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Basic aspects of the seasonal storage of solar heat in the ground

Citation for published version (APA):
Fischer, H. R., van Koppen, C. W. J., & Puts, J. J. (1979). Basic aspects of the seasonal storage of solar heat in the ground. In *Sun 2 : proceedings of the International Solar Energy Society silver jubilee congress, held in Atlanta, Georgia, May 28 - June 1, 1979 : proceedings / Ed. K.W. Boeer, B.H. Glenn* (AS/ESIS : annual meeting; Vol. 1979). Pergamon.

Document status and date:
Published: 01/01/1979

Document Version:
Publisher's PDF, also known as Version of Record (includes final page, issue and volume numbers)

Please check the document version of this publication:

• A submitted manuscript is the version of the article upon submission and before peer-review. There can be important differences between the submitted version and the official published version of record. People interested in the research are advised to contact the author for the final version of the publication, or visit the DOI to the publisher's website.
• The final author version and the galley proof are versions of the publication after peer review.
• The final published version features the final layout of the paper including the volume, issue and page numbers.

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BASIC ASPECTS OF THE SEASONAL STORAGE OF SOLAR HEAT IN THE GROUND.

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INTRODUCTION

It is generally felt that the seasonal storage of solar heat is one of the most important, if not the central problem in the development of solar heating and cooling, in particular for regions with a cloudy climate situated at high latitudes. However, up to now only a very limited number of lay-outs for seasonal storage systems have been put forward that hold a promise of economic viability. One of these lay-outs is depicted in fig. 1. The essential characteristics

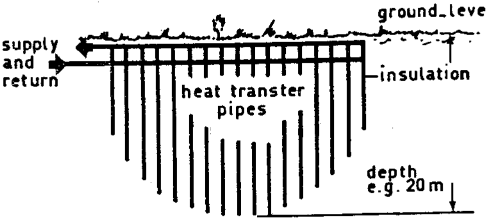

Fig. 1. General lay-out of a seasonal storage in the ground.

of the system are that soft saturated soil is used as the storage medium and that the storage is loaded and unloaded by means of a large number of more or less vertical heat transfer pipes. A thick insulation sheet on top of the storage (not drawn) prevents the heat from leaking away to the atmosphere. The rather good insulating properties of the soil are utilized to reduce heat leakage to the surrounding soil.
In |1| it is shown a.o. that a definite minimum capacity of the storage is required to obtain a sufficiently high storing efficiency; for average thermal properties of the soil the lowest permissible capacity was established at 500.000 kWh. This result was derived by applying the similarity rules of heat conduction obtained in a design study for an all-solar Town Office (Lelystad, the Netherlands). The main data for the seasonal storage concerned were:

- Capacity (net): 683.000 kWh.
- Radius of(hemisperical) outer boundary: 27m
- Radius of (hemispherical) core: 19 m.
- Thickness of insulating "shell" of soil: 27-19=8 m.
- Temperature (average) at outer boundary: 20°C.
- Temperature in core: 70°C at the beginning and 20°C at the end of heating season.
- Heat resistance of insulating sheet: 0.1 m^2 °C W^{-1}.
- Efficiency: 75%
- Investment: Dfl 683,000.-, i.e. Dfl 1.-/kWh cap.
- Thermal properties of saturated soil: conductivity 1.5 W m^{-1} $°C^{-1}$, specific heat 1200 J/kg; density 1600 kg m^{-3}.
- Number of heat transfer pipes: 1350.
- Total length of heat transfer pipes: 24000 m.

The basic questions arising during the design and optimization of a seasonal solar heating installation are related to the volume and the upper and lower temperature limits of the storage, the heat gain of the collectors as related to climate and average collector temperature, the efficiency of the storage and the total heat demand. The selection of the temperature limits is of particular importance because the volume, the heat losses and the efficiency of the storage, the heat gain and the required area of the collectors and consequently the investments are directly dependent on these limits (at a given heat demand). Therefore the temperature limits have been chosen as the primary variables in an engineering model for the first order techno-economic optimization of seasonal solar heating installations, developed at the Eindhoven University of Technology. In the model ample use is made of the similarity rules of heat conduction and of the scaling laws for investments in order to determine the dimensions, thermal characteristics, and costs of the storage. The data mentioned above serve as a fixed point in this connection. The costs of the collectors can be set at any desired value in the model; in our calculations a low price of Dfl 200.-/m^2, including mounting, was chosen, with a view on the large areas involved.
In the first order optimization model all unsteady heat transfer phenomena are reduced

to quasi-stationary problems. A second order optimization requires a more detailed knowledge of the non-steady heat transfer in the storage however. In particular concerning the heat transfer around the pipes, as this determines the maximum allowable mutual distance between the pipes, which is a major cost factor. In the final part of this paper some preliminary results of our research in this field will be presented.

THE FIRST ORDER ENGINEERING MODEL

Defining the capacity (C) of the storage as the amount of heat that can yearly be extracted from it and the efficiency of the storage (η) as the ratio between the amounts of heat extracted and delivered leads to the equation:

$$\eta = C/(C + L) = 1/(1 + L/C) \qquad (1)$$

where L = heat losses per year.

Assuming all linear dimensions of a storage (except the mutual distance of the pipes) to vary proportionally, the capacity of the storage can be written as:

$$C = \gamma \rho c D^3 \, (T_{max} - T_{min}) \qquad (2)$$

where T_{max} = upper limit, and T_{min} = lower limit of the temperature in the core, D = characteristic dimension of the storage, ρ = density, and c = specific heat of the soil, and γ = the proportionality factor.

The heat losses (per year) are proportional to the thermal conductivity of the soil (λ), the outside area of the storage (proportional to D^2) and the (yearly) mean temperature gradients which are proportional to the difference between the annual mean temperature in the core (T_m) and the temperature of the surrounding soil (T_o), and inversely proportional to D. Therefore:

$$L = \varphi \, \lambda D^2 \{(T_m - T_o)/D\} \qquad (3)$$

where φ = the proportionality factor.

From (1), (2) and (3) it follows, after some reduction, that:

$$\frac{1}{\eta} - 1 = \frac{\lambda\varphi}{(\rho c \gamma)^{1/3}} \left(\frac{1}{C}\right)^{2/3} \frac{T_m - T_o}{(T_{max} - T_{min})^{1/3}} \qquad (4)$$

Referring to the data of the Lelystad storage the value of $\varphi\gamma^{-1/3}$ can now be calculated by inserting in (3): $T_m = 45°C$, $T_{max} - T_{min} = 50°C$, C = 683,000 (kWh) and the thermal properties of the soil. The value of $\varphi\gamma^{-1/3}$ being known eq. (3) permits the calculation of the efficiency of all heat stores in the ground that can be considered as (approximately) linear transformations of the Lelystad storage, for arbitrary values of the working temperatures. As an example fig. 2 gives the efficiencies for a storage with a capacity of $3{,}6.10^6$ MJ = 10^6 kWh (temperature of the surrounding soil taken at 10°C). In general

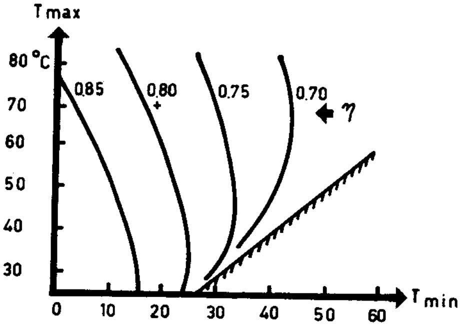

Fig. 2. Efficiency of a thermally stratified seasonal storage in the ground. Capacity = $3{,}6.10^6$ MJ.

increasing the storage temperatures leads to a decrease of the efficiency. However, when the temperature range $T_{max} - T_{min}$ tends to zero the volume, the outside area and consequently the heat losses become so large that this trend is reversed. Increasing T_{max} at a fixed T_{min}, and thereby increasing the temperature range leads to an increase of the efficiency under these conditions.

Incorporation of the collectors in the model.

When incorporating the collectors in the model two special aspects have to be taken in account:
- part of the heat gained in the collectors will be delivered directly to the object to be heated, and not via the storage, and
- the performance of the collectors is influenced by the temperature at which the heat has to be delivered to the storage, that is to say by its working temperatures.

In the development of the model it was assumed that in general 1/3 of the heat gained can be delivered directly to the object (e.g. via a small day to day storage) and the remaining fraction is delivered indirectly via the storage. For climates and objects differing much from those in The Netherlands a different assumption concerning the direct and indirect fraction may have to be made. Denoting the total annual heat demand of the object by H, the collector area by A and the heat gain per unit collector area per year by W, the heat balance of the installation reads:

$$WA = H/3 + 2H/3\eta \qquad (5)$$

The heat gain per unit collector area is a complex function of many variables. In the first order model these complex relations were greatly simplified by relating the heat gain to the average fluid temperature in the collector throughout the year. In its turn the temperature in the collector was estimated to be 10°C higher than the average of the core temperature and the temperature at the outer boundary of the storage. The temperature difference of 10°C is based on the corresponding difference calculated for the Lelystad storage |1|. The temperature at the boundary is also taken into consideration because of the cooling down of the fluid in the heat transfer pipes. At the lower end of these pipes the fluid enters a internal return pipe and is fed back, via the return header, to the collectors. It is to be noted that the difference between the temperature at the outer boundary and the temperature of the surrounding soil is proportional to the difference between the core temperature and the surrounding soil. Starting from the data for the Lelystad storage and a soil temperature of 10°C the average (year) boundary temperature ($\bar{T}_b$) is found to amount to:

$$\bar{T}_b = 20\{(T_{max}+T_{min})/2-10\}/70 \ (^{\circ}C) \qquad (6)$$

So, according to the estimate mentioned above the average (year) collector temperature ($\bar{T}_c$) is given by:

$$\bar{T}_c = \{(T_{max}+T_{min})/2 + \bar{T}_b\}/2+10 \quad (^{\circ}C) \qquad (7)$$

For the Dutch climatic conditions a single glazed, spectral selective flat plate collector (most widely used) is known to deliver about 550 kWh/m^2 yr at a working temperature of 20°C (swimming pools), some 350 kWh/yr at 50°C (hot water supply) and virtually no heat at 140°C. These reference points can be fitted into the following simple relation:

$$W = (140 - \bar{T}_c)^2/24 \qquad (8)$$

It is obvious that this relation is climate-dependent and should be adapted when other climates than the Dutch (NW-European) are considered.
The equations (1) through (8) permit the calculation of the required collector area A at a given H, T_{max} and T_{min}.

Incorporation of economic data in the model

The investment for the storage is less than proportional to the volume of the storage (V) because both the number and total length of the pipes and the area of the insulating sheet are less than proportional to the volume. With the assumed constant temperature difference between the pipes and the soil of 10°C the number of pipes and the total length can be shown to be proportional to $(V)^{1/6}$ and $(V)^{1/2}$ respectively. For a temperature difference proportional to the heat transfer load per unit length of pipe the proportionality exponents become 2/3 and 1 respectively. Taken together these relations suggest a value for the scaling-up exponent in the range of 0.5 to 0.8 to be a fair estimate.
The relative volume of the storage (V'), compared to the Lelystad storage, being given by:

$$V' = (C/683{,}000)\{(70-20)/(T_{max}-T_{min})\} \qquad (9)$$

the investment for the storage (I_s) can be estimated with the equation:

$$I_s = 683{,}000\ (V')^{0.5 \text{ to } 0.8} \quad (\text{Dfl}) \qquad (10)$$

(1 Dfl ≈ .5 US$)

To the investment for the storage the investments for the collectors (I_c) should be added. As stated before it is taken to be Dfl 200.-/m^2, taking the large collector area concerned and future mass production effects into account. So the total investment for the solar seasonal heating installation (I) comes to

$$I = I_s + 200\ A \qquad (\text{Dfl}) \qquad (11)$$

No fixed costs for the control system etc. are included in the investment because these will be relatively small in large systems. Fig. 3 gives the results of some calculations made with a scaling-up exponent of .5. The

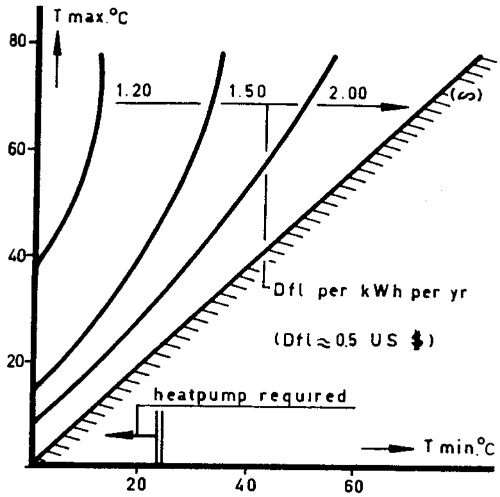

Fig. 3. Investment in a seasonal solar heating installation supplying 10^6 kWh (3600 MJ) per year under Dutch climate conditions (heat pump not included).

investment is expressed in Dfl per kWh per year to allow a quick comparison with energy prices. Calculations with a scaling-up exponent of .8 have also been made but the results differ only slightly from those in fig. 3. The main conclusion to be drawn from fig. 3 is that the seasonal storage should be operated with as low a minimum temperature and as high a maximum temperature as circumstances permit. The beneficial effect of a small volume appears to compensate amply for the decrease in the efficiency of the storage that is associated with a high upper temperature (fig. 2). The main limit to this temperature probably being the increasing vapor pressure of the groundwater as the temperature approaches the boiling point. Only when the minimum temperature is below 20°C (which normally implies that a heat pump has to be used for the extraction of the heat) the optimal maximum temperature shifts to lower values. The disadvantageous effect of a high upper temperature obviously starts dominating in this region.

RESEARCH FOR A SECOND ORDER MODEL

Much research will yet be required for the development of a second order optimization model. The most promising areas for research are

- the unsteady heat transfer from the pipes to the soil under semi-stochastic supply and demand conditions,
- the influence of natural convection phenomena around the pipes and for the storage as a whole,
- the control strategy for the loading and unloading of the storage, and
- the development of quick-fit connections between the heat transfer pipes and the supply and return headers.

On the first three subjects research is currently being executed at the Laboratory for Heat Technology of the E.U.T. Some preliminary results will be presented here.

Unsteady heat transfer

Using both a finite element method and exact solutions for a temperature jump and for harmonic temperature fluctuations more detailed information has recently been obtained concerning the temperature distributions around the heat transfer pipes. Because of the computing time required these calculations were restricted to homogeneous partions of the storage and to an equilateral triangular pipe pattern. In fig. 4 one of the results is depicted. The figure shows the heat up rate of a cylindrical mass of soil after a temperature jump in a coaxial pipe. The dimensionless temperature θ_{av} is the volume averaged temperature increase of the soil related to the change in the temperature of the pipe. The value of 2 of the dimensionless time τ corresponds to about 300 hrs in a real storage. As expected the penetration of the heat is very slow indeed, which points to one of the main problems in the computations: the very large number of time steps that is necessary to obtain reasonable accurate results concerning the temperature distribution around the individual pipes and the temperature required for a further loading of the storage.

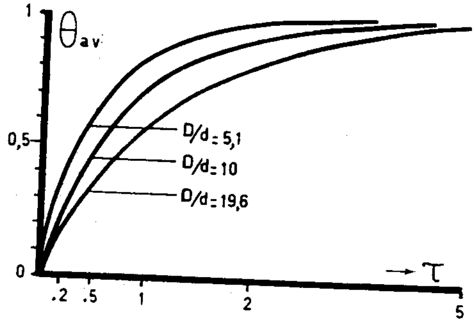

Fig. 4. Heat up rate of a cylindrical mass of soil after a temperature jump in a coaxial pipe. Parameter: cylinder to pipe diameter ratio.

The objective of our research in this field is to obtain a description of the behaviour of storage elements that are rather large in comparison to the mutual distance between the pipes and small in comparison to the storage. It is hoped that the characteristics of these intermediate elements can serve as the starting point for calculations on the storage as a whole.

The influence of natural convection

The problem of natural convection in porous media is not new and much research has been done a.o. in connnection with mining operations and geophysical questions. Starting from this work some results have recently been obtained on the heat transfer, the flow and the temperature distribution around a horizontal pipe, coaxially positioned in a cylindrical porous medium. Fig. 5 gives the ratio in which the natural convection enhances the heat conduction under steady state conditions. The dimensionless number M is defined as:

$$M = \frac{\rho_f g \beta_f \Delta T d_c d_g{}^2}{(\lambda/c)\nu_f} \qquad (12)$$

(ρ = density; β = thermal expansion coefficient; ΔT = temperature difference between pipe and wall of cylinder; d_c radius of cylinder –

radius of pipe; d_g = diameter of grains forming the porous medium; ν = kinematic viscosity; index f = fluid in the pores).

Fig. 5.

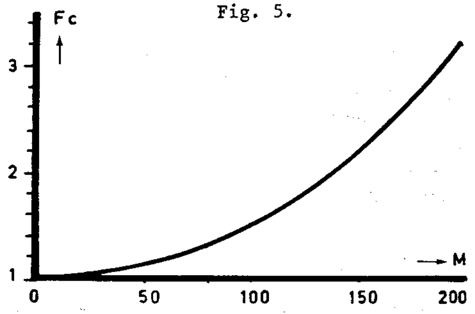

Fig. 5. Convection multiplication factor F_c for the heat transfer from a horizontal pipe into a surrounding porous cylinder. Diameter pipe .04 m, cylinder .60 m. For M see text.

M might be considered as a modified Grashoff number. For groundwater in soil M will only rarely have a value beyond 10, so in this case the influence of the natural convection on the local heat transfer will probably be small. Further investigations for vertical pipes are necessary however.
Fig. 6 gives the streamlines and the temperature distribution around the horizontal pipe for a high value of M.

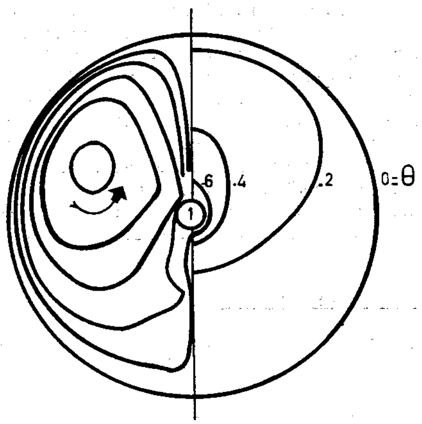

Fig. 6. Stream lines (left) and temperature distribution (right) for combined conductive and convective heat transfer from a horizontal pipe into a surrounding porous medium (M=200). θ = dimensionless temperature.

A fairly strong vertical plume occurs in this case, causing relatively high temperatures in the upper half and relatively low ones in the lower half of the cylinder. Although these effects will probably also be of little significance for heat storage in saturated soil, they may have a considerable influence when the storage of heat in dry sand or a rock bed is considered.

It is clear that the two examples just mentioned represent only a minor fraction of the work that has still to be done for the development of the second order optimization model. However, they may demonstrate the nature of the challenge that lies ahead of the scientists and engineers devoting their efforts to seasonal storage.

ACKNOWLEDGEMENT

The contribution of F.J.A. Martens, student at the E.U.T., concerning the influence of natural convection is gratefully acknowledged.

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