expansion and contraction. Use specific volume/temperature tables for calculation. Each unique circuit must have its own provision for expansion and contraction.
- Three-port valves tend to leak through the closed port. This, together with reversed flows in some modes, can cause potential hydraulic problems. As a general rule, simple circuits and controls are better.

Airflow

- Check that the flow rate through the collector array matches the system design values.
- Check temperature rise across collectors using air mass flow and specific heat.
- Check that duct velocities are within the design limits.
- Check that cold air or water cannot flow from collectors by gravity under "no-sun" conditions.
- Verify duct material and construction methods. Ductwork must be sealed to reduce loss.
- Check duct configuration for balanced flow through collector array.
- Check number of collectors in series. More than two collectors in series can reduce collection efficiency.

Thermal Storage

- Check that thermal storage capacity matches values of collector area, collection temperature, use temperature, and load.
- Verify that thermal inertia does not impede effective operation.
- Check provisions for promoting temperature stratification during both collection and use.
- Check that pipe and duct connections to storage are compatible with the control, ensuring that only the coolest air goes to the collectors and connections.
- If liquid storage is used for high-temperature (above 200°F) applications, check that tank material and construction can withstand the temperature and pressure.
- Check that storage location does not promote unwanted heat loss or gain and that adequate insulation is provided.
- Verify that liquid storage tanks are treated to resist corrosion. This is particularly important in tanks that are partially filled.
- Check that provision is made to protect liquid tanks from exposure to either an overpressure or vacuum.

Uses

Domestic Hot Water

- Characteristics of domestic hot-water loads include short periods of high draw interspersed with long dormant periods. Check that domestic hot-water storage matches solar heat input.
- Check that provisions have been made to prevent reverse heating of the solar thermal storage by the domestic hot-water backup heater.

- Check that the design allows cold makeup water preheating on days of low solar input.
- Verify that the antiscald valve limits domestic hot-water supply to a safe temperature during periods of high solar input.
- Depending on total dissolved solids, city water heated above 150°F may precipitate a calcium carbonate scale. If collectors are used to heat water directly, check provisions for preventing scale formation in absorber plate waterways.
- Check whether the heater is required to have a double wall heat exchanger and that it conforms to appropriate codes if the collector uses nonpotable fluids.

Heating

- Warm air heating systems have the potential of using solar energy directly at moderate temperatures. Check that air volume is sufficient to meet the heating load at low supply temperatures and that the limit thermostat has been reset.
- At times of low solar input, solar heat can still be used to meet part of the load by preheating return air. Check location of solar heating coil in system.
- Baseboard heaters require relatively high supply temperatures for satisfactory operation. Their output varies as the 1.5 power of the log mean temperature difference and falls off drastically at low temperatures. If solar is combined with baseboard heating, check that supply temperature is compatible with heating load.
- Heat exchangers imply an approach temperature difference that must be added to the system operating temperature to derive the minimum collection temperature. Verify calculations.
- Water-to-air heat pumps rely on a constant solar water heat source for operation. When the heat source is depleted, the backup system must be used. Check that storage is adequate.

Cooling

- Solar-activated absorption cooling with fossil fuel backup is currently the only commercially available active cooling. Be assured of all design criteria and a large amount of solar participation. Verify calculations.
- Storing both hot water and chilled water may make better use of available storage capacity.

Controls

- Check that control design matches desired modes of operation.
- Verify that collector loop controls recognize solar input, collector temperature, and storage temperature.
- Verify that controls allow both the collector loop and the usage loop to operate independently.
- Check that control sequences are reversible and will always revert to the most economical mode.
- Check that controls are as simple as possible within the system requirements. Complex controls increase the frequency and possibility of breakdowns.
- Check that all controls are fail-safe.

Performance

- Check building heating, cooling, and domestic hot-water loads as applicable. Verify that building thermal characteristics are acceptable.
- Check solar energy collected on a monthly basis. Compare with loads and verify solar participation.

PHOTOVOLTAIC APPLICATIONS

Photovoltaic (PV) devices, or cells, convert light directly into electricity. Chapter 33 of the 2000 *ASHRAE Handbook—HVAC Systems and Equipment* discusses the fundamentals of their operation. Photovoltaics are used to power communications, equipment

in remote locations, remote monitoring, lighting, water pumping, battery charging, and cathodic protection.

Communications. Photovoltaics can provide reliable power with little maintenance for communications systems, especially those in remote areas (away from the power grid) with extreme weather conditions such as high winds, heavy snows, and ice. Examples include communication relay towers, travelers' information transmitters, cellular telephones, mobile radio systems, emergency call boxes, and military test facilities. These systems range in size from a few watts for call boxes to several kilowatts for microwave repeater stations. For larger systems at remote sites, an engine generator is often combined with the photovoltaic-battery system. These hybrid systems with two or more generators can achieve nearly 100% availability.

Remote Site Electrification. Photovoltaics are used to provide power to rural residences, visitor centers in parks, vacation cabins, island and villages, remote research facilities, and military test areas. The varied loads include lighting, small appliances, water pumps (including circulators on solar water heating systems), and communications equipment. The load demand varies from a few watts to tens of kilowatts.

Remote Monitoring. Photovoltaics provide power at remote sites to sensors, data loggers, and associated transmitters for meteorological monitoring, structural condition measurement, seismic recording, irrigation control, highway/traffic monitoring, and scientific research. Most of these applications require less than 200 W, and many can be powered by a single photovoltaic module. Vandalism may be a problem in some areas, so non-glass-covered modules are sometimes used. Mounting the modules on a tall pole or in an unobtrusive manner may also help avoid damage or theft. The batteries are often located in the same weather-resistant enclosure as the data acquisition/monitoring equipment. This enclosure is sometimes camouflaged or buried for protection. Some data loggers come with their own battery and charge regulator.

Signs and Signals. The most popular application for photovoltaics is warning signs. Typical devices that are powered include navigational beacons, audible signals such as sirens, highway warning signs, railroad signals, and aircraft warning beacons. Because these signals are critical to public safety, they must be operative at all times, and thus the reliability of the photovoltaic system is extremely important. High reliability can be achieved by using large-capacity batteries and no charge control. Many of these systems operate in harsh environments. For maritime applications, special modules are used that are resistant to corrosion from salt water.

Water Pumping and Control. Photovoltaics are used for intermediate-sized water pumping applications—those larger than hand pumps and smaller than large engine-powered pumps. The typical range of sizes for photovoltaic-powered pumps is a few hundred watts to a few kilowatts. Applications include domestic use, water for campgrounds, irrigation, village water supplies, and livestock watering. Photovoltaics for livestock watering can be more economical than maintaining a distribution line to a remote pump on a ranch. Most pumping systems do not use batteries but store the water in holding tanks. The photovoltaic modules are often mounted on tracking frames that maximize energy production by tracking the sun each day.

Rest Room Facilities. Highway rest stops, public beach facilities, outdoor recreation parks, and public campgrounds may have photovoltaics to operate air circulation and ventilation fans, interior and exterior lights, and auxiliary water pumps for sinks and showers. For most of these applications, the initial cost for photovoltaic power is the least expensive option.

Charging Vehicle Batteries. Vehicle batteries self-discharge over time if they are not being used. This is a problem for organizations that maintain a fleet of vehicles such as fire-fighting or snow removal equipment, some of which are infrequently used. Photovoltaic battery chargers can solve this problem by providing a trickle

charging current that keeps the battery at a high state of charge. Often the PV module can be placed inside the windshield and plugged into the cigarette lighter, thus using existing wiring and protection circuits and providing a quick disconnect for the module. The modules are installed on the roof or engine hood of larger vehicles. An application under development is the use of PV modules to charge the batteries in electric vehicles.

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