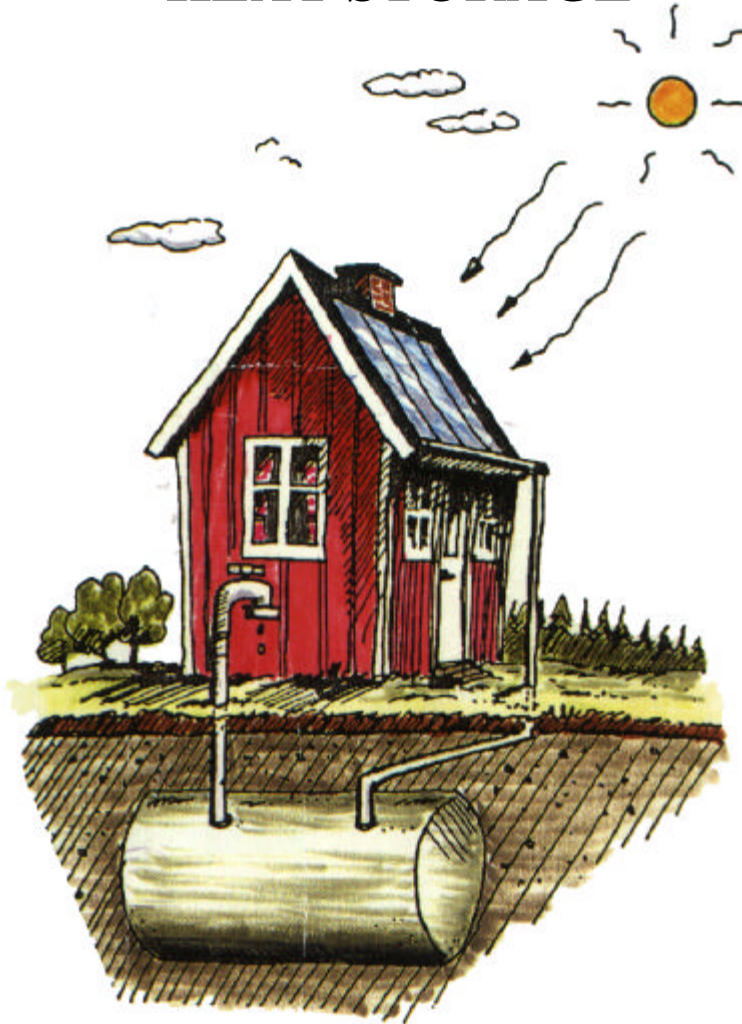


SOLAR ENERGY AND HEAT STORAGE



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SOLAR ENERGY AND HEAT STORAGE

Third Edition

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This textbook was translated and improved by
Environment Canada.

Preface and foreword to the Third Edition

This book was originally published at Luleå University of Technology in 1987 fulfilling a need within Sweden to disseminate the techniques of alternative heating using solar energy in association with energy storage.

In the year 2000 it was realised that the book required updating. This second edition contains a number of changes that bring it up to date, and at the same time, it was made accessible to a much wider audience by being translated into English.

This work for the third edition has been supported by: Environment Canada.

There has been a number of significant changes since the publication of the original edition:

- CFC refrigerants have been phased out.
- Photo-voltaic technology has made a leap forward with the miniaturisation of cells.
- The costs per unit of energy for photovoltaic, solar and wind generation devices has been reduced through advances in technology and mass production.
- The costs of traditional fossil fuels such as natural gas and oil have continued to be variable, but increases have recently been dramatic.
- Deregulation of the electrical generation industry is occurring in a number of countries, with widely variable effects.
- The concept of global warming and climate change resulting from the release of greenhouse gases such as carbon dioxide into the atmosphere is now generally accepted. The concept is fast turning into reality with the consequence that serious efforts are now being made to discourage the use of fossil fuels and encourage alternative, natural ways of creating usable energy from the sun and wind.
- Policy initiatives by many progressive governments have and will continue to aid in the uptake of this new technology, by fostering innovative techniques and making alternative energy systems more economically attractive.

The book contains not only a theoretical treatment of the physics of both heat collection and storage, but also a thorough description of the many alternative methods and materials used for these two processes. In addition, the book is littered with actual examples, and worked and unworked problems that illustrate the practicalities of the subjects covered. / Bo Nordell

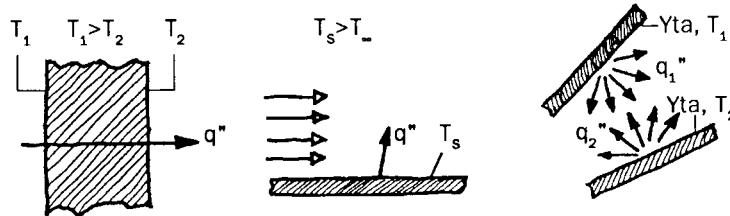
Heat Transfer

INTRODUCTION

The purpose of this section is to increase the understanding concerning heat transfer within media and between different media's. The intention with the presentation is to focus on processes of heat exchange that are important for solar energy, natural energy, and heat storage systems. The presentation is thus not intended to cover the whole field of heat exchange. The interested reader is advised to review other literature for a more thorough covering.

Heat transfer can occur in three different ways, namely through conduction, convection and radiation. The heat exchange is caused by temperature differences within the medium (conduction) or between media (convection, radiation). The different mechanisms for heat transfer are illustrated in Figure 1.

Figure 1. Illustration of heat transfer. Conduction, convection and radiation. (From Incropera et al 1981)



Thermal conduction is the transfer of kinetic energy between molecules. Convection is a fluid motion in liquids or gas, where heat is transported. Convection is therefore macroscopic in its course and depends on several factors such as velocity field, viscosity, specific heat and heat capacity. Heat transfer as radiation takes place through electromagnetic undulating motion.

THERMAL CONDUCTION

Thermal conductivity is a material property which tells how well the material conducts heat, see Figure 2.

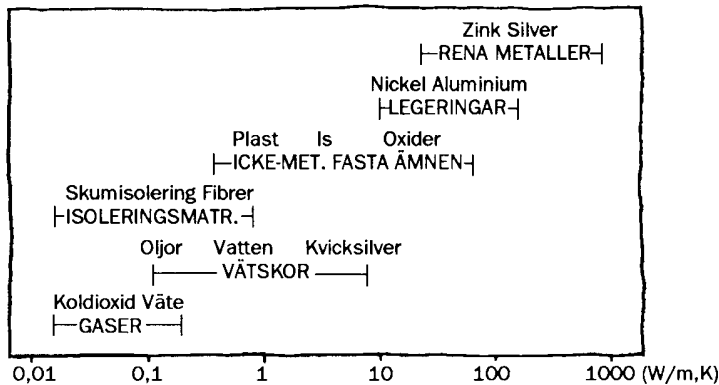


Figure 2. Thermal conductivity (W/m, K) for gases, liquids and solid materials. (From Incropera F et al 1981)

The figure shows that gases have the lowest thermal conductivity, while metals have the best.

The general thermal conduction equation may be written;

$$\frac{d^2 T}{dx^2} + \frac{d^2 T}{dy^2} + \frac{d^2 T}{dz^2} = \frac{1}{a} \cdot \frac{dT}{dt}$$

Equation (1)

where $a = \lambda/c = \text{temperature diffusivity}$

$T = \text{temperature (K)}$

$c = \text{volumetric heat capacity (J/m}^3, \text{K)}$

$\lambda = \text{thermal conductivity (W/m, K)}$

$t = \text{time}$

This equation is therefore the basis for thermal conduction calculations. With stationary problems, the time derivative is zero and the equation gets more simple. A simple stationary case is the thermal conduction through a flat plate. The differential equation describing this process may be written;

$$\frac{d^2T}{dx^2} = 0$$

Equation (2)

The solution to this equation is simple and with limits it is written according to equation (3);

$$Q = A \cdot \frac{\lambda}{d} \cdot (T_1 - T_2)$$

Equation (3)

where Q = heat energy rate (W)

A = area (m²)

λ = thermal conductivity (W/m, K)

d = thickness of the plate (m)

T₁ and T₂ = the temperatures on the plates surfaces

During non-stationary (transient) cases, the time derivative of the general equation (1) must be included. The solution to this equation will be complicated in most cases. To solve the problem one often uses numerical methods, such as the Finite-Difference method (FDM). Transient cases follow for instance a sudden change of a bodies surface temperature, when the temperature varies periodically, or with a sudden supply of heat etc.

CONVECTION

A distinction is being made between natural and forced convection. Natural convection occurs with density differences in the media, who cause circulation. An example of natural convection is the circulation that takes place in a room because of the radiators heating of the air. Forced convection is a result of the outer forces being of such magnitude that the influence of density forces is negligible, such as in pipe flow.

To calculate heat transfer through convection a heat transfer coefficient, α_k , is defined according to equation (4).

$$Q = \mathbf{a}_k \cdot (T_s - T_\infty) \cdot A$$

Equation (4)

where Q = heat energy rate (W)

α_k = heat transfer coefficient (W/m², K)

T_s = surface temperature (°C)

T_∞ = free flow temperature (°C)

A = area (m²)

The heat transfer coefficient is not a material property, as the thermal conductivity is, but a coefficient that depends on the properties of the fluid media and the flow conditions. Low values are typical for gases and minor flow movements, 1-10 W/m², K. High values are typical for liquids flowing at high velocities or when boiling, 1000-50 000 W/m², K.

To decide the heat transfer coefficient, so called dimensionless numbers are used. The Nusselt-number, Nu, is a dimensionless number defined as;

$$Nu = \frac{\mathbf{a}_k \cdot L}{l}$$

Equation (5)

where α_k = heat transfer coefficient (W/m², K)

L = characteristic length (m)

λ = thermal conductivity (W/m, K)

By determining the Nusselt-number the heat transfer coefficient, α_k , may also be determined. It has been found that the Nusselt-number is a function of the Reynolds-number (Re), the Prandtl-number (Pr) and the Grashoff-number (Gr), according to equation 6;

$$\text{Nu} = \text{func} (\text{Re}, \text{Pr}, \text{Gr})$$

Equation (6)

The different numbers are defined according to (7), (8) and (9);

$$\text{Re} = \frac{u \cdot L}{\nu}$$

Equation (7)

where u = average velocity (m/s)

ν = kinematic viscosity (m²/s)

$$\text{Pr} = \frac{\mu}{\lambda} = \frac{\mu}{\rho \cdot c} = \frac{c_f \cdot m}{\lambda}$$

Equation (8)

where ν = kinematic viscosity (m²/s)

λ = thermal diffusivity (m²/s)

c = volumetric heat capacity (J/m³, K)

μ = dynamic viscosity (kg/m.s)

c_f = heat capacity (J/kg, K)

$$\text{Gr} = \frac{g \cdot \beta (T_s - T_\infty) \cdot L^3}{\nu^2}$$

Equation (9)

where β = the volumetric thermal expansion coefficient.

For gases $\beta = 1/T$, where T is the absolute temperature (K). The Reynolds-number describes the velocity field, the Prandtl-number the temperature field, and, finally, the Grashoff number describes the density field.

Therefore the Nusselt-number is a function of the Re-number and the Pr-number at forced convection, equation (10), and a function of the Gr-number and the Pr-number at natural convection, equation (11);

$$Nu = func(Re, Pr) \quad \text{forced convection}$$

Equation (10)

$$Nu = func(Gr, Pr) \quad \text{natural convection}$$

Equation (11)

These numbers have been correlated to each other through theoretical solutions and experiments, for different geometry's and flow conditions. Below, as examples, a few of these relations are expressed.

For turbulent flow in pipes applies;

$$Nu = 0,023 \cdot Re^{0,80} \cdot Pr^{0,33}$$

Equation (12)

Natural convection at a vertical wall;

$$Nu = 0,13 \cdot (Gr \cdot Pr)^{1/3} \quad \text{for } Gr \cdot Pr > 10^8$$

Equation (13)

Laminar flow over a flat plate;

$$Nu = 0,332 \cdot Re^{0,5} \cdot Pr^{0,33}$$

Equation (14)

Several models are offered in the literature to calculate the convective heat transfer from solar collectors. According to McAdams (1954) the heat transfer coefficient can be related to the wind velocity, u , as in equation (15);

$$a_k = 5,7 + 3,8u$$

Equation (15)

where u is the wind velocity in m/s.

EXAMPLE 1

Determine the heat transfer coefficient in a pipe with flowing water, where the pipe diameter is 0,01 m, the water

temperature is 50 °C, and the flow velocity is 2,5 m/s. The water kinematic viscosity is $0,556 \cdot 10^{-6} \text{ m}^2/\text{s}$.

The Reynolds-number is determined (7)

$$\text{Re} = \frac{2,5 \cdot 0,01}{0,556 \cdot 10^{-6}} = 44964 > 2300 \quad \text{i.e. turbulent flow.}$$

Equation (12) can be used

$$\text{Pr}^{50^\circ\text{C}} = 3,58 \text{ (from table)}$$

$$\text{Nu} = 0,023 \cdot (44\,964)^{0,80} \cdot 3,58^{0,33} = 184,8$$

$$\text{Nu} = \frac{\mathbf{a}_k \cdot L}{l}$$

$$\lambda^{50^\circ\text{C}} = 0,647 \text{ W/m, K (from table)} \Rightarrow$$

$$\mathbf{a}_k = \frac{184,8 \cdot 0,647}{0,01} = 11959 \text{ W/m}^2, ^\circ\text{C}.$$

The heat transfer coefficient between the water and the pipe wall will therefore be approximately 12 000 W/m², °C in this case.

RADIATION

For heat transfer by radiation, the energy is transported through electromagnetic undulating motion. The total thermal radiation from an ideal radiation body, the black body, Stefan Boltzmanns law applies;

$$\mathbf{f} = \mathbf{s} \cdot A \cdot T^4$$

Equation (16)

where ϕ = total radiation effect (W)
 σ = the Stefan Boltzmanns constant $5,67 \cdot 10^{-8}$ W/m², K
 A = the body's area (m²)
 T = the absolute temperature (K)

According to Stefan Boltzmanns law, the emitted energy from a black body will depend on the 4th power of the temperature. The black body is an ideal surface that absorbs and emits all radiation. For practical purposes all surfaces are "gray" surfaces, i.e. not ideal. Stefan Boltzmanns law has been modified to deal with gray surfaces (17);

$$f = e \cdot s \cdot A \cdot T^4$$

Equation (17)

where ε is the so called emission number for the surface. The emission number is a material constant similar to the thermal conductivity and is tabled for different types of surfaces, see table 1.

| SURFACE | ε |
|--------------------------|---------------|
| Silver, polished | 0,01-0,02 |
| Steel, sheeting's | 0,65-0,70 |
| Oil colors | 0,90-0,97 |
| Bricks | 0,93-0,97 |
| Tiles | 0,87-0,96 |
| Water, ice, frost | 0,95-0,98 |
| "Absolute black surface" | 1,0 |

Table 1. Examples of emission numbers for some different surfaces.

For calculations it is usually the heat exchange between different bodies that is of interest, not the radiation effect itself. This heat exchange depends, in addition to the surface temperatures, also on the geometric relationship between the surfaces. The heat flow Q (W) between two surfaces may be expressed as;

$$Q = \mathbf{s} \cdot F_{12} \cdot A(T_1^4 - T_2^4)$$

Equation (18)

where F_{12} = the view factor which is a function of the geometric arrangement and the emission numbers for the two surfaces

T_1 and T_2 = the absolute temperature on the respective surface (K)

For radiation between two large parallel surfaces (19) applies;

$$F_{12} = \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1}$$

Equation (19)

where ϵ_1 and ϵ_2 = the emission numbers for each surface respectively.

For radiation from the surface A_1 , fully surrounded by the surface A_2 , applies (20);

$$F_{12} = \frac{1}{\frac{1}{\epsilon_1} + \frac{A_1}{A_2} \left(\frac{1}{\epsilon_2} - 1 \right)}$$

Equation (20)

When the surrounding surface, A_2 , is much larger than A_1 , equation (20) is simplified to (21);

$$F_{12} = e_1$$

Equation (21)

The latter (21) can be used for instance for the calculation of the radiation exchange between a solar collector and the sky. To attain a linear equation, similar to the one for heat transfer through convection, a thought heat transfer coefficient, due to radiation, is often defined according to (22);

$$a_s \cdot (T_1 - T_\infty) = F_{12} \cdot s \cdot (T_1^4 - T_2^4)$$

Equation (22)

where α_s = heat transfer coefficient due to radiation
(W/m², K)
 T_∞ = absolute temperature in the ambient
medium (K)

The heat transfer coefficient due to radiation may therefore be written;

$$a_s = \frac{F_{12} \cdot s \cdot (T_1^4 - T_2^4)}{(T_1 - T_\infty)}$$

Equation (23)

EXAMPLE 2

Determine the heat transfer coefficient, due to radiation, between a solar collector and the sky if; Solar collector temperature = 60 °C; emission number, $\varepsilon = 0,20$; air temperature = 30 °C.

Equation (23) is used;

$$a_s = \frac{F_{12} \cdot s \cdot (T_1^4 - T_2^4)}{(T_1 - T_\infty)}$$

Equation (23)

where $F_{12} = \varepsilon = 0,20$

$$T_1 = 60 + 273 = 333 \text{ K}$$

$$T_\infty = 30 + 273 = 303 \text{ K}$$

T_2 = the temperature of the sky

Several models are available in the literature to calculate the temperature of the sky. Swinbank (1963) related the sky temperature, T_{sky} , to the air temperature, T_a , according to equation (24);

$$T_{\text{sky}} = 0,0552 \cdot T_a^{1,5}$$

Equation (24)

In our example, $T_a = T_\infty = 303 \text{ K}$ and $T_{\text{sky}} = T_2$ becomes (eq. 24) 291 K. Inserted into equation (23) one gets;

$$a_s = \frac{0,20 \cdot 5,67 \cdot 10^{-8} (333^4 - 291^4)}{333 - 303} = 1,9 \text{ W/m}^2, \text{ K}$$

The heat transfer coefficient, due to radiation exchange between the sky and the solar collector, therefore is $1,9 \text{ W/m}^2, \text{ K}$.

EXAMPLE 3

Determine the total heat loss from the solar collector, in Example 2, if the wind velocity above it is 2 m/s.

To determine the total heat loss, also the heat transfer coefficient due to convection must be calculated. According to (15) this is;

$$\alpha_k = 5,7 + 3,8u = 13,3 \text{ W/m}^2, \text{ K}$$

The total heat loss then becomes;

$$Q = \alpha_{\text{tot}} \cdot (T_1 - T_\infty) = (\alpha_s + \alpha_k) \cdot (T_1 - T_\infty)$$

$$Q = (1,9 + 13,3) \cdot 30 = 456 \text{ W/m}^2$$

SUMMARY

It is of great importance to differ between the three ways of heat transport.

Thermal conduction is a transfer of kinetic energy from molecule to molecule. The thermal conductivity informs us how inclined a material is to conduct heat. To some stationary thermal conduction problems, simple analytical solutions to the thermal conduction equation are available. For transient problems, numerical solutions are often necessary to solve the thermal conduction equation.

Convection is a fluid motion in a liquid or gas through which heat is transported. Convection is a macroscopic event and depends on several factors, such as velocity field, viscosity, thermal conductivity etc. The heat transfer coefficient tells us how much heat that is being transferred, and it depends on the mediums properties and the flow conditions. The heat transfer coefficient has been determined analytically and experimentally by correlating dimensionless numbers, for different flow conditions and media properties.

Radiation is a electromagnetic undulating motion. The total thermal radiation from an ideal radiation body, the black body, is described by Stefan Boltzmanns law. This law states that the energy being emitted from a black body is proportional to the 4th power of the absolute temperature.

Solar Energy

INTRODUCTION

The sun was an obvious source of energy also to our forefathers. They worshipped the sun which gave them food and warmth and which let the winter pass into summer. It was early understood how to use the solar energy in an active way. The solar house of Socrates counts among the first known attempts of a practical use of solar energy, see figure 3.

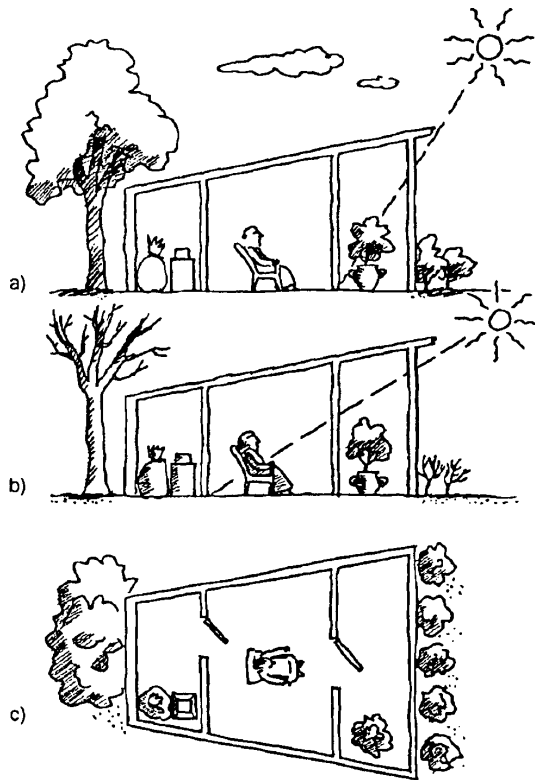


Figure 3. The solar house of Socrates (400 BC) a) summer b) winter c) lay-out (After Peterson et al 1985)

The house was oriented and constructed to receive a large amount of solar radiation during spring and fall. This enabled solar heating of the house even when the sun was too weak to warm the outdoor air. During summer, with the

sun high in the sky, the solar radiation to the house was reduced due to the protruding roof.

Socrates solar house belongs to the so called passive solar heat systems, i.e. a system where the solar energy is utilized without any special machine or apparatus. The passive solar systems, who are mainly studied by architects, will not be further treated in this book. In many cases they complement the active systems to create energy conserving and environmentally friendly heat systems.

The two largest groups of active solar heating systems are concentrating and flat-plate collectors. Designs of these systems emerged already in the 18th and 19th century. 1986/87 there were approximately 65 000 m² of solar collector surface in Sweden. Those collectors delivered 0,02 TWh of heat per year. The difficulties of extracting energy from the sun depends partly on the low degree of effect density - in Sweden the annual average is 100 W/m² - and partly because of the imbalance in supply during summer, and need during winter. Due to this imbalance storage of solar energy would be beneficial, to increase the potential as an all-round energy source.

The scientific contributions, in the field of thermal solar energy, has focused on the reduction of costs for solar collector systems and development of heat storage techniques. In Sweden (1999) research on solar energy is mainly taking place at the following institutions:

- The University of Borlänge, SERC (Solar Energy Research Center)
- The Älvkarleby laboratory. The power production company. Vattenfalls laboratory in Älvkarleby, Gävle.
- The Royal Institute of Technology, Stockholm. Institutionen för Uppvärmnings- och Ventilationsteknik.

- The University of Chalmers, Gothenburg. Institutionen för Installationsteknik.
- SP, Statens provningsanstalt, Borås
- Studsvik Energiteknik AB, Nyköping
- SMHI, Norrköping

ATMOSPHERIC RADIATION PROCESSES

The sun is a giant fusion reactor. The temperature in the center is $8 \cdot 10^6$ - $40 \cdot 10^6$ K. The effective black body temperature at the surface is 5762 K. To the main part the sun consists of helium and hydrogen. It has been calculated that 90% of the energy is generated in the center of the sun (0 - 0,23 of the radius) which contains 40% of the suns mass, see figure 4.

The total emitted radiation from the sun is $3,9 \cdot 10^{26}$ W.

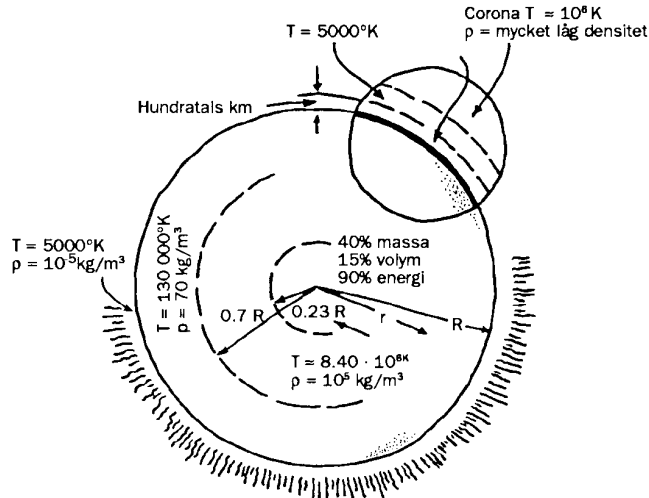


Figure 4. Structure of the sun (From Duffie J.A. et al 1980)

The solar radiation is an electromagnetic undulating motion divided into six groups with increasing wave lengths, as described below. Also see figure 5.

- Gamma radiation
- ultra-violet radiation
- visible radiation
- infrared radiation
- radar waves
- radio waves (short-wave, long-wave)

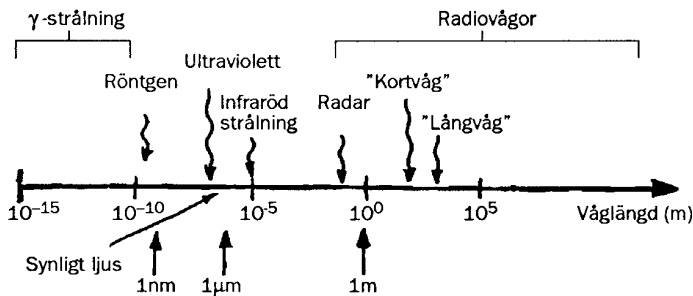


Figure 5.
Electromagnetic radiation divided after wave length.

The most essential parts, from a solar energy point of view, is the visible and the infrared radiation (the thermal radiation). In figure 6 the spectral effect distribution for these wave lengths are shown, outside the atmosphere and on the earth's surface. The span between the curves, in figure 6, represents the part of the radiation that is being reflected or absorbed in the atmosphere a clear day.

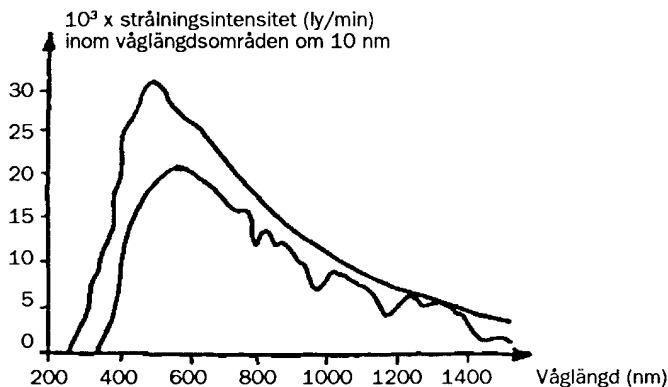


Figure 6. The solar radiation's spectral effect distribution a) outside the atmosphere (upper curve) and b) at the earth surface (lower curve). $1 \text{ ly} = 41,8 \text{ kJ/m}^2$. (After Peterson F. et al 1985)

Figure 7 shows, roughly, the amount of radiation that is being reflected or absorbed in the atmosphere a cloudy and a clear day.

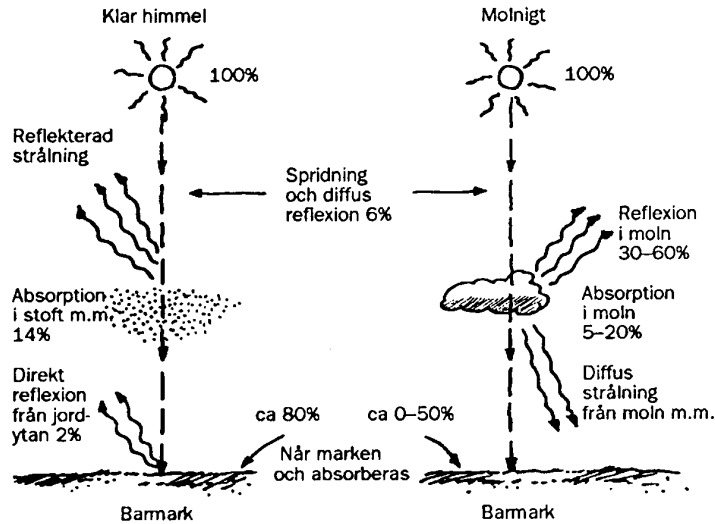


Figure 7. The influence of the atmosphere on solar radiation on the earth surface (After Peterson F. et al 1985).

The air molecules and water vapor will cause scattering and absorption of radiation. The scattering is different for different wavelengths of the light and also depends on the amount of particles the radiation has to pass through. It is this mechanism that explains the blue color of the sky. The diffuse reflection, on the other hand, is the same for all wavelengths and occurs in water drops in the clouds and on larger particles.

The **solar constant** is defined as the amount of energy per time unit that passes a unit of surface placed perpendicular to the direction of radiation, just outside the earth atmosphere, when the earth is on its average distance from the sun. This constant has a value of 1367 W/m^2 . In meteorology, the solar radiation on the ground surface is divided into the components shown in figure 8.

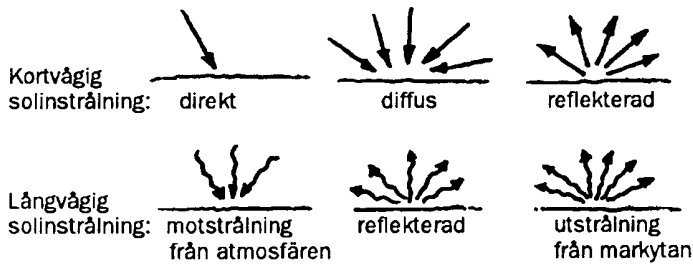


Figure 8. Radiation processes on the ground surface (After Duffie J.A. et al 1980).

Short-wave radiation is the radiation originating from the sun, at a temperature of 5800 K in the wavelength section 0,3-3,0 μm .

Beam radiation is the short-wave radiation that reaches the ground surface without changes in direction (except for diffraction when entering the atmosphere). Diffuse radiation is the part of the short-wave radiation that reaches the ground surface after being scattered in the atmosphere, which means that its direction has changed. The sum of these two, beam and diffuse radiation, is called global solar radiation. The part of the global radiation that is reflected on the ground surface, back into the atmosphere, is called reflected solar radiation.

In addition to the short-wave radiation on the ground surface, there is also a long-wave back-radiation from the atmosphere. As with short-wave radiation, also some part of the long-wave radiation is reflected back to the atmosphere. Finally, a long-wave radiation from the ground surface takes place, in addition to the reflected part. The total radiation to the ground surface is then the sum of all wavelengths, i.e. long-wave plus short-wave radiation.

The most common instruments to measure radiation are; (also see figure 9)

Pyrheliometer (*beam radiation*). The detector is mounted at the bottom of a black-painted tube that is being pointed at the sun at all times.

Pyranometer (*global or diffuse radiation*). The detector consists of a black-painted plate surrounded by a white rim. The black surface absorbs more light and will therefore become warmer than the white rim. The temperature difference depends on the intensity of the (global) solar radiation and is measured with thermo-elements. With a shadow ring, the beam radiation can be excluded and the pyranometer can then be used to measure only diffuse radiation.

Pyrgeometer (*long-wave radiation*)

Pyrradiometer (*total radiation*).

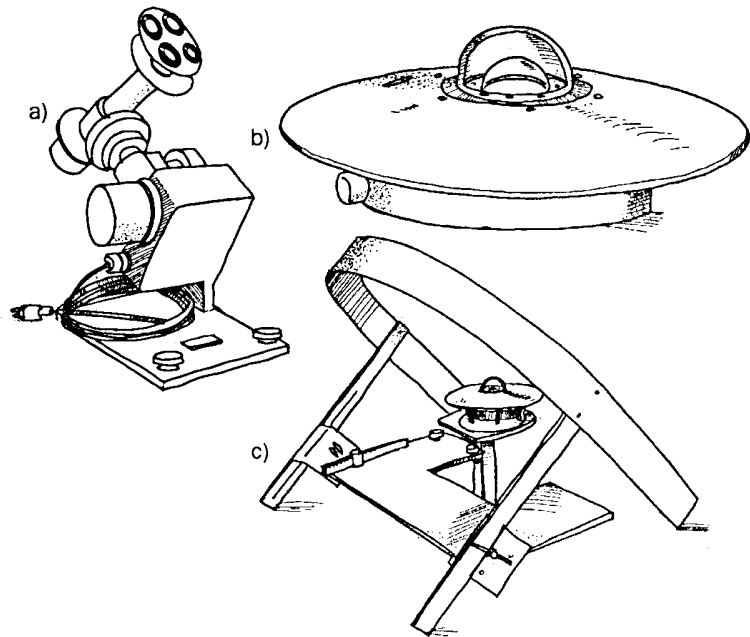


Figure 9. Picture of a sample of instruments to measure radiation
a) Pyrliometer b) Pyranometer for global radiation c) Pyranometer for diffuse radiation (after Duffie J.A. 1980)

The annual global solar radiation and the annual number of solar hours are presented in figure 10. The number of solar hours is at a maximum at the North-East coast while the annual global radiation is larger in Southern Sweden, in the landscape of Skåne. The difference between North and South is explained by the difference in solar height.

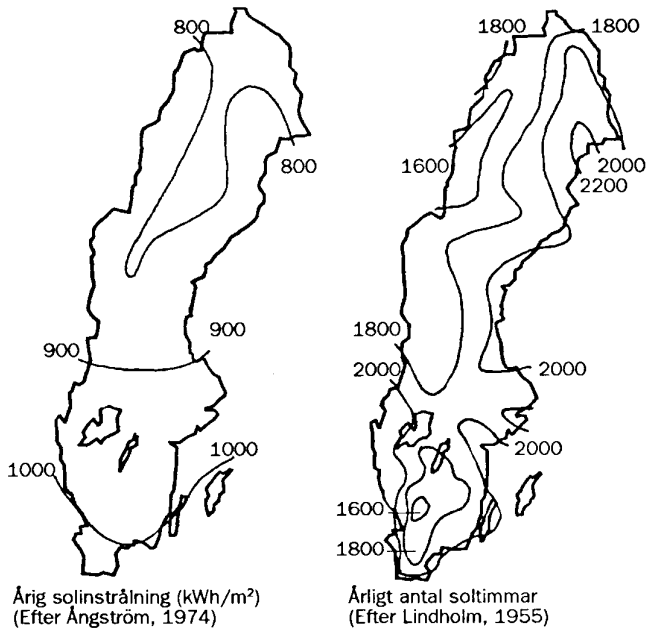


Figure 10. Annual solar radiation and number of solar hours in Sweden.

SOLAR ENERGY

Solar radiation may be converted into usable energy-forms in three principally different ways:

- Photothermal transformation to heat through absorption, e.g. on black surfaces
- Photovoltaic transformation to electrical energy with, for instance, solar cells.
- Photochemically bound energy through photo-synthesis

Photothermal transformation of solar energy to heat is achieved through absorption of radiation on, for instance, black surfaces. The thermal energy can be used for heating or, in a second step, to produce electrical energy.

The efficiency of photovoltaic transformation of solar energy, to electrical energy, is usually between 5-20%.

In nature, plants utilize solar energy to form biologically complex molecules. The efficiency of these processes is below 5%. The energy conversion is, however, taking place in several steps that, individually, has a much larger efficiency.

Photothermal transformation of solar energy

Photothermal transformation of solar energy is accomplished with what we call solar collectors. In a solar collector the incoming radiation is absorbed on an absorber surface and transformed to heat. The heat from the absorber is then conducted to a heat carrier (liquid or gas), usually called a *brine*. The design of the solar collector determines the temperature range within which it can work.

The two dominating groups of solar collectors are concentrating- and flat collectors. In concentrating collectors, the radiation is concentrated to a higher intensity per surface unit. Flat-plate collectors are significantly more simple in design, cheaper and well suited for lower temperature systems, e.g. water heating, house warming etc.

In this chapter, the emphasis is on photothermal transformation of solar energy.

Photovoltaic conversion of solar energy

What we commonly name as solar cells is a device for photovoltaic conversion of solar energy to electricity. The most common solar cells of today are manufactured from single-crystalline or poly-crystalline silicon. A solar cell absorbs light and converts this to electrical energy. Solar cells are made of semi-conductors. The energy of the light is, at the absorption, transferred to electrons in the material. The high-energy, excited, electrons strive to emit their energy-surplus and return to their normal energy level. This can be accomplished in three different ways; emission of light, heat-emission and electrical work. In a semi-conductor, a large part of the absorbed solar energy will be obtained as electrical energy. A solar cell contains a P-N-junction, i.e. semi-conductor silicon. Principally, a solar cell is built up of crystalline silicon as shown in figure 11.

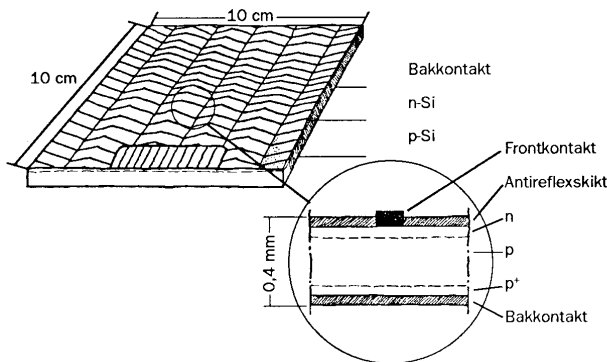


Figure 11. Basic silicon solar cell. (After Sigurd D et al, 1988)

The starting material is P-doped silicon. The surface facing the light has been doped to N-type, usually through thermal diffusion. To conduct the current out of the cell, metal contacts are attached on the front and the back. The front contact is designed as a finger-pattern, so that light can pass into the cell, between the "fingers".

In an illuminated solar cell, free electrons and holes are formed, negatively or positively charged. When the electrons move from P-Si to N-Si, and holes from N-Si to P-Si, a current will flow through a connected, outer electronic circuit. The, by the light, induced current is named photo-current.

The solar light must have a certain energy-level to be able to form free electrons and holes. This minimum energy-level is called the threshold-energy, or band-gap. The silicon cell has a band-gap of 1,1 eV.

The theoretical limit for the efficiency of a simple solar cell is 30%. This is due to the fact that the solar spectrum contains light with different energy-levels, of which some cannot be used efficiently of a semi-conductor with a given band-gap. Theoretical and experimental efficiencies of different solar cells, of varying materials, are presented in table 2.

Table 2. Theoretical maximums and experimentally obtained efficiencies for solar cells of varying materials. (After Sigurd D et al, 1988)

| Photovoltaic type | Efficiency | |
|---|---------------------|--------------------|
| | Theoretical Maximum | Experimental Value |
| | (%) | (%) |
| Crystalline Silica | | |
| • Commercial standard cells | 29 | 15 |
| • Laboratory cell in natural light | 29 | 22 |
| • Concentrating cell (130x) | 32 | 28 |
| AlGaAs / GaAs | | |
| • Commercial cells (space applications) | 31 | 19 |
| • Laboratory cell in natural light | 31 | 24 |
| • Concentrating cell (700x) | 34 | 26 |

The individual solar cells are usually of a size around 0,1 m and will give a voltage of approximately 0,4 V with an amperage of 3 A at full illumination (1000 W/m²). The solar cells are often connected to a module, see figure 10, consisting of several solar cells. The most usual modules based on crystalline silicon have 30-35 connected in series, giving a module-potential of 13-15 V.

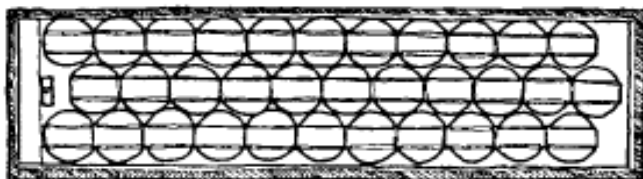


Figure 12. Solar cell panel (After Peterson et al. 1985)

The break through for solar cells came in the space-programs at the end of the 1960's. Since then solar cells have been, and still remain, the most important source of energy in space. Otherwise, the most important market for solar cells is for self-supported systems. Solar cells in lighthouses, radio stations, boats, caravans, summer-houses etc. belong to this group. The consumer market for watches, calculators, radios etc. has grown rapidly and may become the largest in the near future. In addition there are some net-connected smaller solar cell power plants. Table 3 shows a compilation of solar cell deliveries to different kinds of users.

| Sector | 1980 | 1981 | 1982 | 1983 | 1984 | 1985 | 1986 |
|-----------------------------|------|------|------|------|------|------|------|
| Self-sustaining systems | 2.6 | 3.9 | 4.3 | 7.4 | 10.2 | 11.0 | 12.9 |
| Consumer products | 0.4 | 0.9 | 1.5 | 3.6 | 4.9 | 6.7 | 8.9 |
| Net connected small systems | 0.3 | 0.6 | 0.8 | 1.2 | 1.0 | 0.9 | 0.9 |
| Photovoltaic power stations | 0.0 | 0.0 | 1.8 | 8.6 | 5.5 | 1.9 | 5.3 |

Table 3. Solar cell Deliveries to different kinds of users (MW). (After Sigurd et al. 1988).

WHY ARE SOLAR CELLS SO EXPENSIVE?

It is important that the silicon material is extremely pure, at least 99,999999% purity. Pollution's will absorb electrons and holes. Silicon solar cells are thick (300-400 μm), which leads to a large consumption of expensive, high-worthy silicon. Solar cells are also manufactured in a size of 0,1x0,1 m units, which thereafter are treated separately and connected to modules. All put together this leads to high production costs for solar cells. Today, solar cell technology is economically feasible only for self supporting systems. If solar cells were to contribute significantly to the power supply, in the industrialized world, production cost would have to be reduced. It has been calculated that the production costs would have to be around 50 $\$/\text{m}^2$ to make solar cells competitive. The price today (1991) is around 550 $\$/\text{m}^2$. Since the beginning of the 1980 the price has, however, decreased by a factor of 5.

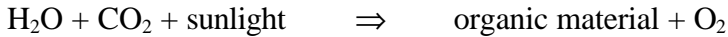
A new generation of solar cells, based on thin-film technology, (app. 1 μm) is under development. This would reduce the consumption of material. The manufacturing process is also new, since the thin-film cells are manufactured as a coating on different materials. Thereby will module-manufacturing be cheaper and more convenient, thus eliminate the small units. The efficiency of thin film cells is today (1991) around 7% and it is a technology with large future potential.

Photochemical conversion of solar energy

In nature, plants utilize solar energy to form biologically high worthy molecules through photosynthesis. The efficiency of this process is below 5%, sometimes only a few tenths of a percent.

A photochemical reaction is a chemical reaction with the help of light.

chlorophyll



The nature's photosynthesis serves many other purposes besides energy storage. It is also the starting point for building high worthy chemical compounds, essential for the life processes. Biomass is one of man's oldest sources of energy. To raise, for instance, forests for fuel production is not new.

In Sweden, the term "Energiskog" (Energy forest) is used to describe the cultivation of trees as an energy resource. Energy forests with a short span of rotation (1-5 years) are, for instance, alder, willow and poplar. The growth is enhanced with synthetic fertilizer, insecticides and earth treatment and may reach 40 tons of dry fuel per hectare (100x100 m) annually.

The photochemical conversion of solar energy to biomass is not further treated in this text.

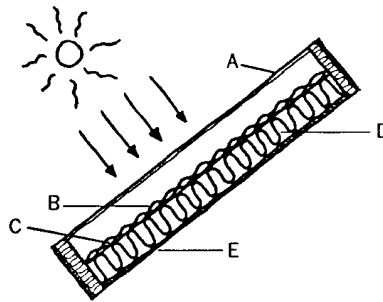
SOLAR COLLECTORS

Flat-plate collectors

A flat-plate thermal solar collector contains an absorber plate that absorbs solar radiant energy and transforms it into heat. The absorber plate consists of a thin plate, often painted black, that contains a system of small channels. In the channel system a heat carrier, gas or liquid, circulates, absorbing some of the generated heat. Heat losses are usually limited with the aid of a transparent cover, glass or plastic. The cover lets the short-wave radiation pass through and prohibits long-wave radiation from the absorber-surface to leave the collector. On the shadow side of the collector,

heat losses are reduced with common insulating material. The insulation and the absorber-surface are protected from rain and other influence of the weather with a housing. In some constructions, a part of the building serves as the housing, for instance when the collectors are integrated into the roof. A cross section of a basic flat-plate collector is shown in Figure 13.

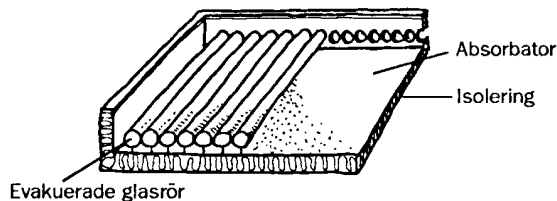
Figure 13. Basic structure of flat-plate collector. A) Cover B) Absorber-surface C) Heat carrier channels D) Back-insulation E) Housing



Flat-plate collectors utilize beam radiation as well as diffuse radiation. They are mainly used for tap-water heating and house warming.

To minimize heat losses the absorbers in flat-plate collectors are sometimes integrated in evacuated tubes. Convection currents, and the accompanying convection losses, are thereby reduced. A basic structure of a collector with evacuated tubes is shown in Figure 14.

Figure 14. Basic structure of flat-plate collector with evacuated tubes.



Heat losses are small when a flat-plate collector is used to produce working temperatures close to the ambient temperature (maximum temperature $\approx 30\text{ }^{\circ}\text{C}$). In those cases there is no need for a cover or back-insulation. During

those circumstances, the efficiency of the collector is high, despite its simplicity. Collectors of this type are useful as pool-heaters, heating of intake-air to ventilation-systems, or as a direct heat source to a heat pump.

When simple collectors are made to produce low temperatures they work as a kind of "solar heat exchangers" that also transfer heat from the ambient air to the system. The collector can be said to work as a energy-absorber for solar energy of varying forms. Figure 15 shows another kind of "solar heat exchanger", a so called rubber-absorber.

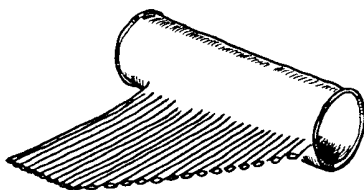


Figure 15. Rubber-absorber = heat carrier channels in a rubber sheeting.

There are also the so called energy-staples, as shown in Figure 16.

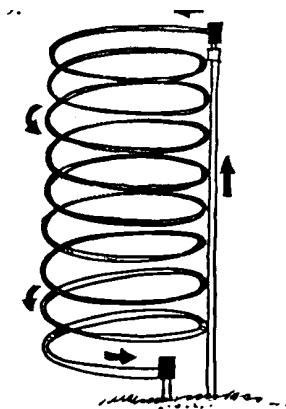
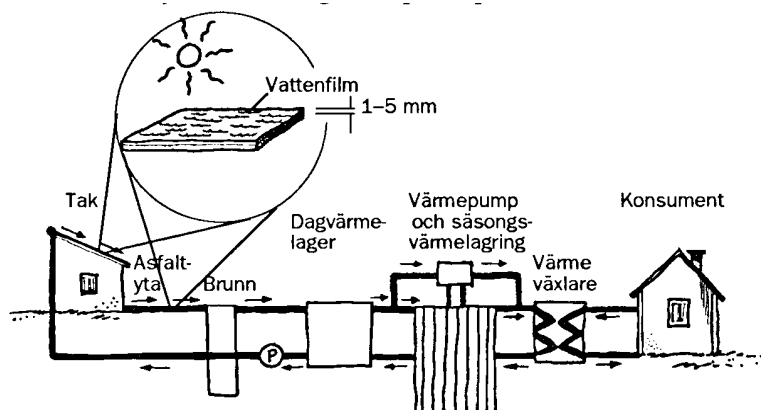


Figure 16. Energy-staple for absorption of solar heat.

Surface-film collectors

Experiments with so called surface-film collectors have been carried out at Luleå University of Technology. The idea behind these collectors is to take advantage of surfaces that are already present in the surroundings (e.g. roofs, asphalt surfaces) and use them as collectors. High costs for collector manufacturing are thereby avoided. The heat from the surface is extracted by a thin layer (film) of water that is flowing on the surface. Also this kind of collector can be said to work as an solar energy absorber, for solar energy of varying forms, since it does not only use radiation, but also condensation heat. Field tests and calculations show a possible annual heat extraction of 400 kWh/m^2 , for low-temperature applications ($\approx 20 \text{ }^\circ\text{C}$). The function of the surface-film collector is presented in Figure 17.

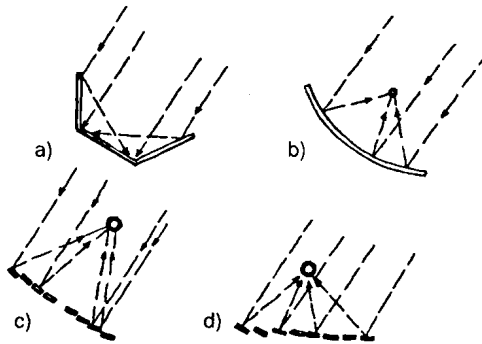
Figure 17. Function of surface-film collectors (From Söderlund M, 1987)



Concentrating collectors

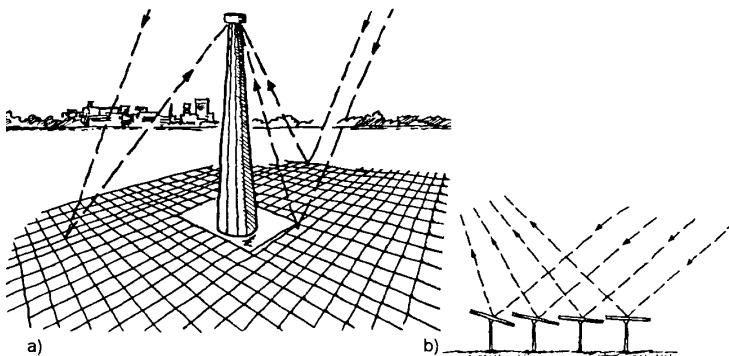
Concentrating collectors, as the name implies, increase the intensity of solar radiation by concentrating the solar radiation. The concentration makes it possible to decrease the size of the collectors absorber and cover. Concentration can be made with lenses, reflectors, mirrors etc. The

collectors must be pointed at, and follow movements of, the sun, since they only use beam radiation. A concentrating collector is mainly attempted for high temperature applications, with temperatures surpassing 80 °C. Different ways of concentrating solar rays are shown in Figure 18.



*Figure 18.
Concentrating of
solar light: a) plane
receiver with plane
reflectors b)
parabolic mirror c)
Fresnel reflector d)
array of heliostats
with central receiver*

In Figure 18 a, a plane receiver with plane reflectors on the borders reflects the radiation towards the receiver. The concentration for this kind of system is rather low, less than a ratio of 4. Figure b shows a parabolic reflector. Figure c shows a Fresnel reflector, i.e. a collection of plane, movable reflectors. Figure d shows the same setup as figure c, but with individually mounted reflectors. The reflected radiation from a number of heliostats are received by a "solar tower", see Figure 19.



*Figure 19. Central
receiver plant (a).
Note that the angle
is different for
different mirrors
(b). (After
Peterson F et al,
1985)*

Moderately concentrating collectors are applied for temperatures around 150 °C. High concentrating collectors require careful following of the sun. The concentration ratio is approximately 40 for a parabolic trench and between 100-1000 for a parabolic bowl and solar towers, see Figure 20.

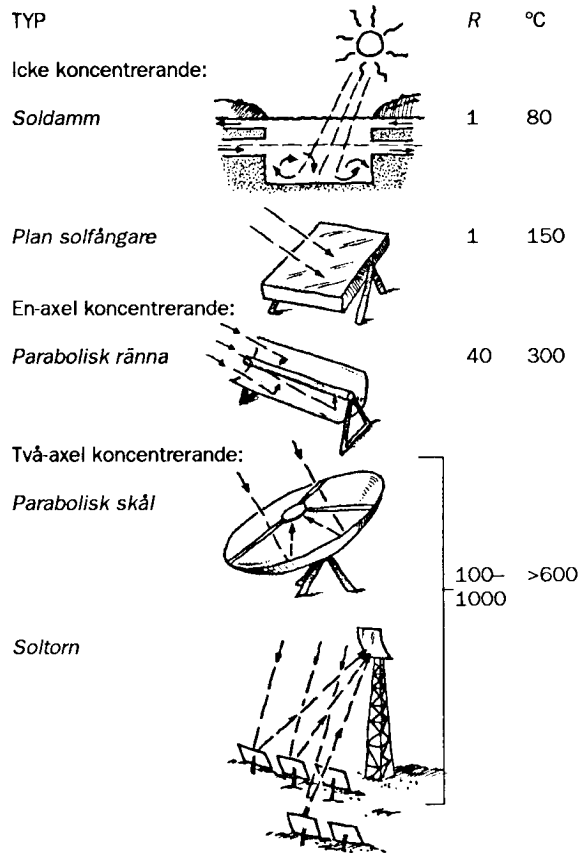


Figure 20. Typical concentration ratios and working temperatures for some collectors. (After Peterson et al 1985).

The heat produced by concentrating collectors is often used in a second step to generate electricity, so called thermal electricity generation. In Figure 21 a summary of performances for varying kinds of collectors is shown, as a function of the working temperature.

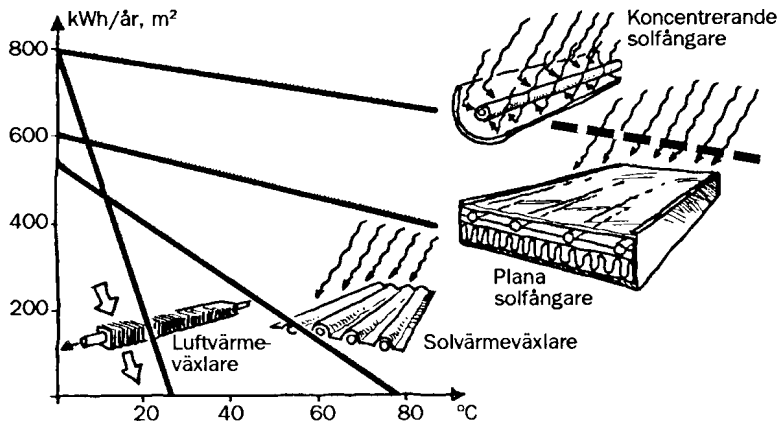


Figure 21. Obtained annual energy as a function of the heat carriers average temperature, for some collector-types. (After Svedinger B, 1981)

From the figure it is seen that concentrating collectors has the best performance per square meter. For lower temperatures of the heat carrier the difference in collected energy decreases. The concentrating collector utilize beam radiation and therefore requires a thorough and careful following of the sun. This kind of collector is of less use in a climate like Sweden's, since the share of diffuse radiation is large in Sweden. The concentrating collector is also considerably more expensive than other collectors and should therefore be used mainly when high temperatures are required.

Solar ponds

A solar pond is used both as a solar collector and as an energy-storage system. Solar ponds have foremost been constructed in Israel. The principle of a solar pond is illustrated in Figure 22.

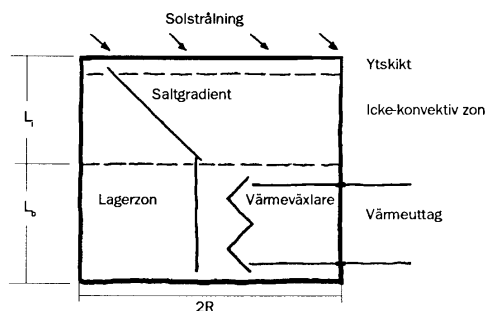
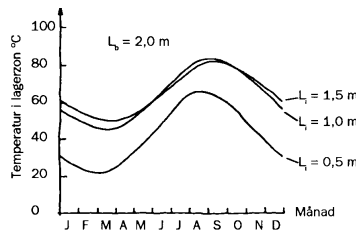


Figure 22. Principle of a solar pond.

The solar radiation is absorbed in the bottom layer of the pond, which has been dyed to a dark color. The bottom layer, which also has a high salt content, is covered with a nonconvecting, insulating layer, with a stabilizing salt-gradient. Heat is therefore transported through the upper layer through conduction. The upper layer will function as an insulating layer and the heat extraction is made through a heat exchanger in the bottom layer. Figure 23 shows calculated temperatures in the ponds bottom layer at stationary conditions, with different thicknesses of the insulating layer.

*Figure 23.
Temperatures in a
solar ponds
bottom layer.*



THEORY OF FLAT-PLATE COLLECTORS

Energy balance of collectors

A flat-plate collector functions according to the principle that the incoming radiation is absorbed and transformed to heat on an absorber surface. A part of the heat is taken up by the heat carrier that circulates in a channel system, integrated in the absorber. This part is the useful energy from the collector, the rest of the energy is lost to the surroundings in the form of convection-, radiation- and conduction-losses, see Figure 24.

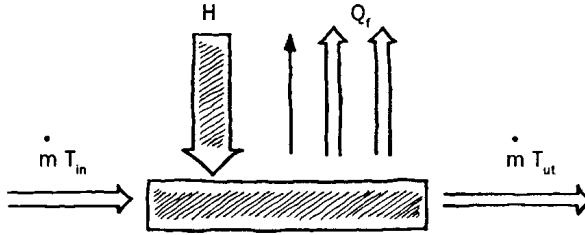


Figure 24. Energy balance for a flat-plate collector.

The loss of energy from a collector, by convection, conduction and heat radiation, can be represented by a heat loss coefficient, the U_L -factor. The heat losses, Q_f , mainly depends on the difference in temperature between the absorber and the surroundings and can be written:

$$Q_f = U_L \cdot (T_a - T_u)$$

Equation (25)

where U_L = heat loss coefficient ($\text{W/m}^2, \text{K}$)

T_a = absorber temperature (K)

T_u = air temperature (K)

The useful extracted energy, at stationary conditions, may be written:

$$Q_u = A \cdot [S - U_L \cdot (T_a - T_u)]$$

Equation (26)

where Q_u = extracted effect (W)

A = collector area (m^2)

S = absorbed radiation in the absorber surface (W/m^2)

The useful effect may also be written as

$$Q_u = \dot{m} \cdot c \cdot (T_{out} - T_{in})$$

Equation (27)

where \dot{m} = mass flow (kg/s)
 c = heat capacity for the heat carrier (J/kg, K)
 T_{out} = outlet temperature from collector (K)
 T_{in} = inlet temperature to collector (K)

The collector is irradiated by the solar radiation intensity H (W/m^2). During the passage through the cover, losses occur due to reflection and absorption. The transmission coefficient, τ , tells how large the part of the radiation that reaches the absorber is. Of this, some is reflected back from the absorber surface. The absorptance, α , tells how much of the radiation that is absorbed in the absorber surface. The absorbed radiation in the absorber surface is then written

$$S = \tau \cdot \alpha \cdot H$$

Equation (28)

The largest part of the heat losses from the absorber is through the cover, losses to the sides are small and the back can be well insulated.

The absorber surface temperature is rarely of primary interest and is also often difficult to measure. Instead, the useful effect, Q_u , is expressed as a function of the heat carrier temperature, T_v . To do so a "collector heat removal factor", F , is introduced. It can be resembled to an efficiency coefficient for a heat exchanger, and tells how much heat that is transferred to the heat carrier, of what is maximally possible. F can be written as

$$F = \frac{\text{Acquired useful effect}}{\text{The useful effect that would have been acquired if the brine would reach the absorbers temperature}}$$

Equation (29)

Equations (25), (26), (27) and (28) can now be written as

$$Q_u = \dot{m} \cdot c(T_{out} - T_{in}) = A \cdot F \cdot [ta \cdot H - U_L \cdot (T_v - T_u)]$$

Equation (30)

where T_v = the heat carriers average temperature in the collector.

From equation (30), the outlet temperature from the collector can be expressed as

$$T_{out} = \frac{A \cdot F}{\dot{m} \cdot c} [ta \cdot H - U_L (T_v - T_u)] + T_{in}$$

Equation (31)

This is the well known Hottel and Whillier model for flat-plate collectors.

The heat loss coefficient, U_L , for glazed collectors is approximately 3-8 W/m², K and for unglazed collectors 12-30 W/m², K. The collectors F-value usually varies between 0,85-0,99. The instantaneous efficiency of the collector, η , is defined as the quotient between the useful effect, Q_u , and the solar radiation, H , on the collector.

$$h = \frac{Q_u}{H \cdot A}$$

Equation (32)

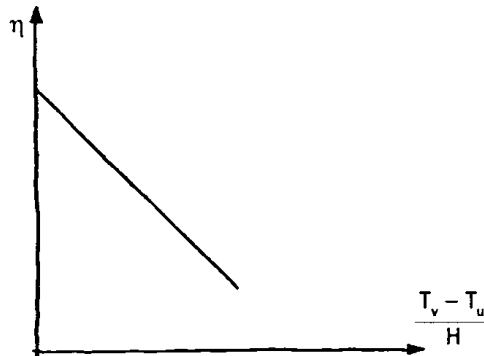
Equation (30) and (32) can be written:

$$h = F[ta - U_L \frac{(T_v - T_u)}{H}]$$

Equation (33)

A good, compact overview of the collector performance is achieved by plotting η as a function of $(T_v - T_u)/H$, in a diagram. See Figure 25.

Figure 25. Principal efficiency curve for flat-plate collector.

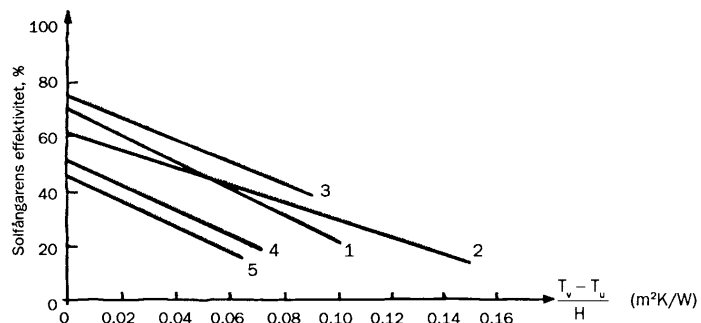


Efficiency curves for different kinds of flat-plate collectors, with air and liquids as heat carriers, are presented in Figure 26.

The efficiency for collectors with air as heat carrier is below those with liquid. This is because the heat transfer between absorber and fluid in an air cooled collector is less than in an liquid cooled collector. Furthermore, the heat capacity ($J/m^3, K$) for air is only $\sim 0,3 \text{ ‰}$ of the heat capacity for water. This causes the F-value of an air cooled collector to be relatively low, and so also the efficiency.

Collectors with water as heat carriers and with one glass has the highest efficiency. The heat losses are small, due to the

Figure 26. Efficiency curves for different types of flat-plate collectors (After Duffie J.A. et al, 1980)



relatively low temperature, at the same time as the transmission factor, τ , is high, with only one glass as a cover.

Collector efficiency

SOLAR RADIATION

The efficiency of a collector depends greatly on the solar radiation, since the absorbed effect in the absorber surface is proportional to the radiation. The amount of radiation is of course mainly dependent of the solar height and the weather, but also on the radiation's angle of incidence towards the collector and of reflecting and absorbing surfaces in the surroundings. In Figure 27, the average daily radiation towards surfaces with different slopes is presented.

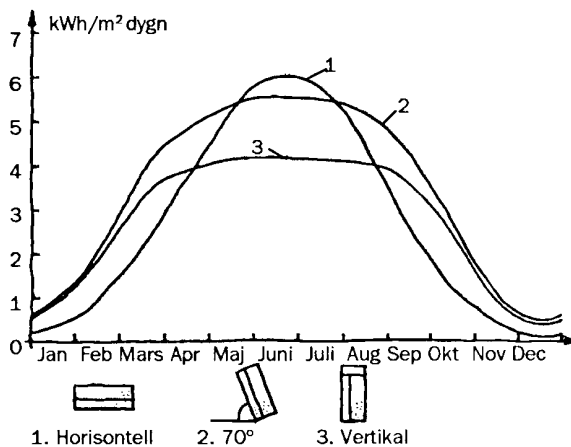


Figure 27. Calculated daily average of radiation in Stockholm, for surfaces facing south. (After Isakson P, 1978).

The radiation on plane surfaces can be increased by placing reflecting surfaces in front of and at the sides of the collector. Calculations show that with a large specular, horizontal reflector in front of a vertical collector, the beam radiation's contribution, to the total incoming radiation on the absorber, can increase with 40-70 % during winter.

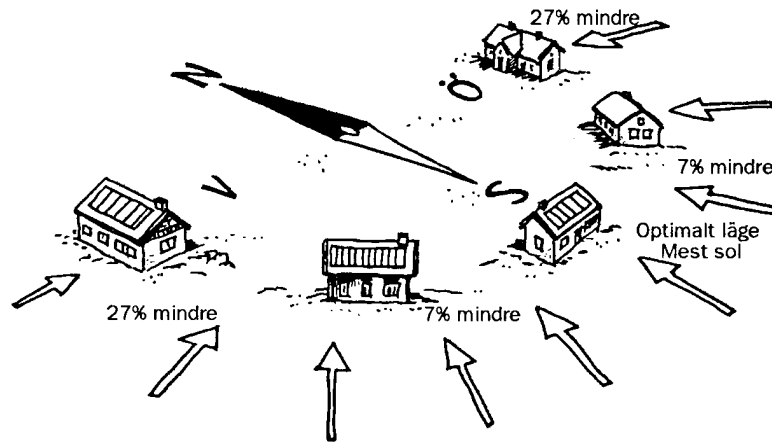
One separates between specular and diffuse reflectors, see Figure 28. Real surfaces are a mixture of these two types.

*Figure 28. Specular and diffuse reflectors
(After Isakson P, 1978)*



The best orientation for a collector is towards the south (in the northern hemisphere), assuming that the average weather conditions are the same in the morning as in the afternoon. Any deviation from the south decreases the energy output, see Figure 29. However, the reduction in energy output is moderate for deviations less than 45° .

Figure 30. Principal figure describing the energy flow and temperature change in the absorber (After Isakson P, 1978)



THE ABSORBER

The collector efficiency is affected by the absorber design, which is described by the F-factor in the Hottel and Whillier equation (30). On the absorber the radiation is transformed to heat which thereafter is conducted to the heat carrier, see Figure 30.

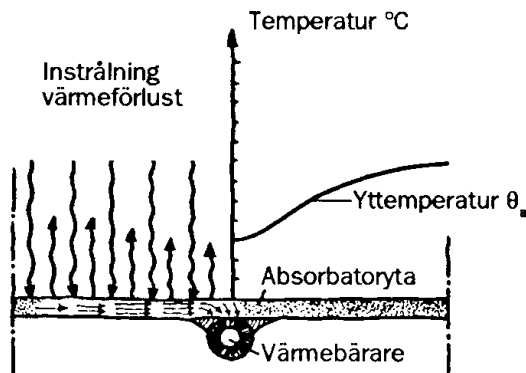


Figure 32. The principle of a selective surface. (After Isakson P, 1978)

For the majority of liquid cooled absorbers $F \geq 0,9$. For some designs, with a very dense channel system, the F-value approaches 1,0. Air cooled absorbers have a lower F-value, caused by the high thermal insulation capacity of air, $F \approx 0,7-0,8$. An ideal absorber shall absorb as much as possible of the incoming light and emit as little as possible to the surroundings. Surfaces with a high absorptance for incoming light and low emittance for outgoing heat radiation are usually called selective black surfaces. In Figure 31, absorptance and emittance for different surfaces is shown.

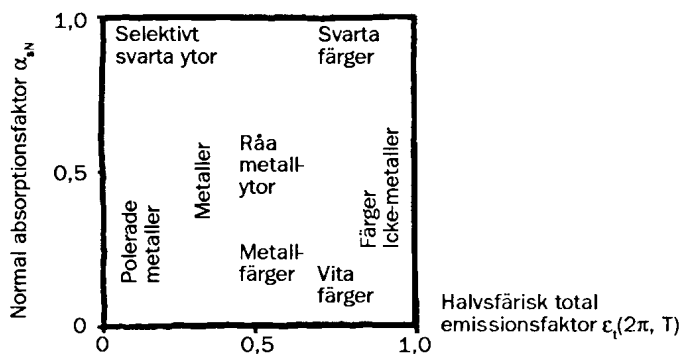


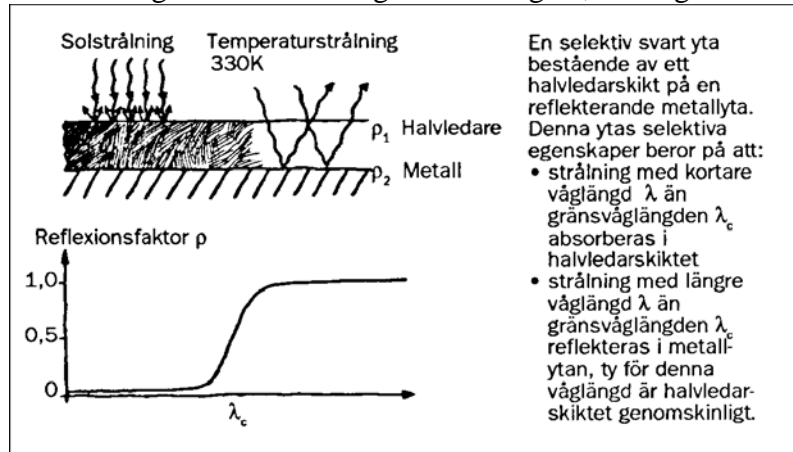
Figure 31. Absorptance and emittance for different kinds of surfaces. (After Isakson P, 1978)

Characterizing for a selective surface is that its properties varies significantly with wavelength and direction of the radiation. The beam radiation reaching the collector has a wavelength-amplitude between $0,3-2,5 \mu\text{m}$. The

temperature radiation, heat radiation, from the collector has wavelengths 4-100 mm. There exist naturally selective surfaces, zinc and copper are two examples, but their absorptance is too low to make them interesting as absorbers.

One of the most common ways to achieve a selective black surface is to combine a reflecting metal surface with a material that absorbs radiation, while at the same time transmitting radiation of longer wavelengths, see Fig 32.

Figure 32. The principle of a selective surface. (After Isakson P 1978)



Absorptance and emittance are shown in Table 4, for a black absorbing lining, consisting of a semi-conducting material, on top of a reflecting metal surface.

Table 4.
Absorptance and emittance for absorber surfaces with black nickel (After Isakson P, 1978)

| Absorber Surface ¹ | Absorption Factor ² | Emission Factor $e \quad \alpha_N$ | Reference ³ |
|-------------------------------|--------------------------------|---------------------------------------|------------------------------|
| Bn Ni | 0.89 | 0.05-0.10 | Edwards (1962), Tabor (1967) |
| Bn Bn Ni | 0.91 | 0.05-0.10 | Edwards (1962), Tabor (1967) |
| Bn Ni Bn Ni | 0.94 | 0.10 | Edwards (1962), Tabor (1967) |
| Bn Zn | 0.88 | 0.15 | Miromit commercial |
| Bn Ni | 0.88 | 0.11 | Pettit & Sowell (1975) |
| Bn Ni | 0.92 | 0.06 | NASA (1973) |
| Bn Bn Ni | 0.95 | 0.07 | Mar et al. (1975) |

Black nickel is a complex of nickel, zinc, sulfides and oxides. This lining is deposited in an electrolytic process on a specular metal surface, that may consist of nickel or zinc.

COVER

As a cover for collectors a full range of different material is used, window glass, glass fiber armoured polyester, polycarbonate, acrylic plastic and several plastic sheeting's. Glass is resistant to high temperatures, better than plastics, and is also more resistant to weather. It is therefore used in most collector constructions.

The covers transmissivity for solar radiation is important for the efficiency of the collector. The cover should have a high transmissivity for solar radiation, while at the same time having a high thermal resistance, to decrease heat losses from the absorber.

GLASS

A part of the radiation that falls on the collector is lost in the cover. The transmission losses in glass depends partly on the reflection on the surfaces, and partly on the absorption in the material itself. The absorption in a 3 mm thick window glass is normally around 6%. The magnitude of the reflection in the glass depends on the radiation's angle of incidence to the collector. The transmission as a function of the angle of incidence is presented in Figure 33.

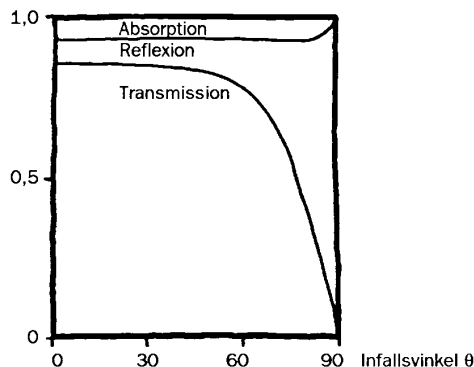


Figure 33. The transmissions dependence on the angle of incidence for common window glass (After Isakson P, 1978)

PLASTICS

Plastics have been used as covers in collectors with varying success. The majority of plastics are sensitive to short-wave radiation and are destroyed after a few years outdoors. Plastics are also more or less sensitive to oxidation, heat, moisture and mechanical strains. An example of a plastic that has been proven to work for solar collector applications is Teflon, with a life-span of ~20 years. The transmissivity for perpendicularly falling solar radiation is 0,97. Acrylic plastic is another example of a material that is very weather resistant and has a relatively high transmissivity, 0,9.

CONVECTION SUPPRESSION

To decrease the convective heat losses, in collectors, a convection suppression layer can be placed between the absorber and the glass cover. This decreases the convection currents in the collector. The convection suppression layer usually consists of a creased or flat plastic, for instance Teflon. An ideal convection suppression layer should not affect the transmission of solar light.

PRACTICAL SOLAR ENERGY APPLICATIONS

One often hear the statement that solar energy is not suitable for a cold climate, like Sweden's. The long winters, the few "sunny hours", and the low temperature is mentioned. However, some of those factors actually increase the interest in solar energy, compared to warmer climates. Low outdoor temperatures and long winters leads to increased energy use for heating and ventilation. A solar collector system comes with high establishment costs but the cost for every kWh generated can be decreased if the system is used for a long time. Flat-plate collectors, which use both diffuse and beam radiation, coupled with low temperature systems (e.g. floor heating, etc.), for heating

and heat storage, will become profitable sooner than they would in warmer countries. Furthermore, the interest in energy questions is larger in countries with a high energy demand than it is in countries in the south, where less energy is used. "Byggforskningsrådet's" (BFR) has made predictions concerning what solar energy systems which stands a good chance to become competitive, in the future Swedish heat market. Systems for solar heated tap water, systems that provide a part of a buildings annual heat demand or systems with heat storage, which can supply heat for almost all of the year, have been found to have a good potential. Other systems of interest are solar collector fields without seasonal storage, connected to district heating and smaller units, where the solar collectors supply around 10 % of the annual need.

SOLAR HEATED TAP WATER

Systems for tap water heating are shown in Figure 34.

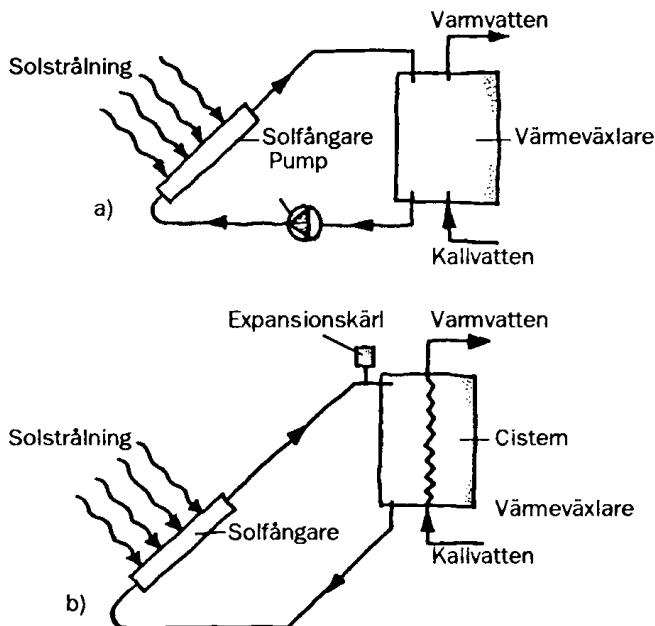


Figure 34. Systems for tap water heating a) Pump system b) Self-circulating system (After Peterson F. et al, 1985)

Picture a) shows a system where the cold water from a water reservoir is pumped to the collector, and heated. The temperature increase in the collector can be determined with the Hottel and Whillier equation (30). This can be written as:

$$\Delta T = \frac{H \cdot A \cdot h}{\dot{m} \cdot c}$$

Equation (34)

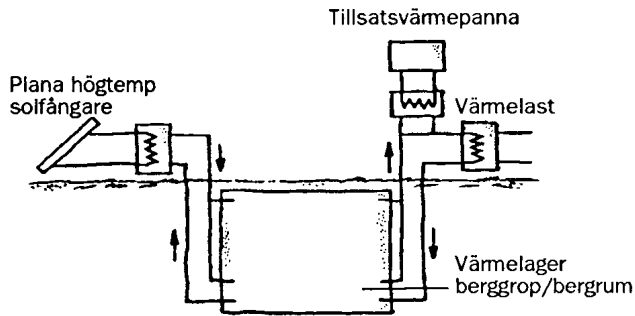
where η = efficiency

H = solar radiation on the collector (W/m^2)

Picture b) shows a self circulation system. The water reservoir is placed above the collector, cold water is gathered in the bottom of the reservoir while hot water stays in the upper part, due to density differences. The flow velocity of the water depends on the amount of heat, where a strong heating gives higher velocities. With a decrease in solar radiation, also the self circulation decreases. The system is thereby self-piloting.

The usable time for tap water heating with solar energy ranges from April to September. If the construction is meant to supply heat during the whole year, then it has to be combined with some other kind of heating system, for instance an electric heater in the reservoir. The solar energy provides the basic heating and with additional heating, a suitable tap water temperature of 45-55 °C is obtained. For a normal one-family house, the required collector surface is approximately 5 m², and the reservoir volume 300 liters, for tap water heating. With respect to freezing-risk in the collector, the heat carrier must have a low freezing point. A mixture of water and glycol is often used.

Large scale experimental plants for seasonal storage of solar heat in clay or blasted rock are currently (1991) in use in Sweden. A simple principal sketch is presented in Figure 35.



*Figure 35.
Principal sketch of
a plant with
seasonal storage of
solar heat. (After
Jilar T, 1987)*

Solar energy is also used to dry hay and grains in farming. For this application air collectors are used. Outdoor air is sucked through longitudinal channels in the roof. The air is heated by the sun, brought into the building and is made to flow through the crops. The efficiency of a solar dryer is approximately 40 %.

ECONOMY, DEVELOPMENT AND POTENTIAL FOR SOLAR ENERGY

Typical for solar heat constructions are high investment costs and, in well made constructions, practically insignificant operating- and maintenance costs. The user cost for extracted solar energy has been steadily reduced during a section of years. The cost for solar heat from large collector fields for district heating systems has been reduced from 1 SEK/kWh in 1982 to 0,30 SEK/kWh in 1987 (with 4 % real interest, and 20 years depreciation time). These cost reductions depends mainly on technical improvements. Solar heat costs can be compared to conventional alternatives for housing heating, which for electricity is 0,25-0,30 SEK/kWh and for municipal heating 0,25-0,35 SEK/kWh. The technical development has mainly been in better system solutions.

In a climate like Swedens, the collector has to work at relatively low outdoor temperatures, so therefore the collectors efficiency should be optimized for such conditions. This climate also requires a collector of a higher mechanical resistance than collectors in warmer climates do. In the last years a development towards highly effective flat-plate collectors in large modules have taken place, specifically suited for connection in large solar plants. A principal figure of such a module is shown in Figure 36.

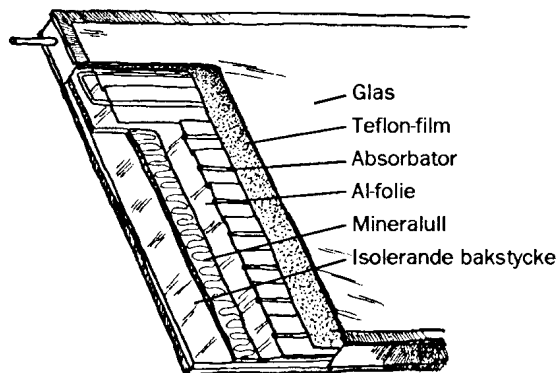


Figure 36. Principal sketch of flat-plate, high temperature collector (After Jilar T, 1987)

It is estimated today (19..) that collectors of this kind could be produced to a cost of 1300 SEK/m². Predictions considering large scale, industrial manufacturing, to much larger volume rates than today, estimates that investment costs for large solar plants could be as low as 800-1000 SEK/m².

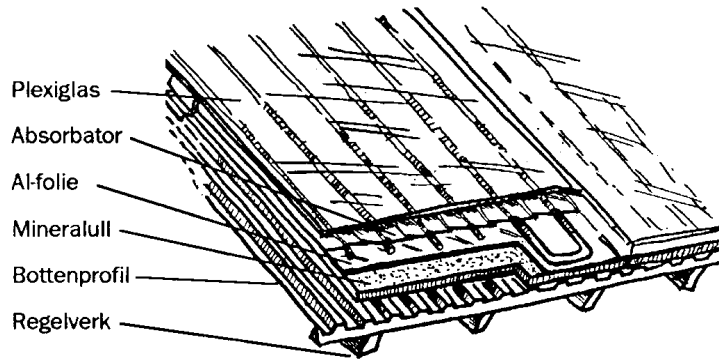
A requirement for a significant increase in solar heat collection in cold climates is that the solar heat can be stored from summer to winter. For research purposes several solar plants with seasonal storage has been built in Sweden. A specification of plants where the solar heat is stored in water is presented in Table 5.

| Plant | Year | Solar Collector | Heat Storage | Heat Load |
|--|------|--|--|---------------------------|
| Studsvik | 1979 | 120 m ² partly conc. | 640 m ³ pit store | Office 200 m ² |
| Ingelstad Ia | 1984 | 1425 m ² plane, high temperature | 5000 m ³ thermally insulated concrete tank | 50 single-family houses |
| Lambohov (HP) | 1980 | 2700 m ³ plane, roof integrated | 10000 m ³ thermally insulated concrete tank | 50 single-family houses |
| Lindälvs-skolan | 1981 | 1500 m ² absorber without glass-cover | 87000 m ³ duct storage in clay | School |
| Kullavik | 1983 | 540 m ² plan, roof integrated | 8000 m ³ duct storage in clay | 50 apartments |
| Lyckebo | 1983 | 28 800 ¹ m ² plane, high temperature | 105 000 m ³ rock cavern | 550 apartments |
| Ingelstad Ib | 1984 | 1425 m ² plane, high temperature | See Ingelstad Ia | |
| Ingelstad Ib+c | 1987 | 3850 m ² , plane | See Ingelstad Ia | |
| Särö | 1990 | 780 m ² plane, high temperature | 1000 m ³ thermally insulated steel tank in rock pit | 40 apartments |
| 1: Only 15% of the solar collectors are constructed. | | | | |

Table 5. Solar plants with seasonal storage in Sweden. Year is date of commissioning of the plant.

The future potential, for seasonally stored solar heat, is estimated to 2-5 TWh/year. The total heat demand in housings is ~75 TWh/year. Seasonally stored solar heat could therefore have a clearly noticeable role in this context. Solar energy technique can already be considered established in a couple of areas. One of those is solar heat production for new built apartment houses. Here so called roof integrated collectors are used, see Figure 37. The heat cover ratio for solar heat is approximately 30 %.

Figure 37. Roof integrated collector, built at the same time as the roof. (After Jilar T, 1987)



Another traditional and well established application is solar heated tap water for smaller houses. Nowadays (1987) solar heat researchers believes that the last years positive development of solar heat technique will make a change in the approach to solar energy. The question will no longer be if solar collectors should be used, but instead how it should be integrated in buildings and how to create suitable financial means.

Natural Heat Systems

INTRODUCTION

In natural heat systems the passively stored solar energy in air, ground and water is used. With the aid of heat pump technique this low tempered heat can be used for heating purposes. Natural heat systems are divided into earth surface heat, ground water heat, rock heat, geothermal heat, air heat and lake heat, see Figure 38.

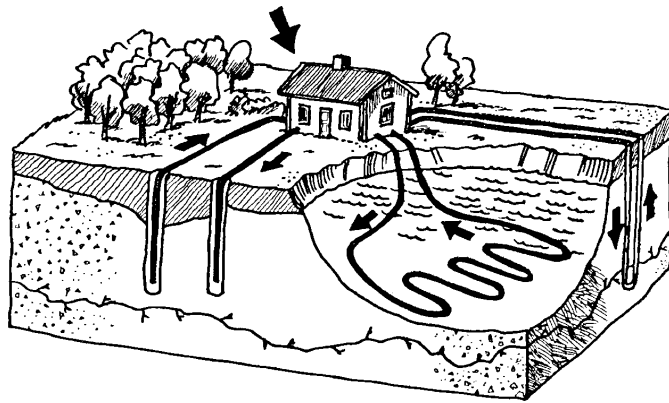


Figure 38. Illustration of natural heat systems.

HEAT PUMP TECHNIQUE

General

Natural heat sources can only be used for heating purposes with the aid of heat pump technique (with the exception of geothermal heat). The principle of a heat pump is therefore here described, before the description of different natural heat sources.

With a heat pump "free energy" from a low temperature heat source can be used. This low temperature heat energy is converted to a higher temperature in the heat pump. To achieve this increase in temperature the heat pump needs

drift energy, for instance electricity. The principle of the heat pump is identical to the refrigerating machine. The term refrigerating machine is used when the primary purpose is to cool something, for instance food. The task for the heat pump is the opposite, namely heating. The following natural laws applies:

- The obtaining of a certain amount of energy, for instance heat, always requires a supply of an equal amount of energy of one or several forms. This is the first postulate of the laws of thermodynamics and is usually referred to as the law of the conservation of energy.
- Heat cannot be converted from a lower to a higher temperature without the sacrifice of energy. This is the second postulate of thermodynamics.

A number of processes can be used for refrigerating and heating machines. Most common is the compressor-driven evaporation process. A heat pump working with the evaporation process utilize the conditions that occur at the transfer from vapor to liquid in cooling agents. By lowering the pressure above a liquid, the liquid can be made to boil at just any temperature and thereby obtain heat of evaporation from the surroundings. The cooling agent circuit of a heat pump is shown in Figure 39.

$$\phi = \frac{Q_1}{E} = \eta \cdot \frac{T_1}{T_1 - T_2}$$

ϕ = heat factor

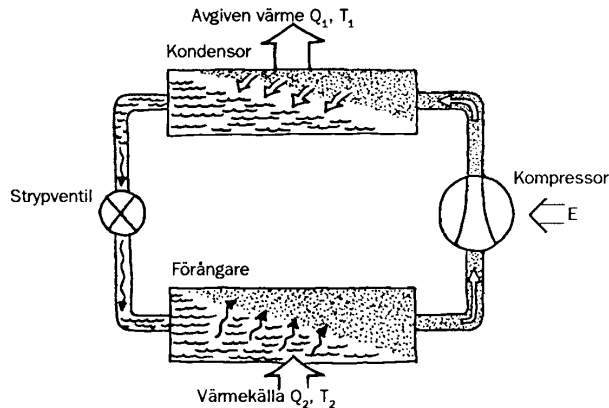
Q = heat amount

T = Temperature (K)

E = Drift energy

η = Efficiency

Figure 39. The cooling agent circuit for a heat pump (After Svedinger B, 1981)



In the evaporation chamber heat is obtained from the heat source, whereby the cooling agent boils at low pressure. The cooling agent vapor is sucked from the evaporation chamber, compressed to a higher pressure, and blown into the condensor. There the cooling agent transforms into a liquid phase by giving off heat to (for instance) the radiator circuit in a house. The higher the pressure, the higher the condensation temperature. The evaporator chamber and the condensor are in practice two heat exchangers. The cooling agent liquid is returned to the evaporation chamber, through a check valve, to be boiled again. The check valve serves an important and complicated function. It shall both lower the pressure on the cooling agent liquid to the low pressure of the evaporation chamber and regulate the amount of cooling agent liquid to the evaporation chamber, so that all liquid has time to evaporate in the chamber. The heat pump process is therefore going to have a high pressure side and a low pressure side. The ideal Carnot-process for the heat pump (or refrigeration machine) is shown in Figure 40.

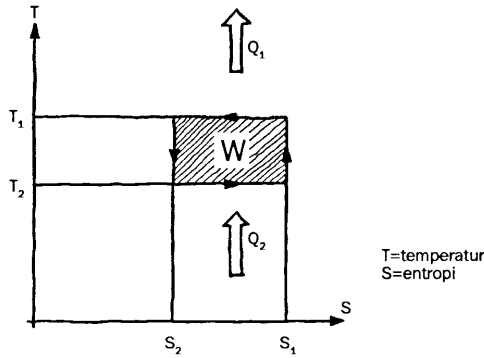


Figure 40. The Carnot-process for a heat pump
(After Beckman O, 1976)

The heat pump is driven by the mechanical work W (the compressor work). Thereby the heat amount Q_2 is taken from the lower temperature T_2 , while the heat amount Q_1 is given off at the higher temperature T_1 . The effectiveness for the heat pump is expressed with the heat factor. This is defined as;

$$\phi = \frac{Q_1}{W}$$

Equation (35)

The heat factor expresses the relationship between given heat amount, Q_1 , and the need for drift energy, W (compressor effect). A heat factor of 3 means that with 1 part of drift energy, 3 parts of heat energy are obtained at the higher temperature. The heat pumps efficiency depends on between which temperature levels it is working. The temperature difference can be said to be the heat pumps "lifting". The ideal Carnot process for a heat pump would give a heat factor expressed as;

$$\phi_{Carnot} = \frac{T_1}{T_1 - T_2}$$

Equation (36)

where T_1 and T_2 express the absolute temperatures on the heat pumps hot and cold side, respectively, in degrees of Kelvin (K). From a theoretical point of view it would be possible to collect heat all the way down to absolute zero, -

273.16 °C. In real heat pump applications one cannot reach the theoretical value of the heat factor as expressed in (36). In practice the real heat factor is approximately half of the theoretically possible;

$$\varnothing_{real} = 0,5 \cdot \varnothing_{Carnot}$$

Equation (37)

The reason for this is losses in the cooling agent cycle, compressor, electric engine, etc. Another important reason is that the thermodynamic processes do not have time to reach equilibrium, in the cyclic work of the heat pump.

Cooling agent

The cooling agent in a heat pump must fulfill certain demands in order to function in the process:

- The cooling agent must not convert into solid form at any actual temperature
- The cooling agent must not disintegrate at any actual temperatures or pressures
- The cooling agent must not cause corrosion on materials in the cooling circuit
- The cooling agents poisonousness and fire- or explosiveness dangers will affect security demands and the construction

It is also desirable that the agents vapor pressure increases as little as possible with increasing temperature and that the pressure at the lowest actual evaporation temperature is slightly above the atmospheric pressure. Pressures below the atmospheric pressure involves the risk that air and moisture is sucked in through leaks in the cooling agent circuit. Fluorine-substituted hydrocarbons are the main cooling agents used in heat pumps and are sold under names

like Freon, Frigen etc. The most common cooling agents for heat pumps have notations like R12, R22, R502 etc., where "R" stands for refrigerant. Vapor pressure curves for some cooling agents are shown in Figure 41.

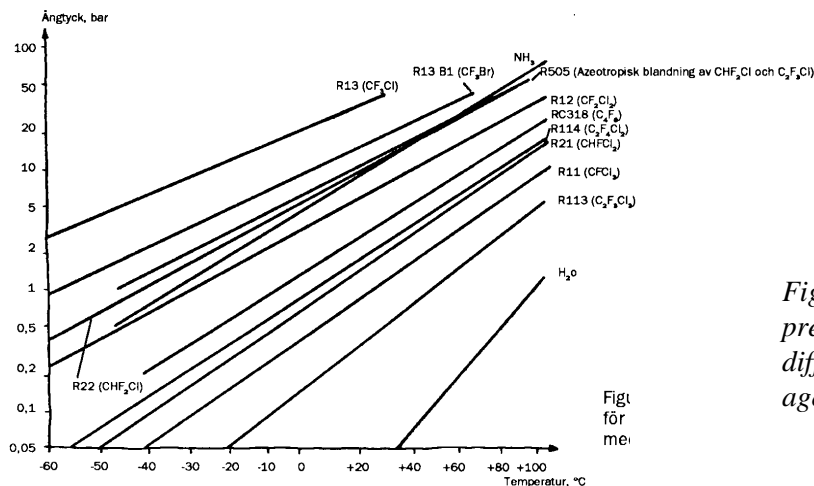
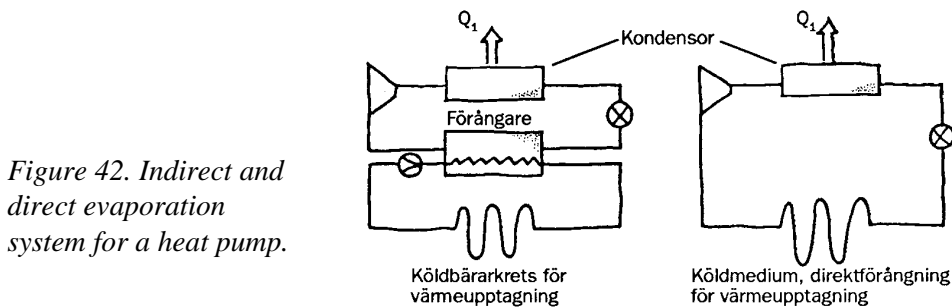


Figure 41. Vapor pressure curves for different cooling agents.

The cooling agent R22 boils at -25 °C at a pressure of 2 bars (the cool side). At a pressure of 20 bars the boiling point is 50 °C (the warm side). Freons will damage the ozone-layer when they are released to the atmosphere. In the Swedish governments environmental proposition of 1988 it is suggested that the usage of fluoride-substituted hydrocarbons should decrease with 25 % to 1990/91 and with another 25 % to 1998/93. The usage of these chemicals should cease entirely from 1994/95.

Heat transfer media

The heat pump can work with direct or indirect evaporation systems. In a direct system the cooling agent circuit is in direct contact with the heat source. The evaporation occurs in contact with the heat source. In an indirect system on the other hand the heat is transported from the heat source to the heat pumps evaporator chamber (where the evaporation takes place) with the aid of a heat transfer medium, a brine, see Figure 42.



The indirect systems require a brine which will transport heat from the heat source to the evaporation chamber of the heat pump. Water is a very good heat transfer medium for temperatures above 0 °C. At temperatures below 0 °C water with an addition of anti-freezing agents is often used. The most common anti-freezing agents are ethylene-glycol, calcium-chloride, propylene-glycol or ethyl-alcohol, see table below. In recent years usage of potash (neutral carbonate of potassium) and water has started.

Table. Different kinds of brines.

WATER (with or without anti-freezing agents)

SALTS:

Sodium-chloride NaCl

Calcium-chloride CaCl₂

ORGANIC ADDITIONS:

Ethylene-glycol (CH₂OH)₂

Propylene-glycol C₃H₈O₂

Ethyl-alcohol C₂H₅OH

An inhibitor to prevent corrosion must often be added to the brine. Examples of inhibitors include nitrite, benzoate, zinc-sulfate, poly-sulfate, amines, aldehydes, and more. Common for all of them is their ability to decrease anode- and/or cathode-reactions at corrosion. A good heat transfer medium should have high density, high specific heat (good storage properties for heat), high heat conductivity (good heat transfer properties), and low viscosity (good pumping properties). The viscosity is often one of the most difficult problems. The reason for this is that the viscosity, and thereby the pressure drop, increases with lowered temperatures. There is also a large risk that the Reynolds number falls below the lower limit for turbulent flow (2300), which leads to laminar flow, and this will significantly decrease the heat transfer coefficient between fluid and surrounding heat source. The viscosity as a function of temperature is shown in Figure 43, for different brines.

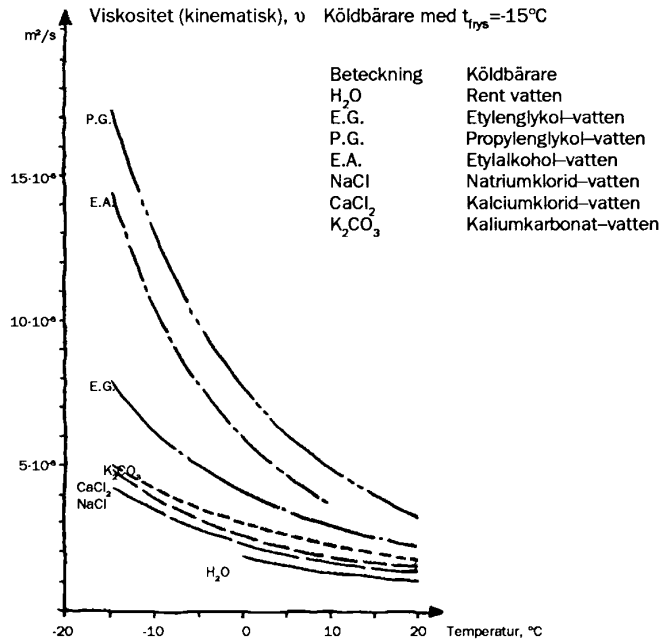


Figure 43. Viscosity as a function of temperature, for different brines.

From a viscosity point of view NaCl is therefore the best brine. Other thermodynamic properties for brines are tabled in "Kyltekniska Föreningens" issue no 5.

Dimensioning

A buildings effect demand is the maximum heat requirement for a given outdoor temperature. For a normal family house the effect demand varies between 6-10 kW. The real effect demand varies from hour to hour during the year. If all effects are sorted from highest to lowest value and plotted in a diagram, with the curve falling from left to right, one has an effect duration-curve, see Figure 44. The total heat demand for a year corresponds to the area under the curve.

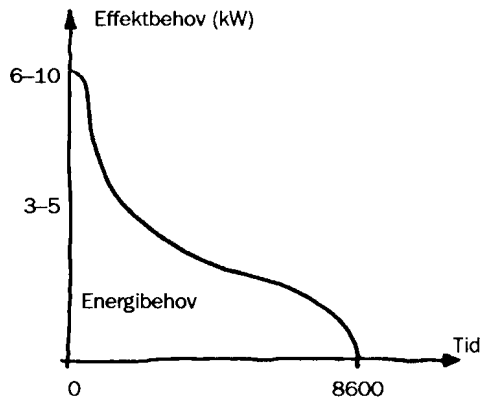


Figure 44. Duration curve for a one family house.

A heat pump has high costs of installation, but low drift costs. To obtain a good economy in a heat pump system one strives to get as long drift times as possible on the system. The cost per produced kWh is thereby reduced. It is possible to cover the entire heat energy demand with the heat pump but this means that the heat pump has an over-capacity during the main part of the year, see Figure 44. The heat pump is normally dimensioned to supply approximately half of the effect demand for the house, see Figure 45. This corresponds to around 70-90 % of the heat energy demand. The remaining effect and energy demand must be covered in another way, for instance with oil, electricity, bio-fuel or carbon. Often already present oil burners are used as a reserve and point coverage.

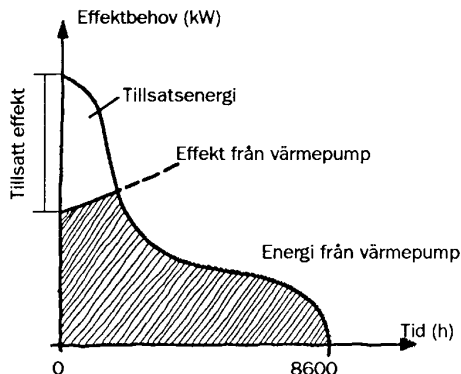


Figure 45. Duration curve with heat pump dimensioning.

NATURAL HEAT SYSTEMS

Soil heat

A soil heat system utilizes the heat that in summer time, through rain and radiation, is stored in surface soil. The heat is extracted with a heat pump and horizontal tubes in the ground. A brine whose temperature is below the soil temperature circulates in the tubes. Heat is thereby conducted from the soil to the tubes. A principal sketch of a soil-heat system is shown in Figure 46.

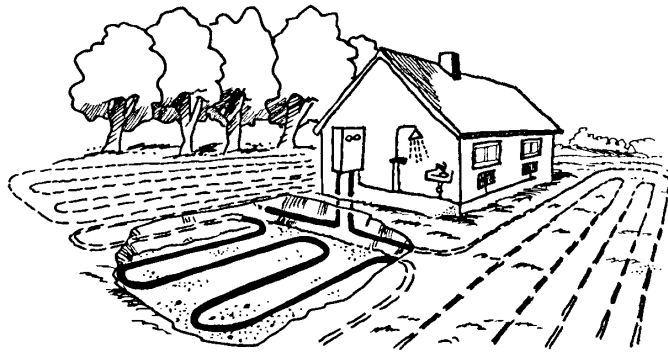


Figure 46. Principal sketch of a soil-heat system (After Svedinger B, 1981)

In the beginning of 1982 there were approximately 7000 soil-heat systems in use in Sweden. A soil consists of particles and pores. The pores can be filled with air and/or water. The water content effect on the soils heat conduction is realized when considering that the heat conductivity for air is $0,024 \text{ W/m, K}$ and for water $0,60 \text{ W/m, K}$. The heat conductivity for ice is $2,1 \text{ W/m, K}$. The variations in water content in a profile of soil depends on the position of the ground water surface and the soils grain size distribution. In Figure 47 the variation of heat conductivity is shown, for sand with varying water content.

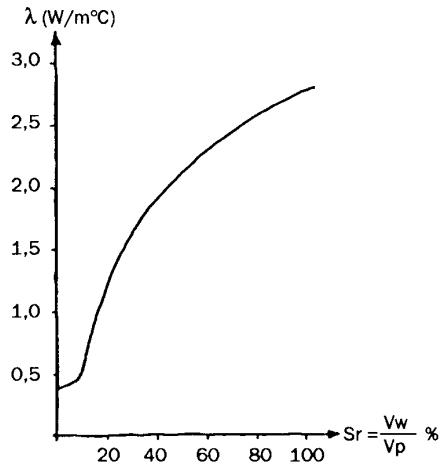
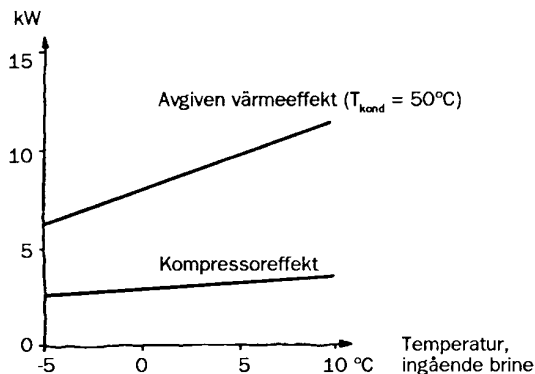


Figure 47. Variation of heat conductivity with water content, for sand. (After Sundberg J, 1982)

Sand is an extreme example and the difficulties in designing the collector are understood from the figure. The soils properties can therefore vary a lot. The required tube-length and surface need for collectors with equal performance can vary with a factor 3, for given climate conditions. For the case with an under-dimensioned system with too high load on the soil-collector, disturbances like too low brine-temperatures, unacceptable frost heave or deteriorated growth-conditions (for plants) may occur. When a soil-heat system is dimensioned to be fully covering it must be able to deliver the required effect the coldest day of the year, and it shall deliver the annual energy demand year after year. The effect criteria means that the maximum effect output must not lead to the brine temperature falling below a given value. This value is determined by the heat pump, see Figure 48.

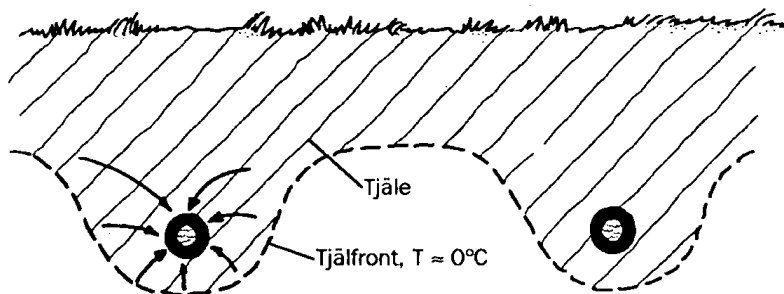
Figure 48. Energy effect output as a function of incoming brine-temperature, for a compressor driven heat pump (JBC 400 M). (After Mogensen P, 1982)



From the figure it can be seen that a lower incoming brine-temperature leads to a lower heat effect yield.

In a soil heat system the major part of the heat estimates from the frozen ground front, see Figure 49. The temperature decrease from the frozen front to the collector tube depends on the heat conductivity of frozen ground.

Figure 49. Heat flow to a soil heat collector tube, the coldest day of the year. (From Mogensen P, 1982)



The most common soil heat systems have plastic tubes (polyethene) that are dug down horizontally to a depth of 0,8-2,0 m. The maximum effect output is 10-30 W/m, depending of soil-type and water content. In a water saturated clay 200-400 m tube on an area of 400-600 m² is required for a small house.

When a soil heat system is constructed, one can differ between construction methods that includes digging and

such that includes ploughing. The methods are showed in principle in Figure 50.

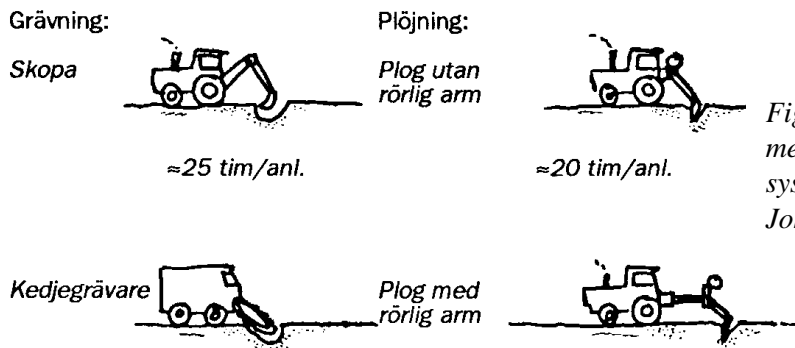


Figure 50. Construction methods for soil heat systems. (After Johansson I, 1982)

A collector can be designed in some alternative ways, see Figure 51. It can consist of a simple loop or several, interconnected loops. There is also a distinction made between simple extension, double extension and several level systems.

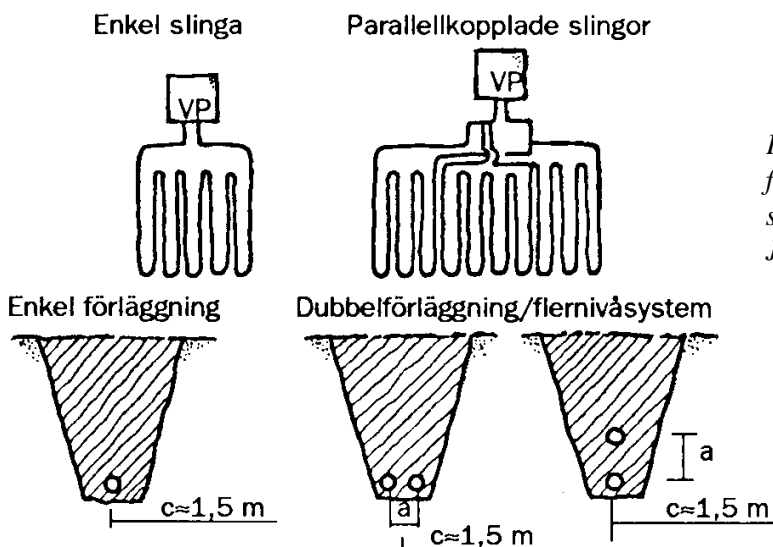


Figure 51. Alternatives for collector design for soil heat systems. (After Johansson I, 1982)

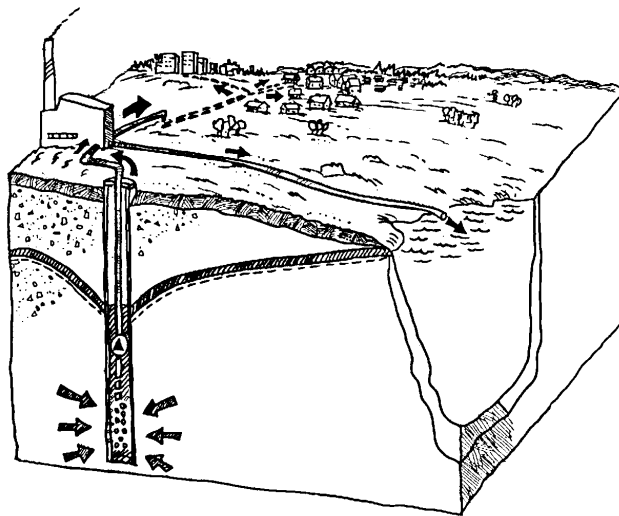
Difficult soil conditions when digging leads to higher expenses for the construction. This may be caused by stony ground, rocks within the digging depth, many trees to take into respect or that the soil has little carrying capacity. From a frost protection point of view, the distance from the cold-loop to a water carrying pipe should be at least 0,50 m.

Soil heat systems affect their surroundings by lowering the ground temperature. The farming conditions and the soil-biological activities are affected by the heat output. Locally, close to the soil heat collector, one can expect that the local climate zone is affected at least one step. This must be considered when a soil heat system is constructed.

Groundwater heat

Groundwater as a heat source to a heat pump is a so called open system. The water is pumped directly to the heat pump evaporation chamber, where the temperature is reduced and the water is returned to a recipient, or injected back to the ground water magazine, see Figure 52.

Figure 52. Alternative collector design for groundwater heat systems (After Johansson I, 1982).



A groundwater heat system is technically a hydraulic problem. The heat output depends on the possible groundwater output. The maximum groundwater output is determined by test-pumping. Considerable groundwater assets are found in glacial-river-sediments and in fractured, porous sedimentary rocks. The possible effect for a groundwater heat system can be calculated if water flow and groundwater temperatures are known. In Figure 53 the effect output is shown as a function of the groundwater temperature decrease and the water flow in the evaporation chamber.

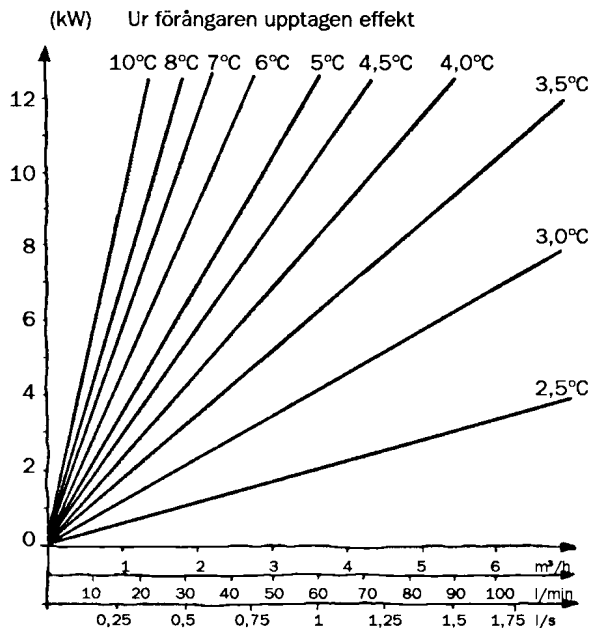


Figure 53. Effect output from groundwater, as a function of flow and groundwater temperature. The diagram requires a cooling to + 2 °C. For a heat performance factor of 3, the heat pump will give 50 % higher effect than what the evaporation chamber takes up. (After Andersson S et al, 1980)

To subtract heat from groundwater, conventional well-drilling equipment is used. The groundwater temperatures in Sweden are presented in Figure 54. In Northern Sweden the possibilities to use groundwater-heat is significantly lower than in Southern Sweden, due to the lower groundwater temperatures.

Grundvattentemperaturens
årsmedelvärden 1968-1975

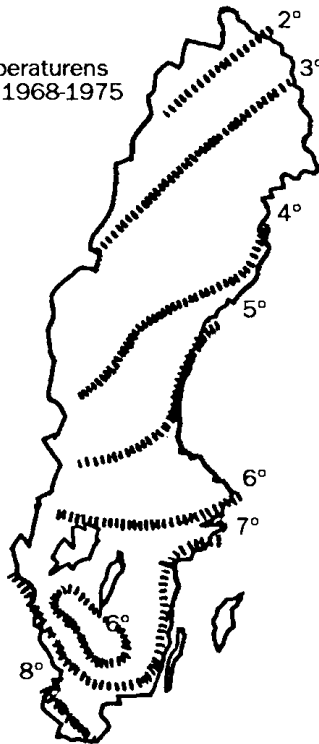


Figure 54. Groundwater temperatures in Sweden. Groundwater temperature variations in Sweden follow the annual mean air temperature. In areas that are rich of snow the groundwater will be somewhat warmer due to the insulating properties of the snow. For Luleå the temperatures are 2,2 °C (air) and 3,6 °C (groundwater) respectively.

Rock heat

In a rock heat system, the stored heat in rock is used as a heat source to a heat pump. The heat energy is transferred to a borehole through conduction in the rock. A distinction is made between open and closed systems, see Figure 55.

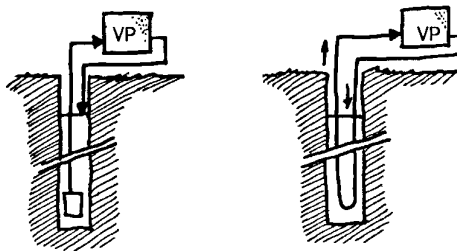


Figure 55. Open and closed rock heat systems.

In a closed system temperatures below 0 °C can be used. Rock heat systems can be made as a single well or a combination of several wells. Closely spaced wells will have a thermal effect on each other. The possible heat output capacity from each well thereby decreases when the number of wells increases if the wells effect each other thermally. The temperature decrease for a single well and for a system of 9 wells are presented in Figure 56.

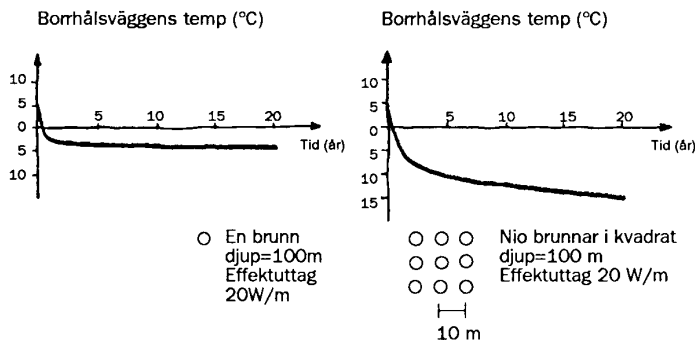


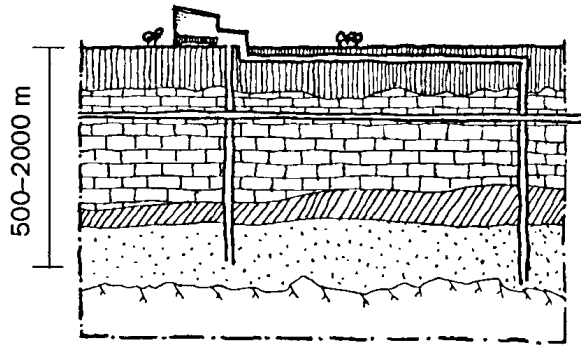
Figure 56. Temperature decrease at a rock heat well and 9 rock heat wells in a system. The effect output for each well is the same in both systems.

For a single well, the main temperature decrease occurs during the first year of running, while the temperature decrease is significant during the whole construction life span for a system with 9 wells. To avoid deterioration of performance, for the construction with 9 wells, it must either be recharged or dimensioned to handle the effect requirement at the end of its drift-time. No negative environmental effects due to the cooling of the rock are to be expected for rock heat systems. There are however risks for groundwater pollution at a leakage of brine.

Geothermal energy

Geothermal energy is the name for extracting hot water from deep situated sandstone formations and use it for heating purposes, see Figure 57.

Figure 57. Principle of geothermal energy.



The temperature on large depths below the ground surface varies significantly between different places. Iceland, for instance, has extremely high temperatures in the deep ground. Generally old rock is colder than new. In Sweden one finds the highest rock temperatures on the island Gotland and in the landscape of Skåne.

The heat flow that emanates from the earth center results in relatively high temperatures in the sandstone 1000 m below the ground surface. The water in the sandstone's pores has the same temperature. The hot water can then be extracted from wells that reaches down to the sandstone formations and the heat in the water can be used for heating purposes by heat exchangers or heat pumps. The cold water is transferred back to the sandstone formation through an injection well.

The Energy department of the City of Lund has built Sweden's first geothermal project. From four wells, each 700 m deep, 20 °C water is extracted and pumped to the constructions heat pumps, 19 and 27 MW respectively. When the heat energy has been extracted 4 °C water is returned to the systems 5 injection wells, see Figure 58.

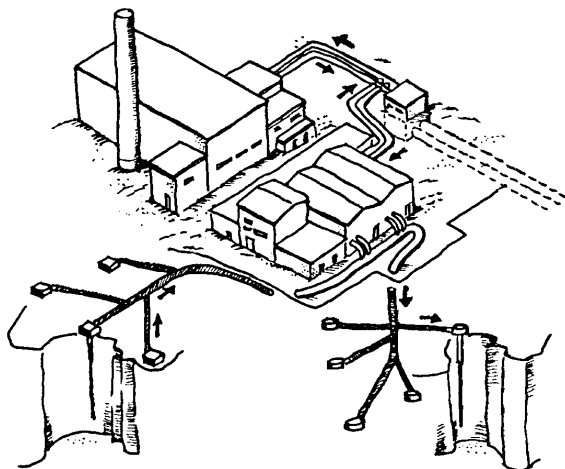


Figure 58. Geothermal construction in the City of Lund.

Immersed centrifugal pumps have been installed in the production wells. The pumps are situated on a depth of 75-80 m. The geothermal water in the holes is artesian with the surface practically at ground level when no pumping occurs. The heat pumps are connected to the municipal heating network, as shown in Figure 59.

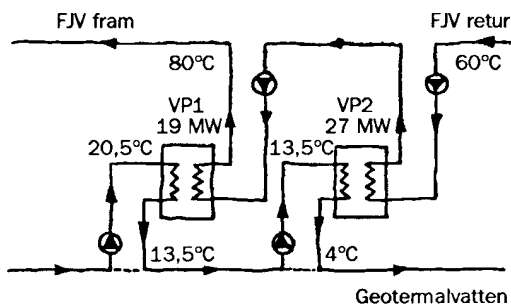


Figure 59. The heat pumps connection to the municipal heating network, in Lund.

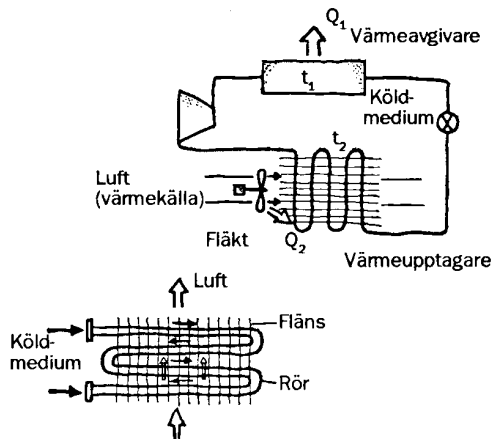
The total theoretical, geothermal energy potential in the sedimentary, water carrying rock down to 2000 m - in the landscape of Skåne - has been estimated to at least 7000 TWh. Only a small fraction of this can be utilized due to practical and economical reasons.

Air heat

Outdoor air, or exhaust air, can be utilized as an energy source to a heat pump. Exhaust air is a splendid source of energy in existing buildings with mechanical air exhausting systems for ventilation. The temperature level is high and constant, which gives suitable drift conditions for a heat pump. The exhaust air in homes contains 20-30 % of the heat demand if the air is cooled to outdoor temperature. In offices, warehouses and many industries the exhaust air amount is large enough to cover the total energy need.

Outdoor air as an energy source to a heat pump has a significant advantage compared to other heat sources; it's available anywhere. The disadvantage is that the temperature is the lowest when heat is best needed. To subtract heat from the air, the air is made to pass a flanged heat exchanger where the brine from the heat pump flows, see Figure 60 (so called direct evaporation system).

Figure 60. Flanged heat exchanger for extraction of heat from air. (After Glas L-O, 1978)



The surface temperature on the flange heat exchanger is sometimes below 0 °C when outdoor air is used as heat source. Frost is thereby deposited on the flanges. If the flanges are not defrosted frequently the spaces between them will grow full of ice. Then air can not pass and give off

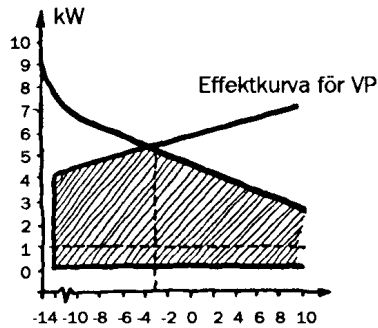


Figure 61. Energy coverage level for an outdoor air heat pump.

heat to the brine. Therefore outdoor air heat pumps are constructed with automatic defrosting. The energy coverage degree for a outdoor air heat pump is presented in a duration diagram in Figure 61.

From the figure it is understood that the effect decreases with decreasing outdoor temperatures. At a certain lowest temperature (usually $-10\text{ }^{\circ}\text{C}$ to $-15\text{ }^{\circ}\text{C}$) the heat pump is shut off and the effect need is supplied with point heat. This means that the point heat must be dimensioned to cover the maximum effect need.

Lake heat

Heat in water and bottom sediments can be used as heat sources to a heat pump and this is called lake heat. A distinction is made between open and closed systems, see Figure 62.

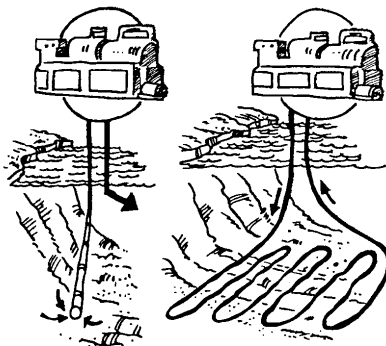
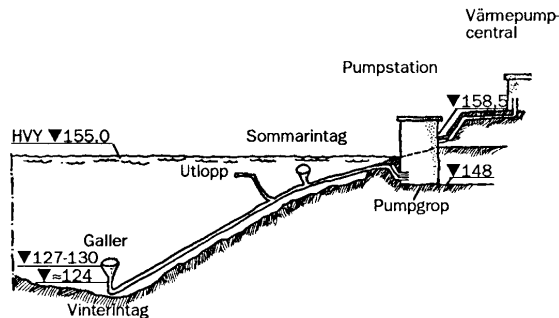


Figure 62. Principle of an open and a closed lake heat system.
(After Svedinger et al, 1991)

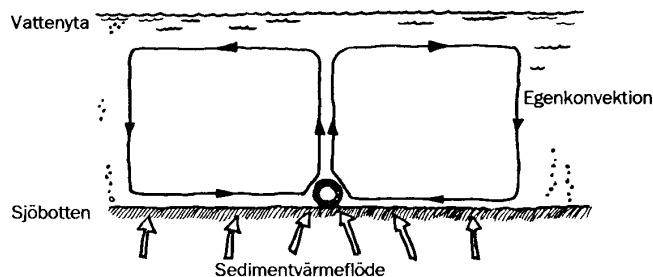
In an open system the lake water is pumped directly to the heat pumps evaporation chamber. Figure 63 shows an example of design of an open lake heat system. Two intake pipes makes it possible to use warm surface water during summer and relatively warm bottom water during winter (since the highest temperature is found at the bottom in an ice covered lake).

Figure 63. Principle for an open lake heat system. (After Petersson L, 1982)



A closed system for lake heat resembles the soil heat systems. The tubes are anchored to the bottom or dug down in the bottom sediments to prevent them from floating when they get covered with ice. The heat transfer occur through conduction from sediments and water, thermal convection and natural turbulence in the water, and through ice formation on the tubes, see Figure 64.

Figure 64. Heat transfer to a lake heat collector. (After Johansson J, 1982)



The heat balance for an ice covered lake is shown in Figure 65. The arrows illustrate the heat addition from the sediments, running water, sewage water and solar radiation through the ice. Heat losses to the ice and through runoff and heat extraction are also shown.

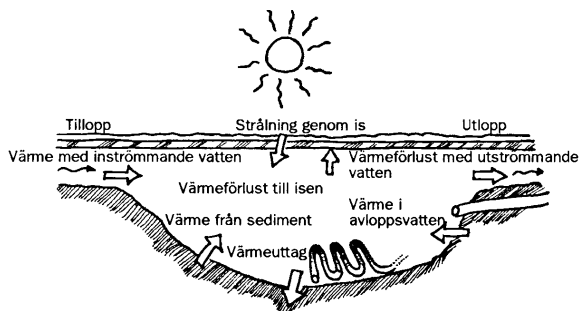


Figure 65. Heat balance for an ice covered lake.

The possible effect output from lake heat collectors varies between 20-40 W per meter tube. Laboratory tests has been performed at Chalmers University of Technology (the Department for Water Construction) to evaluate the heat uptake with a bottom anchored tube, see Figure 66. The figure shows the heat output in relation to the water temperature.

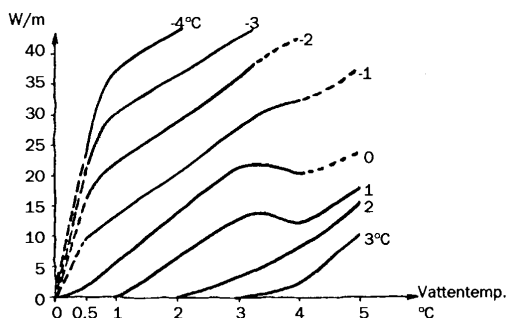


Figure 66. Extracted heat effect from a bottom anchored tube, at different brine temperatures, as a function of the water temperature close to the tube. (After laboratory tests, Svensson T, 1982)

For a brine temperature of -2 °C and the water temperatures +1 °C and +2 °C, the possible effect output is 22 and 29 W/m respectively. Environmental consequences that should

be taken into consideration for lake heat systems, open and closed, are presented in summary in Figure 67.

The ecological impact that can be expected with an open system depends on altered current-conditions in the lake, temperature changes and the pumping of bottom water from the lake. The biological effects that may occur because of this are a change in nutrient transformation and temperature effects on plants and animals. The change in nutrient transformation ought to be the most important factor to consider. The temperature change often stops at some tenth's of a degree, which normally would not cause noticeable environmental consequences. For closed systems the risk for leakage of brine is added, in addition to the already mentioned consequences.

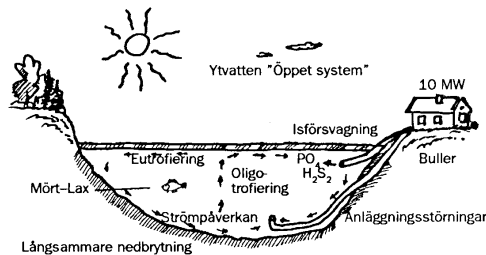
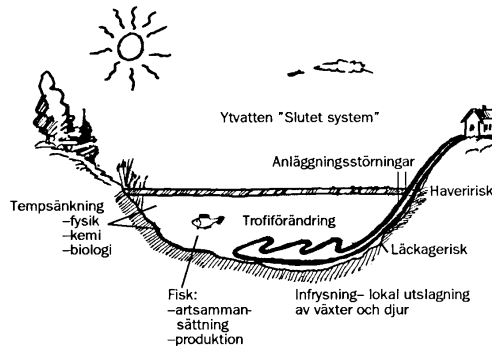


Figure 67. Possible environmental impact with open and closed lake heat systems. (After Dietrichson W, 1982)



NATURAL HEAT THEORY

General

The theoretical description presented below is taken from the handbooks for Ground Heat (in Swedish) by Claesson J, et al, 1985, Lunds University of Technology.

The temperature process in the ground is described by the heat conduction equation. The analyzes that are presented in the Ground Heat handbooks are based on analytical and numerical solutions of this differential equation, with given boundary conditions and other data. In a piece of land without groundwater movements, the temperature T is described with the 3-dimensional, non-stationary heat conduction equation.

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = \frac{1}{a} \cdot \frac{\partial T}{\partial t}$$

Equation (38)

where $a = \lambda/c =$ the temperature conduction coefficient (m^2/s)

$\lambda =$ the heat conductivity ($\text{W}/\text{m}, \text{K}$)

$c =$ the volumetric heat capacity ($\text{J}/\text{m}^3, \text{K}$)

$t =$ time (s)

Superposition

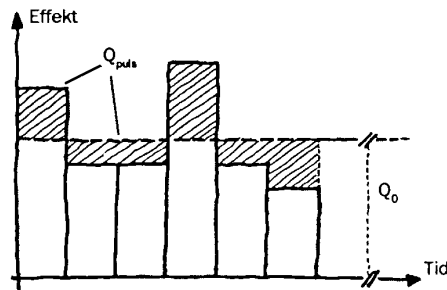
The different forms of the heat conduction equation are basically linear, partial differential equations. This means that different solutions can be superimposed, i.e. if two temperature processes each by themselves fulfills the heat conduction equation this will also be true for the sum of the two temperatures. Complicated temperature processes can be given a relatively simple structure with superposition.

The superposition method has two important limits, it will not work when freezing occurs (for instance soil heat systems), because the phase change has to be taken into account, and neither will it work with temperature processes that occur with flowing groundwater (convective transport).

To describe temperature processes in ground the problem can be separated into three fundamental parts. One is a stationary part which depends on the annual mean effect output and another is a superimposed periodical variation during a year cycle. In addition to those two there is also a transient process during the introduction phase. For a heat storage there is a transient formation of a heat layer around the storage.

Superimposing of the stationary and the transient variations during a year is shown in Figure 68 and equation (39).

Figure 68. Example of superposition technique.



The heat flux, $Q_{flux}(t)$, are added to the stationary heat output, Q_0 , according to (39).

$$Q(t) = Q_0 + Q_{flux}(t)$$

Equation (39)

The transient component, the flux, produces a temperature flux process in the ground with a relatively short reach. The stationary part decides the total heat loss after an introductory transient period and, for a rock heat system, the size of influence between wells.

Rock heat systems

The following presentation will be focused on dimensioning rules for rock heat systems.

The heat output from an rock heat well varies during a year cycle. There is the stationary mean effect and the superimposed flux. During a first time period there is a transient process towards stationary conditions. This will take approximately 20 years. The stationary heat output, $Q(W)$, from a rock heat well can be calculated according to (40):

$$Q = \frac{2p \cdot l \cdot H \cdot (T_{om} - T_R)}{\ln\left(\frac{H}{2R_0}\right)}$$

Equation (40)

where λ = heat conductivity (W/m, K)

H = the wells active depth (m)

T_{om} = undisturbed surrounding temperature (°C)

T_R = stationary mean temperature in the borehole wall (°C)

R_0 = borehole radius (m)

The transient cooling to stationary conditions can be calculated according to (41):

$$T_{om} - T_R(t) = \frac{Q}{2p \cdot l \cdot H} \left[\frac{1}{2} \cdot \left(\ln\left(\frac{4at}{R_0^2}\right) - g \right) \right]$$

Equation (41)

where $t_s = H^2/9a$ (breaking time for stationary conditions)

t = time (s)

a = temperature conduction coefficient = λ/c

c = heat capacity (J/m^3 , K)

γ = a constant = 0,5772 (for $t < t_s$)

EXAMPLE 4

Determine the well temperature for a rock heat well, with the effect output 2000 W, the heat conductivity 3,5 W/m, K, active borehole length of 100 m, undisturbed surrounding temperature 4 °C, borehole radius 0,05 m and the heat capacity 2 200 000 J/m³, K.

After a) 1 year b) 5 years c) 25 years

Solution:

The breaking time, t_s , is:

$$t_s = \frac{H^2}{9a} = \frac{100^2}{9 \cdot 1,6 \cdot 10^{-6}} = 22,0 \text{ years.}$$

Equation (42)

For a) and b) equation (41) is used, for c) equation (40)

Transient cooling:

$$4 - T_R(1 \text{ year}) = \frac{2000}{2p \cdot 3,5 \cdot 100} \cdot \left[\frac{1}{2} \left(\ln \left(\frac{4 \cdot 1,6 \cdot 10^{-6} \cdot 3,1536 \cdot 10^7}{0,05^2} \right) \right) - 0,5772 \right]$$

$$T_R(1 \text{ year}) = -0,87 \text{ °C}$$

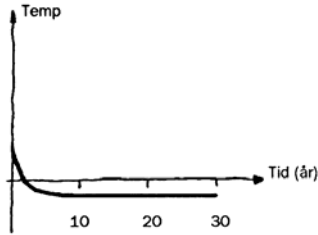
$$4 - T_R(5 \text{ years}) = \frac{2000}{2p \cdot 3,5 \cdot 100} \cdot \left[\frac{1}{2} \left(\ln \left(\frac{4 \cdot 1,6 \cdot 10^{-6} \cdot 1,5768 \cdot 10^8}{0,05^2} \right) \right) - 0,577 \right]$$

$$T_R(5 \text{ years}) = -1,61 \text{ °C}$$

Stationary heat output $t = 25$ years:

$$4 - T_R = \frac{2000 \cdot \ln\left(\frac{100}{2 \cdot 0.05}\right)}{2\pi \cdot 3.5 \cdot 100} = 6.28$$

which gives $T_R = -2,28$ °C



(figure of temperature plot)

From the example one can see that the main temperature decrease for a single well occurs during the first year of drift.

EFFECT FLUX

For a pure effect flux with the constant effect output, q (W/m), from a start time $t = 0$, the 2-dimensional, radial, analytical solution may be written as;

$$q(t) = \begin{cases} 0 & t < 0 \\ q & t > 0 \end{cases}$$

Equation (43)

$$T_{Rq}(t) = -\frac{q}{4\pi \cdot \lambda} \cdot \left[\ln\left(\frac{4 \cdot a \cdot t}{R_0^2}\right) - \gamma \right]$$

Equation (44)

$$\text{when } t > \frac{5R_0^2}{a}$$

where $T_{Rq}(t)$ = temperature at the borehole wall.

The temperature is negative because the solution is only addressing the step-flux. The real well temperature is achieved by superimposing according to (45);

$$T_R(t) = T_{om} + T_{Rq}(t)$$

Equation (45)

This equation is valid for times larger than $5R_0^2/a$. For a rock heat well with the radius $R_0 = 0,055$ m, and the temperature conductivity coefficient, $a = 1,6 \cdot 10^{-6}$ m²/s, this corresponds to a time 2,6 hours. For shorter times the heat capacity of the well water will have an influence. For an optional number of pulses (46), the temperature at the borehole wall at the n:th intervall will be according to equation (47)

$$q(t) = \begin{array}{ll} 0 & t < t_{q0} \\ q_1 & t_{q0} < t < t_{q1} \\ q_2 & t_{q1} < t < t_{q2} \end{array}$$

Equation (46)

$$T_{Rq}(t) = -\frac{q_n}{4p \cdot l} \left[\ln \left(\frac{4a \cdot t_p}{R_0^2} \right) - g \right] - \sum_{i=1}^n \frac{q_i - q_{i-1}}{4p \cdot l} \cdot \ln \left(\frac{t - t_{q,i-1}}{t_p} \right)$$

Equation (47)

$$\left[q_0 = 0; \quad g = 0,5772; \quad t_{q,n-1} + 5 \cdot \frac{R_0^2}{q} < t < t_{q,n} \right]$$

where t_p is an arbitrary reference time.

The effect flux has a limited reach and will only affect the temperature in the borehole wall.

EXAMPLE 5

The effect output for a rock heat well during 3 months is described below. Determine the temperature at the borehole wall for a load of;

| | | |
|----------|-------------------------|------------------------|
| month 1: | $Q(t) = 320 \text{ W}$ | $q = 2,19 \text{ W/m}$ |
| month 2: | $Q(t) = 470 \text{ W}$ | $q = 3,22 \text{ W/m}$ |
| month 3: | $Q(t) = 1170 \text{ W}$ | $q = 8,01 \text{ W/m}$ |

| | |
|-------------------------------------|--|
| Active well depth | $H = 146 \text{ m}$ |
| Heat conductivity | $\lambda = 3,5 \text{ W/m, K}$ |
| Temperature conductivity | $a = 1,6 \cdot 10^{-6} \text{ m}^2/\text{s}$ |
| Reference time | $t_p = 1 \text{ month} =$ $2\,599\,000 \text{ s}$ |
| Borehole radius | $R_0 = 0,05 \text{ m}$ |
| Undisturbed surrounding temperature | $T_{om} = 4 \text{ }^\circ\text{C}$ |

Month 1

$$T_{Rq}(t) = -2,19 \cdot 0,187 - \frac{2,19}{44} \ln\left(\frac{t}{t_p}\right) = -0,409 - 0,0498 \ln\left(\frac{t}{t_p}\right)$$

Month 2

$$\begin{aligned} T_{Rq}(t) &= -3,22 \cdot 0,187 - \frac{2,19}{44} \ln\left(\frac{t}{t_p}\right) - \frac{3,22 - 2,19}{44} \ln\left(\frac{t - t_p}{t_p}\right) = \\ &= -0,602 - 0,0498 \ln\left(\frac{t}{t_p}\right) - 0,0234 \ln\left(\frac{t}{t_p} - 1\right) \end{aligned}$$

Month 3

$$\begin{aligned} T_{Rq}(t) &= -8,01 \cdot 0,187 - \frac{2,19}{44} \ln\left(\frac{t}{t_p}\right) - \frac{3,22 - 2,19}{44} \ln\left(\frac{t - t_p}{t_p}\right) - \frac{8,01 - 3,22}{44} \ln\left(\frac{t - 2t_p}{t_p}\right) = \\ &= -1,49 - 0,0498 \ln\left(\frac{t}{t_p}\right) - 0,0234 \ln\left(\frac{t}{t_p} - 1\right) - 0,109 \ln\left(\frac{t}{t_p} - 2\right) \end{aligned}$$

Table 7 below shows the actual well temperature.

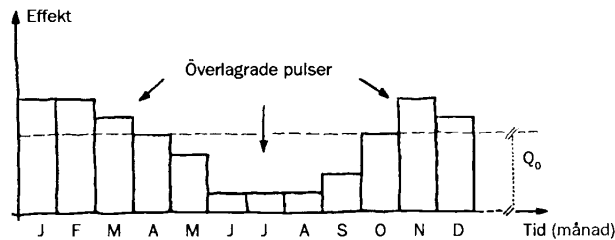
Table 7.

| t/t_p | 0,5 | 1,0 | 1,5 | 2,0 | 2,5 | 3,0 |
|-------------------------------------|------|-------|-------|-------|-------|-------|
| $T_{Rq}(t) \text{ } ^\circ\text{C}$ | 0,37 | -0,40 | -0,60 | -0,63 | -1,47 | -1,56 |
| $T_R(t) \text{ } ^\circ\text{C}$ | 3,63 | 3,60 | 3,40 | 3,37 | 2,53 | 2,44 |

DIMENSIONING OF A ROCK HEAT WELL

The dimensioning rules for rock heat wells are based on given effects from which the well's temperature is calculated. The dimensioning is mainly determined by the lowest well temperature during the year cycle. In Figure 69 it is shown how the effect output during a year cycle can be described.

Figure 69. Effect output during a year cycle.



The solution to the problem is obtained through superposition according to equation (48).

$$T_{TOT}(t) = T_R + T_{Rq}(t)$$

Equation (48)

where $T_{TOT}(t)$ = the total well temperature at time t
 T_R = stationary temperature due to the mean effect output, Q_0

$T_{Rq}(t)$ = temperature decrease due to superimposed flux

The stationary component T_R is given by equation (40). The transient process to stationary conditions is here neglected. If this process is to be taken under consideration, equation (41) should be used to calculate T_R i.e. $T_R(t)$. For times shorter than a year consideration should be taken to the transient process, compare with Example 4. The temperature change due to superimposed flux is treated according to equation (47).

SEVERAL WELLS

Close wells will affect each other thermally. For a system of wells a lower heat output capacity per well is obtained than for independent wells. The influence between the wells is a pronounced long-time effect. For well distances significantly larger than 10 m the influence is very small after the first years effect output. The influence between wells is only affected by the constant mean effect output during the year. The influence between 15 wells at the 25th year is shown in Figure 70. The wells are 150 m deep each, the distance between the wells is 20 m and the energy output is 225 MWh/year. The figure shows the isotherms at a depth of 77 m.

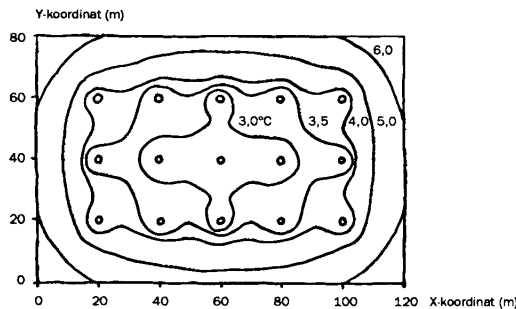


Figure 70. Example of influence between 15 rock heat wells. Isotherms are shown for a horizontal cross section 77 m deep, the 25th year. (After Claesson J et al, 1985)

To decrease the influence between wells the boreholes can be drilled at an angle, so that the distance between holes increases with depth.

The annual heat output per well for six wells relative to a single well is presented in table 8.

Table 8. Annual heat output per well, for six wells compared to an undisturbed well. (After Claesson J et al, 1985)

| Year | 1 | 5 | 25 |
|-------------|----------|----------|-----------|
| B(m) | | | |
| 4 | 0,726 | 0,542 | 0,483 |
| 10 | 0,970 | 0,730 | 0,620 |
| 20 | 0,990 | 0,886 | 0,743 |
| 40 | 1,000 | 0,981 | 0,880 |
| 100 | 1,000 | 0,996 | 0,991 |

The constant mean effect output during the year will determine the influence between wells. The temperature decrease is given by equation (49):

$$T_{om} - T_R(t) = \frac{Q_0}{N} \cdot \frac{1}{2pl \cdot H} \cdot g(t / t_s, R_0 / H, B_{12} / H \dots)$$

Equation (49)

where
$$t_s = \frac{H^2}{9a}$$

where the g-function is the dimensionless temperature response function. For two different borehole radius's (50) applies;

$$g(t / t_s, R_0' / H \dots) = g(t / t_s, R_0 / H \dots) - \ln[(R_0' / H) / (R_0 / H)]$$

Equation (50)

where R_0' = actual borehole radius

The g-function describes the influence between the wells and has to be computer calculated. This has been made by Eskilson (1987). The g-function is presented in form of a diagram, for approximately 40 well-system configurations. Those are presented in Appendix.

EXAMPLE 6

Given 4 wells in a square, distance between them 15 m. Borehole radius 0,055 m. Active depth 150 m. The heat output is constantly 30,4 W/m.

$$\lambda = 3,5 \text{ W/m, K} \quad C = 2,16 \text{ MJ/m}^3, \text{ K}$$

Determine the temperature decrease in the well after;

- a) 5 years b) 25 years

Solution.

Equation (49):

$$T_{om} - T_R(t) = \frac{Q_0}{N} \cdot \frac{1}{2pl \cdot H} \cdot g(t / t_s, R_0 / H, B_{12} / H \dots)$$

$$\frac{Q_0}{N \cdot H} = 30,4 \text{ W/m}$$

$$t_s = \frac{H^2}{9a} = \frac{150^2}{9} \cdot \frac{2160000}{3,5} = 48,9 \text{ years}$$

$$\frac{R_0}{H} = 0,055 / 150 = 0,00037; \quad B / H = 15 / 150 = 0,10$$

$$g\left(\frac{5}{48,9}, 0,0005\right) = (\text{diagram}) = 6,7$$

$$g\left(\frac{25}{48,9}, 0,0005\right) = (\text{diagram}) = 9,1$$

Transformation to the actual borehole radius according to (50):

$$g\left(\frac{5}{48,9}, 0,00037\right) = 6,7 - \ln\left[\frac{0,00037}{0,0005}\right] = 7,0$$

$$g\left(\frac{25}{48,9}, 0,00037\right) = 9,1 - \ln\left[\frac{0,00037}{0,0005}\right] = 9,4$$

The temperature decrease now becomes:

$$T_{om} - T_R(5\text{ years}) = 30,4 \cdot \frac{1}{2p \cdot 3,5} \cdot 7,0 = 9,7 \text{ }^{\circ}\text{C}$$

$$T_{om} - T_R(25\text{ years}) = 30,4 \cdot \frac{1}{2p \cdot 3,5} \cdot 9,4 = 13,0 \text{ }^{\circ}\text{C}$$

Dimensioning of rock heat systems is made the same way as for a single well, see Figure 69. Since the temperature decrease for a system of wells is significant during the entire life of the plant, compare to Figure 56, the solution is superimposed for the transient process to effect flux according to (51)

$$T_{TOT}(t) = T_R(t) + T_{Rq}(t)$$

Equation (51)

where $T_R(t)$ = temperature due to the influence between wells (see equation (49))

$T_{Rq}(t)$ = temperature decrease due to the superimposed flux (see equation (47))

HEAT RESISTANCE IN BOREHOLES

When dimensioning rock heat systems one is interested in learning the fluid temperature. This mainly depends on the heat collector design, the fluid properties and the flow in the channel, i.e. whether laminar or turbulent flow. The heat resistance, m_R (K/(W/m)), between fluid and rock wall is defined by the relationship (52)

$$T_R - T_f = m_R \cdot q$$

Equation (52)

where T_R = the borehole temperature (°C)

T_f = fluid temperature (°C)

q = effect output (W/m)

In an open rock heat system, see Figure 55, typical values of the heat resistance are 0,01-0,10 K/(W/m)). In a closed system they are 0,10-0,20 K/(W/m)).

At a precise dimensioning one wants to know the fluid inlet and outlet temperature, T_{fin} and T_{fout} .

The difference between in- and outlet temperature is given by the heat output, Q (W), according to (53)

$$T_{fout} - T_{fin} = \frac{Q}{C \cdot V}$$

Equation (53)

where C = brine heat capacity (J/m³, K)

V = volume flow (m³/s)

The fluid temperature varies in the well. This temperature variation may be neglected if one defines a constant temperature T_f , according to below. T_f is defined as the average value between in and outlet channel.

$$T_f = \frac{1}{2}(T_{fin} + T_{fout})$$

Equation (54)

Equations (52), (53) and (54) may now be written;

$$T_{fin} = T_R - m_R \cdot \frac{Q}{H} - \frac{Q}{2C \cdot V}$$

Equation (55)

$$T_{fout} = T_R - m_R \cdot \frac{Q}{H} + \frac{Q}{2C \cdot V}$$

Equation (56)

EXAMPLE 7

Determine the fluid temperature for a

a) open rock heat system, $m_R = 0,05 \text{ K/(W/m)}$

b) closed rock heat system, $m_R = 0,15 \text{ K/(W/m)}$

$Q = 2000 \text{ W}$ $H = 100 \text{ m}$ $C = 4\,000\,000 \text{ J/m}^3, \text{ K}$

$V = 0,5 \cdot 10^{-3} \text{ m}^3/\text{s}$ $T_R = 4 \text{ }^\circ\text{C}$

Solution: Equation (55) and (56) are used:

Open system

$$T_{fin} = 4 - 0,05 \cdot \frac{2000}{100} - \frac{2000}{2 \cdot 4 \cdot 10^6 \cdot 0,5 \cdot 10^{-3}} = 2,5 \text{ }^\circ\text{C}$$

$$T_{fout} = 4 - 0,05 \cdot \frac{2000}{100} + \frac{2000}{2 \cdot 4 \cdot 10^6 \cdot 0,5 \cdot 10^{-3}} = 3,5 \text{ }^\circ\text{C}$$

Closed system

$$T_{fin} = 4 - 0,15 \cdot \frac{2000}{100} - \frac{2000}{2 \cdot 4 \cdot 10^6 \cdot 0,5 \cdot 10^{-3}} = 0,5 \text{ }^{\circ}\text{C}$$

$$T_{fout} = 4 - 0,15 \cdot \frac{2000}{100} + \frac{2000}{2 \cdot 4 \cdot 10^6 \cdot 0,5 \cdot 10^{-3}} = 1,5 \text{ }^{\circ}\text{C}$$

In an open rock heat system, with prescribed volume flow and effect output, the outlet water temperature will therefore be 3,5 °C if the temperature of the borehole wall is 4 °C. In a closed system, on the other hand, the outlet brine temperature is 1,5 °C. The difference is explained by the better heat transfer in an open system since the water is in direct contact with the borehole wall. The advantage with the closed system is that the water in the borehole can be frozen, if a brine with anti-freeze agent is used. The phase change energy is thereby utilized.

POTENTIAL AND ECONOMY FOR NATURAL HEAT SYSTEMS

The introduction of heating systems based on natural heat sources, and heat pumps, has increased rapidly in recent years. The number of Swedish plants 1983 was approximately 20 000, distributed according to table 9.

Table 9. Natural heat systems in Sweden 1983.

| | |
|--------------------------------|--------|
| Soil heat | 13 000 |
| Lake heat | 2 000 |
| Groundwater heat and rock heat | 5 000 |

The technique is nowadays well established. A considerable expansion was made 1984-87. The number of heat pumps was 1986 120 000, of which 90 % where for small family houses. Totally those heat pumps produce ~11 TWh heat. Today, 1991, the number of heat pumps are estimated to 200 000.

Table 10. Application and potential for ground heat systems.

A schematic presentation of appliance fields and estimated potential for natural heat systems is shown in table 10.

| | Small houses | Block of flats | Group units | Municipal heat | Gross potential TWh/year | Net-potential TWh/year |
|------------------|--------------|----------------|-------------|----------------|--------------------------|------------------------|
| Soil heat | x | x | o | - | 55 | 1 |
| Lake heat | o | x | x | x | >100 | 15 |
| Groundwater heat | x | x | x | o | 15 | 5 |
| Rock heat | x | x | o | - | 10 | 2 |
| Geothermal | - | - | x | x | >30 | 2 |
| | | | | | Sum: | 25 |

Source: BFR ground heat group 1984-87
 x = Suitable application, good technique and economy
 o = Application uncertain (costs, size, hindrances)
 - = no current application

Gross potential means technically obtainable heat amount. Net potential refers to the obtainable resource and assumes a broad introduction of water carried heating systems.

The cost for some constructed plants are shown in table 11. The costs refers to complete systems, including heat pump, collector, wells, pipes, buildings and taxes and is based on a real interest of 6 % and 15 years pay off.

Table 11. Examples of costs for natural heat based heat pump systems, 1982. (6 % real interest, 15 years pay off)

| | Effect (kW) | Specific investing cost SEK/kW | SEK/ kWh, year | Cost Capital + Drift SEK/kWh, year | Pay-off years | Note Type of project |
|-----------------------------------|------------------------|---|-------------------------------|---|--------------------------|-------------------------------------|
| Soil heat | 10 | 5500 | 2,2 | 0,35 | 14 | Small houses |
| Soil heat | 200 | 4000 | 2,0 | 0,30-0,35 | 13 | Large buildings |
| Lake heat | 750 | 3400 | 0,8 | 0,20-0,25 | 8 | Group central |
| Lake heat | 4000 | 2500 | 0,6 | 0,15-0,20 | 5 | Municipal heat |
| Groundwater r heat | 300 | 4500 | 1,0 | 0,20-0,25 | 8 | Large buildings |
| Groundwater r heat | 1000 | 3200 | 0,5 | 0,15-0,20 | 5 | Group central |
| Rock heat | 10 | 6500 | 2,6 | 0,40 | 16 | Small houses |

Source: BFR ground heat group 1984-87

HEAT STORAGE

GENERAL ON ENERGY STORAGE

The purpose of all energy storage is to save energy from one point of time to another when the energy is to be used. The main purpose for storage is simply economical, it has to be cheaper to produce the energy at another time than at the time of usage. The economically most advantageous alternative is often a compromise where a part of the required energy is taken from the storage and the other part is produced directly.

Depending on energy form, storage can be done in different ways:

| ENERGY FORM | STORAGE TYPE |
|-------------|-----------------------|
| Kinetic | Balance wheel |
| Potential | Hydropower |
| Chemical | Salts |
| Pressure | Pressure accumulator |
| Electrical | Superconductor |
| Thermal | Thermos, heat storage |

Short-term storage of energy, for hours or days, are common in the industry and also in small family houses. Long-time storage occurs for instance in the reservoirs for hydropower. The more frequent usage of short-term storage allows these to have a fairly high production cost per stored energy amount as opposite to long-term heat storages that are charged and emptied once a year. This, together with the size-dependent energy losses, result in that long-time storage of energy necessarily becomes a relatively large-scale activity.

In this text only storage of thermal energy, heat storage, will be treated from now on. Large parts of the texts are obtained from G26:1986, Bygghälsningsrådets (BFR) publication "Energilagring" (Swe.), basis for BFR:s 3-year plan 1987/88-1989/90.

Function of a heat storage

Heat storage is a necessity if a more substantial use of solar energy is to become a reality in Sweden. Technique for storage of heat will also increase the possibilities to utilize waste heat sources. The storage thereby increases the effectiveness of energy systems and can compete with other heat systems.

There are several reasons to use a heat storage in an energy system. One reason is the ability to use energy which is only available during periods with small energy needs, such as solar energy and waste energy. A surplus of energy results in lowered energy prices during such periods. By storing the energy it can thereby be used in a better way. Another reason for a heat storage is its ability to level out effect variations and thereby decrease the top effect for a production plant. This creates a lower cost of heat production.

There are also distribution advantages because of a higher deliverance security. Heat storage is also advantageous from a preparedness point of view.

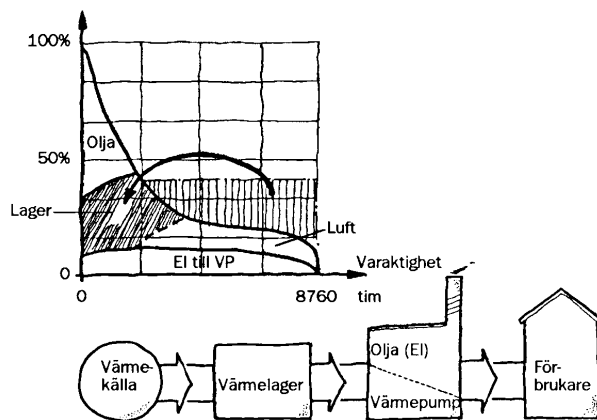
Environmental benefits are directly achieved if the use of fossil fuel can be decreased by storing solar energy. A more even heat production over the entire year is also beneficial for the environment.

Heat can be stored short-term or long-term. This refers to the number of annual turnovers. A short-term storage is turned over perhaps 50-100 times per year, with charging

and discharging during a few hours or days. For long-term storage the heat is charged during a whole season to be used during another season of the year. Usually the storage is from summer to winter.

Usually a distinction is made between low-temperature and high-temperature storage. There is no sharp edge between those two types but usually high or low indicates whether the stored heat can be used directly for heating or if a heat pump is necessary to raise the temperature. This means that a high-temperature storage delivers heat at a temperature above 55 °C while low-temperature storages usually is far below this temperature. A schematic example of a heat storage system is shown in Figure 71.

Figure 71. Example of a heating system based on air-heat pump, long-term storage and top-heat with oil.



A coffee-thermos and a water heater are examples of short-term storages for high temperatures. In the case with the water heater a much larger heat effect would be necessary if the hot water were to be produced instantly at the time of usage. The water is instead heated during a longer time and stored until there is a need for it. A few examples of the advantages with a water heater are presented below.

Example: In a water heater with the volume 150 liters ($0,150 \text{ m}^3$) incoming water is heated from 5 °C to 55 °C.

The heating takes place during night between 11 p.m. and 07.00 a.m. so that shower water is available in the morning. Alternatively the water can be heated directly at the time of showering, for a water flow of 0,2 l/s ($0,2 \cdot 10^{-3} \text{ m}^3/\text{s}$) to a temperature of 40 °C. In the first case an effect of 1,1 kW is required, while in the other case it would require 29,2 kW to manage the hot water demand for the morning shower.

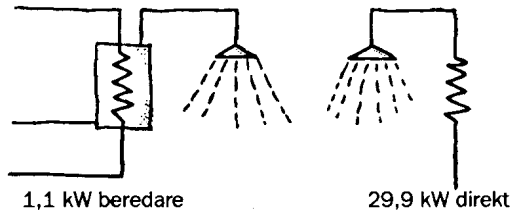


Figure 72. The water heater in a common family house is a short-term storage for high temperatures and results in an improved effect-usage.

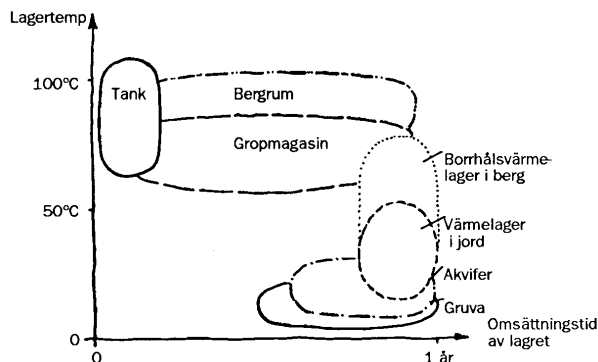
The long-term goal with seasonal storage is to store solar heat from summer to winter. The stored heat shall mainly be used for heating purposes. During longer storage periods heat losses are more important than during short-term storage. Consider the example with the thermos where one knows that the coffee will be hot for a few hours but not for longer times.

Since the heat losses are strongly dependent of the size of the storage long-term storage is large-scale. Large storages demands cheap storage-media. For this and other reasons connected to the storage-size, long-term storages has increasingly been placed below the ground. There are several types of long-term storages. They differ with regard to construction while the storage medium usually is water or ground (soil, rock, sand). There are also storages that use both these storage medias.

| STORAGE TYPE | STORAGE MEDIUM |
|-----------------------------|---------------------|
| Cisterns, pit-storage, mine | Water |
| Underground rock cavity | Water |
| Filled underground cavities | Water + stones |
| Aquifers | Water + sand |
| Borehole storage | ground (rock, clay) |

Common for the water storages are that they can handle high temperature effects both when charging and discharging, since this is only a question of pump-capacity. They can therefore be used both for long- and short-term storage. In cases where heat is stored in clay, soil or rock, the charging- and discharging-sequence is slow because the heat has to be transferred from the ground to the brine (usually water) before it can be utilized. For this reason this type of storage is only suitable for long-term storage. The application-areas for different storage-types are shown in Figure 73.

Figure 73. Application-areas for different kinds of heat storage, depending on their technical and economical prerequisites.



It is not obvious which storage system that should be used in a specific case because this depends on several prerequisites of which the effect-demand may be the most important. For below ground storage the geological conditions may rule out some storage-types. The storage must be designed so that it can be integrated in the buildings area. There is also the economical matter of judgment between the different technically feasible solutions. Drift security- and environmental considerations also has to be made.

Heat storage in water

Heat storage in water is made in cisterns above ground, in underground rock cavities, in stone-filled underground rock cavities and in pit-storages in soil and rock. In all these cases the heat is stored in the form of hot-water that is contained in a closed volume. These storage-types can give and take a large effect since the hot water can be rapidly pumped to/from the storage.

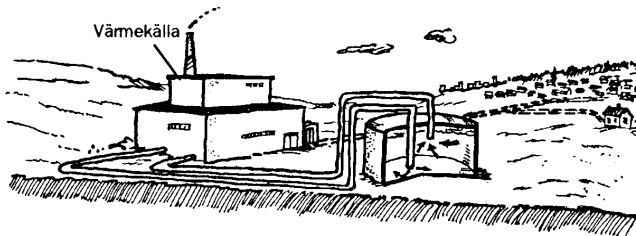


Figure 74. Heat storage in a cistern.

For heat storage above ground a cistern in steel or concrete can be used. The heat losses are minimized through heat insulation of walls, floor and roof. The tank can be pressureless or pressurized. In pressurized systems water temperatures above 100 °C can occur which increases the storage-density [J/m^3]. The production cost for pressurized cisterns are higher. There are security problems in connection with hot water storage above ground. A certain

security distance to surrounding buildings is necessary. The storage cisterns are 10 000 - 100 000 m³ large if they are (also) to be used for long-term storage, which makes them demand a fairly large area.

There are today many cisterns built, often in district heat systems where they work as short-term storages. They are used to even out the effect-load on the municipal heat system (peak-shaving). This is especially for hot tap-water which is mainly consumed during mornings and evenings. The storage tank is charged during the other hours of the day.

Hot water storage in rock cavities is a development of the technique to store oil in rock cavities. There is a large number of such oil-reservoirs in Sweden. In these reservoirs the oil "floats" on a water layer in the cavity's bottom. To make the oil easy to pump it is held warm. With this knowledge of construction technique and heat losses it came natural to consider hot water storage in rock cavities.

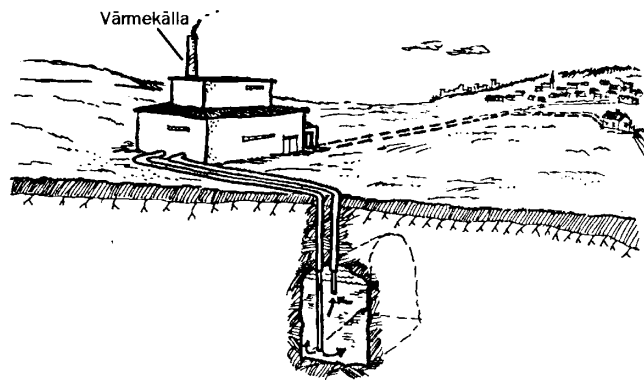


Figure 75. Heat storage in underground rock cavity.

Rock cavities are not heat insulated and must therefore be built large to keep the relative heat losses down. Rock cavity storages are built in a size of a few 100 000 m³. It is the rocks strength that limits the volume of the storage. By leaving blast stone to remain in the cavity this can be built

for larger volumes without construction-strength problems. However, an open rock cavity, filled with water, has a larger heat capacity since water has a heat capacity of 1,16 kWh/m³, °C and the rocks capacity is only (approximately) half of the waters. The heat capacity in the stone-filled cavity storage will then decrease to ca 0,9 kWh/m³.

There are examples of existing rock cavities that has come to use for heat storage. Examples of that are abandoned mines which can be used despite their often unfavorable geometry since construction costs are low. The many drifts and the higher heat conductivity in rocks with high contents of metals makes for large heat losses. At present, when Sweden has an over capacity of rock cavities for oil storage, alternative usage are discussed for these, for instance heat storage. There is one large hot water storage (100 000 m³) in Sweden, connected to the city of Uppsalas municipality heat system.

An alternative to tanks and rock cavities for heat storage is to build a pit-storage in soil or rock. This is a common type of heat storage. Such a storage must be covered and insulated on the ground surface. Small storages must be insulated all way round. One reason for a pit-storage is that it is a simple construction, since pressure forces are taken up by surrounding ground. The problem is to achieve a cheap, watertight seal against the surroundings. For small storage volumes, pit-storage is cheaper than cisterns or rock-cavities. Costs for different kinds of water storages are presented in table 12 below. The cost also includes installation works for connection to the storage.

| Type | Volume m ³ | Storage tempe- rature °C | Production cost SEK/m ³ | SEK/kWh |
|-----------------------------|--------------------------|-----------------------------|---------------------------------------|---------|
| Steel tank | 100 | 55-95 | 1700 | 37 |
| ”- | 5 000 | ”- | 600 | 13 |
| ”- | 50 000 | ”- | 340 | 7,3 |
| Concrete tank | 5 000 | 5-70 | 800 | 10 |
| Pit-storage | 5 000 | 55-95 | 360-400 | 8-9 |
| ”- | 40 000 | ”- | 170-200 | 3,7-4,3 |
| Stone-filled pit | 88 000 | 5-77 | 75 | 1,2 |
| Open rock cavity | 105 000 | 40-90 | 190 | 3,2 |
| ”- | 740 000 | 65-117 | 130 | 2,2 |
| Stone-filled rock cavity | 874 000 | 58-102 | 90 | 2,2 |

After G26:1986

Table 13. Production costs for different kinds of storages (in 1986 years prices)

Heat storage in aquifers

An aquifer is a geological formation from which groundwater can be extracted. The geological formations of most interest for heat storage is of the same type as those most desirable for fresh water extraction. This becomes a problem since aquifers close to built areas often are used as water supplies. It is not possible to combine both functions.

Aquifers has been used in large scale for cooling of industrial buildings in warmer climates. Cold groundwater is pumped up from wells and are returned to the aquifer after heating. Thereby a successive heating of the aquifer occurs. With correct dimensioning such a system can be used for

cooling during summer and heating during winter. During the heating season heat pumps are often used to raise the temperature of the water. This type of systems exist in Sweden.

In an aquifer storage the heat is stored in the groundwater and in the sand or gravel layer which the aquifer is situated in. This type of storage can be made very large and are mostly suited for long-term storage of heat. The heat storage is designed with a system of wells so that the water flow in the ground can be controlled. The heat capacity varies a lot due to the soil porosity. Possible discharge and charge effects depend on the aquifers permeability, i.e. the rate of which water can be extracted and returned.

At the charge hot water is delivered to wells in the aquifer center while, at the same time, cold water is extracted from wells in the periphery. The extracted (cold) water is heated in heat exchangers and returned to the aquifer center. The hot water is thereby flowing towards the aquifers periphery wells whereby the aquifers sand and gravel are heated. As charging occurs the heat front will spread outwards, towards the periphery of the aquifer. The thermal velocity is approximately half of the water velocity. When heat is extracted, water is pumped in the opposite direction.

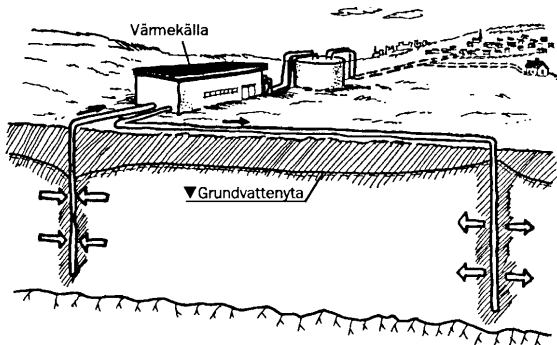


Figure 77. Heat storage in an aquifer with two-well system.

Aquifer storage is best suited for low temperatures, due to water chemical problems that can arise at higher

temperatures. Aquifer storages also often have disc-shaped geometries, which is not the best geometry from a heat loss perspective. The production cost for an aquifer storage consists of wells, pumps and the pipe system that is needed for charge and discharge of heat. The number of wells and borehole depth depends entirely on local geological conditions and is therefore difficult to predict. For three existing plants in Sweden, the heat production cost is 0,15-0,30 SEK/kWh.

Heat storage in the ground

Heat storage in ground refers to heat being stored in soil or rock that is heated during the charging phase and cooled during the discharge phase. In this type of storage heat is transferred to/from the ground by holes or tubes in the ground. Those are usually vertical but can also be angled or horizontal. In borehole heat storage in rock, holes with a diameter of ca 100 mm are usually drilled, to a depth of 50-100 m. In existing storages the borehole distance is ca 4 m. Recent studies shows, however, that it is economically beneficial to increase the borehole depth to ca 125 m, also for relatively small storages.

The heat exchanger system in the boreholes can be open or closed. In the closed case the brine (water) circulates in a closed pipe circuit in the borehole whereby the heat is transferred through the pipe-wall to the surrounding ground by conduction,. In the open system the circulating water is in direct contact with the borehole wall. The heat transfer is thereby improved. The open system has disadvantages, however, since water-chemical problems can occur. In addition to this it is necessary that the groundwater surface must be relatively close to the ground surface. The heat capacity in rock is $0,6 \text{ kWh/m}^3, ^\circ\text{C}$. Borehole heat storages

are estimated to be one of the most promising methods for long term heat storage.

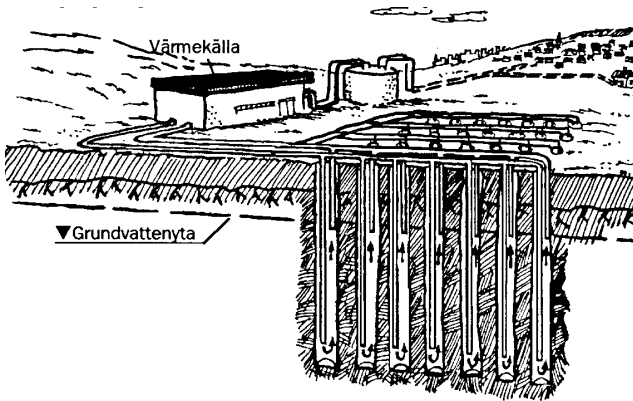


Figure 78. Borheole heat storage in rock, open system.

Closed tube systems are always used in soil whereby the tubes are pressed, flushed or drilled down in the ground. Down-pressing is best in clay while flushing is used for sandy soil. Drilling in soil is normally 5-10 times more expensive than drilling in rock. The soil storage depth is limited by the soil layer thickness. The deepest soil layer in Sweden is 35 m thick. The distance between the vertical tubes is ca 2 m, i.e. half compared to rock. The reason for this is that the heat conductivity in soil is ca 1 W/m, °C while it is 2 W/m, °C in gneiss and 3,5 W/m, °C in granite. The heat storage capacity however is higher in soil, 0,8-1,0 kWh/m³, °C against 0,6 for rock. The variations in soil are large depending on porosity and water content.

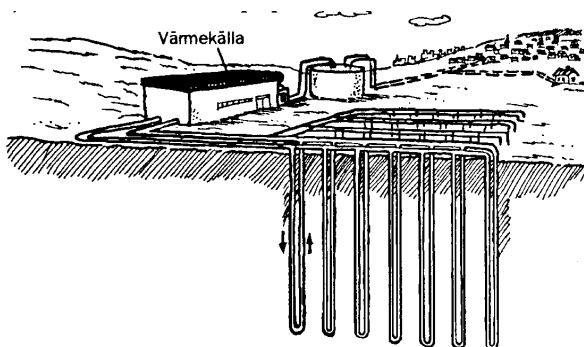
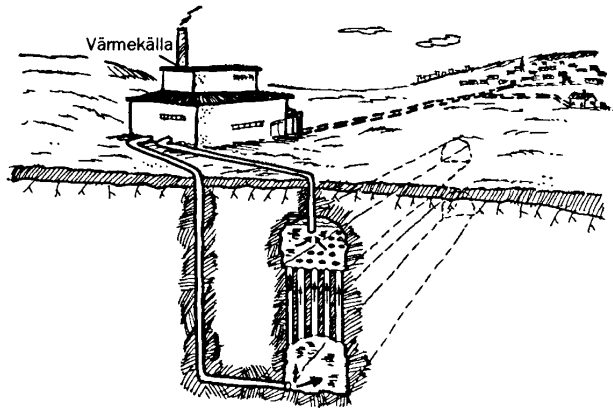


Figure 79. Heat storage in soil, closed system.

The thermal resistance in ground heat storage makes combinations of short- and long-term storage desirable. This can be done if some type of water cistern is used in combination with the ground storage. This gives a possibility of charging and discharging large effects, during shorter times. In such a system it would be possible to handle both day- and seasonal variations in heating need. Plans have been brought forward about constructing storage systems with rock cavity storage and borehole storage in

Figure 80. Heat storage in rock with vertical boreholes, combined with a rock cavity storage.



combination. This would make the system less resistant. A smaller test plant exists in Kerava, Finland.

The costs for a heat storage, in 1986 years prices, are given in table 13 below. In the production cost it is assumed that the heat storage is located close to a heat source or distribution net from a heat source. It should be observed that a specific production cost (SEK/kWh) is not the same as a heat cost but it is the production cost divided by the extracted heat during a year with full drift.

| Type | Volume m ³ | Storage temp. °C | Production cost SEK/m ³ | SEK/kWh |
|---------------|--------------------------|---------------------|---------------------------------------|---------|
| Soil storage* | 100 000 | 4-16 | 8-12 | 0,7-1,1 |
| -''- | 100 000 | 10-50 | 8-12 | 0,2-0,3 |
| Rock storage | 100 000 | 4-16 | 15-30 | 2,1-4,1 |
| -''- | 200 000 | 20-50 | 25-40 | 1,3-2,2 |
| -''- | 2 000 000 | 20-80 | 30-35 | 0,8-1,0 |

* soil with large permeability

Table 14. Specific production cost for different storage types.

Figure 81 shows a map of Sweden with a specification of different heat storage projects. There is a large number of steel-cistern storages not presented in figure 81.

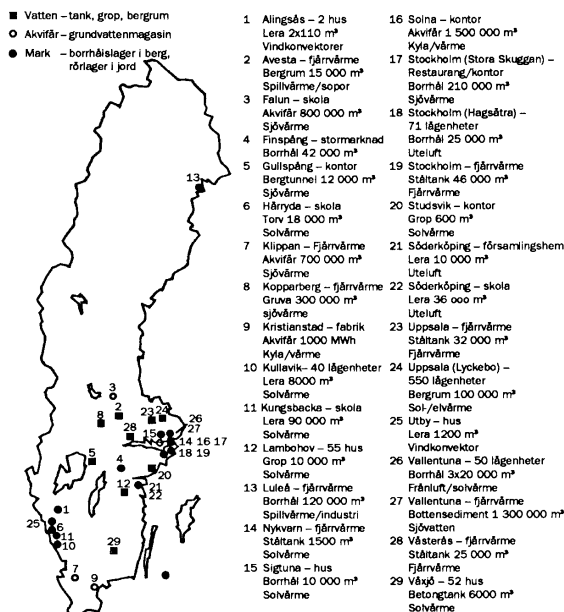


Figure 81. Full-scale heat storages in Sweden.

Potential for heat storage

The potential for heat storage depends partly on technical prerequisites and partly on the actual imbalance between heat demand and existing or possible heat production. Where those coincide, the technical-economical potential is found. This means that good storage conditions must coincide with both energy demand and energy supply.

For ground heat storage it is necessary to find suitable geological conditions. For rock cavities and borehole storages it is possible, with a few exceptions, to find suitable conditions all over Sweden.

For heat storage in clay, however, the geological conditions are very much geographically limited. There are clay-areas in connection to a large number of densely populated areas but the depth of the clay is usually rather small. The conditions for aquifer storage has been studied extensively and over 200 places with suitable geological conditions have been found, close to densely populated areas.

In summary, the nature-conditions are only as an exception a limit to the heat storage potential in Sweden. To better use the 120 municipal heat systems it is estimated that by year 2000 approximately 2800 GWh should be long-term stored in high temperature storages, in 70 cities in Sweden. In those 70 cities and in 40 others there is also a large need of short-term storage. In other words, a large number of potential projects exist around Sweden.

Technical-economical potential

With the energy-prices of today and the costs of storage, the technical-economical potential is small. Heat storage will be economically advantageous only as an exception. The conditions are worst if the storage is charged with solar heat. For large-scale solar heat systems, the heat production

cost of today after storage is ca 0,45 SEK/kWh, which is the double of the mean consumer price for municipal heat.

Future perspectives

Despite the above, a rapid social development takes place that can make solar heat in combination with heat storage something that has a large future potential. The expected and partly announced energy policy will lead to that renewable energy sources will come closer to the conventional systems, from an economical viewpoint. Season-differentiated taxes on electricity and heat, with lower prices during the summer, are examples that strongly benefits long-term heat storage. Actually, this is the basic prerequisite to make long-term storage an alternative. Higher prices for electricity are to be expected. This will be beneficial to all other forms of alternative energy.

The production costs keeps getting reduced for each borehole heat storage being built, at the same time as the conventional plants for heat production are getting more expensive. We have now reached a point where solar heat, in combination with long-term storage, can heat large living areas. It is not the technique but the economy that is holding the expansion back. In a longer perspective however, we have reason to expect a solar heat expansion. If civil alert- and environmental benefits are taken into consideration, then long-term heat storage must become a natural part of future energy systems.

DIMENSIONING OF HEAT STORAGES

This episode is a general description of how storage size, heat losses and temperature fields are calculated. With the aid of computers the heat conduction equation can be solved numerically and thereby the transient heat flow from a heat storage can be calculated. This gives the temperature field in the ground, surrounding the storage. From these calculations, the occurring heat losses can be summarized.

From numerical and analytical calculations the Ground Heat Group, at the Technical University of Lund, has produced simple hand-calculating formulas which will be used in this book. For the specially interested in dimensioning of ground heat systems and heat storage the book "Markvärme, En handbok om termiska analyser, del 1-3 (Johan Claesson et al, Byggforskningsrådet T16-18:1985)" is recommended. The following presentation is mainly based on that book.

Measures of effectiveness

There are some commonly used measures of effectiveness applied to describe the performance of heat storages. The most common is energy efficiency (η_E) which describes heat discharge amount (E_-) in relationship to heat charge amount (E_+).

$$h_E = E_- / E_+$$

Equation (57)

This will however not tell the whole truth about the storage's function. There will always be a temperature loss in the storage which means that the stored energy's temperature (T_+) is higher than the extracted (T_-). To describe this temperature loss the notation temperature

efficiency (η_T) is used. T_+ and T_- are mean temperatures weighted against the energy amount.

$$h_T = (T_- - T_0) / (T_+ - T_0)$$

Equation (58)

where T_0 is the annual mean temperature in the ground. A summarized weighted measure of the heat storage's efficiency is obtained by multiplying η_E and η_T . This product gives the efficiency according to the second proposition of the law of thermodynamics. It is however used very rarely in heat storage applications.

The basic equation for description of heat conductivity is the 3-dimensional, non-stationary heat conductivity equation. The temperature $T(x,y,z,t)$ describes the temperature field in the ground, disregarding the movement of groundwater.

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = \frac{1 \cdot \partial T}{a \cdot \partial t}$$

Equation (59)

$a = \lambda/C$ is the thermal diffusivity (m^2/s) where λ is the heat conductivity and C is the volumetric heat capacity ($J/m^3, K$). For stationary temperature conditions the time derivative is $= 0$, which gives a considerably easier solution of the equation. See the chapter concerning heat transfer.

Thermal properties

A substance's ability to store sensible heat (heat stored as heat) depends on the substance's heat capacity. Heat capacity is a material property which therefore varies between different storage media. Water has a higher storage capacity, c ($kJ/kg, K$), than all other substances. In the following compilation however, the volumetric heat

capacity, C (kJ/m^3 , K or mostly kWh/m^3 , K) will be used. $C = c \cdot \rho$, where ρ is the substance's density. In table 14, below, the thermal properties for water and some different species of stone are presented.

| | ρ (kg/m^3) | λ (W/m, K) | c (J/kg, K) | C (kWh/m^3 , K) |
|-----------|-------------------------------|----------------------------------|-----------------------------|--------------------------------|
| Water | 1000 | 0,6 | 4180 | 1,2 |
| Granite | 2700 | 2,9-4,2 | 830 | 0,6 |
| Pegmatite | 2700 | 2,9-4,2 | 830 | |
| Syenite | 2750 | 2,2-3,3 | 850 | |
| Diorite | 2800 | 2,2-3,3 | 850 | |
| Gabbro | 3000 | 2,2-3,3 | 860 | |
| Diabase | 3000 | 2,2-3,3 | 860 | |
| Quartzite | 2650 | 5,0-7,0 | 790 | |
| Gneiss | 2700 | 2,5-4,7 | 830 | 0,6 |
| Granulite | 2700 | 2,7-4,5 | 830 | |
| Marble | 2700 | 2,5-3,5 | 770 | |

After Lars O Ericsson (1985)

Table 14. Density, heat conductivity and volumetric heat capacity for different materials.

Volumetric heat capacity, see table 14, is a measure of how much heat that must be added a substance per m^3 to increase the temperature with 1°C . Consequently, for water it would require 48 kWh to raise the temperature 40°C for 1 m^3 ($C = 1,2 \text{ kWh/m}^3$, K) while 24 kWh is enough for the corresponding temperature increase for 1 m^3 of gneiss or granite ($C = 0,6 \text{ kWh/m}^3$, K).

EXAMPLE 8

How large volume of water is required to store 1 000 000 kWh if the heat storage's highest mean temperature is 70 °C and it's lowest mean temperature is 30 °C? How large volume of rock is required for a borehole heat storage with the same conditions?

Solution:

$$V = E_+ / (C \cdot \Delta T) \text{ gives}$$

$$V = 1\,000\,000 / (1,2 \cdot 40) = 20,833 \text{ m}^3 \text{ of water.}$$

For granite, with $C = 0,6 \text{ kWh/m}^3 \cdot \text{K}$, twice as large volume is required:

$$V = 1\,000\,000 / (0,6 \cdot 40) = 41,666 \text{ m}^3 \text{ of granite.}$$

In all heat storage there also has to be a compensation made for heat losses that occur during the storage period. Table 14 shows that the heat conductivity varies considerably between different stone species. There is also large variation within each stone specimen, depending on it's mineral composition. For pure quartz the heat conductivity is as high as 7,8 W/m, K and therefore the heat conductivity of a stone specimen will often be dependent on its percentage of quartz. Structures in the rock will also affect the heat conductivity, which therefore can be different in different directions. The volumetric heat capacity is relatively equal for most stone species.

Heat losses

For heat storage in ground there are three fundamental components which describe the temperature field. There is a stationary (time-independent) component and a periodic, transient component (year cycles) which depends on the storage's function. There is also another transient

component which can be significant during the first years of drift-cycles for the heat storage. During this time a heated volume is built up in the ground surrounding the heat storage.

When calculating heat losses the periodic component can be neglected, since its net-outflow is 0. The heat outflow is determined by the mean temperature during the storage cycle. The annual air temperature variation can in some cases affect the heat losses during the year, but it will not affect the annual heat loss. For this reason the annual mean air temperature is used as the ground surface temperature. The annual average temperature on the heat storage surface is also used but this can be difficult to estimate.

Stationary heat loss

SPHERICAL GROUND HEAT STORAGE

With this simple approach using the annual mean temperatures for air, T_0 , and storage surface, T_m , the stationary heat loss can be calculated. It should be observed that no respect has to be taken to the storage medium (water, rock etc.) since we know the mean temperature on the storage surface.

$$Q_m = h \cdot I \cdot (T_m - T_0) \cdot L_s$$

Equation (60)

Q_m (W) indicates the heat flow from the entire storage, where h is a dimensionless heat loss factor and L_s characterizes the length scale of the storage. Equation (60) is general in a sense that the storage area, position and geometry are included in L_s and h . The general character of equation (60) is demonstrated in the following example.

EXAMPLE 9

A house has a heat consumption of 30 000 kWh/year. The average temperature in the house is +20 °C and the annual mean temperature in air is +5 °C.

How large would the heat loss become if the indoor temperature was lowered to +19 °C?

Solution:

We use the fact that $E_m = Q_m \cdot t = h \cdot I \cdot L_s \cdot t \cdot (T_m - T_0)$

Since the geometry of the house, insulation, size and the time we are calculating for, are the same in both the known and the unknown case, we get;

$$h \cdot I \cdot L_s \cdot t = \text{constant}$$

From the known case we determine the constant

$$E_m = 30\,000 = \text{constant} \cdot (20 - 5) \Rightarrow 2\,000$$

This is now used in the unknown case, where the heat consumption is indicated x

$$x = 2\,000 \cdot (19 - 5) = 28\,000 \text{ kWh.}$$

The heat demand is therefore decreased from 30 000 kWh to 28 000 kWh (6,7 %) if the indoor temperature is lowered with 1 °C.

For a spherical storage in an infinite environment the heat loss becomes

$$Q_m = 4p \cdot I \cdot (T_m - T_0) \cdot R$$

Equation (61)

Equation (61) is obtained by setting $h = 4\pi$ and $L_s = R$ in equation (60). By calculating the heat loss per surface unit for the spherical storage, we get

$$\frac{Q_m}{4p \cdot R^2} = \frac{I(T_m - T_0)}{R}$$

Equation (62)

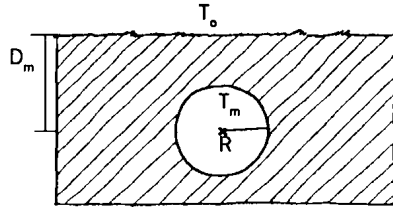
From equation (62) we understand that the spherical environment corresponds to a flat plate with the thickness R , in a 1-dimensional case. Over this plate, with the thickness R , the temperature drops linearly from T_m to T_0 .

Most heat storages are situated on more moderate depths why a correction for the heat loss, with respect to the distance to the ground surface and its temperature, has to be made. Figure 82 shows a spherical layer whose center is at the distance D_m below the ground surface. For this storage the heat loss is

$$Q_m = I \cdot (T_m - T_0) \cdot R \cdot h(D_m / R)$$

Equation (63)

Figure 82. Spherical heat storage with the radius R , at the depth D_m from ground surface to center.



The heat loss factor can be calculated with good approximation according to equation (64), if $D_m/R > 1,5$. When D_m approaches large values, $h(D_m/R)$ approaches the value 4π , i.e. the case where the storage is located in an infinite environment.

$$h(D_m / R) \approx \frac{4p}{1 - R / 2D_m}$$

Equation (64)

EXAMPLE 10

We assume that we have a spherical storage at a very large depth with $R = 29$ m, $T_m = 40$ °C, $T_0 = 3$ °C and $\lambda = 3,5$ W/m, K. Calculate the stationary heat loss.

Solution:

By using equation (63) we get

$$h = 4\pi$$

$$Q_m = 3,5 \cdot (40-3) \cdot 29 \cdot 4\pi = 47 \text{ kW}$$

CYLINDRICAL HEAT STORAGE

This introductory description of heat loss calculations from a spherical storage can be used for approximate calculations, this is for situations where a n estimate of necessary storage size and losses is wanted. There are however no spherical storages In reality a storage will have a more practical geometry. The most common form of ground heat storage is a more or less compact cylinder. The cylinder shaped storage has the radius R and the height H . The storage's upper surface is the distance D below the ground surface, see Figure 83.

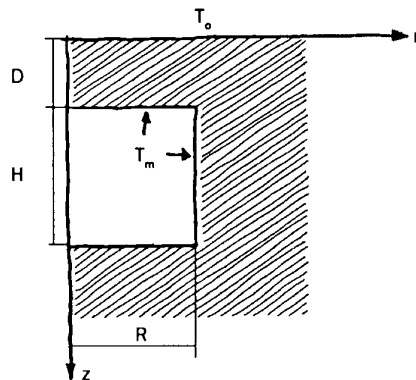


Figure 83. Cylindrical heat storage with stationary temperature conditions.

The general solution, equation (60), is used but the heat loss factor, h , in this case is a function of both R/D and H/D . The stationary heat loss is therefore written

$$Q_m = h(R/D, H/D) \cdot I \cdot (T_m - T_0) \cdot D$$

Equation (65)

In table 15 is the heat loss factor, h , presented for some different cases of cylindrical heat storage.

Table 15. Heat loss factor, h , for a cylinder-formed storage below ground, according to equation (60) and figure 74.

| H/D | | | | | | | |
|--------------|----------|----------|----------|----------|-----------|-----------|-----------|
| 20 | 63 | | | 230 | 569 | 1064 | 1716 |
| 15 | 54 | | | 213 | 547 | 1038 | 1686 |
| 10 | 43 | | | 194 | 521 | 1006 | 1649 |
| 5 | 31 | 57 | 89 | 171 | | | |
| 4 | 28 | 53 | 84 | 165 | | | |
| 2 | 21 | 44 | 73 | 150 | 456 | 923 | 1549 |
| 1 | 17 | 32 | 66 | 140 | | | |
| R/D = | 1 | 2 | 3 | 5 | 10 | 15 | 20 |

EXAMPLE 11

A cylindrical heat storage, see figure 83, has the upper surface 5 m below the ground surface. The storage diameter is 50 m and its height is 50 m. The ground heat conductivity is 3,5 W/m, K. The annual mean temperature above the storage surface is 40 °C while the annual mean temperature at the ground surface is 3 °C. We then have;

$D = 5$ m, $H = 50$ m, $R = 25$ m, and $T_m = 40$ °C, $T_0 = 3$ °C.

The stationary heat loss can be calculated from equation (60) and table 15.

$$H(25/5, 50/5) = 194$$

$$Q_m = 3,5 \cdot (40-3) \cdot 5 \cdot 194 = 125\,615 \text{ W} = 126 \text{ kW}.$$

Counted on the whole year the energy loss is 126 kW multiplied with the number of hours per year (8 760 h) = 1104 MWh. The data supplied in this example are close to the actual data for the borehole heat storage in Luleå, Northern Sweden.

Transient heat losses

The transient heat loss, the heat flow as a function of time, is directly proportional to the grounds heat conductivity and to the temperature difference $T_m - T_0$. In the same way as for the stationary case the heat loss is proportional to the scale-length L_s .

SPHERICAL STORAGE

The transient heat loss for a sphere, on infinite depth, is given by equation (66). The formula is instructive. It tells us that at infinite time, t , the stationary heat flow is obtained as has been earlier expressed by equation (65). The other term in the parenthesis expresses the transient part of the heat loss.

$$Q_{tr} = I \cdot (T_m - T_0) \cdot (4\mathbf{p} \cdot R + 4\mathbf{p} \cdot R^2 / \sqrt{\mathbf{p} \cdot a \cdot t})$$

Equation (66)

The accumulated heat flow, E_{tr} , in equation (67) is obtained after integration of equation (66).

$$E_{tr} = I \cdot (T_m - T_0) \cdot t \cdot 4\mathbf{p} \cdot (R + 2R^2 / \sqrt{\mathbf{p} \cdot a \cdot t})$$

Equation (67)

In the equation above, a ($= \lambda/C$) indicates the thermal diffusivity while t indicates the time that has passed (up to the point where the heat loss is calculated). C , which indicates the volumetric heat capacity, is only of interest for the transient part and is of no importance during stationary conditions. From equation (66) we can also understand that the transient part is equally large as the stationary when the first and second term in the second parenthesis are equal. This occurs at a time t according to equation (67).

$$t = \frac{R^2}{ap}$$

Equation (68)

For a spherical heat storage on a certain depth, D_m , below the ground surface, equation (71) applies during the first time, when the storage temperature is unaffected by the ground surface temperature. After a breaking time, t_b , equation (70) gives an approximate solution, for stationary heat loss. It is evident that t_b must contain the heat storage depth below the ground surface, D_m . See equation (68).

$$t_b = \frac{(2D_m - R)^2}{ap}$$

Equation (69)

$$Q_{ir} \cong \frac{I \cdot (T_m - T_0) \cdot 4p \cdot R}{1 - R / 2D_m} \quad \text{for } t > t_b$$

Equation (70)

$$Q_{ir} \cong I \cdot (T_m - T_0) \cdot 4p \cdot R \cdot (1 + R_v / \sqrt{a \cdot p \cdot t})$$

for $0 < t < t_b$

Equation (71)

EXAMPLE 12

Determine the transient heat loss for a spherical storage with the radius 10 m. The center of the storage is 20 m below the ground surface. $T_m - T_0 = 25\text{ }^{\circ}\text{C}$.

In case a) the storage is in granite ($a = 1,6 \cdot 10^{-6}\text{ m}^2/\text{s}$) with $\lambda = 3,5\text{ W/m, K}$. In case b) the storage is located in a very rich iron ore ($a = 21 \cdot 10^{-6}\text{ m}^2/\text{s}$) with $\lambda = 75\text{ W/m, K}$.

a) storage in granite

b) storage in iron ore

Equation (69)

gives t_b for a) and b)

$$t_b = (2 \cdot 20 - 10)^2 / \pi \cdot 1,6 \cdot 10^{-6} = t_b = 13,6 \cdot 10^6\text{ s} = 0,4\text{ years} \\ = 179 \cdot 10^6\text{ s} = 5,7\text{ years}$$

$t < 5,7\text{ years}$ gives (71) $t < 0,4\text{ years}$ gives (71)

$$Q_{tr} = 236 \cdot \left(1 - \frac{0,22}{\sqrt{t_1}} \right)$$

$$Q_{tr} = 10,99 \cdot \left(1 - \frac{0,8}{\sqrt{t_1}} \right)$$

where t_1 is the time in years

where t_1 is the time in years

| | | | |
|--------------------------|--------------------|----------------------------|---------------------|
| $Q_{tr} (1\text{ year})$ | $= 19,8\text{ kW}$ | $Q_{tr} (0,1\text{ year})$ | $= 400,2\text{ kW}$ |
| $Q_{tr} (2\text{ y})$ | $= 17,2\text{ kW}$ | $Q_{tr} (0,2\text{ y})$ | $= 352,1\text{ kW}$ |
| $Q_{tr} (3\text{ y})$ | $= 16,1\text{ kW}$ | $Q_{tr} (0,3\text{ y})$ | $= 330,8\text{ kW}$ |
| $Q_{tr} (5,7\text{ y})$ | $= 15,1\text{ kW}$ | $Q_{tr} (0,4\text{ y})$ | $= 381,1\text{ kW}$ |

$t > 5,7\text{ years}$ eq. (70) $t > 0,4\text{ years}$ eq. (70)

$$Q_{tr} = 14,7 \text{ kW (stationary)} \quad Q_{tr} = 314 \text{ kW (stationary)}$$

NON-SPHERICAL STORAGES

A spherical storage at a large depth is optimal from a heat loss view. The spherical form however is impractical from a construction view and we will therefore present a general form for calculating heat losses from storages with other geometry's.

Let us therefore consider the heat loss from a surface which has an edge line. The surface is given by two flat parts that connect at a right angle. The total area of the surfaces is A and the edge lines total length is L_e , The transient heat loss during a first time will be:

$$Q_{tr} = I \cdot (T_m - T_0) \cdot (A / \sqrt{p \cdot a \cdot t} + 0,6L_e)$$

Equation (72)

Equation (72) is valid during the first time when

$$\frac{\sqrt{a \cdot t}}{L_1} < 1$$

Equation (73)

where $L_1 = 0,5 \cdot \min(L, B \text{ or } H)$, i.e. that of L , B or H which has the least length. For A and L_e applies for a parallelepiped-shaped storage that

$$A = 2(LB + LH + BH)$$

$$L_e = 4(L + B + H)$$

Equation (74)

For longer times it is reasonable to approximate the heat storage with a sphere. The radius R_v for a sphere with the same volume, V , becomes:

$$R_v = \sqrt[3]{3V / 4p}$$

Equation (75)

The transient heat loss from the heat storage can then, for longer times, be approximated with equation (76)

$$Q_{tr} \cong I \cdot (T_m - T_0) \cdot 4\mathbf{p} \cdot R \cdot (1 + R_v / \sqrt{a \cdot \mathbf{p} \cdot t})$$

Equation (76)

For a parallelepiped-shaped storage with the length L, the height H and the width B, we have

$$R_v = \sqrt[3]{3 \cdot L \cdot B \cdot H / 4\mathbf{p}}$$

Equation (77)

and for the cylinder-shaped storage with the radius R and the height H, applies

$$R_v = \sqrt[3]{3 \cdot H \cdot R^2 / 4}$$

Equation (78)

For this approximation to be valid, 2R/H, B/L and H/L must all be between 1/5 and 5.

Our next case concerns a cylinder-shaped heat storage with the height H and the radius R. We have two flat, circular surfaces, with an area A. Between these and the envelope surface we have two circular edge lines with the length L_e :

$$A = 2 \cdot \mathbf{p} \cdot R^2 \quad L_e = 2 \cdot 2 \cdot \mathbf{p} \cdot R$$

Equation (79)

The transient heat loss from the cylindrical storage is given by equation (80)

$$Q_{tr}^{\perp} = 50 \cdot (6 \cdot 20^2 / \sqrt{\mathbf{p} \cdot t \cdot 10^{-6}} + 0,6 \cdot 240$$

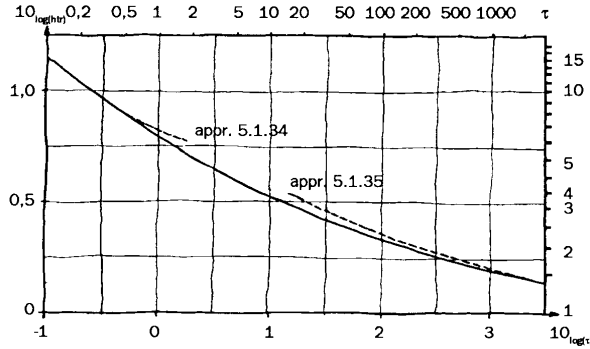
Equation (80)

where $h_{tr}^{cyl}(\tau)$ is obtained from Figure 84, after calculating τ with equation (81)

$$t = \frac{a \cdot t}{R^2}$$

Equation (81)

Figure 84. Transient heat loss factor for a cylinder in an infinite environment. For equation 80.



EXAMPLE 13

Let us calculate the transient heat loss from a cubic-shaped heat storage with the edge length 20 m. The storage is located at a large depth below the ground surface.

$$\lambda \cdot (T_m - T_0) = 50 \text{ W/m. } A = 1,0 \cdot 10^{-6} \text{ m}^2/\text{s.}$$

Equations (72)-(74) gives for short times:

$$Q \frac{1}{tr} = 50 \cdot (6 \cdot 20^2 / \sqrt{p \cdot t \cdot 10^{-6}} + 0,6 \cdot 240)$$

for $t < 100 \cdot 10^6 \text{ s} = 3,2 \text{ years.}$

$$Q \frac{1}{tr} (3,2 \text{ years}) = 14,0 \text{ kW}$$

Equations (76)-(77) gives for times $> 3,2 \text{ years:}$

$$Q \frac{2}{tr} = 50 \cdot \left[4p \cdot 12,4 \cdot (1 + 12,4 / \sqrt{p \cdot t \cdot 10^{-6}}) \right]$$

$$Q \frac{2}{tr} (3,2 \text{ years}) = 13,2 \text{ kW}$$

From the example one can observe that the results are not identical at the breaking time but differ from each other.

The transient heat loss for a storage at moderate depth can be estimated in the following way. The storage center is located at the depth D_m and we assume that the storage's uppermost point is not too close to the ground surface. For short times the ground surface will not affect the transient heat flow. Within this time we can approximately say that $Q_{tr} = Q_{tr}(D_m = \infty)$. For large values of t , $Q_{tr} = Q_m$, i.e. the transient heat loss is equal to the stationary heat loss after a long time. At a certain breaking time, t_b , according to equation (69), these expressions are the same. The transient heat loss is thereby determined by choosing the expression which gives the largest transient heat loss according to equation (82).

$$Q_{tr} \cong \max[Q_{tr}(D_m = \infty), Q_m]$$

Equation (82)

EXAMPLE 14

How large is the heat loss from a spherical heat storage with the radius 10 m? The center is on 20 m depth and we have the following data:

$$R = 10 \text{ m} \quad D = 20 \text{ m} \quad T_m - T_0 = 25 \text{ }^\circ\text{C}$$

$$a = 10^{-6} \text{ m}^2/\text{s} \quad \lambda = 2,0 \text{ W/m, K}$$

and breaking time $t_b = 9,1$ years (according to equation (69))

Solution:

For $0 < 9,1$ years equation (71) applies, which gives

$$Q_{tr} \cong 6,3 \cdot (1 + \sqrt{t_1 / t}) \text{ kW}$$

where $t_1 = 1$ year

For $t > 9,1$ years equation (70) applies (stationary heat loss) which gives:

$$Q_{tr} \cong 8,4 \text{ kW}$$

EXAMPLE 15

We have a cylindrical heat storage with 10 m radius and 20 m height. It's upper surface is 10 m below the ground surface. We choose the following data:

$$R = 10 \text{ m} \qquad H = 20 \text{ m} \qquad D = 10 \text{ m}$$

$$\lambda \cdot (T_m - T_0) = 50 \text{ W/m} \qquad a = 10^{-6} \text{ m}^2/\text{s}$$

The breaking time is calculated with equation (69)

$$t = 31\,847,134 \text{ s} = 1,0 \text{ years}$$

After the breaking time, t , we have the stationary heat loss according to equation (65) and table 15.

$$H(10/10, 20/10) = 21$$

$$Q_{tr}(t > t_b) \cong Q_m = 50 \cdot 10 \cdot 21 = 10,5 \text{ kW}$$

Equations (69)-(71) together with figure 84 gives the transient heat loss solution for short times. At the time 0,5 years, the transient loss becomes:

$$t(\text{equation 81}) = 0,15768 \text{ gives } \log(t) = -0,8$$

$$\log(h_{tr}^{cyl}) = 1,2 \text{ according to Figure 84 this gives } h_{tr}^{cyl} = 15,8$$

$$Q_{tr}(0,5 \text{ years}) = 24 \text{ kW}$$

Temperature fields

It can be of interest to know the temperature in the ground surrounding a ground heat storage. It may consider temperature effects on close buildings, groundwater resources etc. This is here described just briefly but will give

us the capability of judging the temperature effects a storage has on its surroundings.

Let us first consider a spherical storage which is situated so deep below the ground surface that it (the ground surface) cannot affect the stationary temperature conditions outside the storage. The stationary temperature at the distance

$$r = \sqrt{x^2 + y^2 + z^2}$$

outside the spherical storage will then become according to equation (83):

$$T(x, y, z) = T_0 + (T_m - T_0) \cdot R / r$$

Equation (83)

It is more complicated to calculate the 3-dimensional transient temperature field. We will therefore only consider the transient temperatures outside the heat storage for the 1-dimensional, semi-infinite case, see Figure 85.

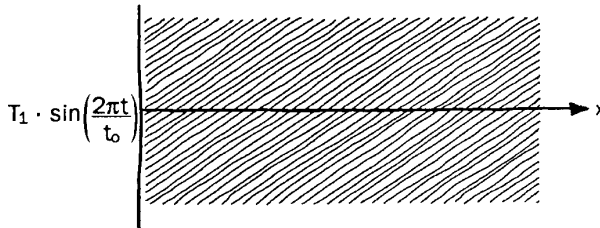


Figure 85. Periodical temperature development. 1-dimensional, semi-infinite case.

At the edge $x = 0$ the temperature varies sinusoidally with the amplitude T_1 according to figure 85. This case is valid in condition that the ground is reaching long enough in the normal direction without any disturbances. The temperature T on the distance x from the temperature source after the time t is given by equation (84). t_0 is the temperature variation period. d_0 will from now on be referred to as the penetration depth, see below.

$$T(x, t) = T_1 \cdot e^{-x/d_0} \cdot \sin(2\mathbf{p} \cdot t / t_0 - x / d_0)$$

Equation (84)

$$\text{where the length } d_0 = \sqrt{a \cdot t_0 / \mathbf{p}}$$

Equation (85)

The edge temperature, presented in complex form is

$$T(0, t) = T_1 \cdot e^{2\mathbf{p} \cdot i t / t_0}$$

Equation (86)

The temperature variation on the surface in figure 85 is rewritten in complex form because it simplifies further calculations. After more mathematical transformation equation (87) is obtained, which describes how the transient temperature is damped with a factor e^{-x/d_0} . The equation therefore describes how the temperature amplitude decreases with increased periodical time (related to d_0 in equation (85)) and depth.

$$|T| = |\hat{T}_1| \cdot e^{-x/d_0}$$

Equation (87)

Depth of penetration

At the edge $x = 0$ the amplitude is $|\hat{T}_1|$. The damping has the length-scale d_0 . This length is a measure of the inwards damping of the transient temperature variation at the surface. We call d_0 the penetration depth for the periodical development.

The damping factor for different depths, x , becomes:

| | |
|----------------------------------|--------------------|
| $x = 0$ gives the damping factor | $e^{-0} = 1$ |
| $x = 0,25 d_0$ | $e^{-0,25} = 0,78$ |
| $x = 0,5 d_0$ | $e^{-0,5} = 0,61$ |
| $x = d_0$ | $e^{-1} = 0,37$ |
| $x = 2d_0$ | $e^{-2} = 0,14$ |
| $x = 3d_0$ | $e^{-3} = 0,05$ |
| $x = 4d_0$ | $e^{-4} = 0,02$ |
| $x = 5d_0$ | $e^{-5} = 0,007$ |

At the depth $x = d_0$ the temperature amplitude has decreased to $0,37 \cdot |T_1|$ and at the depth $x = 3d_0$ only 5% of the temperature disturbance remains.

In table 16 the calculated values of the penetration depth, d_0 , are presented, i.e. the depth at which 37% of the temperature disturbance remains

Table 16. Penetration depth for different periods and different temperature conductivities.

$$\text{Penetration depth } d_0 = \sqrt{a \cdot t_0 / p}$$

| a | | | | | | | | |
|---------------------|--------|-------|-------|-------|--------|---------|--------|-----|
| [m ² /s] | 1 sec | 1 min | 1 h | 1 day | 1 week | 1 month | 1 year | 5 y |
| $1,6 \cdot 10^{-6}$ | 0,0007 | 0,006 | 0,043 | 0,21 | 0,55 | 1,16 | 4,0 | 9,0 |
| $1,0 \cdot 10^{-6}$ | 0,0006 | 0,004 | 0,034 | 0,17 | 0,44 | 0,91 | 3,2 | 7,1 |
| $0,4 \cdot 10^{-6}$ | 0,0004 | 0,003 | 0,021 | 0,10 | 0,28 | 0,58 | 2,0 | 4,5 |

By calculating the penetration depth one can obtain an understanding and feeling for a long row of heat conductivity problems. How thick should a wall be in order

to let the high daytime temperatures heat the building during night, while at the same time the night-cold should penetrate adequately to make the building cooler during daytime?

EXAMPLE 16

At what depth in the ground can the seasonal cycles (winter/summer) not be felt, i.e. on what depth is the temperature disturbance less than $0,1\text{ }^{\circ}\text{C}$?

Solution.

We can assume that the temperature difference between winter and summer is $30\text{ }^{\circ}\text{C}$, i.e. the amplitude is $15\text{ }^{\circ}\text{C}$.

$$a = 1,0 \cdot 10^{-6}$$

$$\text{The damping factor} = 0,1/15 = 0,0067$$

$$e^{-x/d_0} = 0,0067 \text{ gives } -x/d_0 = -5,0106 \text{ and } x = 5,0106 \cdot d_0$$

$$d_0 = \sqrt{a \cdot t_0 / p} \text{ and the period 1 year} = 31\,536\,000 \text{ sec}$$

gives

$$x = 15,9 \text{ m}$$

EXAMPLE 17

How far outside a heat storage in granite is the heat disturbance $5\text{ }^{\circ}\text{C}$, if the storage's mean annual temperature variation is $30\text{ }^{\circ}\text{C}$? $a = 1,6 \cdot 10^{-6}$

Solution:

$$\text{The damping factor} = 5/15 = 0,33 \text{ gives } x = 1,4d_0$$

$$x = 1,4 \cdot \sqrt{1,6 \cdot 10^{-6} \cdot 31,5 \cdot 10^6 / p} = 5,6 \text{ m}$$

EXAMPLE 18

Which reach does 50%, 10% and 1% of the temperature disturbances have in a;

a) granite $\lambda = 3,5 \text{ W/m, K}$ $c = 2\,200\,000 \text{ J/m}^3, \text{ K}$

b) clay $\lambda = 1,0 \text{ W/m, K}$ $c = 3\,500\,000 \text{ J/m}^3, \text{ K}$

after 1 hour, 1 week and 1 year.

Solution:

The thermal diffusivity, $a = \lambda/c$

$$a^{\text{granite}} = 1,60 \cdot 10^{-6} \text{ m}^2/\text{s} \quad a^{\text{clay}} = 2,86 \cdot 10^{-7} \text{ m}^2/\text{s}$$

The reach in [m] for 50%, 10% and 1% of the temperature disturbance:

| Reach (m) | | | | |
|------------------|------------|--------------|---------------|---------------|
| | | 1 h | 1 week | 1 year |
| Granite | $x_{0,5}$ | <u>0,076</u> | 0,984 | 7,103 |
| | $x_{0,1}$ | 0,175 | 2,263 | 16,337 |
| | $x_{0,01}$ | 0,274 | 3,542 | 25,571 |
| Clay | $x_{0,5}$ | <u>0,032</u> | 0,416 | 3,003 |
| | $x_{0,1}$ | 0,074 | 0,957 | 6,907 |
| | $x_{0,01}$ | 0,115 | 1,497 | 10,811 |

From the table it can be seen that temperature disturbances propagates much faster in granite than in clay. If the total temperature disturbance is 10 °C, the reach for a 5 °C temperature disturbance (i.e. 50%) after 1 h is 7,6 cm (0,076 m) in granite and 3,2 cm in clay.

Contact temperature

A somewhat more curious problem, concerning heat transfer, is the contact temperature between two surfaces.

What first springs to mind is why it feels so much colder to walk barefoot on a stone floor than it does on a linoleum carpet, even if both floors have the same temperature. The instant edge temperature, T_f , between two medias with the temperatures T_1 and T_2 are written according to equation (88):

$$T_r = \frac{T_1 \cdot b_1 + T_2 \cdot b_2}{b_1 + b_2}$$

Equation (88)

where b_1 is a function of the material properties density, heat conductivity and heat capacity according to equation (89):

$$b_i = \sqrt{I_i \cdot r_i \cdot c_i} = \frac{I}{\sqrt{a_i}}$$

Equation (89)

Light materials have low values of λ and ρ while c can be relatively high. Put together this gives a low value of b , which explains why it is, for example, warmer to walk barefoot on mineral wool than on stone.

EXAMPLE 19

What is the contact temperature between a plate of mineral wool and a granite-plate? The granite temperature is 37 °C and the mineral wool temperature is 0 °C.

$$\lambda_\mu = 0,04 \text{ W/m, K} \quad a_m = 0,3 \cdot 10^{-6} \text{ m}^2/\text{s} \quad \text{gives} \\ b_m = 73$$

$$\lambda_\gamma = 3,5 \text{ W/m, K} \quad a_g = 1,6 \cdot 10^{-6} \text{ m}^2/\text{s} \quad \text{gives} \\ b_g = 2770$$

$$T_f = (0 \cdot 73 + 37 \cdot 2770) / 2843 = 36 \text{ °C}$$

What are the contact temperatures between a plate of iron and the same granite-plate?

$$\lambda_i = 84 \text{ W/m, K} \quad a_i = 23 \cdot 10^{-6} \text{ m}^2/\text{s} \text{ gives } b_i = 17500$$

$$T_r = (0 \cdot 17500 + 37 \cdot 2770) / 20270 = 5,1 \text{ } ^\circ\text{C}$$

EXERCISES

EXCERCISE 1

Calculate the temperature on a sunlit tin-roof if the air-temperature is $20\text{ }^{\circ}\text{C}$, the solar radiation is 600 W/m^2 and the absorptivity is 0,90. The total heat transfer number is 15 W/m^2 , K. The roof is insulated with 0,05 m cork, whose heat conductivity is 0,04 W/m, K. The temperature on the inside of the roof is $20\text{ }^{\circ}\text{C}$.

EXCERCISE 2

Calculate the heat transfer number caused by radiation from the surface of skin. The skin temperature is approximately $33\text{ }^{\circ}\text{C}$ and the emissivity is 0,92. $T_{\infty} = 20\text{ }^{\circ}\text{C}$.

- a) Temperature of surrounding surfaces is $15\text{ }^{\circ}\text{C}$.
- b) Temperature of surrounding surfaces is $20\text{ }^{\circ}\text{C}$.
- c) Temperature of surrounding surfaces is $25\text{ }^{\circ}\text{C}$.

EXCERCISE 3

Calculate the absorber temperature for a solar collector whose glazing has a transmissivity of 0,90 and the absorptivity 0,95. The heat loss coefficient, U_L -factor, is 8 W/m^2 , K.

The solar radiation is 600 W/m^2 and the air temperature is $20\text{ }^{\circ}\text{C}$.

- a) Determine the absorber temperature when the collector is not in use.
- b) Determine the absorber temperature when the collector has a brine-flow of $0,01 \cdot 10^{-3}\text{ m}^3/\text{s}$, m^2 . The temperature increase is $8\text{ }^{\circ}\text{C}$.

EXCERCISE 4

Calculate the energy-output, the instantaneous efficiency and the total efficiency for a glazed solar collector during 2 days (one in April and one in July) in Stockholm.

The collector has a slope of 45° towards the south and is subjected to solar radiation as shown in the table below. The air temperature is also presented in the table.

The collectors transmission-absorption-coefficient is 0,85. The heat loss coefficient, U_L , is $5 \text{ W/m}^2, \text{ K}$. The F-value is 0,90. The mean temperature of the heat carrier is 40°C and the collector surface area is 1 m^2 .

Climate data for Stockholm in April and July. Radiation towards a southward surface with an angle of 45° .

| Time | APRIL | | JULY | |
|-------|-----------------------------------|-------------------|-----------------------------------|-------------------|
| | Radiation (Wh/m ²) | Air temp. (°C) | Radiation (Wh/m ²) | Air temp. (°C) |
| 06-07 | 85 | 2,9 | 147 | 16,4 |
| 07-08 | 199 | 3,7 | 273 | 17,0 |
| 08-09 | 315 | 4,5 | 393 | 17,8 |
| 09-10 | 422 | 5,1 | 484 | 18,7 |
| 10-11 | 494 | 5,8 | 562 | 19,3 |
| 11-12 | 528 | 6,5 | 592 | 20,0 |
| 12-13 | 520 | 7,1 | 603 | 20,5 |
| 13-14 | 469 | 6,6 | 539 | 20,3 |
| 14-15 | 393 | 6,1 | 462 | 20,0 |
| 15-16 | 306 | 5,7 | 366 | 19,8 |
| 16-17 | 194 | 5,5 | 256 | 19,2 |
| 17-18 | 86 | 5,3 | 141 | 19,0 |

EXERCISE 5

Determine the characteristic solar panel parameters, the U_L -coefficient and the F-value, for a glazed collector whose performance has been measured according to the table below. The transmission-absorption number for the collector is 0,90.

Measured performance for a glazed solar collector

| DATE | TIME | T_{in} (°C) | T_{out} (°C) | T_u (°C) | H (W/m ²) | FLOW (l/s, m ²)* |
|--------|-------|------------------|-------------------|---------------|--------------------------|---------------------------------|
| June 1 | 12.00 | 60,0 | 63,2 | 22,0 | 623 | 0,015 |
| June 2 | 12.00 | 56,3 | 57,5 | 17,0 | 416 | 0,010 |
| June 3 | 12.00 | 42,2 | 47,0 | 20,0 | 617 | 0,015 |

* 1 liter = 0,001 m³

Guide: Determine $(T_v - T_u)/H$ and Q_u/H and draw an efficiency curve.

EXERCISE 6

A solar collector is used to heat tap water according to the figure below. Calculate the temperature increase over the collector if the flow is 1,0 l/min.

$T_v = 40$ °C; $\tau\alpha = 0,90$; F =
 $0,95$; $U_L = 5$ W/m²,
 K ; $T_u = 20$ °C; H = 400 W/m²; C =
4000 J/kg, K

EXERCISE 7

Dimension and calculate the cost for the solar heat system below.

| | |
|---|------------------------------|
| Annual heat demand | 2 300 MWh |
| Coverage of solar heat | 75 % |
| Lowest storage temperature | 40 °C |
| Highest storage temperature | 95 °C |
| Heat exchange, solar panel | 400 kWh/m ² |
| Heat losses in ground pit | 15 % |
| Required storage capacity | 60 % |
| Costs: | |
| Pit storage | 200 - 400 SEK/m ³ |
| Collector | 900 - 1 300 |
| SEK/m ² | |
| 6 % interest and 25 years depreciation time | |

EXERCISE 8

A surface soil heat pump works with the heat factor 2,5.
The added electrical work to the compressor is 3,0 kW.

- Determine the delivered heat effect to the heat system
- Determine the extracted effect in the heat pumps evaporation chamber

EXERCISE 9

A rock heat pump works with the condensation temperature 55 °C and the evaporation temperature -10 °C. Determine the ideal heat factor for the Carnot-process and the "real" heat factor for the heat pump.

EXERCISE 10

One wishes to use the outgoing air from an apartment as a heat source to a heat pump. Calculate the supplied energy

amount to the heat pumps evaporation chamber (annually) if the apartment is 75 m^2 , the height to the ceiling is $2,5 \text{ m}$. The air gets renewed $0,5 \text{ times/h}$. The temperature of the outgoing air is lowered with $15 \text{ }^\circ\text{C}$. The heat capacity for air is $0,00035 \text{ kWh/m}^3, \text{ K}$. The heat factor for the heat pump is $2,5$. How much heat is supplied to the partment every year? Determine supplied heat to the apartment (annually) from the heat pump system!

EXCERCISE 11

For a mineral wool-wall with the thickness $0,15 \text{ m}$, a step-change of the surface temperature occurs with $10 \text{ }^\circ\text{C}$. The thermal diffusivity, a , for mineral wool is $0,3 \cdot 10^{-6} \text{ m}^2/\text{s}$.

Determine the reach for 1% , 10% and 50% of the temperature disturbance after 1 h . Draw a temperature profile for the wall.

EXCERCISE 12

Determine the effect output from a rock heat pump whose stationary borehole temperature is $-4 \text{ }^\circ\text{C}$. The active borehole depth is 120 m , the undisturbed surrounding temperature is $6 \text{ }^\circ\text{C}$. The rock is granite with the thermal conductivity $3,5 \text{ W/m, K}$. The borehole radius is $0,055 \text{ m}$.

EXCERCISE 13

A constant, mean effect output of 15 W/m is taken from a rock heat pump. During 3 cold weeks year 1 and year 10, the effect output is increased to 40 W/m , 30 W/m and 25 W/m according to the figure below. Determine the borehole wall-temperature for those weeks.

H = 120 m
3,5 W/m, K

$a = 1,6 \cdot 10^{-6} \text{ m}^2/\text{s}$

$\lambda =$

$T_{\text{om}} = 6 \text{ }^\circ\text{C}$
week = 604 800 sec

$R_0 = 0,05 \text{ m}$

1

EXERCISE 14

Determine the rock wall temperature year 5 and 25, for two different rock heat systems according to below:

a) 6 boreholes in a circle, angled holes with a slope of 20°

b) 6 boreholes in a rectangle, B = 15 m

H = 150 m
30,4 W/m

$R_0 = 0,055 \text{ m}$

q =

$T_{\text{om}} = 6 \text{ }^\circ\text{C}$
 $2,16 \text{ MJ/m}^3$, K

$\lambda = 3,5 \text{ W/m, K}$

c =

EXERCISE 15

During 3 weeks the 5th and 25th year, the rock heat system in the previous example is subjected to an over-layered load flux, which for the first week is 25 W/m, for the second week 15 W/m and for the third week 10 W/m. Determine the temperature at the borehole wall for respective systems, for these three weeks (compare to Exercise 13).

EXERCISE 16

Determine the borehole walls minimum and maximum temperatures for year 1, 5 and 15, for a rock heat system consisting of 9 wells in a square, B = 10 m. The system is subjected to a load as shown in the figure below.

R = 0,055 m
3,5 W/m, K

$T_{\text{om}} = 6 \text{ }^\circ\text{C}$

$\lambda =$

c = $2,2 \text{ MJ/m}^3$, K

H = 100 m

EXERCISE 17

Calculate the stationary heat loss as a function of the volume for a spherical heat water storage in granite rock. The storage is situated on a large depth. $\lambda = 3,5 \text{ W/m, K}$. The water heat capacity is $1,2 \text{ kWh/m}^3$. The storage mean temperature at the edges is $40 \text{ }^\circ\text{C}$. The storage average temperature varies between $20 \text{ }^\circ\text{C}$ and $60 \text{ }^\circ\text{C}$ during a year-cycle. The undisturbed ground-temperature is $4 \text{ }^\circ\text{C}$.

- a) Absolute heat loss as a function of the storage volume (m^3)
- b) Relative heat loss as a function of stored energy (kWh)
- c) How would the calculated heat losses be affected if the storage medium was rock with a heat capacity of $0,6 \text{ kWh/m}^3$, K?

EXERCISE 18

Calculate the stationary heat loss as a function of depth below ground surface for a spherical heat water storage in granite rock. $\lambda = 3,5 \text{ W/m, K}$. The water heat capacity is $1,2 \text{ kWh/m}^3$. The storage volume is $400\,000 \text{ m}^3$. The storage mean temperature at the edges is $40 \text{ }^\circ\text{C}$. The storage average temperature varies between $20 \text{ }^\circ\text{C}$ and $60 \text{ }^\circ\text{C}$ during a year-cycle. The undisturbed ground-temperature is $4 \text{ }^\circ\text{C}$.

- a) Absolute heat loss as a function of depth.
- b) Relative heat loss as a function of depth.
- c) How would the calculated heat losses be affected if the storage medium was rock with a heat capacity of $0,6 \text{ kWh/m}^3$, K?

EXERCISE 19

Solve Exercise 1 for a cylindrical storage with the diameter equal to the height.

EXERCISE 20

Solve Exercise 2 for a cylindrical storage with the diameter equal to the height.

EXERCISE 21

Draw a plot for the stationary temperature field outside a spherical heat storage.

$$T(x, y, z) = T_o + (T_m - T_o) \cdot R / r$$

$$r = \sqrt{x^2 + y^2 + z^2}$$

Discuss the look of the plot. Try to understand why the curve looks this way. Compare to the 1-dimensional case.