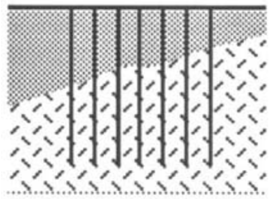
5. GROUND HEAT STORAGE

Heat storage is required when heat demand does not match heat production. Seasonal heat storage is a long term storage (from a few months to a few seasons) devised to store thermal energy collected during the summer for winter use. It can be waste heat, thermal loads from a cooling requirement, solar energy, etc. Seasonal storage of “cold” energy is also a possibility for cooling needs. The advantage of ground heat storage is that large volumes can be realised with a low ground occupation at the surface. The acronym **UTES (underground thermal energy storage)** is dedicated for ground heat storage. General guidelines and detailed information can be found in the SIA documentation D028 (1988). State of the art information and an overview of the past experience can be found in the Giessener Geologische Schriften number 67 (1999).

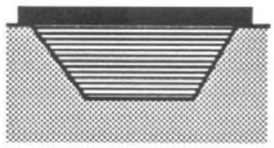
5.1 Storage families

Large heat storages can be categorised in four main families :

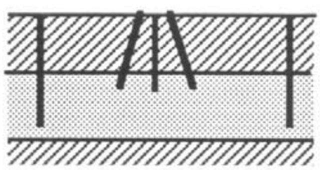
- **Ground diffusive storage**
  The principal heat transport process in the storage is conductive. The storage medium is the ground itself. The ground heat exchanger is vertical and normally formed with borehole heat exchangers. Such a store is also called **borehole thermal energy storage (BTES)**. Very large ground volume can be realised. In soft ground, the heat exchangers can be pushed down or hammered into the ground.



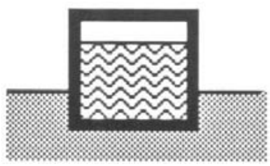
- **Earth storage**
  The principal heat transport process in the storage is also conductive and the storage medium is earth. The ground heat exchanger is horizontal and normally requires the excavation of the storage volume. All the storage sides can be insulated.



- **Aquifer storage**
  Heat transport is both convective and conductive. The storage medium is ground water and the matrix (ground) containing the water. A common application is a doublet for cooling purposes. With high temperature storage in aquifer, chemical problems have to mastered and controlled.



- **Water storage**
  The principal heat transport process in the storage is convective. The storage medium is water. It includes large water tank, on ground or buried, water pit and even rock cavern.



The cheapest storages are in the ground diffusive and aquifer storage families. In this chapter, we will concentrate on the borehole thermal energy storage of the ground diffusive storage family.

5.2 System families

Two main system families can be defined:

- **Seasonal heat storage with heat pump**
  A heat pump is used to extract heat from the store. A thermal recharge of the store is necessary, and is best combined with cooling requirements. In figure 5.1, a system with solar thermal recharge is shown. It can also be waste heat or another source of cheap thermal energy. The storage operates at low temperature, typically between 5 and 35 °C.

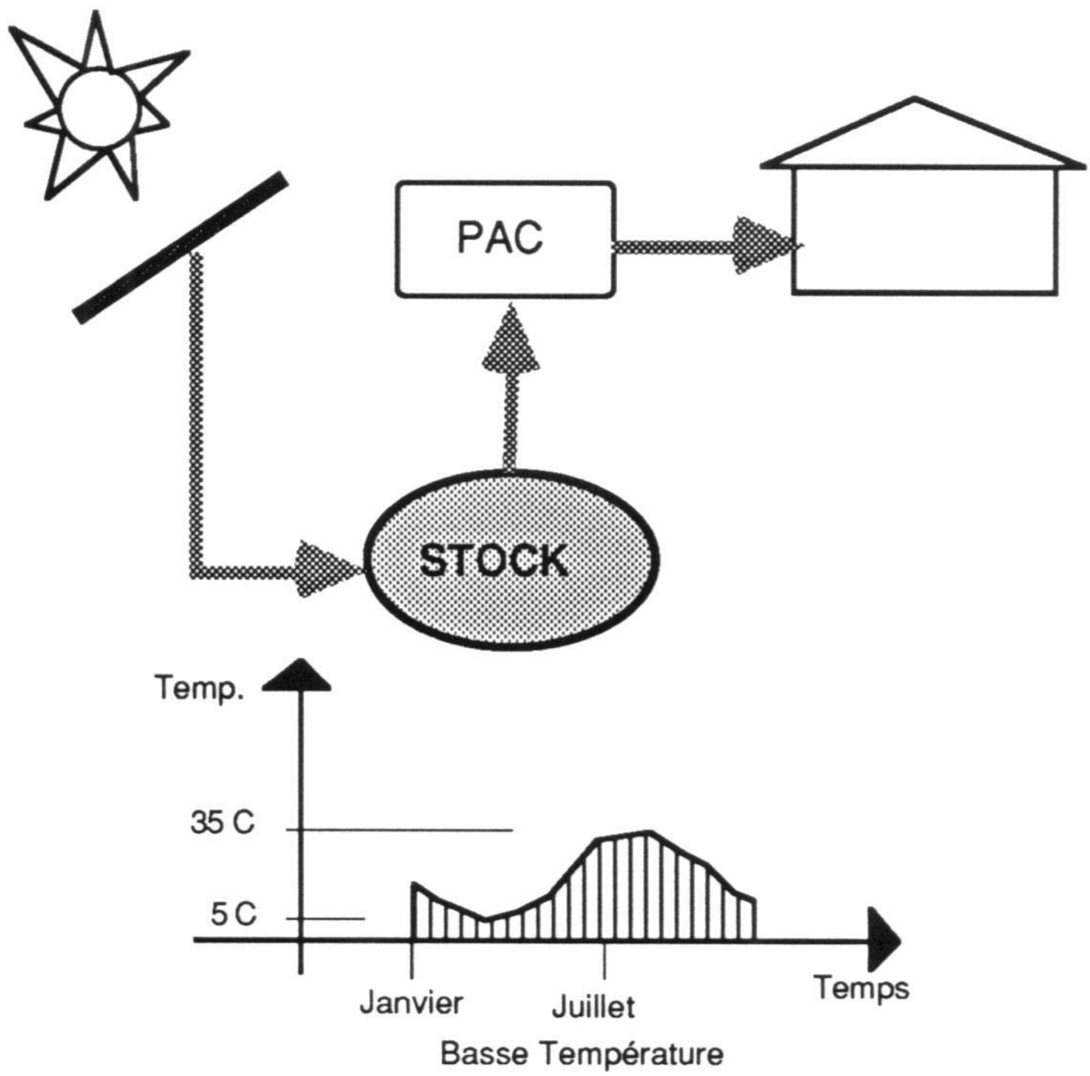


Figure 5.1 Seasonal heat storage in a system with heat pump (source : Hadorn, 1992).

- **Seasonal heat storage without heat pump**
  No heat pump is used in the system. The source of energy (solar energy or waste heat from a thermal process) is used directly when possible (a short-term storage can also be integrated in the system for this purpose) and stored in the seasonal storage otherwise. In figure 5.2, a system with seasonal heat storage of solar energy is shown. It can also be waste heat from a thermal process. Depending on the temperature level of the heat distribution, the seasonal storage normally operates at medium (25°C – 50°C) or high (30°C – 80 °C) temperature.

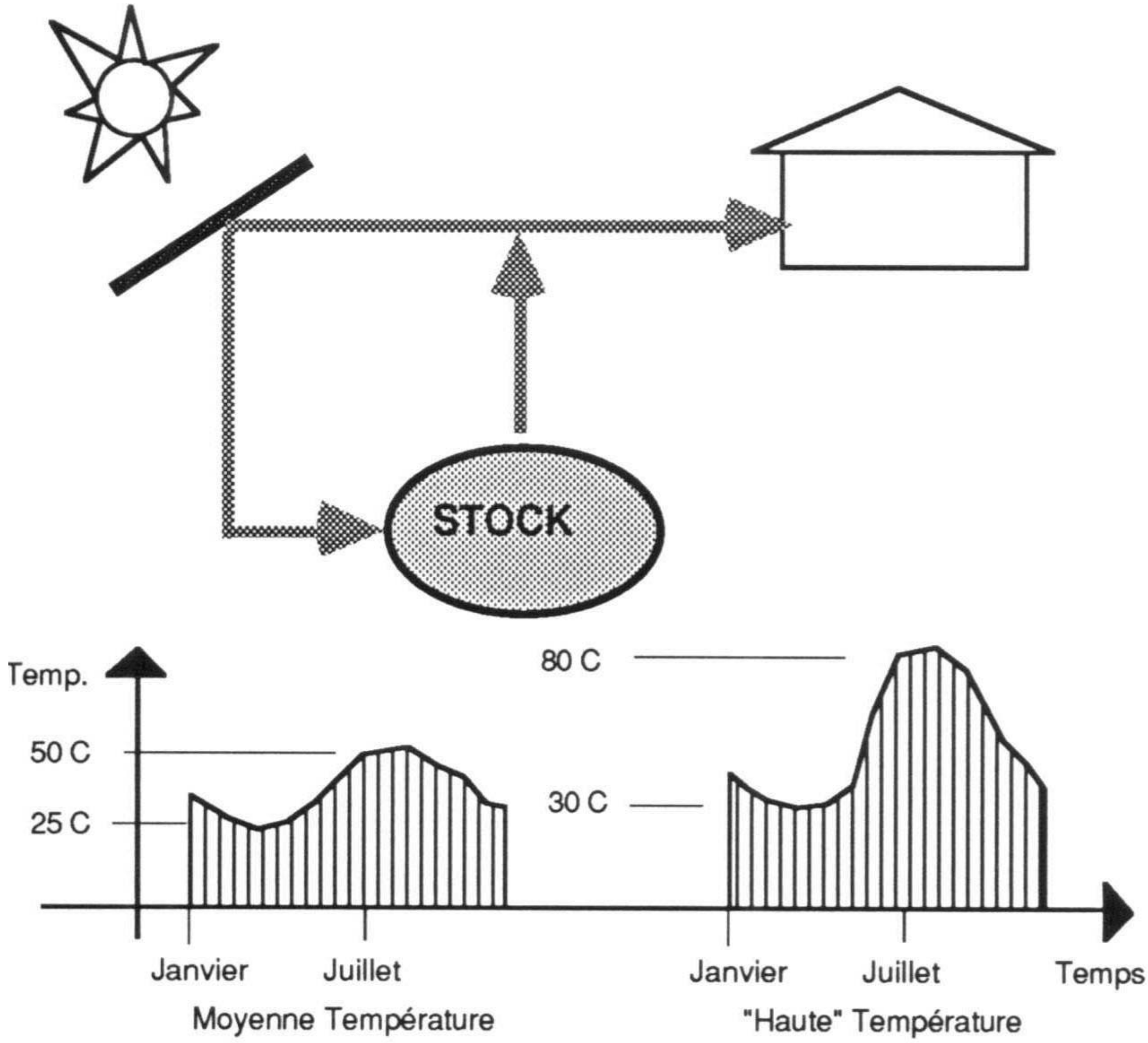


Figure 5.2 Seasonal heat storage in a system without heat pump (source : Hadorn, 1992).

5.3 Borehole thermal energy storage

A borehole thermal energy storage (BTES) is at the same time a heat exchanger and a heat storage. The heat exchanger, called ground heat exchanger, has poor heat transfer thermal characteristics, due to the dominating conductive heat transport process. As a result, a large heat transfer rate often induces a significant temperature loss. Three main properties characterise such a storage on the thermal point of view:

- **The heat transfer capacity**
- **The specific storage capacity**
- **The storage efficiency**

Heat transfer capacity
When a constant heat transfer rate is injected through a ground heat exchanger, a temperature difference will develop between the fluid and the ground. This temperature difference will increase until a steady flux regime is established. At this moment, the average ground temperature in the store increases as fast as the mean fluid temperature. The temperature difference remains stable and constant as long as the heat injection rate lasts.

The **heat transfer capacity UA** is defined for steady flux conditions. It determines the heat transfer rate per temperature difference unit between the heat carrier fluid mean temperature and the storage mean temperature. The transient period until a steady flux regime is obtained can be estimated with relation 5.1 (Hellström, 1991).

$$t_{sf} = 0.065 \frac{A_p}{a} \tag{5.1}$$

- t_{sf} required time until a steady flux regime is obtained (s);
- A_p ground section ascribed to 1 borehole (m^2). With a quadratic borehole arrangement, A_p = B x B, where B is the spacing.
- a ground diffusivity (m^2/s).

Typical values for a ground heat storage are $a = 10^{-6}$ m^2/s and B = 3 m. The steady flux time t_{sf} is about one week.

The heat transfer capacity UA (equation 5.3) depends on the total borehole length and the steady flux thermal resistance, composed by the sum of the borehole thermal resistance and the ground thermal resistance. The ground thermal resistance (in equation 5.2) is calculated for a circular region around each borehole. It can also be used with good precision for a quadratic or an hexagonal region. The condition for the validity of formula 5.2 ($\frac{\sqrt{A_p}}{\sqrt{\pi}\, r_b} \geq 15$) is normally satisfied with ground heat storage.

$$R_{sf} = \frac{1}{2\pi\lambda}\left[\ln\left(\frac{\sqrt{A_p}}{\sqrt{\pi}\, r_b}\right) - 0.75\right] + R_b^* \qquad \text{if} \qquad \frac{\sqrt{A_p}}{\sqrt{\pi}\, r_b} \geq 15 \qquad (5.2)$$

- R_{sf} steady flux thermal resistance (K/(W/m));
- λ mean ground thermal conductivity (W/mK);
- r_b borehole radius (m);
- R_b^* effective borehole thermal resistance of the borehole heat exchanger (K/(W/m)).

$$UA = \frac{n\,H}{R_{sf}} \qquad (5.3)$$

- UA storage heat transfer capacity (W/K);
- n number of borehole heat exchangers (-);
- H mean active length of a borehole heat exchanger (m).

An estimation of the heat transfer rate under steady flux condition is calculated with relation 5.4 for a given temperature loss.

$$P = UA\left(T_f - T_{stk}\right) \qquad (5.4)$$

- P heat transfer rate transferred to/from the storage (W);
- T_f heat carrier fluid mean temperature in the ground heat exchanger; can be estimated with the arithmetic mean of inlet / outlet fluid temperature (°C);
- T_{stk} storage mean temperature (°C).

Specific storage capacity

The **specific storage capacity C_{sp}** is equivalent to the amount of thermal energy necessary to change the storage mean temperature of 1K. It is estimated with the ground volumetric thermal capacity and the storage volume (equation 5.5).

$$C_{sp} = \rho C\ V = \rho C\ n\ A_p\ H \qquad (5.5)$$

- C_{sp} specific storage capacity (J/K);
- ρC mean ground volumetric thermal capacity (J/m^{3}K);
- V storage volume, defined by $n\, A_p\, H$ (m^3).

Another interesting quantity is the **storage capacity C**. It is the maximum amount of thermal energy that can be stored. It depends on the minimum and maximum storage mean temperature during a cycle (one year). It clearly depends on the integration of the storage in the system, the system type and operation. In particular, the maximum storage temperature is conditioned by the temperature level of the heat source and the storage heat transfer capacity. For system without heat pump, an important parameter for the minimum storage temperature is the return fluid temperature from the heat distribution.

$$C = C_{sp} \; (T_{stk\text{-}max} - T_{stk\text{-}min}) = \rho C \; V \; (T_{stk\text{-}max} - T_{stk\text{-}min}) \tag{5.6}$$

- C storage capacity (J);
- $T_{stk\text{-}max}$ maximum storage mean temperature (°C);
- $T_{stk\text{-}min}$ minimum storage mean temperature (°C).

An index associated to the storage capacity is the **equivalent cycle index EC**. This index is defined with relation 5.7.

$$EC = Q_{ext} / C \tag{5.7}$$

- EC equivalent cycle index (-);
- Q_{ext} annual thermal energy extracted from the store (J).

This index indicates how many time the storage has been “recycled” during a year. For most of the long term storage in the world, this index lies between 1.5 and 2. For a purely seasonal heat storage EC is equal to 1. It is much higher for a short-term storage, and would be 365 for an ideal daily storage, fully used all over the year. This index shows the necessity of a low cost for a seasonal heat storage.

Storage efficiency

The **storage efficiency** η is defined by the ratio of the annual extracted energy by the annual injected energy in the storage.

$$\eta = Q_{ext} / Q_{inj} \tag{5.8}$$

- η seasonal storage efficiency (-);
- Q_{inj} annual thermal energy injected in the store (J).

Assuming that the storage temperature returns to the same value after 1 cycle (1 year), the storage efficiency can also be calculated with relation 5.9.

$$\eta = Q_{ext} / (Q_{ext} + Q_{loss}) = (Q_{inj} - Q_{loss}) / Q_{inj} = 1 - Q_{loss} / Q_{inj} \tag{5.9}$$

- Q_{loss} annual storage heat losses (J). If the storage temperature returns to the same value after 1 year, a storage heat balance gives $Q_{inj} = Q_{ext} + Q_{loss}$.

The storage efficiency depends on the annual storage heat losses Q_{loss} and on its mode of utilisation Q_{ext}. As for the storage capacity, the storage efficiency depends on the integration of the storage in the system, the system type and operation. For low temperature seasonal storage application, storage efficiencies of 60 – 90 % can be reached. For medium and “high” temperature storage, the storage efficiency strongly depends on the relative importance of heat losses to the energy stored (which is in fact the ratio Q_{loss}/Q_{inj}). The magnitude of the ratio is decreasing with increasing storage size, as heat losses are increasing with the storage envelope surface (proportional to the square of a length) and stored energy is increasing with the storage volume (proportional to the cube of a length). For “small” seasonal storage (volume in the range 10’000 – 20’000 m^3), storage efficiencies of 30 to 60% can be realised. With larger volume (> 20’000 m^3), storage efficiencies of 50 – 80% can be expected.

Storage heat losses depend mainly on the mean annual storage temperature $T_{stk\text{-}moy}$, the mean ambient temperature T_o, an equivalent heat loss factor U and the area of the store border A. The average heat loss factor is essentially conditioned by the store design (insulation of upper parts of storage border, geometry, etc.), the ground properties and is time-dependent. A transient thermal process usually lasts a few years until a steady-state thermal process is established. Forced and free convection in the ground results in increased heat losses. In the case of a dominant conductive thermal process, storage heat losses can be expressed with relation 5.10 for steady state conditions.

$$Q_{loss} = U\ A\ (T_{stk\text{-}moy} - T_o)\, t_{year} \tag{5.10}$$

- Q_{loss} annual storage heat losses (J);
- U equivalent mean storage heat loss factor (W/m^{2}K);
- $T_{stk\text{-}moy}$ mean annual storage temperature (°C);
- T_o mean annual ambient temperature (°C);
- t_{year} duration of one year (s) ($\approx \pi\ 10^7$ s per year).

Heat losses can be reduced with:

- storage insulation at the top (more important for small storage);
- storage shape (vertical extension about twice the storage diameter, and not one, as would be the case without ground);
- low temperature heat distribution (result in a lower mean annual storage temperature);

The design of a borehole thermal energy storage requires dynamic system simulations, especially for a system without heat pump. It is important to simulate the store as part of the system and take into account both short term and long term thermal processes.

5.4 System examples

Examples of borehole thermal energy storage are numerous and spread worldwide. Storages of up to 1’000’000 m^3 have been built (Sanner and Stile, 1995). In this section, two examples are presented: a system with heat pump/cooling machine (the D4 centre) and a system without (ice-melting system for a bridge at Serso, Därlingen). Two response tests for the D4 centre were performed (see chapter 4, section 4.6).

Le centre D4
Sur la commune de Root près de Lucerne, la suva réalise la première étape du centre d’entreprises et d’innovation D4. D4 désigne les 4 dimensions homme, haute technologie, environnement et temps. Le centre D4 sera justement occupé par des PME actives dans les branches high-tech, technologies de l’environnement et service de santé. Dès le début du projet en 1990, la suva a fixé comme objectif de doter le centre D4 d’un concept énergétique respectueux de l’environnement. En plus d’un usage rationnel et optimal de l’énergie, le recours aux énergies renouvelables est explicitement spécifié. Elles doivent couvrir au moins 50% des demandes d’énergie de chauffage et de refroidissement restantes. Compte tenu de la difficulté de connaître avec exactitude les demandes de chaleur, et en particulier la demande de refroidissement (qui dépend également des besoins particuliers des utilisateurs qui ne sont pas encore connus), le concept énergétique doit avoir un caractère flexible et polyvalent. Il fera intervenir une toiture solaire, une pompe à chaleur/machine frigorifique combinée et un stockage diffusif de chaleur dans le terrain (B+B Energietechnik et al., 1999). L’énergie thermique est transférée au stockage au moyen d’un échangeur de chaleur souterrain, formé par un ensemble de sondes géothermiques régulièrement espacées. Le stockage diffusif jouera un rôle clef dans le concept énergétique et permettra de satisfaire aussi bien des besoins de chauffage que de refroidissement. Les sondes géothermiques atteindront une profondeur de 160m.

Trois concepts de système ont été évalués et ont permis d’optimiser le stockage de chaleur diffusif dans le terrain en tant que partie intégrante du système thermique. Les trois concepts de système sont:

- cas 1: refroidissement direct sur le stockage diffusif; seule une partie des besoins de refroidissement est satisfaite. La pompe à chaleur est dimensionnée en fonction de l'importance de la recharge thermique du stockage effectuée par le refroidissement direct.

- cas 2: la totalité des besoins de refroidissement sont injectés dans le stockage de chaleur par l'intermédiaire d'une machine frigorifique. La pompe à chaleur, dimensionnée pour « vider » le stockage en hiver, permettra de couvrir une plus grande fraction de la demande de chaleur que dans le cas 1.

- cas 3: la taille de la pompe à chaleur est fixée arbitrairement à 1.5 fois celle du cas 2. La recharge thermique du terrain est effectuée par les rejets de chaleur de la machine frigorifique et par l'énergie thermique collectée par des absorbeurs solaires.

Des contraintes sur la température du fluide circulant dans les sondes sont imposées. Une température minimale de 3 °C est prescrite, puisqu'il n'est pas prévu d'ajouter de l'antigel à l'eau qui circulera dans les sondes. Une température maximale de 50 °C est tolérée, afin de ne pas risquer l'endommagement des tubes en polyéthylène utilisés dans les sondes. Pour les trois cas, le nombre et l'espacement adéquat des sondes est à peu près le même. La 3^{e} variante a été choisie. Les demandes annuelles de chauffage et de refroidissement estimées pour le dimensionnement du système sont de respectivement 1'510 MWh/an et 730 MWh/an.

La pompe à chaleur couplée au stockage, avec une puissance thermique nominale de 450 kW, permet de couvrir 90% de la demande de chauffage annuelle des bâtiments. Il en résulte une extraction annuelle de 910 MWh du stockage, qui doit être compensée par une injection estivale de 1'270 MWh. Cette solution implique donc que l'énergie annuelle injectée dans le stockage soit environ 40% plus grande que celle qui est extraite. Elle sera couverte par les besoins de refroidissement des bâtiments et la toiture solaire. La température moyenne du stockage, initialement de 12 °C, augmentera au cours des années pour se stabiliser vers 18 °C. Les caractéristiques et les performances thermiques calculées du stockage sont énumérées dans la table 5.1.

Stockage de chaleur		volume de stockage:	**330’000 m^3**
49 sondes de 160 m		capacité de transfert de chaleur:	**30 kW/K**
espacement de 6.5 m		capacité spécifique du stockage:	**200 MWh/K**
Bilan stockage	Extraite	Injectée	Ratio injecté/extrait
Energie	910 MWh/an 116 kWh/m/an	1’270 MWh/an 162 kWh/m/an	1.39 Efficacité stockage
Puissance maximum	300 kW 38 W/m	- -	**72%**
Température		moyenne annuelle stockage	18 °C
Demande d’énergie satisfaite		chauffage 1’370 MWh/an (91 %)	refroidissement 730 MWh/an (100 %)

Table 5.1 Caractéristiques et performances thermiques du stockage après 10 ans de fonctionnement pour le concept n° 3: recharge complémentaire estivale avec toiture solaire.

Serso

Le projet Serso est né de l’idée de vouloir dégivrer un pont avec de l’énergie solaire. Il en résulte un concept qui met en œuvre un stockage saisonnier de chaleur dans le terrain. L’énergie solaire est captée par le pont en été, stockée dans le terrain par l’intermédiaire d’un ensemble de sondes géothermiques, puis restituée en hiver pour le dégivrage du pont. Le pont est équipé de serpentins qui permettent de collecter les gains solaires estivaux et de chauffer la chaussée en hiver pour empêcher la formation de glace ou de givre. A l’exception de l’énergie électrique nécessaire au fonctionnement des pompes, le système est conçu pour fonctionner sans énergie auxiliaire. En raison du niveau de température extrêmement bas pour le dégivrage du pont, ce système est une exception aux catégories présentées au début du chapitre. C’est un système sans pompe à chaleur, mais le stockage de chaleur diffusif fonctionne malgré tout à basse température (entre 5 et 15 °C). Cette installation a été mise en route en 1994 et a fait l’objet d’une campagne de mesure détaillée (Hopkirk et al., 1995).

Les caractéristiques du stockage sont données dans la table 5.2. Il est isolé en surface par une couche de 25 cm d’épaisseur de morceaux de mousse de verre. Elle est recouverte par une autre couche de 30 cm avec des matériaux de très faible perméabilité pour limiter les infiltrations d’eau par la surface. Puis une couche de drainage de 30 cm d’épaisseur et finalement une couche de terre recouvre le tout. La surface occupée par le stockage peut à nouveau être cultivée.

Stockage de chaleur	
Nombre de sondes géothermiques (n)	91
Longueur active des sondes (H)	65 m
Espacement des sondes (B)	3 m (arrangement hexagonal)
Surface de terrain occupée par sonde (B^2 x $\sqrt{3}$/2)	7.8 m^2/sonde
Volume de stockage (B^2 x $\sqrt{3}$/2 x N x H)	46'100 m^3
Diamètre des sondes (forage)	11.5 cm
Type de sonde	Double-U
Diamètre intérieur des tubes formant les U	26 mm
Matériau de remplissage	Bentonite et ciment
Résistance thermique d'une sonde R_b (1)	0.12 K/(W/m)
Résistance thermique interne d'une sonde R_a (1)	0.44 K/(W/m)

(1) calculé avec le programme EED (Hellström and Sanner, 2000) et les paramètres supplémentaires suivants :

diamètre externe / interne du tube en plastique	32 / 26 mm
conductibilité thermique du tube en plastique	0.42 W/mK
conductibilité thermique du matériau de remplissage	0.8 W/mK
espacement axe – axe de deux tubes opposés	75 mm
débit de fluide par sonde	440 litres/h
type de fluide	éthylène glycol
point de congélation du fluide	-21 °C

Table 5.2 Caractéristiques du stockage saisonnier de l'installation de Serso.

La figure 5.3 montre deux années de mesures. Les mesures de température du terrain dans le volume du stockage ont été utilisées pour calculer sa température moyenne. Cette dernière est reportée en fonction du bilan cumulé de l'énergie injectée moins l'énergie soutirée du stockage.

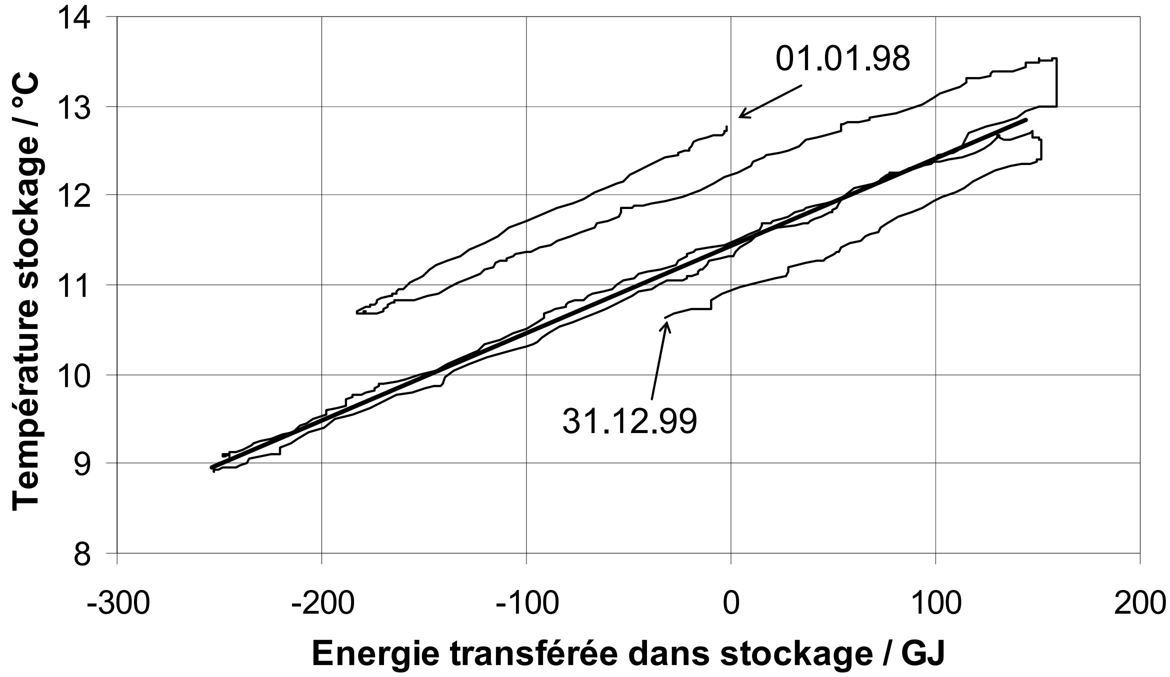


Figure 5.3 Température moyenne du stockage représentée en fonction de l'énergie nette transférée par l'échangeur souterrain (cumul énergie injectée – énergie soutirée) (source : Pahud, 2001c).

Si le stockage n'a pas de pertes thermiques, un cycle de charge – décharge fera déplacer les points de la courbe sur une droite dont la pente (en K/GJ) est l'inverse de sa capacité thermique spécifique (en GJ/K). Les pertes thermiques du stockage ont pour effet de déplacer les points sur la droite. A l'inverse ils seront déplacés à gauche, si les pertes thermiques sont en réalité des gains thermiques. Lors de l'hiver 98 – 99, l'extraction importante de chaleur a eu pour résultat de sensiblement abaisser la température du stockage, si bien que les pertes du stockage ont été réduites à zéro, voir inversées. Au cours de la décharge du stockage pendant l'hiver 98-99 et de sa recharge le printemps – été suivant, les pertes thermiques du stockage ont été faibles relativement aux énergies transférées. Cette période a l'avantage de permettre une estimation directe de la capacité thermique spécifique du stockage. La régression linéaire montrée dans la figure 5.3 permet de l'estimer à environ **100 GJ/K**, soit près de **30 MWh/K**. En divisant la capacité thermique spécifique du stockage par son volume, on obtient la capacité thermique volumétrique moyenne du terrain. On trouve 2.2 MJ/m^{3}K, qui est une valeur tout à fait acceptable pour de la molasse. La conductibilité thermique du terrain a été mesurée en laboratoire à 4.5 W/mK sur des échantillons de la couche de molasse qui se trouve à 10 m de la surface. La capacité de transfert de chaleur de l'échangeur souterrain est estimée à **28 kW/K**.

5.5 CSHPSS system

CSHPSS system is an acronym for central solar heating plant with a seasonal storage. A CSHPSS system with a BTES is a CSHPSS system whose seasonal storage is a borehole thermal energy store. Such a system operates without a heat pump. Neckarsulm in Germany is an example. The system involves 2'700 m^2 of flat plate solar collectors, a 20'000 m^3 BTES with 168 boreholes of 30 m long each and an auxiliary gas burner to feed a heat distribution network for up to 1'300 flats and terraced houses (Seiwald and al, 1999). A solar fraction of about 50% is expected.

In this section, design guidelines for a CSHPSS system with a BTES are presented. They were obtained by dynamic thermal simulations of the whole system (Pahud, 1996b; Pahud, 2000).

Methodology

A system must be completely defined before its thermal performances can be assessed. In other terms, the system layout, which determines how the subsystems are connected together, and the system control strategy, which determines the system operation, should be known in advance, in addition to the many parameters that define each subsystem. Furthermore, the conditions that drive the system, i.e. the weather data and the heat load, are set to a particular climate and type of consumer.

A reference system is defined by fixing its system layout and control strategy. The collector area is used as a scaling factor for the design of the subsystems. Five main system parameters are varied and expressed in relation to the collector area when possible. They are the collector area (m^2), the specific buffer store volume (litre per m^2), the specific duct store volume (m^3 per m^2), the specific total borehole length (m per m^2) and the shape factor of the duct store (m per m), defined by the ratio between its vertical extension and its diameter; (the duct store volume is taken as being a vertical cylinder).

For each set of parameters, the thermal performances of the system are simulated with the dynamic model of the system over several years. The delivered heat in the distribution network that originates from the solar part of the system, called **solar heat**, is thus known year after year. The average yearly value, calculated for the life-time of the system, takes into account a cold start of the stores and the ground. Cost functions for the collector field, buffer store and duct store are used to establish a yearly cost of the solar part for each of the systems. This yearly cost takes into account the investment and operational costs. Combined with the average yearly solar heat simulated with the dynamic programme, the cost of the solar energy delivered in the distribution network, called **solar cost**, is calculated for a variety of systems. They all satisfy a known fraction of the annual heat demand, which is called the **solar fraction**. A cluster of points is obtained when the solar cost is plotted in relation to the solar fraction. The lower points provide optimal system designs in relation to the solar fraction. The whole procedure is repeated for different annual heating requirements, so that the influence of the annual heat quantity and the heat distribution temperature levels can be explored.

System layout
The system is formed by a solar collector array, a short-term water store (buffer store), a long-term borehole thermal energy store (duct store), an auxiliary heater (boiler) and a heat distribution network to provide heat and hot water to consumers. The system layout is shown in figure 5.4.

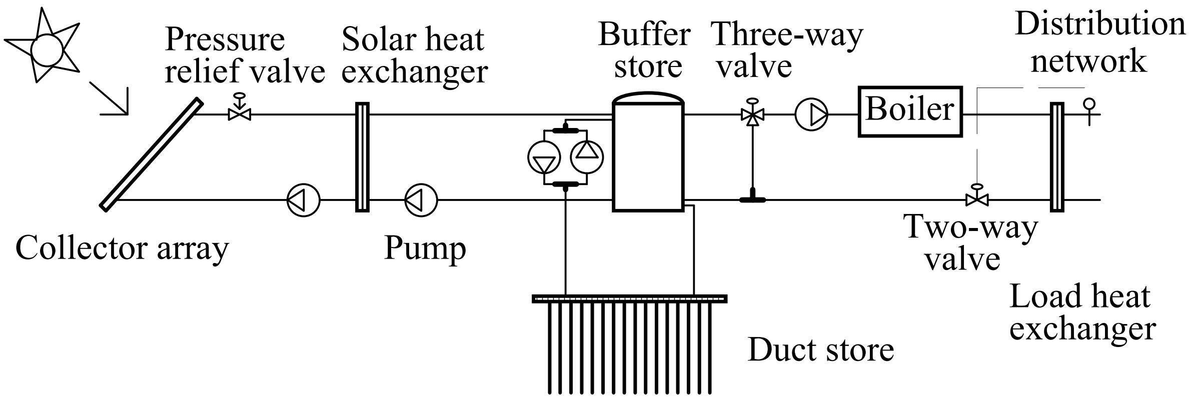


Figure 5.4 Analysed system layout of the CSHPSS system with a BTES.

Weather conditions and heat demand
The meteorological conditions are chosen to correspond to typical Swiss plateau conditions (north of the Alps). Various heat demands are defined. They depend on the quantity of annual energy (from 500 to 5'000 MWh per year), the forward distribution temperatures (medium: 50 to 55 °C and low: 25 to 30 °C) and the proportion of annual energy used for hot water (hw) and space heating (sh).

Simulation results
Simulation showed that a BTES can be justified from an economical point of view for solar fraction greater than 50%. For a smaller solar fraction, a system with only a water store has a cheaper solar cost.

Large scale solar heating is an important factor for cost reduction (large storage), together with low temperature heat distribution. In figure 5.5, solar cost for a solar fraction of 70% are shown in relation to the annual heat demand and for different heat distribution temperature.

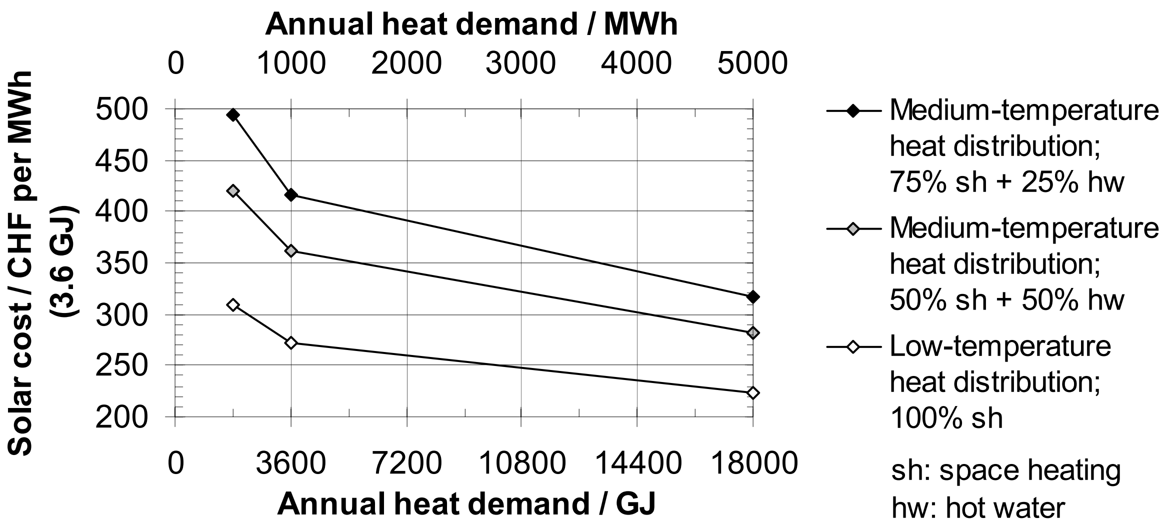


Figure 5.5 Influence of the load type on the solar cost. A large annual heat demand and a low temperature heat distribution are major factors for a significant solar cost reduction.

A ground heat storage volume greater than 20'000 m^3 requires an annual thermal energy demand of at least 1'000 MWh/year. It corresponds to about 100 – 150 low energy houses (annual heat demand for space heating and domestic hot water of 50 – 80 kWh/m^{2}y, and a heated floor area of 100 – 150 m^2 per house). A system designed for a solar fraction of 70% has following sizing values:

- **Collector area**
  2 – 3 m^2/(MWh/y) of annual heat demand, or about 20 – 30 m^2 per house;

- **Buffer water tank volume**
  110 – 130 litre/m^2 of collector area;

- **Borehole thermal energy store volume**
  11 – 13 m^3/m^2 of collector area with low heat distribution (space heating only);
  6 – 8 m^3/m^2 of collector area with medium heat distribution and 75% sh + 25% hw;
  4 – 6 m^3/m^2 of collector area with medium heat distribution and 50% sh + 50% hw;

- **Borehole spacing**
  2.3 – 2.7 m (ground thermal conductivity of 2.5 W/mK).

- **Ground storage shape factor**
  ~ 2 (ratio store vertical extension over store diameter).

System thermal behaviour
In figure 5.6, a system monthly heat balance is shown.

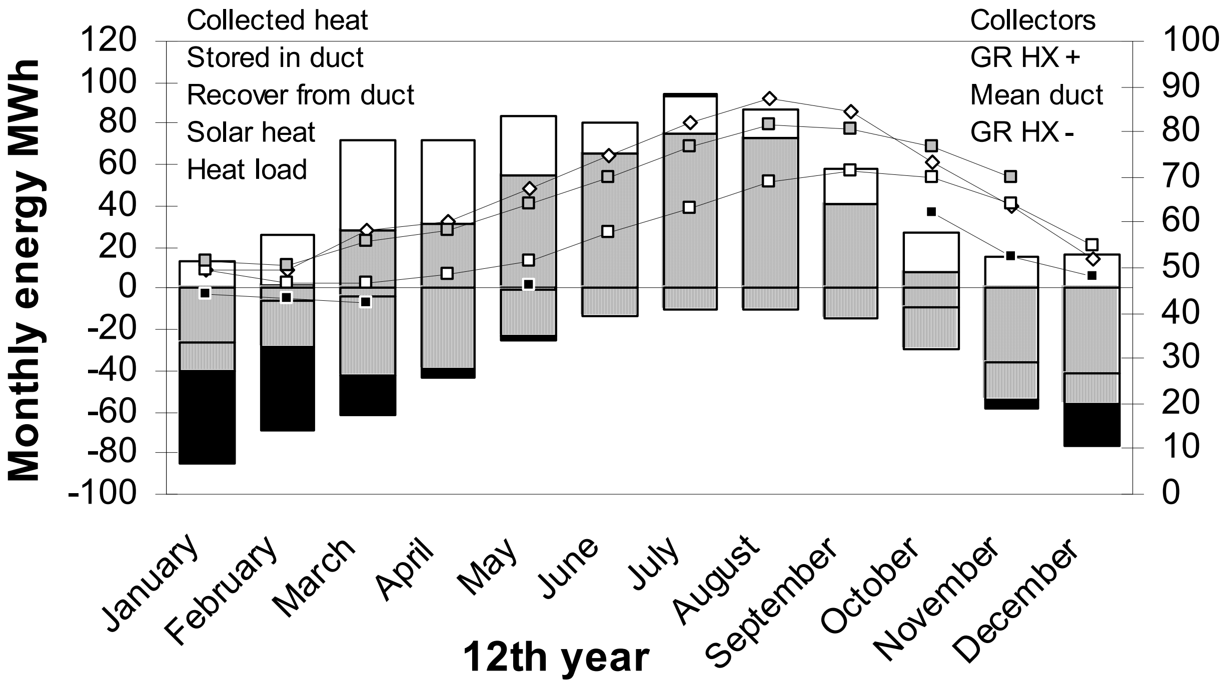


Figure 5.6 Monthly heat balance for the system designed for the small heat load, 500 MWh/y, 75% sh + 25% dhw and a medium temperature distribution. The solar fraction is 70%.

Figure 5.6 clearly show the task of each store: the short-term heat storage requirements are mainly covered by the buffer store, whereas the borehole heat store is principally used for seasonal heat storage requirements. The temperature loss between the temperature level in the collector array and the mean temperature of the duct store is mostly significant when the duct store is loaded. For all the optimal simulated systems, a monthly loss of around 20 K is calculated, of which about 15 K is caused by the ground heat exchanger. Another 5 to 10 K is lost when heat is recovered from the ground storage.

System control strategy
The thermal performances of the systems that have a solar fraction of 70% reveal two main operation modes: a “summer” mode, observed from early June until late September, and a “winter” mode, from early December until late February. During the “summer” mode, heat always flows from the buffer store to the duct store, and inversely from the duct store to the buffer store during the “winter” mode. This also confirms the fact that the duct store is principally used for seasonal heat storage requirements. There are also two transition periods where both modes are present; (in the spring from March to May and during the autumn from October to November). These modes can be deduced from figure 5.7, which shows the temperature evolution of the two stores for the twelfth-operation year. During these transition periods, the operation strategy of the system may have some influence on the thermal performances of the system. Is it better to keep as much heat as possible in the buffer store, so that the heat load can be met by solar heat as often as possible, or to transfer heat to the duct store as soon as possible, in order to enhance the efficiency of the collector array? Depending on the weather forecast, each alternative has its advantages. If the next day is sunny, it might be better to load the duct store in order to make “room” in the buffer store for the solar gains to come. On the other hand, if the next day is cloudy, it might be preferable to keep the heat in the buffer store to have it available for the heat load. It should be remembered that once a heat quantity is transferred to the duct store, it probably will not be available to the heat load the next day, due to the large temperature losses caused by the ground heat exchanger.

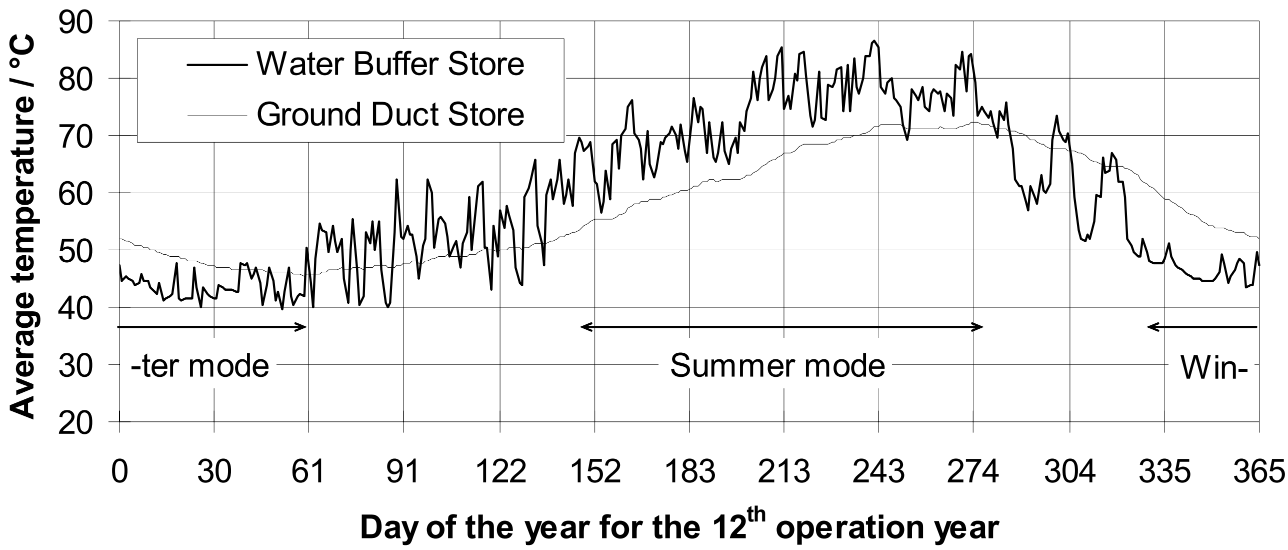


Figure 5.7 Evolution of the mean buffer and borehole heat store temperatures for the 12th year of operation. System designed for the small heat load, 500 MWh/y, 75% sh + 25% dhw and a medium temperature distribution. The solar fraction is 70%.

The optimisation of the system control can be achieved with the help of a new generation of simulation tools applied to solar heating with seasonal heat storage (Rüdiger, 1997). Numerical optimisation procedures are integrated together with the dynamic models describing the system. An optimum system design is directly calculated, given the objectives (for example the solar fraction), the optimisation criteria and the constraints on the variables. A multi-parameter optimisation of a system can be realised in one run. Such a simulation tool cannot yet provide as detailed simulations as TRNSYS can, but it has successfully been used for the simulation of a central solar heating plant with a water tank in Särö, Sweden (Rüdiger, 1997). The methodology has been further developed to simulate a CSHPSS-system with a ground heat storage, including a buffer store in the system design (Rüdiger and Hellström, 1997). Preliminary simulations have shown that relative to a simple system control that would transfer heat between the buffer and the duct store as soon as it is possible, an optimum system control would increase the annual solar heat of a typical solar heating system by about 10%. This optimum control is established using knowledge of the weather in the near future, so that the best decision can be anticipated at the right time. So far, it has been difficult to reproduce the optimum system control with a simple control criterion. However, the optimum system control suggests that the buffer store should cover the short-term heat storage requirement as much as possible, while the mean temperature level of the buffer store is kept as low as possible.