CHAPTER II

LITERATURE REVIEW

Introduction

Energy storage plays important roles in conserving available energy and improving its utilization, since many energy sources are intermittent in nature. Short term storage of only a few hours is essential in most applications, however, long term storage of a few months may be required in some applications. Solar energy is available only during the day, and hence, its application requires an efficient thermal energy storage so that the excess heat collected during sunshine hours may be stored for later use during the night. Similar problems arise in heat recovery systems where the waste heat availability and utilization periods are different requiring some thermal energy storage. Also, electrical energy consumption varies significantly during the day and night, especially in extremely cold and hot climate countries where the major part of the variation is due to domestic space heating and air conditioning. Such variation leads to an off peak period, usually after midnight until early morning. Accordingly, power stations have to be designed for capacities sufficient to meet the peak load. Otherwise, very efficient power distribution would be required. Better power generation management can be achieved if some of the peak load could be shifted to the off peak load period, which can be achieved by thermal storage of heat or coolness. Hence, the successful application of load shifting and solar energy depends to a large extent on the method of energy storage used.

The most commonly used method of thermal energy storage in all the above mentioned applications is the sensible heat method. In solar heating systems, water is still used for heat storage in liquid based systems, while a rock bed is used for air based systems. In the application of load leveling, heat is usually stored in a refractory bricks storage heater, known as a night storage heater [13].

Solar thermal power plant [13]

Solar thermal systems are non-polluting and offer significant protection of the environment. The reduction of greenhouse gasses pollution is the main advantage of utilizing solar energy. Therefore, solar thermal systems should be employed whenever possible in order to achieve a sustain-able future.

Conversion of solar to mechanical and electrical energy has been the objective of experiments for more than a century, starting from 1872 when Mouchot exhibited a steam-powered printing press at the Paris Exposition. The idea is to use concentrating collectors to produce and supply steam to heat engines. Much of the early attention to solar thermal–mechanical systems was for small scale applications (up to 100 kW) and most of them were designed for water pumping. Since 1975 there have been several large-scale power systems constructed and operated. Commercial plants of 30 and 80 MW electric (peak) generating capacity are nowadays in operation for more than a decade.

The process of conversion of solar to mechanical and electrical energy by thermal means is fundamentally similar to the traditional thermal processes. These systems differ from the ones considered so far as these operate at much higher temperatures. This section is concerned with generation of mechanical and electrical energy from solar energy by processes based mainly on concentrating collectors and heat engines.

The basic process for conversion of solar to mechanical energy is shown schematically in Fig.1. Energy is collected by concentrating collectors, stored (if appropriate), and used to operate a heat engine [14].

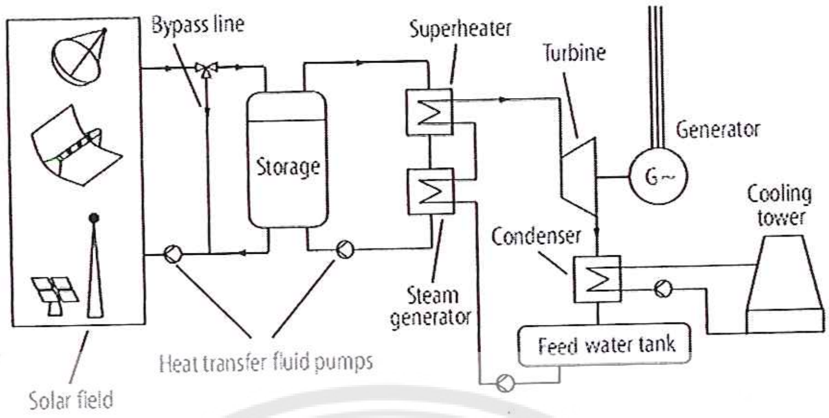


Figure 1 Principle of a solar thermal power plant

Source: A. Neumann, 2006

The main problem of these systems is that the efficiency of the collector is reduced as its operating temperature increases, whereas the efficiency of the heat engine increases as its operating temperature increases. The maximum operating temperature of stationary collectors is low relative to desirable input temperatures of heat engines, therefore concentrating collectors are used exclusively for such applications. Identifying the best available sites for the erection of solar thermal power plants is a basic issue of project development. Recently the planning tool STEPS was developed by the German Aerospace Centre (DLR), which uses satellite and Geographic Information System (GIS) data in order to select a suitable site. The factors taken into account are the slope of the terrain, land use (forest, desert, etc.), geo-morphological features, hydrographical features, the proximity to infrastructure (power lines, roads, etc.) and of course solar irradiation of the area. Three system architectures have been used for such applications, the PTC system, the power tower system, and the dish system.

1. Solar power tower

Solar power towers generate electric power from sunlight by focusing concentrated solar radiation on a tower-mounted heat exchanger (receiver). The system uses hundreds to thousands of sun-tracking mirrors called heliostats to reflect the incident sunlight onto the receiver. These plants are best suited for utility-scale applications in the 30 to 400 MW, range.

In a molten-salt solar power tower, liquid salt at 290 °C is pumped from a cold storage tank through the receiver where it is heated to 565 °C and then on to a hot storage tank for storage. When power is needed from the plant, hot salt is pumped to a steam generating system that produces superheated steam for a conventional Rankine-cycle turbine/generator system. From the steam generator, the salt is returned to the cold tank where it is stored and eventually reheated in the receiver. Figure 2 is a molten salt power tower system schematic diagram. [15]

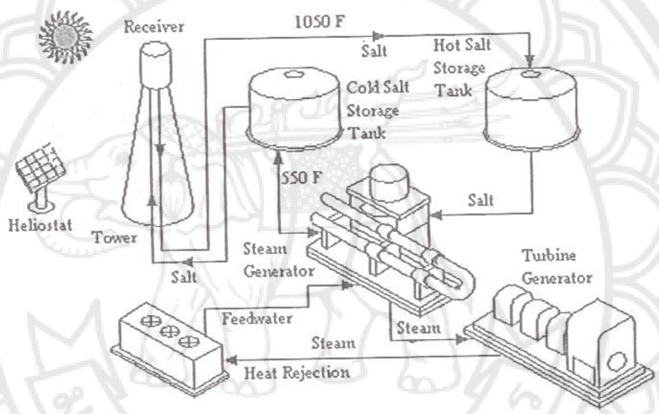


Figure 2 Molten salt power tower system schematic diagram

Source: Peter Heller, 1998

The largest solar power tower is the Solar One, a pilot demonstration plant was operated in Mojave Desert near Barstow, Southern California, USA from 1982 to 1988 with steam as the heat transfer medium. Rebuilt to the 10 MW_e Solar Two plant, it was successfully operated with a molten salt-in-tube receiver system and a two-tank molten salt storage system from 1997 to 1999 as seen in Figure 3. [16]

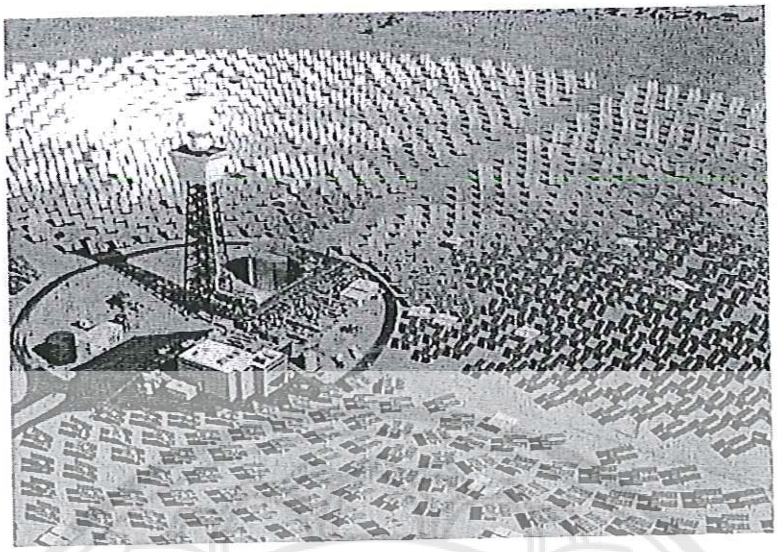


Figure 3 Solar two, the prototype power tower with 1926 mirrors

Source: Solar Energy Development (Solar Energy Development PEIS), 2009

Determining the optimum storage size to meet power dispatch requirements is an important part of the system design process. Storage tanks can be designed with sufficient capacity to power a turbine at full output for up to 13 hours. [15] The heliostat field that surrounds the tower is laid out to optimize the annual performance of the plant. The field and the receiver are also sized depending on the needs of the utility. In a typical installation, solar energy collection occurs at a rate that exceeds the maximum required for providing steam to the turbine. Consequently, the thermal storage system can be charged at the same time that the plant is producing power at full capacity. The ratio of the thermal power provided by the collector system (the heliostat field and receiver) to the peak thermal power required by the turbine generator is called the solar multiple. With a solar multiple of approximately 2.7, a molten-salt power tower, located in the California Mojave desert, can be designed for an annual capacity factor of about 65%. Consequently, a power tower could potentially operate for 65% of the year without the need for a back-up fuel source. Without energy storage solar thermal power technologies are limited to annual capacity factors near 25%. Because of the storage, power output from the turbine generator remains constant through fluctuations in solar intensity and until all of the energy stored in the hot tank is depleted. Energy storage and dispatch ability are very important for the success of solar power tower technology, and molten salt is believed to be the key cost effective energy storage. [15].

The Solar One thermal storage system stored heat from solar-produced steam in a tank filled with rocks and sand using oil as the HTF. The system extended the plant's power-generation capability into the night and provided heat for generating low-grade steam for keeping parts of the plant warm during off-hours and for morning startup. Unfortunately, the thermal storage system was inefficient and could not compensate for the power fluctuations caused by cloud cover. Using what was learned from Solar One, Solar Two became the next generation. Solar Two used slight modifications of the Solar One tower, heliostats, and turbine/generator. The major changes in the new design were a new molten-salt heat transfer system replacing the direct heating of water into steam and a new control system. Using the molten-salt heat transfer system required a new receiver, thermal storage, piping, and steam generator. Solar Two operates in the following manner as shown in Figure 4.

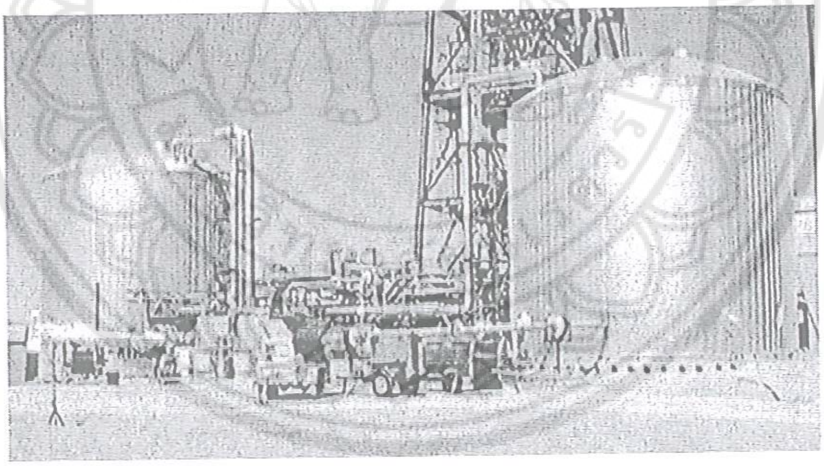


Figure 4 Molten salt storage tanks at Solar Two, California

Source: Rainer Tamme, 1998

2. **Dish/stirling technology**

Dish systems use parabolic reflectors in the shape of a dish to focus the sun's rays onto a dish-mounted receiver at its focal point as shown in Figure 5. In the receiver a HTF takes over the solar energy and transfers it to the power conversion system, which may be mounted in one unit together with the receiver (e.g. receiver/Stirling engine generator unit) or at the ground. Due to its ideal optical parabolic configuration and its two axes control for tracking the sun, dish collectors achieve the highest solar flux concentration, and therefore the highest performance of all concentrator types in terms of peak solar concentration and of system efficiency. These collector systems are restricted to unit capacities of some $10\,KW_e$ for geometrical and physical reasons. Parabolic dishes can reach 1000 $^{\circ}C$, have high optical efficiency and low start up losses, which make them the most efficient (29,4% record solar to electricity conversion) of all solar thermal power technologies. In addition, the modular design of dish/engine systems make them a good match for both remote power needs in the kilowatt range as well as hybrid end of the line grid-connected utility applications in the megawatt range.

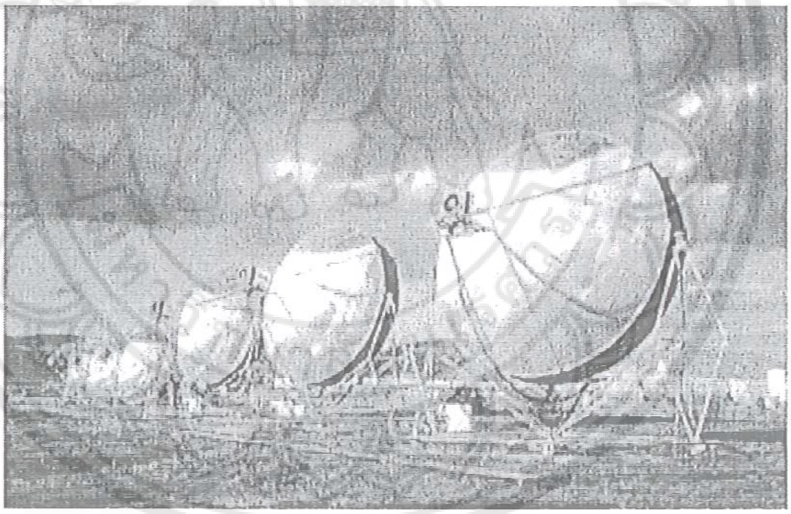


Figure 5 The six 9 kWe dish/stirling operation on the Plataforma solar de Almeria

Source: Peter Heller, 1998

Solar thermal power is a relatively new technology that has shown significant improvement since its inception about 20 years ago. Of the three mirror types, parabolic trough systems are at the commercial stage, whereas power tower and parabolic dish systems are still at the demonstration stage, although there are plans for commercial schemes in the near future.

3. Parabolic trough power plant

A parabolic trough system consists of a large number of single-axis tracking parabolic trough solar collectors. The solar field is composed of many parallel rows of solar collectors aligned on a north-south horizontal axis as seen in Figure 6 The reason for a North-South orientation is to create an optimal orientation for solar irradiation capture from sunrise to sunset. Each solar collector has a linear parabolic-shaped reflector that focuses the sun's direct beam radiation on a linear receiver located at the focus of the parabola. The collectors track the sun from east to west during the day to ensure that the sun is continuously focused on the linear receiver. A HTF is heated as it circulates through the receiver and returns to a series of heat exchangers in the power block where the fluid is used to generate high-pressure superheated steam. The superheated steam is then fed to a conventional reheat steam turbine/generator to produce electricity. After passing the generator, the steam is condensed in a standard condenser and returned to the heat exchangers via condense and feed water pumps to be transformed back into steam. Condenser cooling is provided by mechanical draft wet cooling towers. After passing through the HTF side of the solar heat exchangers, the cooled HTF is re-circulated through the solar field. The hot oil converts water to steam driving the steam turbine generator of a conventional power block. The superheated steam is then fed to a conventional reheat steam turbine/generator to produce electricity. After passing through the HTF side of the solar heat exchanger, the cooled HTF is re-circulated through the solar field.

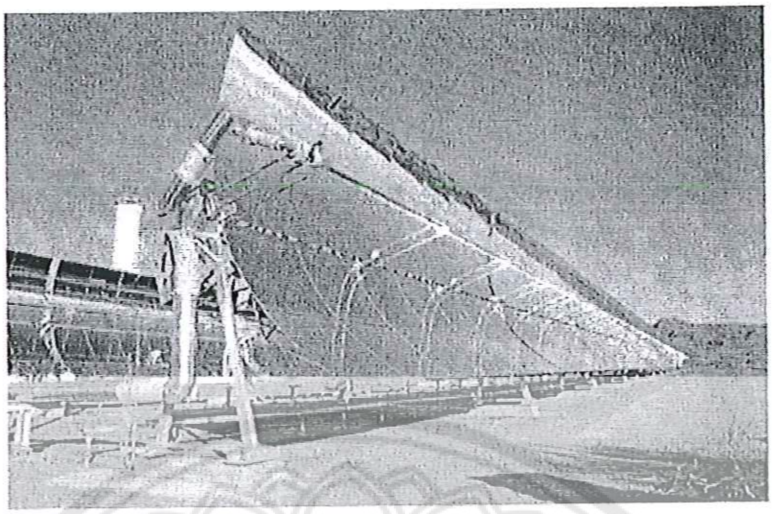


Figure 6 Principle of parabolic trough solar collector

Source: Wolf-Dieter Steinmann, Doerte Laing and Rainer Tamme, 2004

The power generation system consists of a conventional Rankine cycle reheat steam turbine with feed water heaters desecrators, etc. The condenser cooling water is cooled in forced draft cooling towers.

Although many solar technologies have been demonstrated, parabolic trough solar thermal electric power plant technology represents one of the major renewable energy success stories of the last two decades. Parabolic troughs are one of the lowest cost solar electric power options available today and have significant potential for further cost reduction.

However, use of solar energy presently poses technical problems, primarily because of inefficient collection and storage. Therefore, developing efficient and inexpensive energy storage devices is as important an endeavor as developing new sources of energy. The solar energy has certain positive and negative characteristics when applied to thermal processes. The major technical obstacle for solar thermal application is the intermittent nature of the source, both on daily and short time scale. The intermittent nature leads to a storage requirement, not present in non-solar systems. Terrestrial solar energy systems require either an energy storage system or an alternate source of energy to maintain the power supply, when there is insufficient or no sunlight. Solar energy is a time-dependent energy source. The demand of energy consumption follow their own time-dependent pattern and solar energy may not match this at all, hence another product of solar energy is necessary to match the demand. Energy storage can reduce the time or rate of mismatch between the energy supply and the energy demand, thereby playing a vital role in energy conservation. Non-storage processes waste energy but energy storage results in a saving of premium fuels. In addition, the imbalance which occurs between the availability of energy and the requirements for its use, which exists in virtually any system, can be balanced by using energy storage devices, with a consequent saving in capital costs. Energy storage improves the performance of energy systems by providing a consistent supply and increasing reliability. For example, storage would improve the performance of a power generating plant by load leveling. The higher efficiency would lead to energy conservation and improve cost effectiveness. Solar energy resource can only provide energy intermittently

Thermal Energy Storage system (TES)

The solar irradiance illuminating a fixed location on the Earth surface is subject to daily and yearly variations and fluctuations due to cloud coverage. When using this highly variable power source for supplying electric energy to humanity, a mismatch will occur due to the requested power production profile.

Conventional power plants operate as base load plants giving a constant power production over time or as backup plants which deliver the power on demand during high load conditions. Coal fired or nuclear power plants are candidates for base load plants. Gas turbines have a very short startup time and therefore can be used as backup sources. The solar input with its variability will neither fulfill the base load nor the power on demand operation. There are two ways to solve this problem. First, one can implement a power co-production like fossil co-firing in a solar plant. This means that during low solar irradiance periods (cloud transients or at night) a burner supplies the power. This implies that the burner must be able to operate efficiently in any part-load condition between 0-100%, which is a non-standard requirement. The second option is an energy storage system. Adding this type of storage system to a solar power plant is useful for increasing the efficiency of the power generation block because transients and part-load operation may be avoided and it can run more often under nominal conditions.

Thermal Energy Storage systems (TES) have the ability to store high or low-temperature energy for later use [16]. For example, the solar energy can be stored for overnight heating, the summer heat stored for winter use, etc. Thus, this system have potential applications in active and passive solar heating, water heating, air conditioning etc., and is regarded as an economical and safe energy storage technology. TES systems are attractive means for bridging the mismatch that usually exists between the availability of energy source and the demand for energy. For example, in solar applications, the sun is available only during the daylight hours or when it is not cloudy, but the demand for energy usually arises at other times, such as at night. Thus, they are useful in meeting the energy needs at all times [17].

Solar heat can be stored during the day in concrete, ceramics or phase change media. At night, it can be extracted from the storage to run the power block. Fossil and renewable fuels like oil, gas and organic waste can be used for co-firing the plant, providing power by demand, as base or peak load

A key issue in the design of a thermal energy storage system is its thermal capacity the amount of energy that it can store and provide. However selection of the appropriate system depends on many cost-benefit considerations.

The cost of a TES system mainly depends on the following items:

1. The storage material itself
2. The heat exchanger for charging and discharging the system
3. The cost for the space and/or enclosure for the TES

From the technical point of view, the crucial requirements are:

1. High energy density (per-unit mass or per-unit volume) in the storage material
2. Good heat transfer between heat transfer fluid (HTF) and the storage medium.
3. Mechanical and chemical stability of storage material
4. Compatibility between HTF, heat exchanger and/or storage medium
5. Complete reversibility for a large number of charging/discharging cycles
6. Thermal losses

7. Ease of control

1. Need of thermal energy storage [18].

Energy storage can even out mismatches between the demand and the supply of energy, thereby improving the system performance and reducing total cost. The type and extent of mismatch varies from system to system, influencing the type and size of storage (Figure 9.) Consider the following cases:

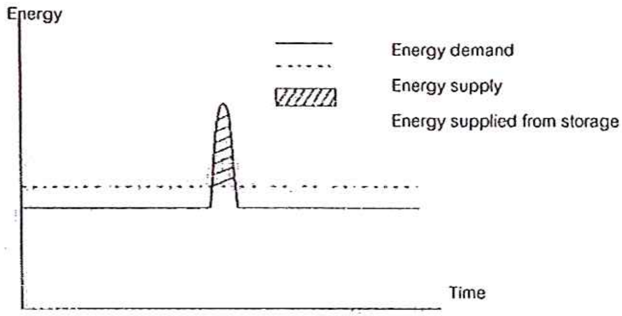
1.1 Sometimes the energy supply from the source may be constant, but there may be a sharp peak load of short duration as shown in Figure 1a. In this case the amount of energy needed to be stored is small. However, since the storage has to supply this energy in a very short time, the rate of energy transfer (the power) involved is high. If storage is not provided, a source with a much larger power rating would be required to meet this sharp demand, raising the total system cost considerably.

1.2 The energy supply may be a variable one, as in the case of solar energy, and the load may be constant as shown in Figure 1b. Since the energy supply is zero at night, considerable amount of energy must be stored during day time to meet the demand at night. The energy source must be sufficiently large to meet the load in the day time also to supply energy to the storage for the night.

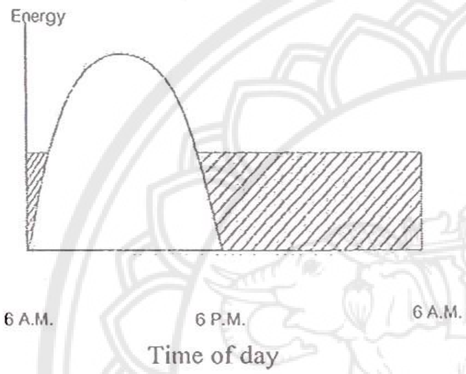
1.3 Both the supply and the demand may vary, and the supply may peak at a time quite distant from the peak demand, as in the case of solar space heating. Consider the extreme case where the space heating load occurs only at night, as shown in Figure 1c. In this case, the entire energy collected by solar collectors in the day time goes to the storage, and the demand at night is met from the storage, thus requiring an energy storage much larger than in case (2), for the same total diurnal load.

1.4 In another extreme case similar to case (3), where solar energy is available only in summer and the heat load is to be met only in the winter. In this case the energy collected during an entire season goes to the storage and is discharged from the storage in winter only and the time for which it must remain stored is very large. This type of storage is termed as seasonal or long term storage.

We have seen that apart from the amount of energy storage required, there are other characteristics of the mismatch, such as the time interval between the energy supply and demand, and the rate of energy withdrawal (or power) required to meet the demand.



(a)



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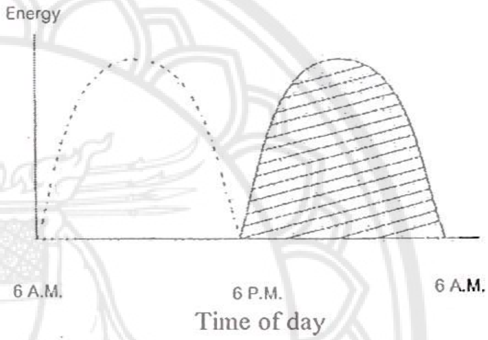

(c)

Figure 7 Mismatch in the energy supply and energy demand.

(a) Constant energy supply, sharp pulse (peak load) in energy Demand.

(b) Constant energy demand, variable energy supply lasting for only half of the diurnal (24 hour) cycle.

(c) Variable energy supply and demand, with a phase difference, the entire demand being met by storage of energy supply.

Source: H. P. Garg and J. Prakash, 1997

2. Size and duration of storage

The amount of energy storage provided is dictated, of course, by the cost. It may be possible, for example, to meet the energy demand during a week long spell of rain by energy stored from solar collectors. However, in such cases, a very large storage would be required and its utilization would be limited to a few such long spells of overcast days in a year. Investment on so large a storage would not be justified; it would be more economical to supply the energy demand from an auxiliary energy source (back-up unit). It should be borne in mind that storing up of large amounts of solar energy would be possible only by using a much larger solar collector (over sizing) to collect energy in surplus of the average demand. The investment on an oversized collector is not likely to be paid back.

The size of storage is related to the energy density or the amount of energy stored per unit mass (or per unit volume) of storage material. Larger energy density may offer two advantages: (1) the size of storage is reduced, and (2) lesser quantity of storage material and packaging material are required, thereby reducing costs. A smaller weight of storage has some advantages in both habitat and transports applications. The weight may not be very important in case of an underground storage or storage in the basement, but may become critical in case of roof-top storage.

An important characteristic of the storage material is its 'volumetric energy capacity', or the amount of energy stored per unit volume. The energy density may be expressed in Kilojoules per Kilogram (kJ/kg), Mega-joules per Kilogram (MJ/kg), or Kilowatt-hours per Kilogram (kWh/kg), while the volumetric energy density may be expressed per cubic meter as kJ/m^3, MJ/m^3, or kWh/m^3. Denser materials have smaller volumes, and correspondingly an advantage of larger energy capacity per unit volume. The space available is limited both in transport and in habitat applications. The volume occupied by the presently available storage systems is considerable, and may be an important factor in limiting the size of storage provided. The cost of floor space or volumetric space should be one of the parameters in optimizing the size of storage.

Duration of storage is another parameter to be taken into account while dealing with storage. If energy is converted into a fuel such as hydrogen, it can be stored almost indefinitely. However, if energy is stored as thermal energy, one has to ensure that the thermal losses during the length of time for which the energy is to be stored are within acceptable limits.

In case of solar thermal energy storage for space heating or cooling, the short term storage means storing the solar heat to meet the energy requirements for a 24 hour period, i.e. day-night period, or sometimes for a few days. Short term storage is a dynamic system that undergoes a daily charge/discharge cycle and therefore plays significant role in the performance of solar heating system. Long term storage, on the other hand, is to store sufficient heat to meet the building heating or cooling demand for a whole year, i.e. summer stored heat can be used for winter heating. Long term storage is called annual or seasonal storage in solar terminology.

The optimum size of solar thermal storage for space heating/cooling system depends on the type of application, and is a function of several parameters, such as type of material, storage temperature, storage heat losses, costs of the storage medium and container, type of heat exchanger and pumps, cost of auxiliary energy if any, climatic data such as solar radiation, ambient temperature, wind speed, sky conditions etc., heating or cooling load, collector type, its area and efficiency, solar fraction of the total heat load, thermo-physical properties of the storage materials etc.

3. Techniques for thermal energy storage

A thermal energy storage system consists of mainly three components: 1) the storage medium, 2) energy transfer mechanism and 3) containment system. The energy storage medium stores the thermal energy either in the form of sensible heat or in the form of latent heat of fusion or vaporization or in hybrid system or in the form of chemical reversible reactions. The purpose of the energy transfer mechanism is to supply (charge) or to extract (discharge) heat from the storage medium efficiently. The containment system holds the storage medium and the energy transfer equipment and insulates the system from the surroundings. The containment may be pressurized or non pressurized type. Sometimes the heat exchanger carrying the heat transfer fluid is under pressure and the containment works without pressure. The performance of thermal energy storage depends on many sub-systems such as solar collector, heat exchanger, input and output loads, auxiliary energy supply, temperatures of operation, and the control mechanism. The parameters required for thermal storage design are storage concept definition, storage material used, degree of reliability needed, the nature of the loads expected and the manner the heat is supplied and extracted from the storage medium.

All the three forms of thermal energy storage are sensible, latent and chemical are used to store the heat at high temperature. An overview of major technique of storage of solar thermal energy is shown in Figure 12. The requirements of a good storage are low cost, high heat capacity, good thermal stability and easy availability, low vapors pressure at operating temperature, safe, easy handling and compatible with containment material. Water is the most suited material for thermal energy storage but requires high pressure tank for high temperature storage and therefore not preferred.

Thermal energy can be stored in well insulated fluids or solids. It is generally stored either as:

1. Sensible heat storage

In the sensible heat storage, energy is stored by changing the temperature of a storage medium. The thermal property of material that is relevant here is the specific heat capacity of the material. Examples of this type of storage material are water, air, rock, sand or soil etc. Water, in particular is a favorable candidate because of its high specific heat (4.186 KJ/Kg °C). Water is used in solar energy applications where solar energy is collected during the day to heat water which can store more energy than other materials without overheating. At night, the warm water may be circulated for home heating.

Sensible heat Q, is stored in material of mass m and specific heat, c_p by raising the temperature of the storage material from T_1 to T_2 and is expressed by equation

$$Q = \int_{T_1}^{T_2} mc_p dT$$

Or

$$Q = \int_{T_1}^{T_2} \rho V c_p dt \qquad \text{[Eq.1]}$$

Where ρ and V are density and volume of the storage material, respectively. For moderate temperature change, such as for solar space and water heating system, the density and specific heat may be considered constants. Therefore $Q=\rho V c_p \Delta T$.

2. Latent heat storage

In this type of heat storage, energy is stored as latent heat in suitable substances during a phase change, usually, from a solid to a liquid phase at a desired temperature [16, 17]. The energy that is absorbed during the melting (solid --> liquid) process is stored as "latent heat of fusion" and is released during the freezing (liquid --> solid) process. Hence, these substances are also known as phase change materials or heat-of-fusion materials. Typical examples of such materials are: water/ice, organic compounds such as, paraffin, fatty acids, salt hydrates such as, Glauber Salt ($Na_2SO_4.10H_2O$), certain polymers etc. The thermal characteristics that are relevant here are the latent heat of melting/fusion, and the melting point of the material. The latent heat storage materials have high energy storage capacities than sensible storage materials. [Glauber salt: 250 kJ/kg, paraffin wax 200 kJ/kg, fatty acids > 180 kJ/kg] [16,18]. The energy is delivered over a narrow temperature range making these materials attractive for smaller and lighter storage devices and lower storage losses.

Thermal energy can stored as latent heat in a material that undergoes phase transformation at a temperature that is useful for the application. If a material with phase change temperature T_m is heated from T_1 to T_2 such that $T_1 < T_m < T_2$ the thermal energy Q stored in a mass m of the material is given by equation

$$Q = \int_{T_1}^{T_m} mc_p dT + m\lambda + \int_{T_m}^{T_2} mc_p dT \qquad \text{[Eq.2]}$$

Where λ is heat of phase transformation.

Four type of phase transformations useful for latent heat storage are: solid - liquid , liquid - vapor, solid - vapor, and solid - solid . Since phase transformation is an isothermal process, thermal energy is stored and retrieved at a fixed temperature known as the transition temperature. Some common phase change materials (PCM) used for thermal storage are paraffin waxes, non-paraffin, inorganic salts (both anhydrous and hydrated) and eutectics of organic and/or inorganic compounds.

Most common PCM used for solar energy storage undergo solid $\leftrightarrow$ liquid transformation. For such material, the thermal energy stored may be written from equation as, approximately,

$$Q = m[\bar{c}_{ps}(T_m - T_1) + \lambda + \bar{c}_{pl}(T_2 - T_m)] \qquad \text{[Eq.3]}$$

Where $\bar{c}_{ps}$ and $\bar{c}_{pl}$ are the average specific heats in the solid and liquid phases, respectively.

3. Thermochemical Energy Storage

Thermochemical energy can be stored as heat of reaction in reversible chemical reaction. In this mode of storage, the reaction in the forward direction is endothermic(storage of heat),while the reverse action is exothermic (releaser of heat)

$$A + \Delta H \leftrightarrow B + C$$

The amount of heat Q stored in a chemical reaction depends on the heat of reaction and the extent of conversion as given by equation

$$Q = a_r m \Delta H \qquad \text{[Eq.4]}$$

where a_r = fraction reaction, ΔH = heat of reaction per unit mass and m = mass

Chemical reaction is generally a highly energetic process. Therefore, a large amount of heat can be stored in a small quantity of a material. Another advantage of thermochemical storage is that the products of reaction can be stored at room temperature and need not be insulated. [19]

In the first type of the storage, the temperature of the medium changes during charging or discharging of the storage, whereas in the second type the temperature of the medium remains more or less constant since it undergoes a phase transformation.

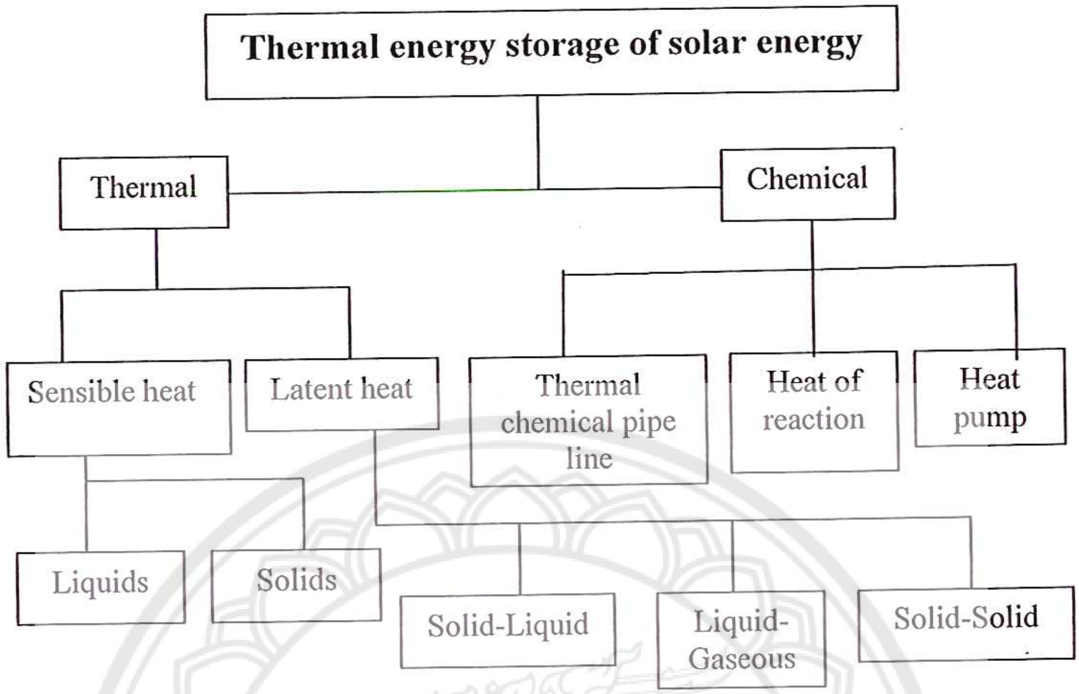

Figure 8 Storage of solar thermal energy

Source: H. P. Garg and J. Prakash, 1997

There are four main items affecting the cost of a thermal storage:

1. the thermal heat storage material
2. the insulating packaging
3. the space occupied by the storage device
4. the heat exchanger for charging and discharging the storage

Desired characteristics of the thermal storage

1. Compact, large storage capacity per unit mass and volume.
2. High storage efficiency.
3. Heat storage medium with suitable properties in the operating temperature range.
4. Uniform temperature
5. Capacity to charge and discharge with the largest heat input/ output rates but without temperature gradients.
6. Complete reversibility.

7. Ability to undergo large number of charging/discharging cycles without loss of performance and storage capacity.

8. Small self-discharging rate i.e. negligible heat losses to the surroundings.

9. High speed of charging and discharging.

10. Long life.

11. Inexpensive.

12. Should not be corrosive.

13. No fire and toxicity hazard.

In small heat storage, the surface area to volume ratio is large and therefore, the cost of insulating material is an important factor. Phase change storages with higher energy densities are more attractive for small storage. In larger heat storage, on the other hand, the cost of storage material is more important and sensible heat storage is very attractive . Thermal power plants are generally based on a Rankine Cycle with steam as the working fluid.

4. Storage Concepts

Storage concepts can be classified as active or passive systems. Active storage is mainly characterized by forced convection heat transfer into the storage material. The storage medium itself circulates through a heat exchanger. This heat exchanger can also be a solar receiver or a steam generator.

The main characteristic of a passive system is that a heat transfer medium passes through storage only for charging and discharging. The heat transfer medium itself does not circulate.

4.1 Active Thermal Energy storage

Active thermal systems typically utilize tank storage. They can be designed as one tank or two tank systems.

Active storage is again subdivided into direct and indirect systems. In a direct system the heat transfer fluid, which collects the solar heat, serves also as the storage medium, while in an indirect system, a second medium is used for storing the heat.

Two prominent examples of two-tank systems for solar electric applications are the storage systems of the SEGS I and Solar Two plants Figure 9. shows a Schematic of a concentrated solar thermal trough power plant with thermal storage. An initial experience with a small-scale two-tank molten salt system has already been described [20]

A two-tank system (Figure 9.) uses one tank for cold HTF coming from the steam generator and one tank for the hot HTF coming directly out of the solar receiver before it is fed to the steam generator. The advantage of this system is that cold and hot HTF are stored separately. The main disadvantage is the need for a second tank. In this type of system, the storage tanks are directly coupled to the HTF pressure levels (which is not necessarily a disadvantage).

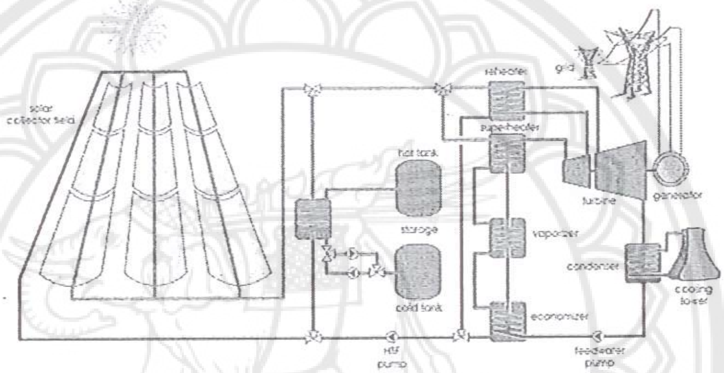

Figure 9 Schematic of a concentrated solar thermal trough power plant with thermal storage

Source: Pilkington Solar International, 2000

The single-tank system reduces storage volume and cost by eliminating a second tank. However, in a single-tank system it is more difficult to separate the hot and cold HTF. Because of the density difference between hot and cold fluid, the HTF naturally stratifies in the tank, from coolest layers at the bottom to warmest layers at the top. These systems are called thermocline storage (Figure 9). Experience with thermocline storage was described by [20]. Maintaining the thermal stratification requires a controlled charging and discharging procedure, and appropriate methods or devices to avoid mixing. Filling the storage tank with a second solid storage material (rock, iron, sand etc.) can help to achieve the stratification.

4.2 Passive Thermal Energy Storage

Passive systems are generally dual medium storage systems. The HTF carries energy received from the energy source to the storage medium during charging and receives energy from the storage material when discharging. These systems are also called regenerators.

The storage medium can be a solid, liquid, or PCM. In general, a chemical storage system employs at least two media.

The main disadvantage of regenerators is that the HTF temperature decreases during discharging as the storage material cools down. Another problem is the internal heat transfer. Especially for solid materials, the heat transfer is rather low, and there is usually no direct contact between the HTF and the storage material as the heat is transferred via a heat exchanger.

5. Storage Media [20]

Thermal storage can utilize sensible or latent heat mechanisms or heat coming from chemical reactions.

Sensible heat is the means of storing energy by increasing the temperature of a solid or liquid. Latent heat, on the other hand, is the means of storing energy via the heat of transition from a solid to liquid state. For example, molten salt has more energy per unit mass than solid salt.

Table 1 shows the characteristics of candidate solid and liquid sensible heat storage materials and potential phase change (latent) heat storage media for a SEGS plant. For each material, the low and high temperature limits are given these limits, combined with the average mass density and heat capacity, lead to a volume-specific heat capacity in kWh_t per cubic meter. The table also presents the approximate costs of the storage media in dollars per kilogram, finally arriving at unit costs in $\$/kWh_t$.

The average thermal (heat) conductivity given in the table has a strong influence on the heat transfer design and heat transfer surface requirements of the storage system, particularly for solid media (high conductivity is preferable). High volumetric heat capacity is desirable because it leads to lower storage system size, reducing external piping and structural costs. Low unit costs obviously mean lower overall costs for a given thermal capacity.

reducing external piping and structural costs. Low unit costs obviously mean lower overall costs for a given thermal capacity.

5.1 Sensible Heat Storage media

Thermal energy can be stored in the sensible heat (temperature change) of substances that experience a change in internal energy. The stored energy is calculated by the product of its mass, the average specific heat, and the temperature change. Besides the density and the specific heat of the storage material, other properties are important for sensible heat storage: operational temperatures, thermal conductivity and diffusivity, vapor pressure, compatibility among materials, stability, heat loss coefficient as a function of the surface areas to volume ratio, and cost.

5.1.1 Solid Media

For thermal storage, solid media usually are used in packed beds, requiring a fluid to exchange heat. When the fluid heat capacity is very low (e.g., when using air) the solid is the only storage material; but when the fluid is a liquid, its capacity is not negligible, and the system is called a dual storage system. Packed beds favor thermal stratification, which has advantages. Stored energy can easily be extracted from the warmer strata, and cold fluid can be taken from the colder strata and fed into the collector field.

An advantage of a dual system is the use of inexpensive solids such as rock, sand, or concrete for storage materials in conjunction with more expensive heat transfer fluids like thermal oil. However, pressure drop and, thus, parasitic energy consumption may be high in a dual system. This has to be considered in the storage design.

Table 1 Candidate Storage Media for SEGS Plants

Storage Medium	Temperature Cold	Hot (˚C)	Average density (ρ) (kg/m^3)	Average heat conductivity (W/mK)	Average heat capacity (C_p) (kJ/kgK)	Volume specific heat capacity (kWh_t/m)	Media costs /kWh_t ($/$kWh_t$)
Solid media							
Sand-rock-mineral oil	200	300	1,700	1.0	1.3	60	4.2
Reinforced concrete	200	400	2,200	1.5	0.85	100	1.0
NaCl (solid)	200	500	2,160	7.0	0.85	150	1.5
Cast iron	200	400	7,200	37.0	0.56	160	32.0
Cast steel	200	700	7,800	40.0	0.60	450	60.0
Silica fire bricks	200	700	1,820	1.5	1.00	150	7.0
Magnesia fire bricks	200	1,200	3,000	5.0	1.15	600	6.0
Liquid media							
Mineral oil	200	300	770	0.12	2.6	55	4.2
Synthetic oil	250	350	900	0.11	2.3	57	43.0
Silicone oil	300	400	900	0.10	2.1	52	80.0
Nitrite salts	250	450	1,825	0.57	1.5	152	12.0
Nitrate salts	265	565	1,870	0.52	1.6	250	5.2
Carbonate salts	450	850	2,100	2.0	1.8	430	11.0
Liquid sodium	270	530	850	71.0	1.3	80	21.0

Table 1 (**cont.**)

Storage Medium	Temperature Cold Hot (˚C)	Average density (ρ) (kg/m^3)	Average heat conducti-vity (W/mK)	Average heat capacity (C_p) (kJ/kgK)	Volume specific heat capacity (kWh$_t$/m)	Media costs /kWh$_t$ ($/kWh$_t$)
			Phase change media			
$NaNO_3$	308	2,257	0.5	200	125	3.6
KNO_3	333	2,110	0.5	267	156	4.1
KOH	380	2,044	0.5	150	85	24.0
Salt-ceramics	500-850	2,600	0.5	420	300	17.0
NaCl	802	2,160	5.0	520	280	1.2
Na_2CO_3	854	2,533	2.0	276	194	2.6
K_2CO_3	897	2,290	2.0	236	150	9.1

Source: Pilkington Solar International, 2000

The cold-to-hot temperature limits of some solid media in Table 2 are greater than could be utilized in a SEGS plant because parabolic trough solar fields are limited to maximum outlet temperatures of about 400 ˚C. Table 2 shows the effect on solid media by imposing this temperature limit on the storage medium temperature range, the unit heat capacities, and media costs.

Table 2 Solid Storage Media for SEGS Plants

Storage Medium	Heat Capacity (kWh_t / m^3)	Media Cost ($\$ / kWh_t$)
Reinforced concrete	100	1
NaCl (solid)	100	2
Cast iron	160	32
Cast steel	180	150
Silica fire bricks	60	18
Magnesia fire bricks	120	30

Source: Pilkington Solar International, 2000

Using these values and judging the options against the guidelines discussed above, the sand-rock-oil combination is eliminated because it is limited to 300 °C. Reinforced concrete and salt have low cost and acceptable heat capacity but very low thermal conductivity. Silica and magnesia fire bricks, usually identified with high temperature thermal storage, offer no advantages over concrete and salt at these lower temperatures. Cast steel is too expensive, but cast iron offers a very high heat capacity and thermal conductivity at moderate cost.

5.1.2 Liquid Media

Liquid media maintain natural thermal stratification because of density differences between hot and cold fluid. To use this characteristic requires that the hot fluid be supplied to the upper part of a storage system during charging and the cold fluid be extracted from the bottom part during discharging, or using another mechanism to ensure that the fluid enters the storage at the appropriate level in accordance with its temperature (density) in order to avoid mixing. This can be done by some stratification devices (floating entry, mantle heat exchange, etc.).

The heat transfer fluid in a SEGS plant operates between the temperatures of 300 °C and 400 °C, approximately. Applying these limitations on temperature, and dropping mineral oil because it cannot operate at the upper temperature requirement gives the results shown in Table 3.

Table 3 Liquid Storage Media for SEGS Plants

Storage Medium	Heat Capacity (kWh_t / m^3)	Media Cost $(\$/kWh_t)$
Synthetic oil	57	43
Silicone oil	52	80
Nitrite salts	76	24
Nitrate salts	83	16
Carbonate salts	108	44
Liquid sodium	31	55

Source: Pilkington Solar International, 2000

Both the oils and salts are feasible. The salts, however, generally have a higher melting point and parasitic heating is required to keep them liquid at night, during low insulation periods, or during plant shutdowns. Silicone oil is quite expensive, though it does have environmental benefits because it is a non-hazardous material, whereas synthetic oils may be classified as hazardous materials. Nitrites in salts present potential corrosion problems, though these are probably acceptable at the temperatures required here. (The U.S. Solar Two project has selected a eutectic of nitrate salts because of the corrosively of nitrite salts at central receiver system temperature levels.)

5.2 Latent Heat Storage media

Thermal energy can be stored nearly isothermally in some substances as the latent heat of phase change, that is, as heat of fusion (solid-liquid transition), heat of vaporization (liquid-vapor), or heat of solid-solid crystalline phase transformation. All substances with these characteristics are called phase change materials (PCM). Because the latent heat of fusion between the liquid and solid states of materials is rather high compared to the sensible heat, storage systems utilizing PCM can be reduced in size compared to single-phase sensible heating systems. However, heat transfer design and media selection are more difficult, and experience with low-temperature salts has shown that the performance of the materials can degrade after a moderate number of freeze-melt cycles. LUZ International Ltd. proposed evaluation of an innovative phase-change salt concept to the solar community that used a series of salts in a "cascade" design (to be discussed later).

Table 1 showed, for a number of potential salts, the temperature at which the phase change takes place as well as the heat capacity (heat of fusion). Data for the salts shown in that table that are applicable to SEGS plants are shown in Table 4 below. It can be seen that the heat capacities, at least for the nitrites, are high and unit costs are comparatively low.

5.3 Chemical Storage media

A third storage mechanism is by means of chemical reactions. For this type of storage it is necessary that the chemical reactions involved are completely reversible. The heat produced by the solar receiver is used to excite an endothermic chemical reaction. If this reaction is completely reversible the heat can be recovered completely by the reversed reaction. Often catalysts are necessary to release the heat. This is even more advantageous as the reaction can then be controlled by the catalyst.

Table 4 Latent Heat Storage Media for SEGS Plants

Storage Medium	Heat Capacity (kWh_t/m^3)	Media Cost ($\$/kWh_t$)
$NaNO_3$	125	4
KNO_3	156	4
KOH	85	24

Source: Pilkington Solar International, 2000

Commonly cited advantages of TES in a reversible thermochemical reaction (RTR) are high storage energy densities, indefinitely long storage duration at near ambient temperature, and heat-pumping capability. Drawbacks may include complexity, uncertainties in the thermodynamic properties of the reaction components and of the reaction ís kinetics under the wide range of operating conditions, high cost, toxicity, and flammability.

Although RTRs have several advantages concerning their thermodynamic characteristics, development is at a very early stage. To date, no viable prototype plant has been built.

Heat transfer of storage system

1. hermal conduction

The temperature distribution existing within a material can at most depend on three space variables and on time. If indeed the temperature is a function of time, the problem is called unsteady and the temperature distribution is referred to as being transient. If the temperature is not a function of time, the problem is referred to as steady or the temperature distribution as being in a steady state. If temperature depends only on a single space coordinate, the problem, or the temperature distribution, is referred to as one dimensional. When temperature depends on two or three space variables, the problem is referred to as a two or three dimensional problem, respectively. In a one-dimensional, unsteady problem, temperature is a function of one space variable and of time.

In our analysis will assume that thermal conductivity is constant. Even though thermal conductivity for most materials does vary with temperature, the dependence in a majority of cases is not a strong one. A material in which thermal conductivity does not vary with direction is call isotropic. On the other hand, a fibrous material such as wood will have a thermal conductivity in the grain direction that is different from that measured across the grain. Thus wood is anisotropic.

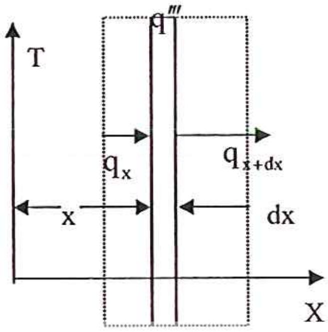

Figure 10 A one-dimensional system in rectangular coordinates

Source: William S. Janna, 1986

Let us initially consider a one-dimensional coordinate system, as shown in Figure 10. The system consists of a plane wall on which we impose between of the T and x axis. A slice of the material dx thick is selected for study. The fact energy can be generated within the material is taken into account by the term q'''. Typically, internal heat generation within a solid passing through the material, or by a nuclear reaction. Some of the energy passing through the control volume may be stored, thus increasing the internal energy of the material, which is sensed physically as an increase in the temperature of the material. This is the case for an unsteady problem.

For any geometry we can write

Rate of energy conducted into control volume	+	Rate of energy generated inside control volume	=	Rate of energy conducted out of control volume	+	Rate of energy stored inside control volume

For the system of Figure 14 this equation becomes

$$q_x + q'''Adx = q_{x+dx} + \rho Adx\frac{\partial u}{\partial t} \qquad \text{[Eq.5]}$$

Where A is the cross-section area, Adx is the volume of the element, q''' is the internal heat generated per unit volume, ρ is density, ρAdx is mass, and $\frac{\partial u}{\partial t}$ represent the rate of change in internal energy per unit mass of the control volume. The rate of energy conducted into the control volume is q_x.

The rate of energy conduced out of control volume becomes

$$q_{x+dx} = q_x + dq_x$$

where denote a change as dq_x . Alternatively, could write

$$q_{x+dx} = q_x + \frac{dq_x}{dx}dx \qquad \text{[Eq.6]}$$

to denote more generally that the change is a function of the x coordinate. If it is anticipated that q_x will be a function of more than one variable, the following partial derivation notation is appropriate:

$$q_{x+dx} = q_x + \frac{dq_x}{dx}dx \qquad \text{[Eq.7]}$$

Another way of obtaining this expression is to expand q_{x+dx} in a Taylor series and write only the first two terms. Of the various expressions written above for q_{x+dx} , Equation 7 is form to be used here. Combining with Equation (1) gives

$$q_x + q'''Adx = q_x + \frac{\partial q_x}{\partial x}dx + \rho Adx\frac{\partial u}{\partial t}$$

so that

$$-\frac{\partial q_x \partial x}{\partial x} + q'''Adx = \rho Adx\frac{\partial u}{\partial t} \qquad \text{[Eq.8]}$$

This is the general energy equation for unsteady heat conduction in one dimension with internal heat generation. It is desirable to substitute something more easily measured for the various terms in this equation. Fourier's Equation. Fourier's Equation is used for the heat conducted:

$$q_x = -KA\frac{\partial T}{\partial x} \qquad \text{[Eq.9]}$$

where k is thermal conductivity, T is temperature, and because we anticipate that temperature will be a function of more than one variable, the partial derivative notation is used.

Conduction is the principal mode of energy transfer in solids; the above equation reduces to

$$du = dh \qquad \text{[Eq.10]}$$

Substituting from the definition of specific heat, obtain

$$c_v dT = c_p dT$$

or

$$c_v = c_p = c$$

where c_v is the specific heat at constant volume and c_p is the specific heat at constant pressure. They are equal in a solid. So the internal energy change can be written as

$$\frac{\partial u}{\partial t} = c\frac{\partial T}{\partial t} \qquad \text{[Eq.11]}$$

combining Equation 9 and 10 with 11 yields

$$-\frac{\partial}{\partial x}(-kA\frac{\partial T}{\partial x})dx + q'''Adx = (\rho Adx)c\frac{\partial T}{\partial t}$$

Dividing by Adx and simplifying, we obtain

$$\frac{\partial}{\partial x}(k\frac{\partial T}{\partial x})+q'''=\rho c\frac{\partial T}{\partial t} \quad \text{[Eq.12]}$$

For constant thermal conductivity the above becomes

$$\frac{\partial^2 T}{\partial x^2}+\frac{q'''}{k}=\frac{\rho c}{k}\frac{\partial T}{\partial t}=\frac{1}{\alpha}\frac{\partial T}{\partial t} \quad \text{[Eq.13]}$$

which is the general energy equation for unsteady heat conduction in one dimension with internal heat generation, written in terms of temperature. The term $\alpha=\frac{k}{\rho c}$ is introduced as the thermal diffusivity of the material, with dimensions of L^2/T.

2. Convection Heat Transfer

Convection is the mode of heat transfer associated with fluid motion. If the fluid motion is due to an external motive source such as a fan or pump, the term forced convection applies. On the other hand, if fluid motion is due predominantly to the presence of a thermally induced density gradient, then the term natural convection is appropriate.

Consider a plate immersed in a uniform flow as shown in Figure 11. The plate is heated and has a uniform wall temperature of T_w. The uniform flow velocity is U_∞, and the fluid temperature far from the plate is T_∞. At any location we sketch the velocity distribution on the axes labeled V vs y. The velocity at the wall is zero due to the nonslip condition ; that is , fluid adheres to the wall due to friction or viscous effects. The velocity increases with increasing y from zero at the wall to nearly the free-stream value at some vertical distance away, known as the hydrodynamic boundary layer. Also sketch a temperature distribution, T vs y, which is seen to decrease from T_w at the wall to T_∞ at some distance from the wall (assuming $T_\infty < T_w$).

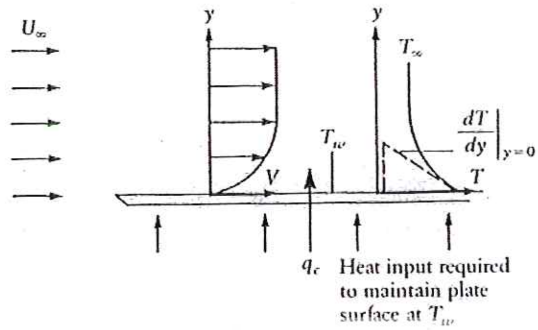

Figure 11 Uniform flow past a heated plate

Source: William S. Janna, 1986

Heat is transferred from the wall to the fluid. Within the fluid the mechanism of heat transfer at the wall is conduction because the fluid velocity at the wall is zero. However, the rate of heat transfer depends on the slope of the T vs y curve at the wall $-dT/dy$ at $y=0$. A steeper slope is indicative of a greater temperature difference and is highly dependent on the flow velocity. The flow velocity will influence the distance from the wall that we must travel before sense that the temperature is T_∞

The heat transferred by convection is found to be proportional to the temperature difference. In the case of Figure 11

$$\frac{q_c}{A} \propto T_w - T_\infty$$

Introducing proportionality constant, get

$$q_c = \bar{h}_c A(T_w - T_\infty) \qquad \text{[Eq.14]}$$

In which $\bar{h}_c$ is called the average convection eat transfer coefficient or the film conductance. This coefficient accounts for the overall effects embodied in the process of convection heat transfer.

3. The Overall Heat – Transfer Coefficient [21]

In many problem having a polar cylindrical geometry, heat transfer by convection is quite common. Examples include steam flowing through an insulated pipe, cold Freon flowing through a copper tube, and heated crude oil flowing up from underground at an oil well. It is therefore convenient to include convective effects in the one-dimensional conduction problem for cylinders.

Figure 12. is a sketch of a pipe or tube containing a fluid at temperature $T_{\infty 1}$. Heat is transferred to the pipe by convection, through the pipe wall by conduction, then to the fluid outside, which is at temperature $T_{\infty 2}$. Also shown in the figure is the thermal circuit. From previous discussion can write equations for each resistance as

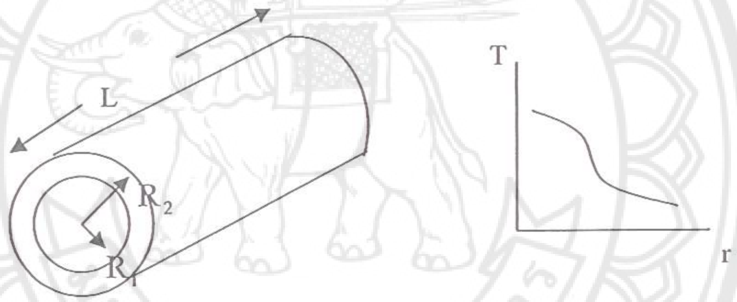

Figure 12 Heat flow through a cylinder with convection

Source: William S. Janna, 1986

$$R_{c1} = \frac{1}{\bar{h}_{c1}A_1} \qquad \text{[Eq.15]}$$

$$R_k = \frac{\ln(R_2/R_1)}{2\pi kL} \qquad \text{[Eq.16]}$$

$$R_{c1} = \frac{1}{\bar{h}_{c2}A_2} \qquad \text{[Eq.17]}$$

In these equations the convection coefficient, assumed constant, on the inside and outside surfaces are $\bar{h}_{c1}$ and $\bar{h}_{c2}$, respectively. The areas for heat transfer on the inside and outside surfaces are

$$A_1 = 2\pi R_1 L \quad \text{and} \quad A_2 = 2\pi R_2 L \qquad \text{[Eq.18]}$$

The heat flow from fluid to fluid, based on the overall temperature difference, is

$$q_r = \frac{T_{\infty 1} - T_{\infty 2}}{\frac{1}{\bar{h}_{c1} 2\pi R_1 L} + \frac{\ln(R_2/R_1)}{2\pi k L} + \frac{1}{\bar{h}_{c2} 2\pi R_2 L}} \qquad \text{[Eq.19]}$$

or

$$q_r = \frac{2\pi L(T_{\infty 1} - T_{\infty 2})}{\frac{1}{\bar{h}_{c1} R_1} + \frac{\ln(R_2/R_1)}{k} + \frac{1}{\bar{h}_{c2} R_2}} \qquad \text{[Eq.20]}$$

In some problem it is desirable to express this equation in terms of an overall heat-transfer coefficient that accounts for the combined effects of convection at both surfaces and conduction:

$$q_r = UA(T_{\infty 1} - T_{\infty 2}) \qquad \text{[Eq.21]}$$

Where U is an overall heat-transfer coefficient and A is some area on which to base U. The question arises as to which area to select. It is customary practice to select either the outside surface area or the inside surface area of the cylinder. Based on A_2, U_2 can be determined by equation 20 and 21

$$U_2(2\pi R_2 L)(T_{\infty 1} - T_{\infty 2}) = \frac{2\pi L(T_{\infty 1} - T_{\infty 2})}{\frac{1}{\bar{h}_{c1}R_1} + \frac{\ln(R_2/R_1)}{k} + \frac{1}{\bar{h}_{c2}R_2}} \quad \text{[Eq.22]}$$

Solving ,

$$U_2 = \frac{1}{\frac{R_2}{\bar{h}_{c1}R_1} + \frac{R_2\ln(R_2/R_1)}{k} + \frac{1}{\bar{h}_{c2}}} \quad \text{[Eq.23]}$$

$$U_1 = \frac{1}{\frac{1}{\bar{h}_{c1}} + \frac{R_1\ln(R_2/R_1)}{k} + \frac{R_1}{\bar{h}_{c2}R_2}} \quad \text{[Eq.24]}$$

As seen in Equation 23 and 24 the overall coefficient is independent of length.

4. Radial Systems of Cylinders [22]

Consider a long cylinder of inside radius r_i, outside radius r_0 and length L such as the one shown in Figure 13.

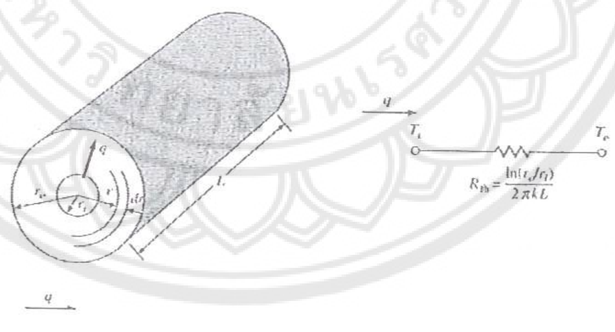

Figure 13 One-dimensional heat flow through a hollow cylinder and electrical analog

Source: J.P Holman, 1997

We expose this cylinder to a temperature differential $T_i - T_0$ and ask what the heat flow will be. For a cylinder with length very large compared to diameter, it may be assumed that the heat flows only in a radial direction, so that the only space coordinate needed to specify the system is r. Again, Fourier's law is used by inserting the proper area relation. The area for heat flow in the cylindrical system is

$$A_r = 2\pi rL$$

So that Fourier's law is written

$$q_r = -kA_r \frac{dT}{dr} \qquad \text{[Eq.25]}$$

or

$$q_r = -2\pi krL \frac{dT}{dr} \qquad \text{[Eq.26]}$$

with the boundary conditions

$$T = T_i \quad \text{at } r = r_i$$

$$T = T_0 \quad \text{at } r = r_0$$

The solution to Eq. 26 is

$$q = \frac{2\pi kL(T_i - T_0)}{\ln(r_0 / r_i)} \qquad \text{[Eq.27]}$$

and the thermal resistance in this case is

$$R_{th} = \frac{\ln(r_0 / r_i)}{2\pi kL} \qquad \text{[Eq.28]}$$

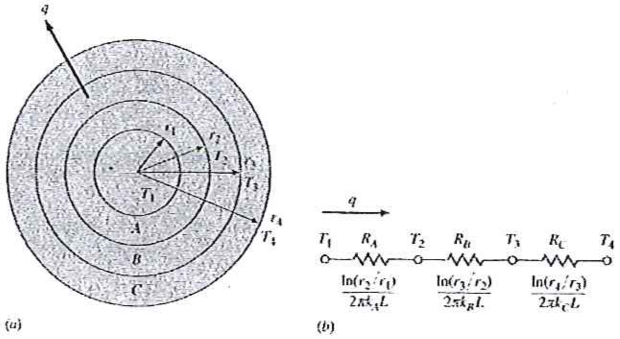

Figure 14 One-dimensional heat flow through multiple cylindrical section and electrical analog

Source: J.P. Holman, 1997

The thermal- resistance concept may be used for multiple-layer cylindrical walls just as it was used for plane walls. For the three-layer system shown in Figure 14 the solution is

$$q = \frac{2\pi L(T_1 - T_4)}{\ln(r_2/r_1)/k_A + \ln(r_3/r_2)/k_B + \ln(r_4/r_3)/k_C} \quad \text{[Eq.29]}$$

5. Laminar and turbulent flow in tubes

Flow in the tube can be laminar or turbulent, depending on the flow conditions Fluid flow is streamline and thus laminar at low velocities, but turns turbulent as the velocity is increased beyond a critical value. Transition from laminar to turbulent flow does not occur suddenly, rather, it occurs over some range of velocity where the flow fluctuates between laminar and turbulent flows before it becomes fully turbulent. Most pipe flows encountered is practice are turbulent. Laminar flow is encountered when highly viscous fluids such as oils flow in small diameter tubes or narrow passages. The hydrodynamic entry length is usually taken to be the distance from the tube entrance where the friction coefficient reaches within about 2% of fully developed value. In laminar flow, the hydrodynamic and thermal entry lengths are given approximately as following.

$$L_H = 0.05 \text{ ReD}$$
$$L_T = 0.05 \text{ RePr} \qquad \text{[Eq.30]}$$

In the entrance region the flow in tubes is not developed, Nusselt recommended the following equation

for $10 < L/d < 400$

$$Nu = 0.036 Re^{0.8} Pr^{1/3} \left(\frac{d}{L}\right)^{0.055} \qquad \text{[Eq.31]}$$

where L is the length of the tube and d is the tube diameter.

Re is the Reynolds number ($Re = \frac{4\dot{m}}{\pi D \mu}$)

Pr is the Prandtl number ($Pr = \frac{\mu C_p}{k}$).

The properties in equation 31 are evaluated as the mean bulk temperature the above equations offer simplicity in computation, but errors on the order of ±25% are not uncommon. Petukhov has developed a more accurate, although more complicated, expression for fully developed turbulent flow in smooth tubes. [13]

$$Nu = \frac{(f/8) Re Pr}{1.07 + 12.7\sqrt{f/8}\left(Pr^{2/3} - 1\right)} \left(\frac{\mu_b}{\mu_w}\right)^n \qquad \text{[Eq.32]}$$

where $n = 0.11$ for $T_w > T_b$, $n = 0.25$ for $T_w < T_b$ and $n = 0$ for constant heat flux or for gases. All properties are evaluated at $T_f = (T_w + T_b)/2$ except for μ_b and μ_w. The friction may be obtained from the following equation for smooth tubes.

$$f = (1.82 \log Re - 1.64)^{-2} \qquad \text{[Eq.33]}$$

Equation 32 is applicable for the following ranges.

$0.5 < Pr < 200$ for 6% accuracy and $0.5 < Pr < 2000$ for 10% accuracy

$10^4 < Re < 5x10^6$, and $0.08 < \mu_b / \mu_w < 40$

To obtain agreement with data for smaller Reynolds number, Gnielinski modified the correlation and proposed an expression of the form

$$Nu = \frac{(f/8)(Re-1000)Pr}{1+12.7\sqrt{f/8}\left(Pr^{2/3}-1\right)} \qquad [Eq.34]$$

The correlation is valid for $0.5 < Pr < 200$ and $2300 < Re < 5x10^6$. We note that, unless specifically developed for the transition region. ($2300 < Re < 10^4$)

Storage performance of sensible heat storage media [23]

Sensible heat storage media may be classified on the basis of the heat storage media as liquid, solid and dual media. An example of liquid media storage is water, being inexpensive and widely available. For the storage of hot water; steel, concrete, fiberglass, fiberglass reinforced plastic, or vinyl lined wooden tanks may be employed. If the tank is cylindrical, vertical mounting is generally preferred as compared to horizontal due to the requirement of the latter to resist beam bending or buckling. Fiberglass or fiberglass reinforced plastic tanks have the major advantage that they are corrosion resistance.

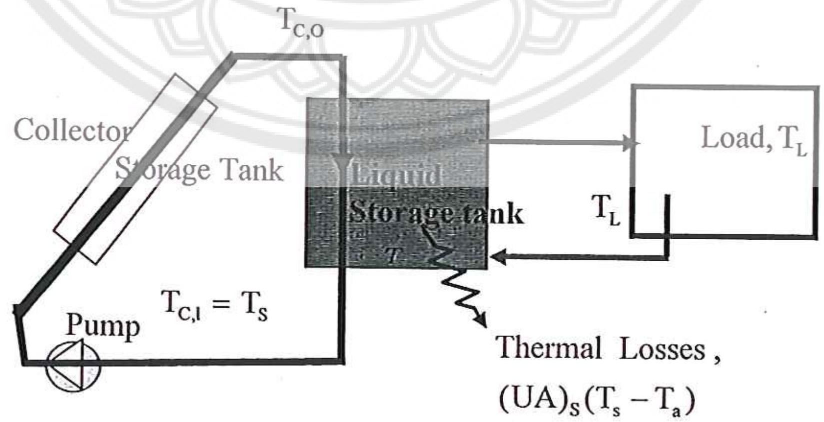

Figure 15 Essentials of mixed liquid storage systems

Source: Paykoc and Kakac, 1987

From Figure 15, assuming if the storage tank has a uniform temperature T_s, energy balance on a storage tank gives:

$$(MC_P)_s \frac{dT_s}{dt} = \dot{Q}_c - \dot{Q}_L - (UA)_s(T_s - T_a) \qquad \text{[Eq.35]}$$

where M is the mass of media in storage tank, kg

$\dot{Q}_c$ is the rate of heat addition by the collector, J/s

$\dot{Q}_L$ is the rate of heat removal by the load, J/s

U is the overall heat transfer collection between media in the tank, W/m^2K

T_a is the ambient at temperature, K

and A is the surface area of the storage tank, m^2.

If for a reasonable time period Δt, the rates of heat addition and removal are assumed to be constant, then Eq. 35 can be written for each time interval as

$$T_{s,new} = T_{s,old} + \frac{\Delta t}{(mC_P)_s}\left[\dot{Q}_c - \dot{Q}_L - (UA)_s(T_{s,old} - T_a)\right]$$

[Eq.36]

Taking, for example, one hour time intervals, one can estimate the temperature of the storage after several hours if hourly heat addition and removal are known. This method gives successful results when one hour intervals are taken.

1. Storage tank without heat exchanger

This type, the liquid flows directly into and out of storage, the solar collector inlet temperature, T_{ci} is the same as temperature in the storage T_s, the energy rate is given as

$$\dot{Q}_c = A_c F_R [(\tau\alpha) I(t)_c - (UA)_s (T_s - T_a)] \qquad \text{[Eq.35]}$$

where F_R is heat removal factor, A_c is area of collector, m^2

$\tau\alpha$ is transmissivity and absorptivity respectively

It has been assumed here, that there is no drop temperature of the fluid between the collector and the tank.

The rate of heat addition $\dot{Q}_c$ by the collector can be written as

$$\dot{Q}_c = (\dot{m} C_p)_c (T_{c,o} - T_s) \qquad \text{[Eq.36]}$$

Where $\dot{m}$ is the flow rate in kg/s, and C_p is the specific heat of HTF flowing through the collector. In writing equation (34) is assumed that there is no temperature drop of the fluid between the collector and the tank.

Similarly, the rate of heat removal by the load $\dot{Q}_L$ can be written as:

$$\dot{Q}_L = (\dot{m} C_p)_L (T_s - T_L) \qquad \text{[Eq.37]}$$

Where $\dot{m}$ is the flow rate of load stream and it is again assumed that there is no temperature drop between the tank and the load.

2. Storage tank with heat exchanger

A well mixed solid storage with heat exchanger for both the collector and the load circuits is shown in Figure 16. In this case T_{ci} is not the same as T_s

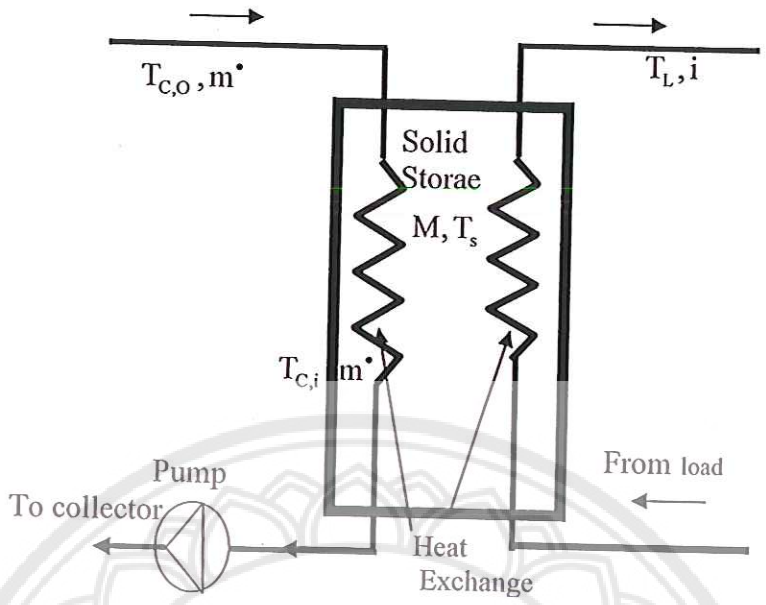

Figure 16 Fully mixed storage with heat exchangers

Source: Paykoc and Kakac, 1987

When the fluid in the collector and in the distributing circuit is not the same as in the storage, then fully mixed tanks are used with heat exchangers as illustrated in Figure 16. The rate of heat addition by the collector can be written as;

$$\dot{Q}_c = \left(\dot{m}C_p\right)_c \left(T_{c,o} - T_{c,i}\right) \quad \text{[Eq.38]}$$

It is assumed that the overall heat transfer coefficient is constant. By writing down the basic equations of heat exchange between the collector fluid and the liquid stored through the heat exchanger, one can get the following relation:

$$\frac{T_{c,o} - T_{c,i}}{T_{c,o} - T_s} = 1 - \exp\left[-\frac{(UA)_c}{\left(\dot{m}C_p\right)_c}\right] \quad \text{[Eq.39]}$$

Similarly, energy withdrawal rate by the load is

$$\dot{Q}_L = \left(\dot{m}C_p\right)_L \left(T_{L,i} - T_{L,o}\right) \qquad \text{[Eq.40]}$$

The temperature difference $(T_{L,i} - T_{L,o})$ of the load stream as it flows through the heat exchanger can be found from the following relation:

$$\frac{T_{L,i} - T_{L,o}}{T_s - T_{L,o}} = 1 - \exp\left[-\frac{(UA)_L}{\left(\dot{m}C_p\right)_L}\right] \qquad \text{[Eq.41]}$$

The heat exchanger can also be located outside the storage tank and they are used in application of solar water heating and storage systems.

Properties of Sodium chloride, Sodium chloride and Potassium nitrate

1. Sodium Chloride

Sodium chloride, also known as salt, common salt, table salt or halite, is an ionic compound with the formula NaCl. Sodium chloride is the salt most responsible for the salinity of the ocean and of the extracellular fluid of many multicellular organisms. As the major ingredient in edible salt, it is commonly used as a condiment and food preservative. Sodium chloride forms crystals with face-centered cubic symmetry. In these, the larger chloride ions are arranged in a cubic close-packing, while the smaller sodium ions, fill all the cubic gaps (octahedral voids) between them. Each ion is surrounded by six ions of the other kind; the surrounding ions are located at the vertices of a regular octahedron. This same basic structure is found in many other minerals and is commonly known as the halite or rock-salt crystal structure. It can be represented as a face-centered cubic (fcc) lattice with a two-atom basis or as two interpenetrating face centered cubic lattices. The first atom is located at each lattice point, and the second atom is located half way between lattice points along the fcc unit cell edge.

Thermal conductivity of pure NaCl as a function of temperature has a maximum of 2.03 W/(cm K) at 8 K and decreases to 0.069 at 314 K (41 °C). It also decreases with doping. Small particles of sea salt are the dominant cloud condensation nuclei well out at sea, which allow the formation of clouds in otherwise non-polluted air.

Table 5 Properties of sodium chloride

Properties of sodium chloride	
Solubility in water	359 g L^{-1}
Boiling point	1413 °C, 1686 K, 2575 °F
Melting point	801 °C, 1074 K, 1474 °F
Density	2.165 g cm^{-3}
Molecular	NaCl
Properties of sodium chloride	
Appearance	Colorless crystals
Exact mass	57.958622382 g mol^{-1}
Molar mass	58.44 g mol^{-1}
Molecular formula	NaCl
Odor	Odorless
Solubility in ammonia	21.5 g L^{-1}
Solubility in methanol	14.9 g L^{-1}
Acidity (pK_a)	6.7–7.3
Basicity (pK_b)	6.7–7.3
Refractive index (η_D)	1.5442 (at 589 nm)

2. Sodium Nitrate

$NaNO_3$, a salt; colorless crystals. Density, 2.257 g/cm3; melting point, 308 °C (decomposes into $NaNO_2$ and O_2 at higher temperatures).Solubility in water, 47.6 percent at 25°C and 64.3 percent at 100°C. It is a strong oxidizing agent. Sodium nitrat e occurs naturally as Chilesaltpeter. It is prepared industrially by reaction of nitric oxid es and Na2CO3 solutions, with subsequent oxidation of the resulting $NaNO_2$. Itisused as a minor additive in the processing of meats (since it readily reduces to sodium nitrit e) In agriculture, sodium nitrate is used as a nitrogen fertilizer. It contains 16 percent nitrogenand not more than 2 percent moisture; it ishygroscopic, agglutinates very little during storage, and spreads readily. It is applied as a base fertilizer, and also as a row fertilizer and topdressing for various types of soil and all crops. It is most effective wh en applied under sugar beets and root vegetables (which require relativelylarge quantit ies of Na), and also wheat and barley, particularly (because of its physiological alkalin ity) on acidic soddy-podzolic soils.

Table 6 Properties of Sodium nitrate

Properties of Sodium nitrate	
Molar mass	84.9947 g/mol
Appearance	White powder or colorless crystals
Odor	sweet
Density	2.257 g/cm3, solid
Melting point	308 °C (586 °F; 581 K)
Boiling point	380 °C (716 °F; 653 K) decomp
Solubility in water	73 g/100 mL (0°C) 91.2 g/100 mL (25 °C) 180 g/100 mL (100 °C)
Refractive index(η_D)	1.587 (trigonal) 1.336 (rhombohedral)
Specific heat capacity C	93.05 J/mol K
Std molar entropy S^o298	116 J·mol−1·K−1
Std enthalpy of formation ΔfH^o298	−467 kJ·mol−1
Gibbs free energy ΔG	-365.9 kJ/mol

3. Potassium Nitrate

Potassium nitrate's colorless crystals dissolve easily in water and remain stable unless mixed with other substances, when it can become explosive because of its strong oxidizing properties. Its melting point occurs at 334 degrees centigrade and it reaches boiling at 400 degrees centigrade. It produces orange flames in less-pure forms and lilac flames in purer forms. Hazardous signs must be displayed by trucks carrying this oxidizer. Physicochemical Properties: Potassium Nitrate is white odorless crystal powerder or granular. Specific Density: 2.109, Melting Point :334 ˚C Thermal Decomposition : 400 ˚C ,it easily dissolve in water solubility: 320 g/L (20 ˚C) as strong oxidizer ,it can be burning and exploding when mixed with organism. Applications and Usages: Potassium Nitrate is mainly applied in fireworks and black powder; salt bath of heat treatment; glass refining agent and motor light glass drugs and catalyzer ceramic and enamel cigarette pater, concentrating agent compound fertilizer and foliar spray fertilizer in agriculture.

Table 7 Properties of potassium nitrate

Properties of potassium nitrate	
Molecular formula	$KNO3$
Molar mass	101.1032 g/mol
Appearance	white solid
Odor	odorless
Density	2.109 g/cm3 (16 °C)
Melting point	334 °C
Boiling point	decomposes at 400 °C
Solubility in water	133 g/L (0 °C) 316 g/L (20 °C) 2460 g/L (100 °C)

Table 7 (cont.)

Properties of potassium nitrate	
Basicity (pKb)	15.3
Refractive index(η_D)	1.5056
Specific heat capacity C	95.06 J/mol K
Std enthalpy of formation ΔfHo298	-494.00 kJ/mol

Review of related of research

1. Storage tank

The most advanced thermal energy storage for solar thermal power plants is a two-tank storage system where the heat tansfer fluid (HTF) also serves as storage medium. This concept was successfully demonstrated in a commercial trough plant (13.8 MW_e SEGS I plant; 120 MW_{ht} storage capacity) and a demonstration tower plant (10 MW_e Solar Two; 105 MW_{ht} storage capacity). However, the HTF used in state-of-the-art parabolic trough power plants(30 -80 MW_e) is expensive, dramatically increasing the cost of larger HTF storage systems. An engineering study was carried out to evaluate a concept, where another (less expensive) liquid medium such as molten salt is utilized as storage medium rather than the HTF itself. Detailed performance and cost analyses were conducted to evaluate the economic value of this concept. The analyses are mainly based on the operation experience from the SEGS plants and the Solar Two project. The study concluded that the specific cost for a two-tank molten salt storage is in the range of US$ 30–40/kWhth depending on storage size. Since the salt storage was operated successfully in the Solar Two project, no major barriers were identified to realize this concept in the first commercial parabolic trough power plant [23].

Study storage by Cascaded latent heat storage for Solar Thermal Power Stations Solar Electricity at Day and Night. Thermal energy storage for parabolic trough solar power stations work in the temperature range from 250 °C to 450 °C. preferable phase change materials for this temperature range are KNO_3, KNO_3/KCl

and $NaNO_3$. Three test modules, filled with these salts, were connected as a cascaded system and experimentally investigate [24].

Advanced Thermal Energy Storage Technology for Parabolic Trough. The availability of storage capacity play an important role for the economic success of solar thermal power plants. For today's parabolic trough power plants, sensible heat storage systems with operation temperature between 300 °C and 390 °C can be used. A solid media sensible heat storage system is developed and will be tested in a parabolic trough test loop at PSA, Spain. A simulation tool for the analysis of the transient performance of solid media sensible heat storage systems has been implemented. The computer results show the influence of various parameters describing the storage system. While the effected geometry of the storage system demands the analysis of the complete power plant and not only storage unit. Then the capacity of the system is defined by the electric work produced by the power plant, during a discharge process of the storage unit. The choice of the operation strategy for the storage system proves to be essential for the economic optimization[25].

2. Storage media

The experimental and numerical results from the investigation of cascaded latent heat storages with alkali nitrate salts like $NaNO_3$, KNO_3 and others more. The experiments were conducted with vertical shell and tube type heat exchanger devices under realistic operation parameters. The experimental results were used for a numerical model to simulate different CLHS configurations. Dymola/Modelica was used to conduct the simulation. The outcome of this work shows on the one hand, that the design of CLHS for this temperature range is more complex than for the temperature range up to 100 °C. And on the other hand, the low heat conductivity of available PCM is an obstacle which must be overcome to make full use of this promising storage technology [26].

3. Heat transfer

A phase change thermal energy storage system of the shell and tube type is numerically modeled. The heat transfer fluid is flowing by forced convection. The phase change material is in the shell around the HTF carrying tubes. The momentum equations of the flowing fluid in the tubes are solved to obtain the velocity field which is used to solve the energy equation of the HTF. The three energy equations, namely the energy equation of the heat transfer fluid, the energy equation of the tube wall and the energy equation of the phase change material, are solved as one domain The solution of the system consists of solving the equations of the heat transfer fluid (HTF), the pipe wall and the phase change material (PCM) as one domain. The control volume finite difference approach is used to solve the equations describing the phase change thermal storage system. The SIMPLE scheme is used to solve for the pressure and velocity fields of the HTF. The radial temperature distribution, the phase change interface position and the latent and sensible heat accumulated along the system axial length are shown for different Reynolds and Stefan numbers, phase change temperature range and time periods. Also shown in this paper is the possibility of using the numerical model to solve problems with a single temperature phase change. The results obtained are important parameters for PCM thermal system performance investigation and design [27].

Thermal performance of a PCM storage unit. This paper present a two-dimension model for the phase change, conduction based heat transfer problem around a tube immersed in the PCM. The energy equation is written in the enthalpy from, and tube. The numerical solution is based upon the average control volume technique and ADI finite difference representation. The result obtrained show the effects of the variation of the ratio of the radius of the inner to the outer tube. Biot number, Stafan number and the working fluid inlet temperature on the solidified mass fraction, NTU and effectiveness [28].

Most of the work mentioned is mainly on the solution techniques, there is a lack of systematic studies on the various parameters that influence the melting process, especially the melting fraction and the melting time (that is, the time required to finish melting). The main objective of the present study is, by numerical simulation of a very simple melting problem of a finite slab on which is imposed constant flux heating at its surfaces, to investigate the influences of various parameters on the melting process. In order to take account of the starting period of thermal energy storage systems, a permit stage is also included. Melting in a finite slab with a second kind boundary condition is studied numerically in order to simulate the charging process of a thermal energy storage system. A dimensionless model is given, from which it is concluded that the main factors that influence the melting process are the dimensionless heating flux, the modified Stefan number, the relative thermal diffusivity and the relative thermal conductivity. The influence of preheating or solid sub-cooling is studied. It is found that though preheating does not have very important effects on the melting time, it does influence the interface marching velocity significantly. The melt fraction and the melting time are calculated extensively for various dimensionless numbers. The numerical results show that the ratio of the thermal conductivity of the solid to that of the liquid has little effect on the melting time, and the time for finishing melting can be expressed as a function of the dimensionless heating flux, the modified Stefan number and the relative thermal diffusivity, and the possible function form is suggested [29].

Study heat transfer characteristics during melting of $MgCl_2 \cdot 6H_2O$ have been determined for circular finned and unfinned-tube systems. The effects of inlet temperature and flow rate of heat transfer characteristics have been determined. Heat transfer from the tube to the phase change material is strongly influenced by natural convection of melted PCM, especially in the unfinned-tube system. The measure melting front velocities in the unfinned – tube system agree well with analytical predications. The amount of heat storage in three difference systems have been correlated in terms of Fourier, Stefan, and Reynolds numbers to provides basic data for designing unfinned and finned- tube heat storage units [30].

Finite–element analysis of cyclic heat transfer in a shell and tube latent heat energy storage exchange. The physical module of a shell and tube latent heat storage exchanger consists of a tube which is surrounded by an external coaxial cylinder made up of a phase change material. A heat transfer fluid flows through the tube to store or extract thermal energy from the PCM. Two alternative operation modes are possible. One option is to introduce hot and cold fluids from different ends of the tube. A finite element model is developed to simulate the inherently cyclic thermal processes. Involved as a result of the alternating melting and freezing processes in the storage exchanger. Computations have been carried out to investigate the effects of the two different operation modes. Numerical results provide guidance for selection of the appropriate mode [31].

Experiment and analytical phase change heat transfer. An experiment and analytical study was performed to evaluate the heat transfer results during melting and solidification of a phase change medium. The experiment work included determination of the thermal conductivities for the solid as well as the liquid phase of a phase change fluoride salt. A two dimension, transient ax symmetric analytical model was developed to study the phase change transfer process in an energy storage system. The results of previously conducted experimental tests on the system were compared and correlated with the numerical results. Agreement was very good in most cases [32].

Study thermal energy storage using phase change material. A two-dimensional melting process of a solid phase change material is investigated theoretically. The material contained in rectangular enclosure heated from one side, while all the other sides are assumed to adiabatic ones. In this study convection mode was considered to be the solid surface at the bottom where conduction mode was only taken into considerations. It was found that the obtained results are good agreement with previous ones. Finally the present analysis was use to predict the melted fraction of the phase change material and hence the amount of storage energy [33].