Analysis of Performance

A perfect thermal-storage system would return all of the energy supplied to it at the temperature at which the energy was originally supplied. In real systems, of course, energy is lost through the containers and the transport system, and the temperatures of fluids which discharge the storage are lower than those used to "charge" the storage with thermal energy. This temperature reduction results not only from a net loss of energy to the surroundings, but also from the mixing of hot and cold fluids in the storage vessel (if a single tank is used), unless stratification is achieved, and from the temperature difference which must be maintained across any heat exchangers which are employed.

One advantage of chemical storage systems is that the products of the reactions can usually be stored at ambient temperatures. In chemical systems, however, temperature losses are the necessary result of the fact that reversible reactions must operate with the forward reaction occurring at a temperature different from the reverse reaction. Energy is also lost during the process of cooling the hot chemicals emerging from the high-temperature reaction chambers down to ambient temperature. Some chemical reactions also require the addition of "process heat" to stimulate intermediate processes, and this heat requirement lowers the overall "efficiency" of the storage process.

The significance of energy lost through insulation, parasitic loads due to pumps, process heat, etc., is clear. This is energy which cannot be recovered for useful work. On the other hand, the significance of storage temperature reductions which do not result in a net loss of energy requires more explanation: 1) heat engines are more efficient at high temperatures, and 2) the performance of solar collectors decreases as the exit temperature of fluids produced by the collectors increases; thus collector performance is maximized if the storage system can be charged from the collectors at a temperature close to the temperature at which the fluids will ultimately be used. In general, systems which use the stored heat only for space heating and domestic hot water will not be as sensitive to temperature degradation as systems which generate electricity or which drive air-conditioning systems directly, but size and cost of even simple systems can be reduced if the temperatures can be kept high.

The designs chosen will depend crucially on the economics of the system. Insulation should be added until the incremental value of the energy saved by insulation equals the incremental cost of adding insulation, Similarly, the sophistication of systems designed to maintain temperatures at constant levels should be increased until the incremental value of the energy provided by these systems is less than the incremental investment needed to prevent the additional drop in temperature.

STORAGE TEMPERATURE LOSS AND HEAT ENGINE EFFICIENCY

In the immediately following analysis, losses through insulation and temperature drops across heat exchangers are ignored for simplicity; they are included later. A detailed analysis of the effect of heat exchangers is contained in appendix A of Bramlette.[1]

The performance of heat engines can be approximated by assuming that their mechanical or electrical output is some fraction (f_e) of their ideal Carnot efficiency.

$$Q_{out} = f_e Q_{in} (1 - T_C / T_h)$$

(constant-temperature storage or two-tank system)

where Q_{out} is the mechanical or electrical energy generated by the heat engine, Q_{in}

[1] T Bramlette, et al (Sandia La boratories, Livermore), *Survey of High- Temperature Thermal Energy Storage, SAN* D75-8063, March 1976

was the energy originally sent to the storage, T_c is the temperature at which heat is rejected from the engine, T_h is the temperature at which energy is sent to the heat engine (also the temperature of the storage medium), and T_c and T_h are absolute temperatures.

If the temperature of the storage (T_σ) drops from T_h to T_h during discharge because of mixing or other effects, the energy produced by the heat engine will be given by:

$$Q_{out} = f_e \int_0^{Q_{in}} (1 - T_c/T_\sigma)dQ$$

If there is perfect mixing in the storage medium (i.e., a constant temperature in the tank) and no phase change (no losses) is encountered:

$$Q_{out} = f_e Q_{in} \left[\frac{T_h - T'_h - T_c \ln(T_h/T'_h)}{T_h - T'_h}\right] \quad \text{(perfect mixing)}$$

If there is perfect mixing in the storage medium and N phase changes are encountered:

$$Q_{out} = f_e Q_{in} \left[\frac{T_h - T'_h - T_c \ln(T_h/T'_h) + \sum_{i=1}^{N} [1 - T_c/T_{pc}(i)]\, \Delta H_{pc}(i)/C_p}{T_h - T'_h + \sum_{i=1}^{N} \Delta H_{pc}(i)/C_p}\right]$$

where $T_{pc}(i)$ and $\Delta H_{pc}(i)$ are the temperature and the enthalpy change associated with the i^{th} phase change. C_p is the specific heat. The specific heat is assumed to be constant for all temperatures in this simple approximation.

In the case of a thermochemical reaction, it is assumed that the reaction proceeds to 95-percent completion at a temperature T_h in the forward direction (in the solar collector), and proceeds to 95-percent completion at a temperature T_h in the reverse direction at the user site. In this case:

$$Q_{out} = f_e Q_{in} (1 - T_c/T'_h)$$

It is now possible to define a uniform figure of merit or "temperature efficiency" (η_T) for all of these types of thermal-storage techniques used with heat engines, It will simply be the ratio of Q_{out} for the technique in question to Q_{out} for a constant-tern perature storage system.

$$\eta_T = \begin{cases} \dfrac{1 - \dfrac{T_c}{T_h - T'_h} \ln(T_h/T'_h)}{(1 - T_c/T_h)} & \text{(perfect mixing)} \\[2ex] 1.0 & \text{(two-tank storage)} \\[2ex] \left[\dfrac{T_h - T'_h - T_c \ln\left[\dfrac{T_h}{T'_h}\right] + \sum\limits_{i=1}^{N}\left(1 - \dfrac{T_c}{T_{pc}(i)}\right)\dfrac{\Delta H_{pc}(i)}{C_p}}{\left[1 - \dfrac{T_c}{T_h}\right]\left[T_h - T'_h + \sum\limits_{i=1}^{N}\dfrac{\Delta H_{pc}(i)}{C_p}\right]}\right] & \text{(phase change with perfect mixing)} \\[2ex] \dfrac{T_h(T'_h - T_c)}{T'_h(T_h - T_c)} & \text{(chemical reaction)} \end{cases}$$

The average efficiency of a heat engine used with thermal storage is just:

$$\eta_T\ f_e(1-T_c/T_h)$$

if η_e is the heat engine efficiency at temperature T_h. Note that the heat engine will operate more efficiently than this average figure when the storage is fully charged and the temperature is closer to T_h than to T_h'. This "temperature efficiency" is not the same as "round-trip storage efficiency" (Q_{out}/Q_{in}), which is the ratio of the amount of heat which can be recovered from a fully charged system divided by the amount of heat which is required to charge it. "Round- trip storage efficiency" affects al l energy storage systems, while "temperature efficiency" is of concern primarily for systems which drive heat engines from stored heat. While a temperature drop sometimes signals a loss of thermal energy, the "temperature efficiency" is a measure of how the engine performance is reduced as a result of heat being delivered at a lower temperature, and has nothing to do with how much energy was first pumped into storage or how much subsequently leaked out.

SENSIBLE- AND LATENT-HEAT STORAGE LOSSES

In storage systems using sensible and latent heat there are three primary loss mechanisms: losses through the insulation; temperature degradation in heat exchangers; and temperature degradation due to mixing.

In all of the direct thermal-storage applications considered, it will be assumed that the storage vessels are cylindrical with a radius R and a height L which is equal to 4R.

Losses Through the Insulation

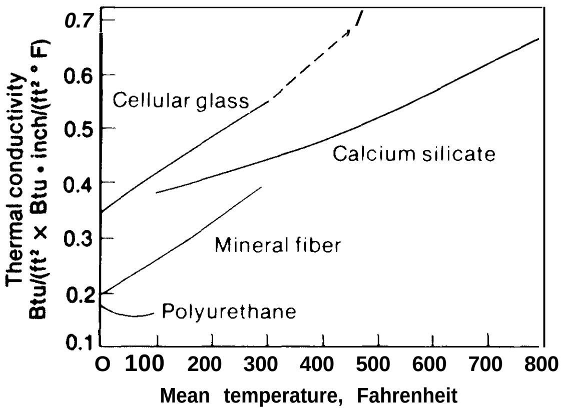
The losses from such a tank will, in general, depend on the thickness (t_i) and conductivity (k) of the insulation and the heat transfer coefficient (h) giving the conductive, convective, and radiative losses of the outer insulation surface to the air or ground. The heat loss rate (Q) from the cylinder just described would be approximately:

$$\dot{Q} = 2\pi(T_s - T_a)k\left[\frac{L}{\ln(R_o/R_i) + k/(hR_o)} + \frac{R_o^2}{t_i + k/h}\right]$$

where T_s is the average temperature of the storage material and T_a is the ambient temperature; R_o is the radius of the cylinder with insulation, and R_i is the radius inside the insulation.

This equation can be simplified in most cases of practical interest by assuming that the thickness of insulation (t_i) is small compared to R_o [i. e.: (R_o-R_i) is much less than R_i] and that the quantity k/h is small compared with t_i. This last condition is equivalent to an assumption that the air or earth surrounding the tank provides no insulation, and is, therefore, a conservative assumption. Figure XI- B-1 shows that most common insulations have k less than 0.05 Btu/hr ft°F. h will range between the value for air with free convection (l-5 Btu/hr ft²°F) to that of water with forced convection (50 to 100 Btu/hr ft²°F). [1]

Figure XI- B-1 .—Thermal Conductivity of Insulation



SOURCES:
1. *ASHRAE Handbook of Fundamentals*. 1972, page 293.
2. "Foamglass Insulation" Brochure FI-132 (REV). Pittsburgh Corning Corp., April 1975.

[1] F T Kreith, *Principles of Heat Transfer,* second edition, 1965, p 15

It can be seen that k/h is small for either assumption. In the remainder of this discussion, a cylindrical storage container with length L = 4R is assumed, unless stated otherwise. With the assumptions $t_i/R \ll 1$ and $t_i \ll k/h$, the heat flow Q from the cylinder can be approximated as follows:

$$\dot{Q} = 10\pi(T_s - T_a)R^2 k/t_i$$

As noted earlier, in an actual design, the insulation thickness would be chosen so that the marginal cost of adding additional insulation would be equal to the marginal cost of energy saved by adding this insulation. For simplicity, the Insulation used in the calculations which follow is based *on* the premise that only 5 percent of the stored energy will be lost during the desired storage interval of Ø hours (Ø would be 24 hours for daily storage, 168 hours for weekly storage, etc.). ($Q\Theta = 0.05E_s$) This fraction can then be used to determine the desired thickness of insulation.

The energy stored (E_s) can be written approximately as follows (assuming that the specific heat does not change significantly with temperature).

$$E_s = 4\pi R^3 \varrho \begin{cases} C_v(T_h - T'_h) & \text{sensible heat} \\ C_v(T_h - T_{pc}) + \Delta h_{pc} + C_v(T_{pc} - T'_h) & \text{sensible heat and latent heat} \\ \Delta h_{pc} & \text{latent heat} \end{cases}$$

Alternatively, the volume required to store a given amount of energy is:

$$V_s = \frac{E_s}{\varrho} \begin{cases} [C_v(T_h - T'_h)]^{-1} & \text{sensible heat} \\ [C_v(T_h - T'_h + \Delta h_{pc}]^{-1} & \text{sensible heat plus latent heat} \\ [\Delta h_{pc}]^{-1} & \text{latent heat} \end{cases}$$

Here T_h and T'_h are the high- and low- temperature limits of the storage cycle and ϱ is the density of the material. C_v and Δh_{pc} are the specific heat and latent heat of phase change per unit volume The desired thickness for the insulation can then be given as follows.

$$t_i = \frac{50k(T_s - T_a)\,\Theta}{R\varrho} \begin{cases} \dfrac{1}{C(T_h - T'_h)} & \text{sensible heat} \\ (1/\Delta h_{pc}) & \text{latent heat} \end{cases}$$

Earth Insulation

In practice, the soil itself will provide some insulation for a buried tank. The insulating properties of the soil will depend strongly on local conditions, in particular, ground water flows, The heat conductivity of soil, for example, varies from 0.1 Btu/hr ft°F for dry sand or soil to 2 for wet sand or soil.[3] Assuming that the storage tank is a simple hemisphere, it can be shown that the heat leaving through the earth when equilibrium is reached with soil at temperature T_∞ and the outer tank temperature T_o is given by:[4]

$$Q = 2\pi r_s k_s (T_o - T_\infty) = 2\pi r_s^2 U_s (T_o - T_\infty)$$

where r_s is the radius of the storage and k_s is the soil's heat conductivity. If we write the heat flow through the insulation as $U_i(T_s - T_o)$ per unit tank area, where T, is the temperature inside the tank:

$$\dot{Q} = 2\pi r_s^2 \frac{T_s - T_\infty}{(1/U_s) + (1/U_i)}$$

Note that it is assumed that losses to the surface are minimai. The system can be well insulated on top, buried deeply, or placed under the building to minimize surface losses.

As shown in the equation above, Shelton found that the thermal losses to the ground were proportional to the perimeter rather than the surface area of the hemlsphere He also showed that the earth is equlvalent to a layer of insulation of thickness $k_i R/k_s$, where k_i is the thermal conductivity of the insulation being compared and R is the radius of the storage tank. The thermal conductiv ty

[3] Jay Shelton, 'Underground Storage of Heat in Solar Heating Systems,' *Solar Energy* 17(2), 1975, p. 138

[4] Ibid

of different kinds of soil can vary by more than an order of magnitude, depending on soil composition and moisture content (see table XI-B-1). A hot storage tank under or near a building or paved area tends to dry out the soil surrounding it. Moderately dry soil of a type typically found near building sites has a conductivity of 6.0 Btu in/hr ft² "F. (About 40 times the conductivity of polyurethane.)

If it is computed that a thickness t_i of insulation with conductivity k_i must be used to reduce losses to acceptable levels without using earth insulation, burying the tank in soil can reduce the required thickness to t_i' where

$$t_i' = \begin{cases} t_i - k_i R/k_s & \text{if } t_i > k_i R/k_s \\ 0 & \text{otherwise} \end{cases}$$

For large storage tanks, the thermal losses are small even if no insulation other than the soil is used. Figure XI-B-2 shows the annual thermal losses from an uninsulated storage tank (as a percentage of the storage capacity) for a hot water storage system operating between 1200 and 2000 F. While the losses are large for small systems, by the time the storage capacity reaches 1 million kWh (typical of the size needed to provide 100 percent solar heating and hot water for a high rise apartment), the annual losses have dropped to 6 percent of the storage capacity.

Shelton also shows, however, that the heat loss through the soil is much greater than the equilibrium for the first few months after the system is installed, as the soil

Table XI-B-1 .—Thermal properties of soils

Soil description	k Btu/hr ft² °F/ft	$\alpha(=k/C_V)$ ft²/hr °F	C_V Btu/ft³	ρ lb/ft³
Very dry soil[a]	0.1-0.2			
Wet soil[a]	0.7-2.0			
Dry sand[b]	0.1	0.0054	19	
Wetsand[b]	1	0.042	25	
Sandy clay (15% moisture)[c]	0.6	0.015	37	
Concrete[c]	1.6	0.046	34	
Organic soil[a]	0.8	0.021	36	
Wet marshy soil[a]	0.7	0.012	54	
Granite rock[a]	0.26	0.008	31	
Concreted	0.54	0.025	22	144
Dry earth, packed[d]	0.037			95
Sand[d]	0.19	0.011	18	95
Moist high-conductivy soil[e]	1.2	0.026	46	102
Moist medium-low conductivity soil[e]	0.4	0.013	30	100
Assorted soils[f]	0.3-1.3			90-110

SOURCE. Reproduced from Shelton (op cit.) p 143,

aR. E, Munn, *Descriptive Micrometeorology*, 34. Academic Press, New York (1966).

bR Geiger, Constant values for estimation of heat economy for the agricultural meteorologist *Met Rundschau*, Heft 11 /12 (1948)

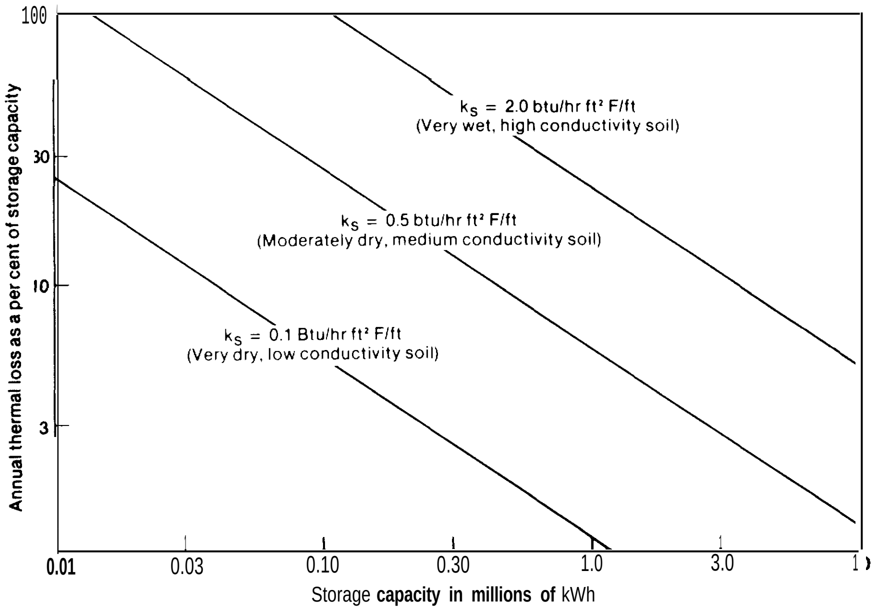
cH Lettov, Theory of surface temperature and heat transfer oscillations near a level ground Surface. *Trans* Am *Geophys U* 32, 189 (1951)

d Geiger, op cits, P P 571-572

'L R Ingersoll, F T Adler, H. J. Plass and A. C. Ingersoll, Theory of earth heat exchangers for the heat pump, *Trans Arrr Soc Heating Ventilating Engrs* 57, 167 (1951)

f Geiger, op cits , p 374

Figure XI-B-2.-Annual Thermal Losses From Uninsulated Storage Tank



NOTE. The thermal losses plotted above are based on the results of Shelton using water as the storage medium for a wide range of soil conductivities Shelton's results were modified by OTA to assume a n average storage temperature of 160°F, a ground temperature of 50°F, and a storage temperature swing of AT= 80°F.

'Shelton, op. cit Table XI. B-1.

warms up and dries out. Figure XI-B-3 illustrates this transient behavior.[5] He concludes that roughly a year is required for the soil to develop its full insulating value.[6] For simplicity, this assessment assumes the equilibrium value.

Thus, the total volume of insulation which must be purchased is the greater of

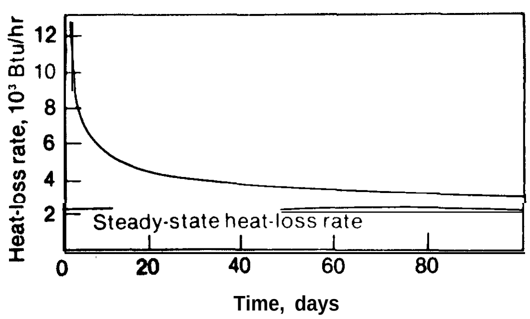
$$V_i = \begin{cases} 10\pi R^2(t_i - kR/6) \\ \text{zero} \end{cases}$$

where k is in Btu in/ft²hr °F. This shows that the insulation cost per kWh of the energy stored is:

$$\frac{C_1\Theta}{E^{2/3}} - C_2$$

(where C_1 and C_2 are positive constants) or zero, whichever is larger.

Figure XI-B-3.– Rate of Heat=Loss to Ground



Rate of heat-loss to ground as a function of time after initial start-up. It is assumed that there is no heat demand, so that all the collected solar heat is available to maintain the storage temperature at near its peak level. This curve corresponds to a 6 ft radius ground or gravel hemisphere storage region with a maximum temperature of 170°F with initial surrounding ground temperature of 50°F, and a solar-heat collection rate of 94,000 Btu/hr for eight hours *every* day (except when maximum storage temperature would be exceeded). The ground's volume heat capacity is 20 Btu/ft° F and its conductivity is 0-50 Btu/hr ft²⁰F/ft.

SOURCE:
J. Shelton (Williams College, Williamstown MA). "Underground Storage of Heat in Solar Heating Systems," ***Solar Energy.*** Volume 17, No. 2, May 1975, pages 137.143

[5] Ibid
[6] Ibid

Shelton's early calculation only assessed the thermal energy lost into the earth and did not include an estimate of the losses from heated earth into the atmosphere. When ground-to-air losses were taken into account in a detailed analog simulation of a 27,000-gallon storage pond, it was found that total losses were approximately twice those caculated by Shelton.' It should be possible to keep seasonal losses in thermal storage ponds at acceptable levels even if these added effects are considered, however. Clearly, this is an area where further analysis is warranted

Reduction in Temperature Due to Mixing in the Storage Vessel

It can be very difficult to compute the temperature gradient which can be maintained in a storage vessel. The result depends strongly on the design of the system (e.g., use of multiple interior baffles, pumping rates, tank orientation). A variety of clever approaches are currently being examined and much more interesting work can be expected. I n the absence of the necessary empirical or theoretical work, it is necessary to make some very crude approximations.

Multiple-Tank Systems

The problem of temperature degradation due to mixing is a problem only for techniques employing sensible heat for storage It is clear that the best way to eliminate mixing is to use separate tanks for the hot and cold liquids. The major difficulty with this is, of course, the added expense of another tank.

In the case of a two-tank system of capacity E_s, each tank is as large as the one- tank volume calculated earlier. The volume of heat-storage liquid is the same as before, and it is pumped back and forth between the half-empty tanks as the storage is charged and discharged Calculation of insulation thickness is similar to the earlier case.

$$t_i(\text{hot}) = \frac{50(T_h - T_a)k\Theta}{R\varrho C(T_h - T_c)}$$

$$t_i(\text{cold}) = \frac{50(T_c - T_a)k\Theta}{R\varrho C(T_h - T_c)}$$

Even though each tank is half full, it is still assumed that heat is lost through the entire wall The insulating value of the earth is still equal to a thickness kR/6 of insulation. The additional cost of the extra tank could be justified if the added performance permitted would lead to significant energy saving through increased heat-engine performance or if the cold tank can be enough colder than the lower limit of mixing storage that the storage capacity of the fluid is significantly increased. The use of multiple tanks in the same manner, but with only one tank empty at any time, would result in lower costs. The analysis in this paper assumes that multiple-tank hot water systems for low-temperature storage are 50 percent more expensive than single-tank systems of the same capacity. However, hot oil storage has been costed on the basis of two tanks

Solid Storage Media

Another approach is to use a solid material such as stone or steel as the storage medium. As noted earlier, some thought has been given to the possibility of using steel ingots or other material for storing energy in specific heat at temperatures as high as 1,0000 F. One concept would use large steel rectangular cylinders stacked together in an insulated container building. Heat would be added and removed by passing steam/water through a series of holes penetrating each steel cylinder. The heat transfer from the hot end of the steel cylinders and the cold ends should be slow enough to permit a temperature gradient of several hundred degrees to be maintained across the storage. This

[7] J Taylor Beard, F A Iachetta, L U Lilleleht, and J W Dickey, *Annual Collection and Storage of Solar Energy for the Heat/rig of Buildings, Report No. 2,* prepared for ERDA under contract No E-(40-I)-51 36, July 1977

would permit the withdrawal of high temperature steam at relatively constant temperature.

An ideal material for such a storage system would have the following characteristics:

- Very low cost,
- High specific heat,
- Low thermal transfer from the hot to the cold side of storage,
- High thermal transfer from the charging tubes to the storage medium, and
- Ability to withstand constant thermal cycling without degradation

The search for an Ideal material is in a very prellminary stage and many promising concepts have not progressed beyond speculation.

Temperature gradients can, of course, also be maintained in tanks of liquids, since hot liquids are usually lighter and rise to the top of a tank. This natural "thermocline" can be assisted by insulating baffles and other devices for reducing convection. The table indicates that liquids are capable of maintaining large gradients if convection can be eliminated.

Reduction in Temperature Due to Heat Exchangers

The rate of heat transfer in a counterflow heat exchanger In which no phase change occurs in the materlals and in which changes in the specific heat are negligible can be written as follows:

$$Q = UA \frac{\Delta T_h - \Delta T_c}{\ln(\Delta T_h / \Delta T_c)} \quad (= UA\Delta T \text{ when } \Delta T_h = \Delta T_c)$$

where *A* is the area of the heat exchanger, ΔT_h is the temperature difference between the two liquids at the hot side of the heat exchanger, ΔT_c the temperature difference at the cold side, and U is the overall thermal conductivity between the fluids, The size, and thus the cost of the heat exchanger, will determine the temperature difference ΔT for a given rate of heat exchange. In an ideal system design, the heat exchange area would be chosen so that the marginal cost of increasing the area would equal the marginal value of extra useful energy resulting from a higher temperature emerging from storage.

If one or both of the liquids passing through the heat exchanger undergoes a phase change, the analysls becomes significantly more complex. This situation could occur if a phase-change materlal is used for storage or if water is boiled to steam while passing through the exchanger Designing heat-exchanger systems using materials which sol id if y can be a serious difficulty.

If heat exchangers are used both for charging and discharging, two separate temperature reductions would occur one during charging of the storage and one during discharge. One of these heat exchangers can be eliminated if the same fluid is used in the storage and collector system.

Systems of the type shown in figure XI-13 have a great advantage in that they do not require any heat exchangers (the whole storage device being, in effect, a large heat exchanger). Liquid thermal-storage systems can also be built without heat exchangers if the storage medium is also the heat-transfer fluid.