# SIMULATION OF CENTRAL SOLAR HEATING PLANTS USING A DUCT STORE: AN APPLICATION FOR SWITZERLAND

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# 1. INTRODUCTION

Unlike conventional technologies, the design of a central solar heating plant with a seasonal storage (CSHPSS) which uses a duct store in the ground, is based on heat output (kWh) rather than heat demand (kW). It requires design procedures that account for the different thermal processes involved. It is important to assess both short-term and long-term performances. For example, the heat transfer along the ducts in a ground heat store is influenced by different factors, such as the loading conditions of the last days of operation of the system, which determine the temperature profile in the ground around the ducts, and the average ground temperature in the store, which normally varies on a seasonal basis. A transient effect, due to the warming of the surrounding ground, is usually observed during the first years of operation, and affects the thermal performances of the system. In order to obtain an accurate evaluation of the system heat balance, system performances have to be calculated with high resolution over several years, if not the life-time of the system. The final design is generally established with the help of detailed computer simulations, and relies on the ability of the computer programme to reproduce the actual characteristics of the planned system.

Simulation tools which use the TRNSYS programme have been developed for the simulation of CSHPSS systems with a ground duct store. TRNSYS is a well-known modular programme for the simulation of partial or complete energy systems. The simulated systems involve the following subsystems: a collector array, a short-term water buffer store, a ground duct store, an auxiliary heat supply and a heat distribution network to the consumer. State of the art system component models could be used thanks to the TRNSYS programme, which represents a standard basis for user-developed simulation components; (advanced collector and tank components have been used). The world-wide use of TRNSYS by consultant engineers and researchers improves international co-operation and the transfer of technological expertise.

Based on the results of a previous study (Pahud, 1995), the widely used duct store component DST has been improved, so as to be able to take into account finer phenomena related to the heat transfer from the fluid to the ground (local problem). Other components were also created for the needs of the simulation tools (pressure drop caused by the fluid circulation in a ground heat exchanger and pumping power, variable flow rate control, heat load component, etc.). The simulation tools are created for different system layouts, depending on the use or not of a water buffer store and a ground duct store. They also provide different levels of complexity in the way such systems can be simulated. For the purpose of a more general study, where only the main system parameters are investigated, finer effects, such as the optimum flow rate for a particular type of ground heat exchanger, are probably not of primary importance. On the other hand, such effects are susceptible to be investigated, once an optimum system sizing has been found. The simulation tools permit the user to assess the influence of specific problems, which should help the optimisation of a particular component simulated as part of the whole system. The simulation tools may also be used as a basis for other studies which would, for example, assess the influence of the system operation strategy, or the influence of the system layout. The flexibility of the TRNSYS programme makes the tools easily adaptable to a particular problem. This work may also be applied to other fields:

- seasonal duct storage without a solar heat source (waste heat source, etc.);
- duct cold storage and low temperature applications;
- multiple heat extraction boreholes, etc.

The simulation tools are applied to a system defined for typical Swiss conditions. Optimal ratio between the main system parameters are sought for sizing purposes. The influence of the load size, load type (size of the proportion of domestic hot water) and load temperature levels are investigated. The performed simulations have helped to characterise and highlight important points in the design of such systems.

This study has been carried out in the Department of Mathematical Physics at Lund University (Sweden), financed by a grant from the Swiss National Science Foundation. It brings to a close a two year research period spent in the Ground Heat System group.

## 2. OBJECTIVES

The objectives can be divided into two main sections. The first section is devoted to the development of reliable and accurate simulation tools for central solar heating plants with a seasonal duct store in the ground. It involves the following tasks:

- the set up of different system designs, comprising the complete system layout and connections between the different subsystems of such solar heating plants;

- the build up of the simulation tools using TRNSYS. The design tools are destined to be used at different levels of complexity, depending of the purpose of the study to be done.

- improvement of the existing modules when required (e.g. the duct store module), and creation of new modules for specific needs (e.g. load model, pressure drop in a ground heat exchanger, etc.);

- having reasonably fast simulation tools for practical use (according to the system layout);

- verification with more detailed programmes; (which tend to be time consuming and not as flexible).

The second section is concerned with the characterisation of such systems, for the particular case of typical Swiss conditions. This section involves the following tasks:

- characterisation of typical Swiss conditions for such a solar heating plant (type of heat load, weather data, ground properties, etc.);

- to perform a "case study" for typical Swiss conditions; combine today's costs and thermal performances in order to establish an optimal ratio between the different subsystems' size; assess the solar cost of such systems;

- to assess the influence of the load size, the proportion of domestic hot water in the load and the temperature levels of the heat distributed to the consumer;

- comparison with a solar heating system without ground duct store;

- characterisation of the behaviour of such systems so as to develop a better control strategy.

The availability of these design tools, together with the treated examples, should hopefully help in the choice and design of future solar heating plants.

## 3. SYSTEM DESIGNS AND OPERATION

Three different system designs are defined, depending on the use or not of a short-term water buffer tank or a seasonal heat storage in the ground. The system designs are voluntarily very simple. This makes a simple system control possible, which is an important factor for satisfactory system operation and system reliability.

### 3.1 System Design without a Short-Term Buffer Tank

The design is very similar to that used extensively in the analysis of systems using duct store with the MINSUN programme (Mazzarella, 1989, 1991). An important difference is that direct connection between the collector array and the distribution network is taken into consideration in the present study. The dynamic behaviour of the interaction between the solar collectors and the duct store is also taken into account more accurately, as a smaller time-step is used in the simulations (≈1 hour instead of ≈1 day). The layout of the system is shown in Fig. 3.1.

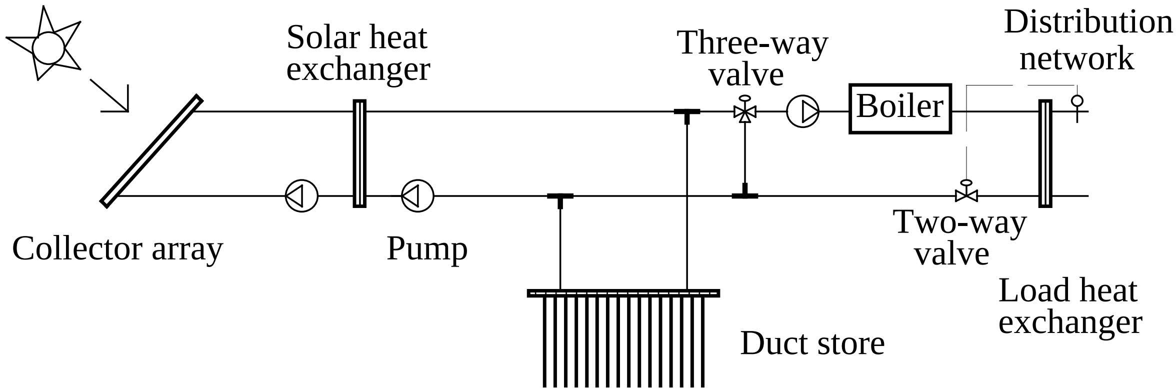



Fig 3.1 Design of the system without a short-term buffer tank.

On/off controllers with dead-band temperature differences control the two pumps of the collector subsystem. The flow rate is set to a constant value when usable solar gains are available. The flow rates have the same value on both sides of the solar heat exchanger; (they are called "collector flow rates").

The three-way valve permits the disconnection of the solar collectors and the duct store from the rest of the system, if the forward fluid temperature (before the boiler) is lower than the return fluid temperature from the load heat exchanger. The load heat exchanger separates the heat distribution network from the central heating plant. The inlet temperature on the primary side can not fall below a value which is shifted by some additional Kelvins (typically 5K) relative to the requested outlet temperature on the secondary side. The boiler is used if necessary. A two-way valve, controlled by the forward fluid temperature in the distribution network, reduces to the maximum the flow rate on the primary side, thus making the lowest return temperature to the solar heating system possible. The pump is switched off if the heat demand is null.

The flow rate through the duct store results from the difference between the collector flow rate and the return flow rate from the load subsystem. Typically, the duct store is loaded when the collector pumps are on, and the fluid circulates from the centre to the border of the store. When they are off, the fluid circulation is reversed and the duct store is unloaded if possible.

## 3.2 System Design with a Short-Term Buffer Tank

The water buffer tank is inserted in the system so that the collector array (via the solar heat exchanger), the duct store and the load subsystem are directly connected to it. These three subsystems can be operated independently. The layout of the system is shown in Fig. 3.2.

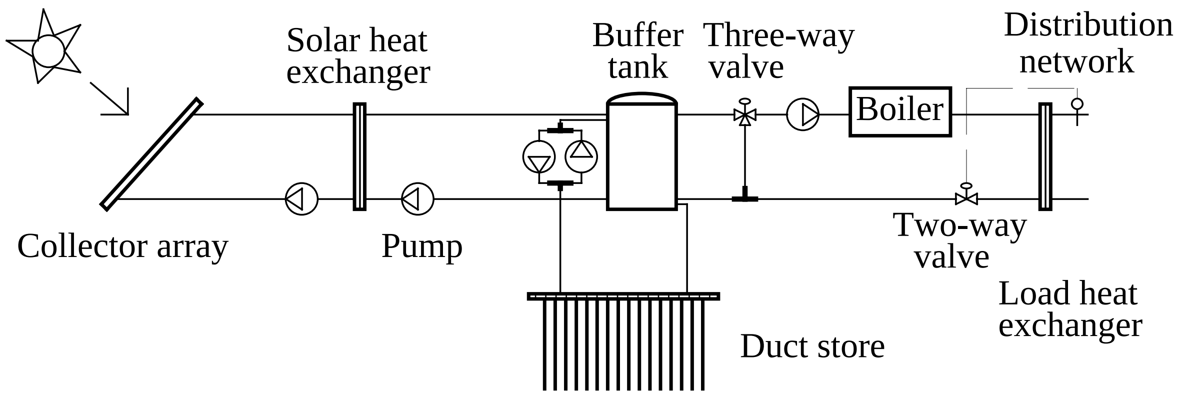



Fig 3.2 Design of the system with a short-term buffer tank.

The collector array and the load subsystem are operated as in the system without buffer tank. Two additional pumps are required for the operation of the duct store: one pump when the duct store is loading the buffer; (inlet at the top and outlet at the bottom of the buffer), and the other when the buffer is loading the duct store; (the heat carrier fluid is circulated in the opposite direction). If heat can be transferred from the buffer tank to the duct store, this latter is loaded with a flow rate set to the half of the nominal flow rate in the collectors. If the duct store can load the buffer tank, then the flow rate through the duct store is set to the flow rate value in the hot side of the load heat exchanger. This system control defines the *reference strategy*. Extensive system simulations are performed with this design.

## 3.3 System Design without a Seasonal Ground Heat Store

For comparison purposes, a system without a ground heat storage is also examined. The system layout is shown in Fig. 3.3. The same system operation is used for the collector array and the load subsystem.

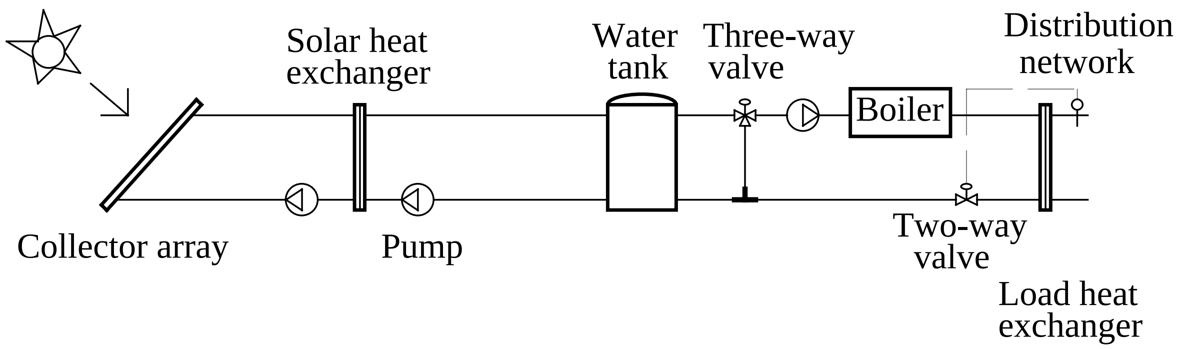



Fig 3.3 Design of the system without a seasonal ground heat store.

# 4. SUBSYSTEM COSTS

The cost of the whole solar heating system is divided into the costs of the three main subsystems: the collector array, the short-term water buffer store and the duct store in the ground. These costs do not involve the cost of the auxiliary boiler as they are used to estimate the solar cost (see section 4.4 for the solar cost). The costs given in this chapter should be representative of typical costs found in Switzerland.

## 4.1 The Collector Array

The cost of the collector array include all the costs involved for the build up of the collector subsystem, from the solar collectors to the connecting pipes and the heat exchanger that will transfer the solar heat to other subsystems. The simulated solar collectors correspond to high performance single-glazed flat-plate collectors. They probably would be mounted on the roof. The cost function **COLCOST**, expressed in Swiss francs (CHF), is voluntarily simple, and set as a fixed price per collector area (COLAREA):

$$\text{COLCOST} = 600\ \text{CHF/m}^2 \cdot \text{COLAREA} \tag{4.1}$$

This cost corresponds roughly to a collector array of 1'000 - 2'000 m$^2$ of collectors. This cost is also used for larger collector areas, although further cost reductions are expected due to the effect of a larger scale installation.

## 4.2 The Short-Term Water Buffer Store

The price of the water tank includes its installation, the thermal insulation and the hydraulic connections to the other subsystems. The specific price of a water tank is strongly dependent on its size, especially when the tank volume tends to be small. The cost function **BUFCOST** is based on the following specific prices for installed steel tanks: 1'400 -1'500 CHF/m$^3$ for a 1 m$^3$ tank, 500 CHF/m$^3$ for a 500 m$^3$ tank and 300 CHF/m$^3$ for a 10'000 m$^3$ tank. The following cost function fits the cost values (Hadorn, 1995):

$$\text{BUFCOST} = \left(\text{Cb} + \frac{(\text{Co} - \text{Cb})}{(\text{BUFVOL} / \text{Vo})^{\beta}}\right) \cdot \text{BUFVOL} \tag{4.2}$$

where:

| | | | |
|---|---|---|---|
| BUFCOST: | | [CHF], | cost of a water tank store of volume BUFVOL; |
| BUFVOL: | | [m$^3$], | volume of the water tank; |
| Cb: | 100 | [CHF/m$^3$], | asymptotic specific cost for an infinite volume; |
| Co: | 1'500 | [CHF/m$^3$], | specific cost for a tank volume of Vo; |
| Vo: | 1 | [m$^3$], | tank volume corresponding to the specific cost Co; |
| β: | 0.2 | [-], | adjustable exponent. |

The shape of the specific cost function is shown in Fig. 4.1 for a large range of tank volumes. The specific price of a tank volume is calculated to 980, 650 and 450 CHF/m$^3$, for tank volumes of respectively 10, 100 and 1'000 m$^3$.

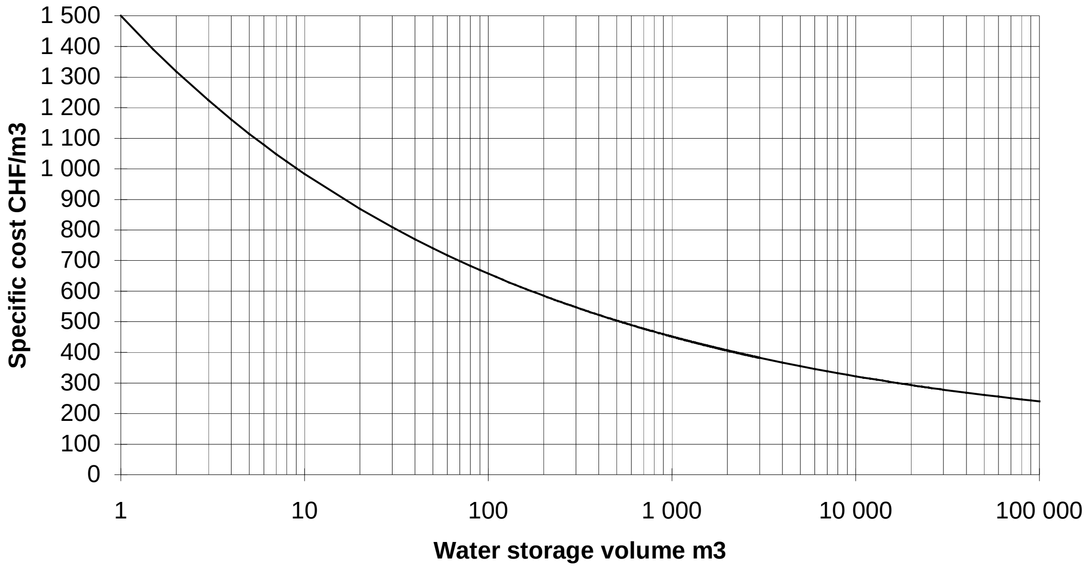



Fig. 4.1 Specific cost of an installed water tank store shown in relation to the tank volume.

## 4.3 The Ground Duct Store

The cost of the duct store includes all the costs involved in the realisation of the store, from the ground drilling to the flow circuits that will transport the heat carrier fluid to and from the store. The primary cost is the specific cost of an equipped borehole, BORECOST, expressed in CHF per meter borehole. This cost includes drilling, pipes, pipe installation and refilling if the space between the pipes and the borehole wall has to be filled. Based on the experience of the construction of duct stores in Sweden (Nordel, 1994), other costs should be considered so that the number of boreholes is penalised. These costs may involve a fixed initial cost per borehole, the land area used by the top of the store, top insulation, drilling in the soil layer covering the store, the connecting and collecting pipes (see below) and so on. The cost function **DUCTCOST** used in this study is defined by:

$$
\begin{aligned}
\text{DUCTCOST} = \; & \text{BORECOST} \cdot \text{NBORE} \cdot \text{HEIGHT} \; + \\
& \text{BORECOSTTOP} \cdot \text{NBORE} \cdot \text{DEPTHTOP} \; + \\
& \text{FIXBORE} \cdot \text{NBORE} \; + \\
& (\text{LANDCOST} + \text{INSULCOST} \cdot \text{INSULTHICK}) \cdot \text{AREALAND} \; + \\
& \text{CONNCOST} \cdot \text{NBORE} \cdot \text{BORESPACING} \; + \\
& \text{COLLCOST} \cdot 4 \cdot (\sqrt{\text{NBORE}} - 1) \cdot \text{BORESPACING} \; + \\
& \text{INITIALCOST}
\end{aligned}
\qquad (4.3)
$$

where:

| | | |
|---|---|---|
| BORECOST: | 80 | [CHF/m], specific cost of an equipped borehole; |
| NBORE: | | [-], number of boreholes forming the ground heat exchanger; |
| HEIGHT: | | [m], active length of one borehole, i. e., borehole length where the heat transfer with the ground takes place; |
| BORECOSTTOP: | 100 | [CHF/m], specific cost of the equipped borehole in the top soil layer (including pipe insulation); the heat transfer from the borehole to the ground is assumed to be negligible in this ground layer; |
| DEPTHTOP: | | [m], depth of the top soil layer covering the store; |
| FIXBORE: | 200 | [CHF], fixed initial cost per borehole; |
| LANDCOST: | 100 | [CHF/m$^2$], cost of the ground per unit area on top of the store; |
| INSULTHICK: | | [m], thickness of the insulation placed on top of the store; |
| INSULCOST: | 500 | [CHF/m$^3$], insulation cost per unit volume; |
| AREALAND: | | [m$^2$], area occupied by the top of the store and the top insulation; |
| BORESPACING: | | [m], borehole spacing corresponding to a quadratic pattern; |
| CONNCOST: | 80 | [CHF/m], connecting pipe cost per unit length of connection; these pipes conduct the heat carrier fluid between the boreholes and the collecting pipes; |
| COLLCOST: | 80 | [CHF/m], specific cost of the collecting pipes; the collecting pipes distribute and collect the heat carrier fluid to and from the connecting pipes. For a quadratic pattern, their total length amounts to about 4 times the side of the square area occupied by the top of the store, i. e. $4 \cdot (\sqrt{\text{NBORE}} - 1)$; |
| INITIALCOST: | 4'000 | [CHF], initial cost for the building of the store. |

The land area AREALAND and the borehole spacing BORESPACING are calculated with the following two relations, by assuming a quadratic arrangement of the boreholes:

$$
\text{AREALAND} = \left( \sqrt{\frac{\text{DUCTVOL}}{\text{HEIGHT}}} + 2 \cdot \text{HEIGHT} \cdot \text{FRISO} \right)^2 \qquad (4.4)
$$

$$\text{BORESPACING} = \sqrt{\frac{\text{DUCTVOL}}{\text{HEIGHT} \cdot \text{NBORE}}} \quad (4.5)$$

where

DUCTVOL: [$m^3$], volume of the duct store, defined as NBORE times the ground volume ascribed to one borehole; the boreholes are assumed to be uniformly placed within the store volume;

FRISO: [-], ratio related to the top insulation of the store. The top insulation may extend horizontally beyond the border of the store. The width of the insulation overlay, divided by the active length of one borehole, HEIGHT, defines the ratio FRISO; FRISO is set to zero if no insulation is used.

The mentioned specific cost for an equipped borehole, BORECOST, corresponds to a typical cost for drilling in soils such as moraine, limestone or sandstone. The price includes polyethylene double-U pipes, and a refilling of the borehole with bentonite or another more conductive material. These borehole installations are used in low temperature applications. This cost is used in this study, although high temperatures may cause a problem for the pipe material used (polyethylene). Pipes in a material that can withstand high temperatures may result in a higher cost for the borehole installation. Another consideration for pipe material is the importance of the oxygen diffusion, which may increase with higher temperatures. Oxygen diffusion has to be reduced as much as possible if the heat carrier fluid enters in contact with other components made of steel (heat exchanger, water tank, etc.).

In Fig. 4.2, the specific cost of the duct store is shown in relation to the store volume. The cost is calculated for 3 different shapes of the store, defined with the ratio HEIGHT over DIAMETER. The diameter is defined as the diameter of the disk whose area is the same as the store section. The specific cost is calculated with the above mentioned costs. A borehole spacing of 2.5 m is assumed. A layer of 20 cm of insulation covers the top of the store and extends to 5% of its vertical extension (height). The top of the store is at the ground surface. The graphs show the influence of having fewer but deeper boreholes on the specific cost (the ratio HEIGHT over DIAMETER becomes greater). They also indicate the increase of the specific cost with smaller storage volumes, which is particularly important for storage volumes smaller than 10'000 - 20'000 $m^3$. For a duct store of 20'000 $m^3$, the top insulation amounts to 20% of the total storage price. A earth layer on top of the store can replace the insulation. By keeping the specific cost of the 20'000 $m^3$ storage constant, the earth layer can be 9, 12 or 16 m thick if the ratio HEIGHT over DIAMETER is respectively 1, 2 and 3. This increase is due to the insulation layer that extends to 5% of the storage vertical extension, which increases with increasing ratio HEIGHT over DIAMETER.

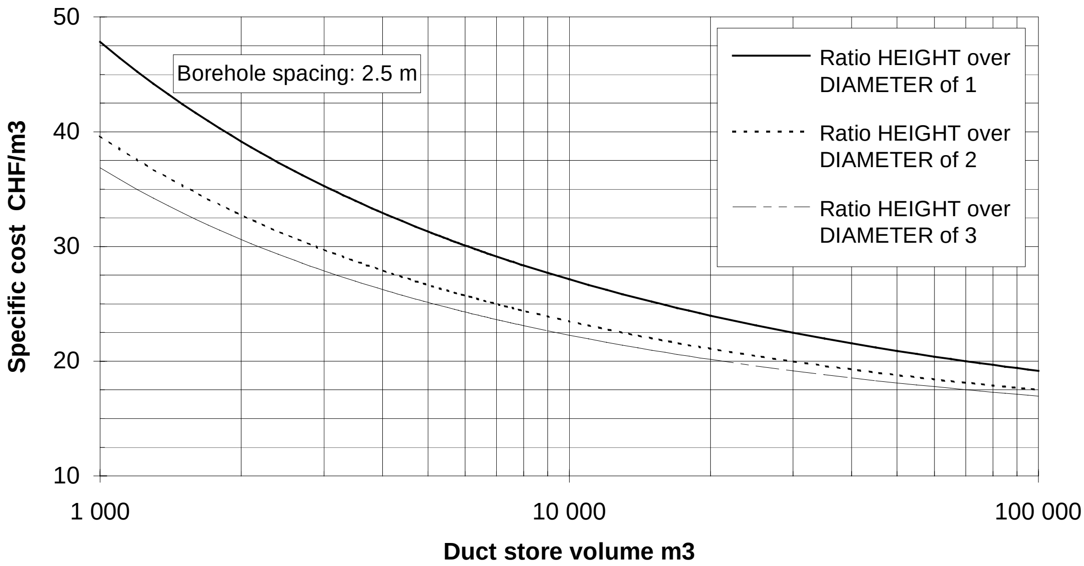



Fig. 4.2 Specific cost of a duct store shown in relation to the tank volume. The cost is calculated for 3 different shapes of the store, defined with the ratio HEIGHT over DIAMETER. The diameter is defined as the diameter of the disk whose area is the same as the store section. The borehole spacing is set to 2.5 m; a layer of 20 cm of insulation covers the top of the store and extends to 5% of its vertical extension; the top of the store is at the ground surface.

## 4.4 The Solar Cost

The solar cost is defined as the annualised cost of solar heat per MWh supplied to the heat load, including operating and capital cost for the collectors and the heat storages.

$$\text{Solar Cost} = \frac{\text{Annuity} \cdot \text{SolarCapitalCost}}{\text{SolarFraction} \cdot \text{AnnualHeatLoad}} \qquad (4.6)$$

where

**Annual Heat Load**: annual head demand of the consumer, for the space heating and the domestic hot water preparation. In this study, the distribution heat losses are included in the annual heat load.

**Solar Fraction**: fraction of the annual heat load that is covered by solar heat. The solar heat is the energy produced by the solar part of the system, i. e., the solar collectors, the buffer store and the duct store. In this study, the annual solar heat is the mean value during the first 25 years of system operation, and takes into account the cold start of the storages.

**Solar Capital Cost**: capital cost of the three main subsystems forming the solar part of the whole system, i. e., capital cost of the solar collector array, the short-term buffer store and the seasonal ground heat storage.

The **annuity** factor can be calculated with:

$$\text{Annuity} = q^n \cdot \frac{(q - 1)}{(q^n - 1)} + f_{op} \qquad (4.7)$$

Where: $q = 1 + i / 100$

i : mortgage interest rate

n : number of years for the mortgage

$f_{op}$ : operation cost expressed as a percentage of the solar capital cost

Example: i = 8%

n = 25 years

$f_{op}$ = 1% => Annuity = 0.1

A **annuity factor of 0.1** is assumed in this study. The transformation of the solar cost so that it would correspond to another annuity factor can thus easily be performed in one's head.

The primary variables to optimise are the collector area, the duct store volume, the borehole spacing and depth, and the buffer tank volume. Other optimisations may involve the system layout and operation strategy, as well as the optimisation of a particular component as part of the whole system. There are also physical constraints on the system to optimise. The fluid in the collector array must not boil, as the system would be less reliable, requiring (much?) more maintenance to make it run. The maximum permitted temperature in the storage and or the pipes of the ground heat exchanger may also be a constraint on the optimisation of the system. Finally, there are practical constraints such as the maximum available area for the collectors, the maximum room available for a buffer store and the local geology for a seasonal storage in the ground.

Theoretically, the solar cost can be minimised for any solar fraction, given all the thermal characteristics of the different subsystems, the physical and practical constraints and all the different costs. Practically, it is very difficult to have an accurate picture of the whole project. New technologies are often developed on site, principally due to the lack of experience. There are uncertainties on some parameters used in the simulation programme, that can affect the overall thermal performance of the system. The actual thermal characteristics of a component can be quite different from the design values, and may significantly affect the overall thermal performances of the system.

Sensitivity analyses are required to explore the influence of possible variations of the major parameters. Experience is also required to anticipate the influence of the variation of some specific parameters. For example, the overall heat transfer coefficient of a conventional heat exchanger may have a strong negative influence on the solar fraction of a solar heating system, if its value is undersized. On the other hand, an oversized value will result in additional costs without a significant increase of the solar fraction. Practical experience has shown that the actual value of the overall heat transfer coefficient may in some cases be half of the design value, mainly due to fouling (Mermoud et al., 1991; Dalenbäck, 1993). Depending on the importance of the influence, a heat exchanger should be adequately oversized. In some cases, it may be necessary to have the possibility to clean it.

## 4.5 Cost Reduction Margin

It is difficult to transpose the costs from one country to another. Nevertheless, typical costs found in Sweden are given here, to show that lower costs are already possible today. The exchange rate between the Swedish crown (SEK) and the Swiss franc (CHF) is fixed to 5.3 SEK/CHF, and corresponds to the current exchange rate used in July 1996.

**Collector array:**

For a total collector area of 1'000 to 2'000 $m^2$, formed with large module collectors mounted on the roof, the collector array cost is less than 400 CHF/$m^2$. A CSHPSS system is currently being built in Germany with such prices for the collector array. If roof integrated collector modules are used, the marginal cost for the collector array, compared to the cost of a normal roof, is about 200 CHF/$m^2$ (Dalenbäck, 1996). Thus, the cost of the collector subsystem can be one third to one half cheaper than of the price assumed in this study.

**Water store:**

Typical prices for water store of 100 $m^3$ are less than 600 CHF/$m^3$, and 250 CHF/$m^3$ for 1'000 $m^3$ (Dalenbäck, 1996). These prices do not include inside heat exchangers, as is the case with the prices assumed in this study. They are respectively 10% and 40% cheaper than the prices assumed for the same storage volumes.

**Duct store:**

Comparison of the prices are even more complicated with a duct store, due to the influence of the type of ground and the local geology. Drilling in granite costs less than 40 CHF per meter borehole on average, for depths up to 150 - 200 m. The borehole is normally filled with the ground water, and a single U-pipe in plastic is inserted. Such a borehole equipped with a single U-pipe would cost about 40 -45 CHF/m (Hellström, 1996).

# 5. WEATHER DATA AND HEAT LOADS

In a system using a duct store, the heat rate exchanged with the store is mainly determined by the difference between the temperature level of the heat carrier fluid and the ground in the immediate vicinity of the borehole. The precision of the calculation will depend on the ability to reproduce the rapid temperature variation of the heat carrier fluid heated by the solar collectors. The time-step at which weather data is provided will also have an effect on the temperature variations and thus the calculated performances. Hourly weather values are necessary for detailed system simulations. An hourly time resolution is also fine enough for accurate results (Pahud, 1996b).

The hourly weather data values are generated with the help of METEON95 (Meteonorm, 1995), a powerful programme for the calculation of climatic data, based on series of 10 years of measurements in many different places and correlations. Weather values of any location in Switzerland can be generated. Data for 95 European towns are also included. The incident radiation on any plane is calculated, given the location, the situation (town, open area, valley bottom, etc.) and the horizon.

The hourly load data are calculated by using a simple load model (Pahud, 1995). It is a one-node model based on the steady-state heat losses of a building and a solar effective area for the collection of the passive gains. An effective heat capacity is used for the storage of the passive solar gains. The model is used in its simplest form, without explicitly calculating the passive solar gains. In this case the heat demand is calculated according to the degree-day approach, with an outdoor temperature limit above which the heating is stopped.

The domestic hot water demand is given with a representative daily profile. The forward and return fluid temperatures of the heat distribution network are prescribed and depend on the outdoor temperature. (See Annex A.4 for the TRNSYS component of the heat load).

## 5.1 The Weather Data

The reference weather data file is chosen for Geneva. The data values are generated for a collector plane facing the south and titled to 45°. In Meteon95, the Anetz site of Genève-Cointrin is chosen (latitude: 46.15°; longitude: -6.07°; altitude: 420 m), corresponding to an open area. A constant horizon of 10° is prescribed. The standard output is selected and the following 6 variables are generated on a hourly basis: the global horizontal insolation H-Gh, the diffuse horizontal insolation H-Dh, the global insolation on the tilted plane H-Gkhor (with horizon effects), the diffuse insolation on the tilted plane H-Dkhor (with horizon effects), the beam insolation at normal incidence H-Bnhor (with horizon effects) and the outdoor temperature Ta. The monthly values are given in Table 5.1.

| Month | H-Gh [kWh/m²] | H-Dh [kWh/m²] | H-Gkhor [kWh/m²] | H-Dkhor [kWh/m²] | H-Bnhor [kWh/m²] | Ta [°C] |
|---|---|---|---|---|---|---|
| Jan. | 29 | 22 | 42 | 23 | 23 | 1.0 |
| Feb. | 46 | 33 | 61 | 35 | 34 | 1.8 |
| Mar. | 92 | 52 | 115 | 56 | 76 | 5.6 |
| Apr. | 122 | 75 | 126 | 73 | 79 | 8.8 |
| May | 156 | 91 | 145 | 85 | 99 | 13.2 |
| June | 172 | 95 | 153 | 88 | 113 | 16.6 |
| July | 189 | 92 | 174 | 89 | 149 | 20.3 |
| Aug. | 161 | 73 | 168 | 75 | 140 | 19.6 |
| Sep. | 116 | 64 | 138 | 67 | 93 | 15.9 |
| Oct. | 69 | 45 | 89 | 48 | 54 | 11.0 |
| Nov. | 31 | 22 | 45 | 23 | 28 | 5.3 |
| Dec. | 24 | 16 | 39 | 19 | 24 | 2.6 |
| Year | 1'205 | 678 | 1'295 | 682 | 912 | 10.1 |

Table 5.1 Monthly weather data values for Geneva. See text for symbol definitions.

Other towns such as Basel, Zurich, Bern, Lausanne (Pully) and Lugano have an annual global incident insolation that ranges from 1'200 to 1'350 kWh/m$^2$; (in a plane tilted to 45° and facing south). The mean annual air temperature lies between 9 and 12 °C. Rather different climates can be found in the Alps. For example, Sion (altitude: 480 m) and Davos (altitude: 1'590 m) have an annual insolation of respectively 1'540 and 1'680 kWh/m$^2$ in the same plane. The mean annual air temperature is respectively 9.4 and 3.2 °C.

## 5.2 The Heat Loads

The annual heat load is the annual energy used for the heating of one or more buildings and for the preparation of the domestic hot water. The distribution heat losses are also included. A heat load of 500 MWh/year is considered as a typical size for Switzerland. Nevertheless, it is still a small size for a central solar heating plant with a seasonal storage, especially if heat is stored directly in the ground. In order to explore the influence of the load size on the solar heating system, annual heat loads of 500, 1'000 and 5'000 MWh are investigated. The proportion of energy used for the domestic hot water (dhw) and the space heating (sh), and the temperature levels of the heat distributed are also examined. Seven different loads are defined.

*5.2.1 Small load (25% dhw, 75% sh), moderate-temperature heat distribution*

*Small load* designates a load of **500 MWh/year**. Seventy-five percents of the annual load is used for the heating of the buildings. The rest is shared between the preparation of the domestic hot water and the distribution losses, set to 6% of the annual load. If single-family houses are considered, an annual heat consumption of 3'500 kWh/house can be assumed for the domestic hot water, or 35 kWh/m$^2$year (125 MJ/m$^2$year) with a floor area of 100 m$^2$. The *small load* would then correspond to **27 single-family houses**. The heating heat demand of a house would amount to 140 kWh/m$^2$year or 500 MJ/m$^2$year, and correspond to a conventional house. Although the level of insulation is not so high, the temperature levels of the heat distribution are assumed to be moderate. The forward and return fluid temperatures from and to the central heating plant, as shown in Fig. 5.1, define the *moderate-temperature heat distribution*.

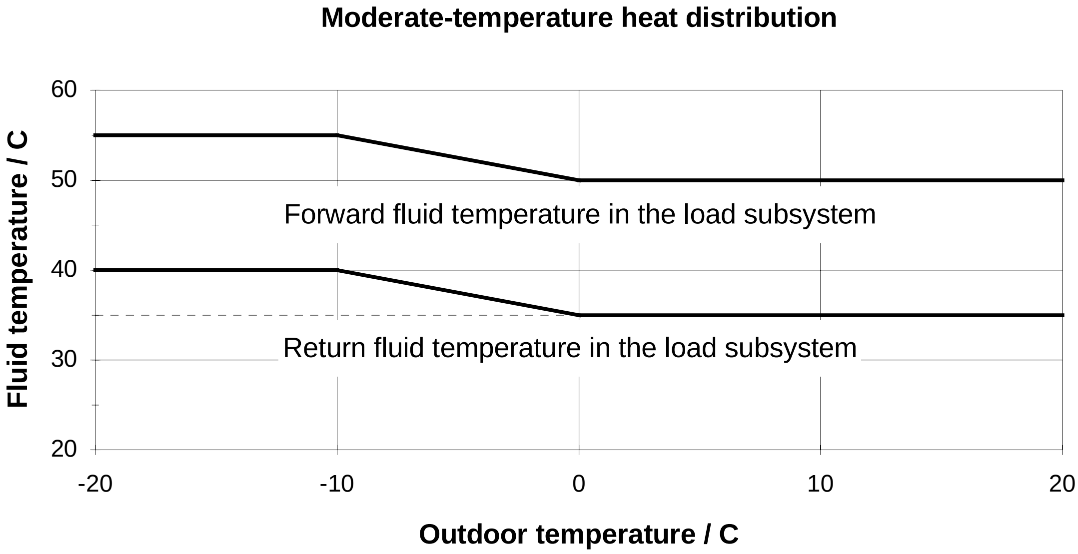



Fig. 5.1 Forward and return fluid temperature defining the moderate-temperature heat distribution.

*5.2.2 Medium load (25% dhw, 75% sh), moderate-temperature heat distribution*

Same heat load as the *small load* (25% dhw, 75% sh), but two times larger (**1'000 MWh/year**). This load is equivalent to approximately **50 single-family houses**. The temperature levels of the heat distribution correspond to the moderate-temperature heat distribution (see Fig. 5.1).

*5.2.3 Large load (25% dhw, 75% sh), moderate-temperature heat distribution*

Same heat load as the *small load* (25% dhw, 75% sh), but ten times larger (**5'000 MWh/year**). This load is equivalent to approximately **270 single-family houses**. The temperature levels of the heat distribution correspond to the moderate-temperature heat distribution (see Fig. 5.1).

*5.2.4 Small load (50% dhw, 50% sh), moderate-temperature heat distribution*

Same heat load as the *small load* (**500 MWh/year**), but the space heating requirement represents only 50% of the annual load. The distribution heat losses are also set to 6% of the annual load, although they should be larger than those with 25% dhw, due to a larger extension of the distribution network. The rest, 44%, is the energy required for the preparation of the domestic hot water. Using the same assumptions as in section 5.2.1, this heat load corresponds to approximately **60 single-family houses**. The space heating heat demand would be 40 kWh/$m^2$year (140 MJ/$m^2$year), which is close to the heating requirements of a modern low energy house.

*5.2.5 Medium load (50% dhw, 50% sh), moderate-temperature heat distribution*

Same heat load as the *small load* (50% dhw, 50% sh), but two times larger (**1'000 MWh/year**). This load is equivalent to approximately **125 single-family houses**. The temperature levels of the heat distribution correspond to the moderate-temperature heat distribution (see Fig. 5.1).

*5.2.6 Large load (50% dhw, 50% sh), moderate-temperature heat distribution*

Same heat load as the *small load* (50% dhw, 50% sh), but ten times larger (**5'000 MWh/year**). This load is equivalent to approximately **630 single-family houses**. The temperature levels of the heat distribution correspond to the moderate-temperature heat distribution (see Fig. 5.1).

*5.2.7 Small load (100% sh), low temperature-heat distribution*

Same heat load as the *small load* (**500 MWh/year**), but only space heating is considered, so that an extremely low temperature heat distribution can be assumed. If low energy houses are considered, this load is equivalent to approximately **120 single-family houses**. The forward and return fluid temperatures from and to the central heating plant, as shown in Fig. 5.2, define the *low-temperature heat distribution*.

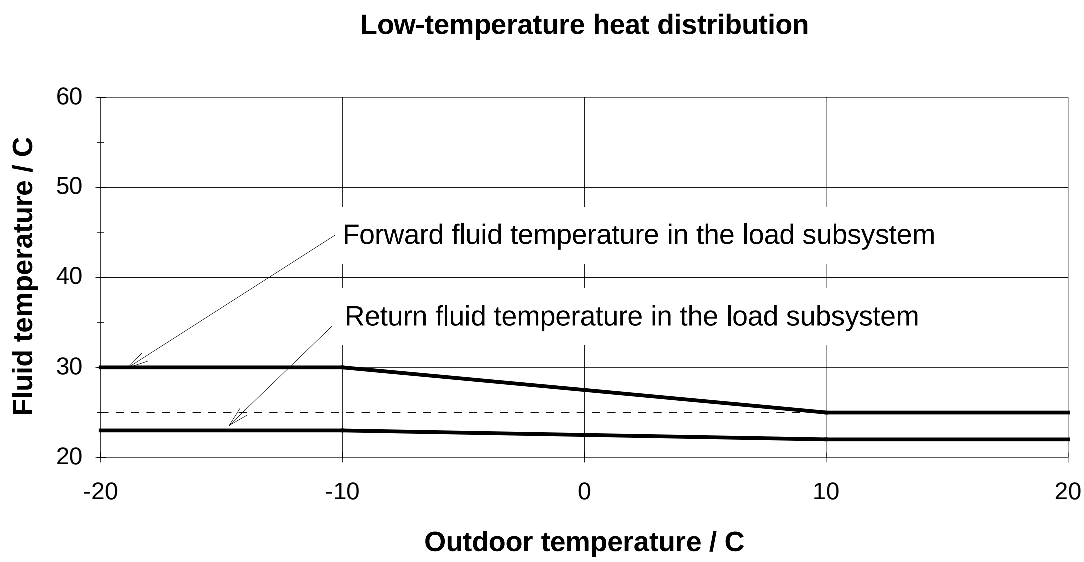



Fig. 5.2 Forward and return fluid temperatures defining the low-temperature heat distribution.

# 6. SYSTEM SIMULATION TOOLS

TRNSYS 13.1 (Klein et al. 1990) is used for the simulation of the CSHPSS systems (Central Solar Heating Plant with a Seasonal Storage). TRNSYS is a widely used, modular thermal process simulation programme. Subroutines are available that represent, for example, the components of typical solar energy systems (such as solar collectors, water storages, pumps, pipes, valves, controllers, etc.). Users can readily write their own component subroutines if they have a particular need. Seasonal storage models (water in a rock cavern, SST; duct storage in the ground, DST), were developed at Lund University (Eftring and Helltröm, 1989; Hellström, 1989), and have also been implemented in TRNSYS as seasonal storage components (Mazzarella 1993a, 1993b; Hellsröm et al., 1996). By a simple language, a TRNSYS deck "connects" the components together in a manner analogous to piping, ducting, and wiring in a physical system. The programmer also supplies values for all of the parameters describing the components to be used. The programme does the necessary simultaneous solutions of the algebraic and differential equations which represent the components and organises the inputs and outputs.

The TRNSYS decks of the systems described in chapter 3 are based on the results of detailed studies which dealt with the simulation of a CSHPSS system with a seasonal duct store (Pahud, 1995). These studies have led to a new improved version of the ground duct store component DST for TRNSYS (Pahud and Hellström, 1996). Accurate system simulation tools using TRNSYS were also developed and validated with more detailed programmes.

## 6.1 The User-Written TRNSYS Components

The user-written TRNSYS components comprise all the TRNSYS components that are used in this study, but were not sold with the components included in TRNSYS 13.1. In particular, all the main components of the system (collector array, buffer store and ground duct store) are "user-written", developed, validated and documented by their respective authors. Some other components, developed especially for this study, are documented in this report.

### *6.1.1 Collector component*

A collector component (TYPE 52), based on the Matched Flow Collector model (MFC), is chosen for the simulation of the collector field (Isaksson, 1995). The version 1.0β, which dates from August 1993, is used. The main reasons for its choice are the possibilities to take into consideration a quadratic temperature dependence of the overall loss coefficient as well as the heat capacitive effects of a solar collector field.

### *6.1.2 Buffer store component*

The tank component (TYPE 74) developed at the ITW of the Stuttgart University (Druck and Pauschinger, 1994), is chosen for the possibility of handling up to five flow loops (or hydraulic circuits) connected to the tank. The other features of the component are not used (up to three internal heat exchangers, an internal auxiliary heater, etc.).

The earliest choice was the Multi-Flow Stratified Thermal Storage Model with Full Mixed Layers, XST (TYPE 62), developed by Mazzarella (1993a). A seasonal storage in or on the ground is simulated, including the thermal process in the surrounding ground. Up to 4 flow loops can be connected to the store. The XST component is better suited to seasonal heat storage. In relation to the annual system heat demand and the collector area, the small size of the buffer tank makes it function in short-term storage mode. Due to a technical reason in the XST computer code, the volume of water moved during 1 TRNSYS time-step has always to be smaller than the volume of one water layer in the storage. In consequence, the number of nodes, which defines the number of horizontal water layers that form the storage volume, has to be reduced to its minimum value and the TRNSYS simulation time-step reduced to a very small value. The execution time is thus significantly increased. For example, a buffer size of 20 litres/$m^2$ of collector area can be simulated with only 3 nodes and a TRNSYS simulation time-step of 0.1 hour, if a flow rate of 25 litres/$m^2$hour is set in the collector flow loop connected to the storage.

The TRNSYS decks exist with both XST and the TYPE 74 for the buffer tank. Annual results calculated with both decks are very close, provided that equivalent parameters are given to the buffer components. Parameters for the storage heat losses calculation should be adjusted so that the annual heat losses are approximately the same. With the use of TYPE 74, the TRNSYS simulation time-step is set to 0.25 instead of 0.1 hour. The execution time is about 40% shorter.

*6.1.3 Duct store component*

As previously mentioned, the duct storage model was developed at Lund University (Sweden). It can be used for the simulation of thermal processes that involve heat storage and/or cold storage in the ground. It was chosen in 1981 by the participants of the International Energy Agency, Solar R&D Task VII (Central Solar Heating Plant with Seasonal Storage) for the simulation of duct ground heat storages. A simpler but faster version was implemented in the MINSUN programme (Mazzarella, 1991), a simulation tool for the optimisation of a CSHPSS system. A TRNSYS version based on this faster version has been implemented by Mazzarella (1993b). With the new DST version for TRNSYS (Hellström et al., 1996), the easy utilisation of the simple version is combined with the additional features of the more detailed original DST programme (Hellström, 1989). This version also offers the possibility of a detailed computation of the local heat transfer along the flow path within the storage region (Pahud and Hellström, 1996). The local solutions, which take into account the short-time effects of the injection/extraction through the ducts, may take into account a radial stratification of the store temperatures (due to a coupling in series of the ducts (or boreholes)), as well as an increased resolution in the vertical direction. The local heat transfer resistance from the fluid to the ground (or borehole thermal resistance) may depend on the flow conditions. It may also take into account the unfavourable internal heat transfer between the downward and upward flow channels in a borehole. The effect of a regional ground water flow is not taken into consideration.

If the local heat transfer resistances depend on the flow conditions, a two-entry table, containing the temperature- and flow-dependent thermal resistances, is given as input to the DST component. A computer programme, called BOR (Pahud, 1996a), has been developed for the calculation of these local heat transfer resistances and the generation of the input data file. This programme, based on the Earth Energy Designer programme (Hellström and Sanner, 1994), has a user-friendly interface for the set up of the ground heat exchanger design. Coaxial pipe, single, double and triple U-pipe in a borehole are implemented in the programme. Data bases for the thermal properties of various ground types and materials, fluid properties of different heat carrier fluids, as well as the characteristics of common pipes, help the user to build his own ground heat exchanger. Graphical visualisations of the calculated quantities are also provided in the programme. The programme allows the user to generate a library of a large range of different ground heat exchangers, that can then be simulated in a system by using the DST component for TRNSYS.

*6.1.4 Pressure drop component*

In a ground heat system, the electric energy required for the circulation of the fluid may, in some cases, be significant relatively to the annual heat recovered from the duct store. In the final design of a duct store, the electric power consumed by the pumps should be assessed, as it will affect the design of the hydraulic circuits (number of boreholes connected in series, flow rate values, etc.). The consumed electric power provides a criterion which penalises large flow rates, just as laminar flow and the internal heat exchange in a borehole penalise low flow rates.

A pressure drop component (TYPE 64), developed especially for this study (see Annex A1), calculates the pressure drop and head loss caused by the fluid circulation in a ground heat exchanger. The heat rate dissipated by the fluid circulation and the electric power consumed by the pump are also assessed. The collecting and connecting pipes on ground surface may be included in the calculations.

*6.1.5 Duct store controller*

A duct store controller component (TYPE 61) is developed to control the operation of the duct store. Its main purpose is to combine the control signals of the ON/OFF controllers of the duct pumps, with a criterion based on the electric power consumed. If the electric power, multiplied with a user-given weight factor, is larger than the thermal heat rate transferred through the ground heat exchanger, the pump is switched off (see Annex A2). This component may also be used to implement a more complex strategy of the duct store operation.

*6.1.6 Variable flow rate component*

The component simulates a pump and a two-way valve controlled by a temperature, according to the system designs shown in chapter 3. This component is used together with a heat exchanger, as it controls the flow rate of the hot side, so that the outlet fluid temperature of the cold side is adjusted to a desired value. This component ensures the lowest possible flow rate in the hot side.

Such a component has been developed by Jaboyedoff (1993). It can be used with any type of heat exchanger (counter-flow, parallel-flow, etc.). On the other hand, the component may need too many iterations at a TRNSYS simulation time-step to converge, and may result in a significant increase of the execution time. In some cases, the convergence criterion can not be met.

A more efficient variable flow rate has been developed (see Annex A3). This component can only be used with a counter-flow heat exchanger, as the mathematical model used to calculate the heat transfer in the heat exchanger must correspond to that assumed in the variable flow rate. The flow rate in the hot site is calculated to its exact value rather than being estimated by successive iterations.

*6.1.7 Load component*

The load component (TYPE 59) is based on a simple load model, intended to generate hourly values for the heat demand of a building (Pahud, 1995). The model is used in its simplest form. The passive solar gains are not calculated, but implicitly taken into account with an outdoor temperature limit above which the heating is stopped (Degree-Day approach).

A representative daily profile is used for the hot water heat demand. The forward and return fluid temperatures of the heat distribution network are prescribed and depend on the outdoor temperature. The load component is described in Annex A4.

*6.1.8 Miscellaneous Components*

Some TRNSYS components were modified or corrected, so that they are adapted and "bug-free" for the kind of thermal processes simulated. They are the standard auxiliary heater component (TYPE 6), the temperature controlled flow diverter component (TYPE 11), the heat exchanger component (TYPE 5) and the algebraic operations component (TYPE 15). The modifications of the first three components were performed by Dalenbäck (1993), in the framework of a detailed study of a solar heating plant with a water storage. The modifications are reported in Annex A5.

## 6.2 The TRNSYS Decks

The system layouts are shown according to PRESIM (1991), a user-friendly pre-processor, devised to develop representations of real systems by assembling components in an interactive graphical mode. PRESIM produces a TRNSYS deck that can be computed by TRNSYS. The components designed to produce the output data are not shown. The system design with a buffer tank is the most extensively used. The corresponding TRNSYS deck and the quantities calculated with the output components are described in the next section.

*6.2.1 Deck of the system with the buffer tank*

In Fig. 6.1, the TRNSYS deck of the system design with a short-term buffer tank is shown. For better clarity in the layout of the system, the buffer tank is represented with the interface of the XST component, although the simulations were performed with the tank component TYPE 74.

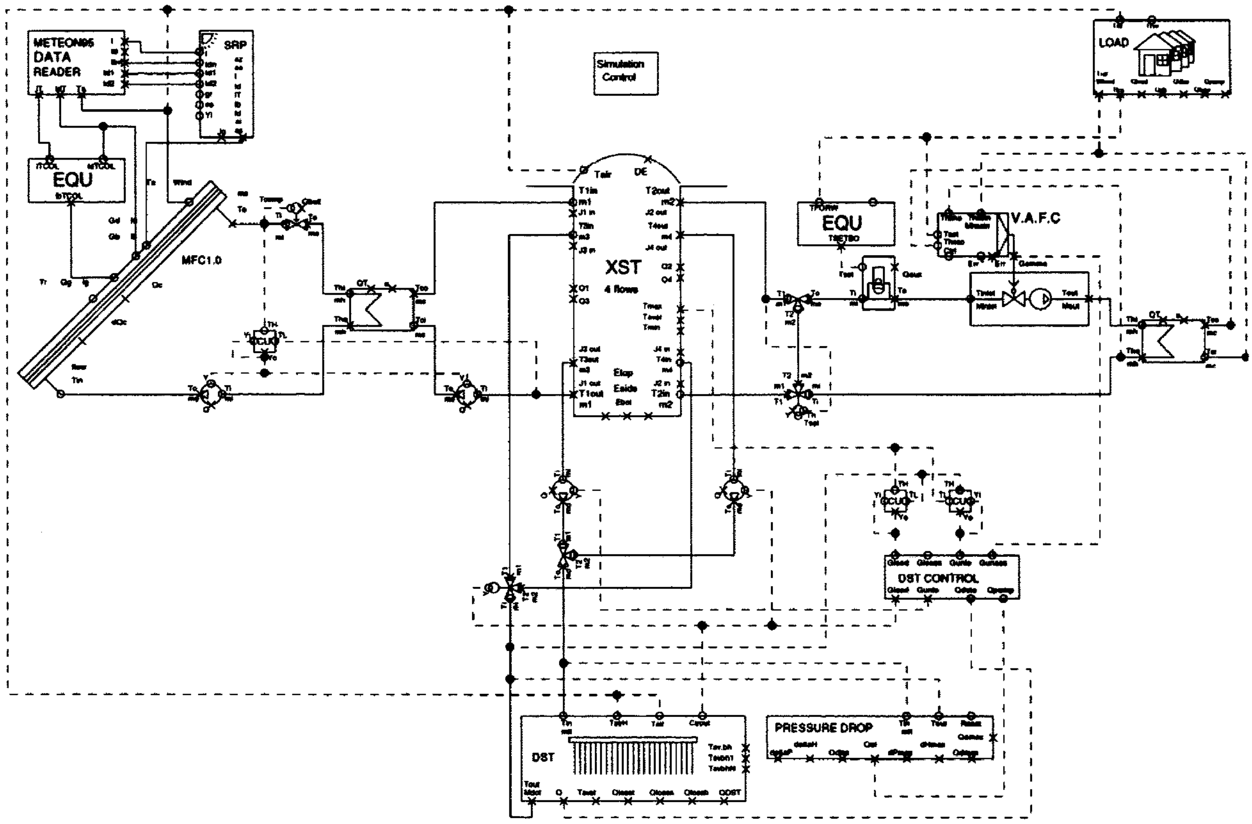



Fig. 6.1. PRESIM representation of the system design with a short-term buffer tank.

The heat capacity and losses of the main pipes of the collector array are included in the collector parameters. In consequence, the collected energy corresponds to the useful energy available at the solar heat exchanger. For typical pipe characteristics, the annual collected energy simulated in this way is close to the value obtained by simulating the collectors and the forward and return pipes to the solar heat exchanger (see Annex A.6). An advantage of including the pipes' thermal characteristics in the collector parameters is to make them dependant on the collector area. The simulation of the control of the collector array is also simpler, as the two flow loops on both sides of the solar heat exchanger can be controlled together (one ON/OFF controller instead of two).

A pressure relief valve limits the inlet temperature of the hot side of the heat exchanger to 100 °C. Overheating of the collector array will result in wasted heat, dissipated in the environment. It should be noted that the collector array is not stopped if overheating is taking place. In a real system, overheating may paralyse the collector operation, if vaporisation of the heat carrier fluid occurs.

The collector array, the duct store and the load subsystem are operated according to the *reference strategy* defined in section 3.2. Three ON/OFF controllers switch on or off the collector and duct pumps. The control function is based on the difference of two temperatures which are compared with two prescribed temperature differences, or dead-band temperature differences; one to enable and the other disable the control signal. The dead-band temperature differences are set so that a hysteresis effect is provided.

The solar controller controls the two solar pumps of the two flow-circuits on both sides of the solar heat exchanger. The upper and lower temperatures of the solar controller are chosen as the outlet fluid temperature from the collector array and the outlet fluid temperature from the bottom of the buffer tank.

When the duct store is loaded, water is taken at the top of the buffer, pushed through the ground heat exchanger and re-injected at the bottom of the buffer. The loading controller controls the duct loading mode. According to the reference strategy, the flow rate is set to half of the nominal flow rate through the collector array. The upper and lower temperatures of the loading controller are chosen as the maximum fluid temperature in the buffer tank and the outlet fluid temperature from the duct store. When the duct store is unloaded, the fluid circulates in the opposite direction. Water is taken out at the bottom of the buffer, pushed through the ground heat exchanger and re-injected at the top of the buffer. The unloading controller controls the duct unloading mode. According to the reference strategy, the flow rate is set to the actual flow rate in the flow loop of the load subsystem. This is realised by combining the control signal of the unloading controller with the variable flow rate signal (product of the two signals). The upper and lower temperatures of the unloading controller are chosen as the outlet fluid temperature from the duct store and the maximum fluid temperature in the buffer tank. The flow rate through the duct store is stopped if the heat rate transferred through the ground heat exchanger is smaller than the electric power consumed by the pump.

In the load subsystem, the temperature controlled flow diverter disconnects the buffer tank from the load subsystem, if the forward temperature (before the boiler) is lower than the return temperature from the load heat exchanger. Mixing never occurs, as the desired fluid temperature (before the boiler) is voluntarily set to an unattainable value. The boiler is only used to keep the fluid temperature above a minimum value, set typically to 5 K above the prescribed forward fluid temperature in the distribution network. The auxiliary heat provided by the boiler is the net auxiliary heat required to meet the heat demand, and does not include the overall efficiency of the process. The variable flow rate component adjusts the flow rate so that the forward fluid temperature in the distribution network matches the prescribed value. The flow rate can not exceed a maximum value, set typically to the maximum flow rate in the distribution network. If the overall UA-value of the heat exchanger is too small in relation to the heat rate to be transfered, or if the inlet temperature in the hot side is set too close to the required temperature in the cold side, the criterion can not be met. In consequence, the flow rate in the hot side will be set to its maximum value, the forward fluid temperature will be lower than the prescribed value and the heat load will not be fully met. It is also possible to set the parameters so that the system behaves as if there were no heat exchanger. (For example, the inlet temperature in the hot side is set 0.01 K higher than the required temperature in the cold side and the UA-value of the heat exchanger is set to an "infinite" value. The maximum flow rate in the hot side has to be at least as large as the maximum flow rate in the distribution network).

In order to control the heat balance of the subsystems and to provide a detailed view of the system's thermal performances, many output quantities are produced. Heat quantities, integrated during one month or one year, of each heat flux through the system are calculated. They are used to check the heat balance of each subsystem. Mean temperature levels of the main heat fluxes are also computed. Maximum and minimum fluid temperatures at some locations (collectors, buffer store, duct store, etc.) are determined and printed at regular intervals. The temperature evolution at different levels in the buffer store and the duct store may also be printed. The calculation of the output quantities involves 5 simulation summary components, 2 printer components, some algebraic operator and quantity integrator components. See Annex A.7 for a description of the calculated output quantities.

*6.2.2 Decks of the system without buffer tank and without duct store*

In Fig. 6.2, the TRNSYS deck of the system design without a short-term buffer tank is shown. The TRNSYS deck of the system design without duct store is shown in Fig. 6.3. The descriptions given for the system design with a short-term buffer tank apply for these two systems as well.

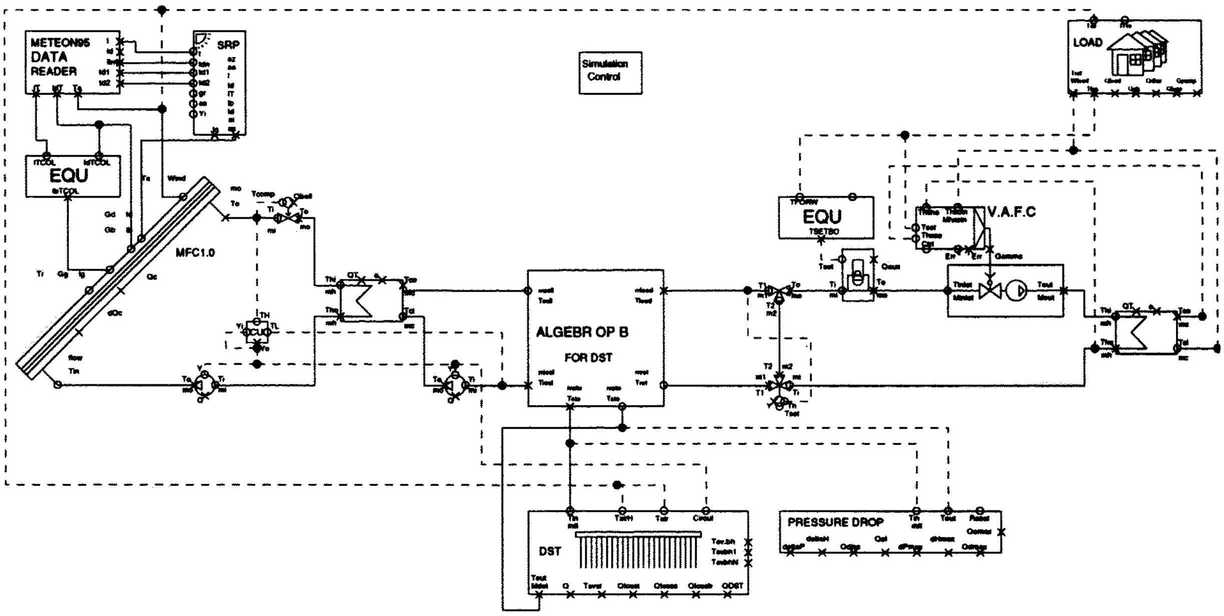



Fig. 6.2. PRESIM representation of the system design without a short-term buffer tank.

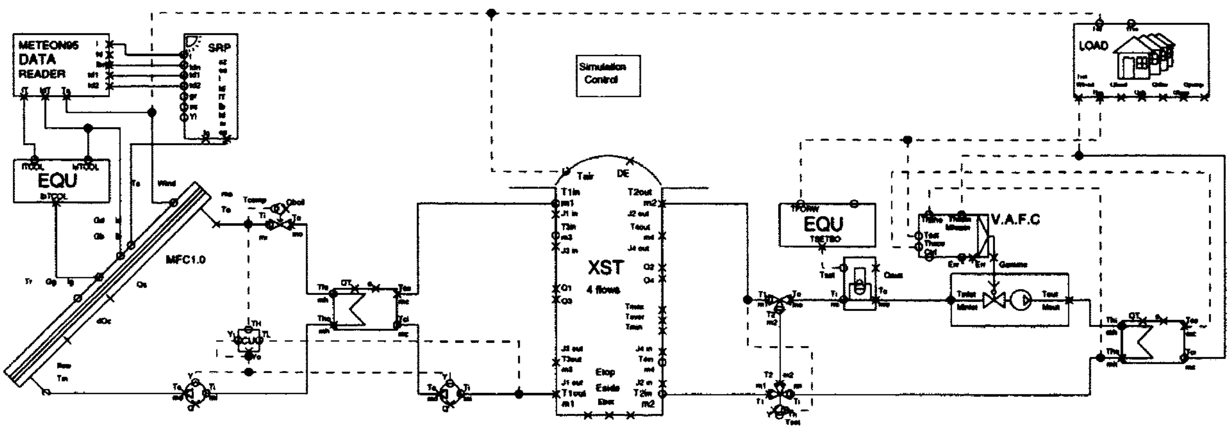



Fig. 6.3. PRESIM representation of the system design without a duct store.

See annex A.7 for a description of the output quantities.

# 7. SYSTEM THERMAL CHARACTERISTICS

The thermal characteristics of the subsystems are, as closely as possible, set to correspond to common values used in solar heating systems. Given the weather data and the heat load, five of the main parameters are varied: they are the **collector area**, the **buffer store volume**, the **duct store volume**, the **duct vertical extension** and the **number of boreholes**. All the other parameters are fixed or depend on the size of the different subsystems. The size-dependent parameters are expressed in terms of other parameters. For example, the overall UA-value of the solar heat exchanger depends on the collector area; it is given in W/K per unit collector area.

## 7.1 Collector Array Parameters

The parameters define the characteristics of the collectors, the main pipes, the pressure relief valve, the solar heat exchanger and the solar controller. The collector thermal characteristics correspond to commonly used flat plate collectors in Switzerland (Agena). They are summarised in Table 7.1.

| COLLECTOR ARRAY PARAMETERS: | |
|---|---|
| Location: GENEVA | latitude: **46.2°** North |
| | longitude: **6.1°** East |
| Orientation: | azimuth: **0°** (South facing) |
| | tilt angle (with respect to horizontal plane): **45°** |
| Area: | **COAREA** $m^2$ |
| Average transmittance-absorptance product: | |
| | **0.81** (-) |
| Overall loss coefficient: | **4.0** $W/m^2K$ + **0.3** $W/m^2K$ (pipes) |
| | **0.006** $W/m^2K^2$ |
| Collector heat capacity: | **15** $kJ/m^2K$ + **10** $kJ/m^2K$ (pipes) |
| Incidence angle modifier: | **0.11** (parameter bo in $1 - bo\,(1/\cos\theta - 1)$) |
| Specific flow rate: | **0.007** kg/sec /$m^2$ of collector area |
| Heat carrier fluid: | density: **1'050** $kg/m^3$ |
| | specific heat: **3.8** kJ/kgK |
| Pressure relief valve: | limit the inlet fluid temperature in the heat exchanger to **100** °C |
| Solar controller: | dead-band temperature differences: **2 - 14** K |
| Solar heat exchanger: | UA-value: **100** W/K /$m^2$ of collector area |
| | counter-flow heat exchanger |
| Specific flow rate, cold side: | **0.007** kg/sec /$m^2$ of collector area |
| Heat carrier fluid, cold side: | density: **1'000** $kg/m^3$ (water) |
| | specific heat: **4.19** kJ/kgK (water) |

Table 7.1 Parameters defining the collector array subsystem.

## 7.2 Short-Term Water Buffer Tank

The parameters define the characteristics of the short-term buffer tank and the two ON/OFF controllers for the operation of the duct store. They are summarised in Table 7.2.

| SHORT-TERM WATER BUFFER TANK: | |
|---|---|
| Volume: | **BUFFV** $m^3$ (cylindrical) |
| Vertical extension: | set equal to the buffer's diameter |
| | number of nodes when simulated: **3** |
| Storage medium: | water |
| Initial store temperature: | **10** °C |
| Connecting pipes: | top and bottom of the buffer |
| Insulation: | thermal conductivity: **0.05** W/mK |
| | thickness: **0.2** m |
| | location: uniformly placed on buffer envelope |
| Loading controller (duct store): | |
| | dead-band temperature differences: **1 - 5** K |
| Loading flow rate: | half of the nominal flow rate in the collector array |
| Unloading controller (duct store): | |
| | dead-band temperature differences: **1 - 5** K |
| Unloading flow rate: | equal to the flow rate in the load subsystem |

Table 7.2 Parameters defining the short-term buffer tank.

## 7.3 Duct Store Subsystem

The parameters define the characteristics of the duct store, the ground properties and the pipe dimensions for the calculation of the pressure drop. They are summarised in Table 7.3.

| DUCT HEAT STORAGE | | |
|---|---|---|
| Volume: | **DUCTV** $m^3$ | |
| Vertical extension: | **DUCTH** m | |
| Number of boreholes: | **NBORE** | |
| Distance between ground surface and upper side: **1** m | | |
| Insulation: | thermal conductivity: **0.05** W/mK | |
| | thickness: **0.2** m | |
| | location: | on top (the horizontal extension from the edge of the store equals **5%** of its vertical extension) |
| Storage medium: | ground: | thermal conductivity: **2.5** W/mK |
| | | volumetric heat capacity: **2.3** MJ/$m^3$K |
| Initial store and ground temperature: | | **10** °C |

Table 7.3a Parameters defining the duct store subsystem.

| DUCT HEAT STORAGE (continued) | |
|---|---|
| Ground heat exchanger: | heat carrier fluid: water<br>boreholes connected in series of **3**<br>borehole diameter: **0.115** m<br>borehole thermal resistance (Rb): **0.100** K/(W/m)<br>internal thermal resistance (Ra): **0.396** K/(W/m) |
| Pressure drop parameters: | double-U pipe installation<br>pipe internal diameter: **0.026** m<br>effective roughness: **1.5** $\mathbf{10^{-6}}$ (plastic, smooth pipe)<br>sum of losses values due to bends, fittings, etc. per borehole: **3**<br>pipes on ground surface: not taken into account<br>overall efficiency of the "pump + electric motor": **0.4** |

Table 7.3b Parameters defining the duct store subsystem (continued).

## 7.4 Load Subsystem

The boiler can provide the auxiliary heat with the required heat rate to meet the requisite temperature level. In a real system, a small water store may be included in the "boiler" part, in order to satisfy the short-term peak power loads. Another possibility is to allocate the top part of the buffer tank to the boiler.

The different investigated load types are described in chapter 4. The parameters of the load model component are adjusted so that the annual load is reproduced with the desired proportion of space heating and domestic hot water energy, given the hourly weather data. A spread-sheet EXCEL is used, which calculates and adds the 8'760 load values. A TRNSYS deck, involving only the weather data and the load model, allows the user to simulate the load type and determine the maximum flow rate in the distribution network. The parameters shown in Table 7.4 define the characteristics of the variable flow rate component and the load heat exchanger. With a low temperature distribution, the parameters are set so that a system without load heat exchanger is simulated.

| LOAD SUBSYSTEM | | |
|---|---|---|
| | Moderate temperature heat distribution: | Low temperature heat distribution: |
| Variable flow rate component: | | |
| maximum flow rate: | **FLOMAX** | **FLOMAX** |
| (FLOMAX: maximum flow rate in the heat distribution network) | | |
| Inlet fluid temperature, hot side: (boiler used if necessary) | | |
| difference with the prescribed forward fluid temperature in the distribution network: | +**5** K | +**0.01** K |
| Load heat exchanger: (counter-flow) | | |
| UA-value per annual MWh load | **90** W/K /MWh | $\mathbf{10^{12}}$ W/K /MWh |

Table 7.4 Parameters defining the load subsystem. With a low temperature heat distribution, the parameters correspond to a system without load heat exchanger.

# 8. SYSTEM SIMULATION RESULTS

Several thousand runs have been performed. Programmes were developed to automate the generation of TRNSYS input files, create batch files for the execution of the TRNSYS programme with different input files and read and process the output results contained in a set of output files. The cost functions are calculated when the output results are processed.

The calculated solar fraction corresponds to a mean value over 25 years, including the effect of a cold start; (cold storage and ground temperatures when the operation of the system is started). In order to save computation time, the first 12 operation years are simulated, and the following 13 are assumed to be well represented by the 12th operation year. The weather data, set for a typical year (see chapter 5), is repeated every year.

## 8.1 Solar Cost for the Different Load Types

The solar cost (see section 4.4) is shown in relation to the solar fraction. Five main parameters (collector area, buffer store volume, duct store volume, duct vertical extension and number of boreholes) are varied, so that the minimum solar cost can be found for a given solar fraction. All the other parameters are fixed as described in chapter 7. An **expansion path**, defined as the curve that gives the minimum solar cost for any solar fraction, can be drawn from the graphs. The systems that lie on the expansion path are established without any constraint on the fluid temperatures or the system parameters. For example, the fluid temperature in the collector array may rise above 100 °C. Nevertheless, heat would be dissipated in the environment in this case, as a pressure relief valve limits the inlet fluid temperature in the solar heat exchanger to a maximum of 100 °C. The five main parameters are varied in the following ranges:

- collector area: 2, 2.5, 3, 3.5, 4 $m^2$ per MWh of total annual heat load;
- duct store volume: 2, 4, 6, 8, (10, 12) $m^3$ per $m^2$ of collector area;
- number of boreholes: adjusted so that the total borehole length is equal to: 0.5, 1, 1.5, 2 m per $m^2$ of collector area.
- buffer store volume: 80, 120, 160 litres per $m^2$ of collector area;
- duct vertical extension: 2, 2.5, 3 m per m of duct storage diameter (see section 4.3 for the definition of the ratio height over diameter);

If necessary, the duct volume is increased to 10 and 12 $m^3$ per $m^2$ of collector area, but only for systems whose total borehole length is equal to 1.5 and 2 m per $m^2$ of collector area. From Fig. 8.1 to 8.7, the solar cost is shown for the seven different load types defined in section 5.2; (refer to this section for the characteristics of the heat loads and the temperature levels of the heat distributions). The expansion path can be drawn in the graphs *a)* of Fig. 8.1 - 8.7, as being the curve that would follow the lower lines at a small distance below the cluster of points. In the graphs *b)* to *e)*, sensitivity to the varied parameter is indicated by the lines, as they connect the solar cost of systems whose remaining parameters are identical. When the total borehole length is varied, the duct store volume is kept constant and vice-versa. As a result, an increase of the borehole length is equivalent to a decrease of the borehole spacing. An increase of the duct store volume induces an increase of the borehole spacing. It also reduces the number of boreholes, as the ratio duct vertical extension over duct diameter is kept constant together with the total borehole length. This latter is actually slightly varied so that the number of boreholes is an integer value.

**Load: 500 MWh/y (75% sh, 25% dhw), moderate-temperature heat distribution**

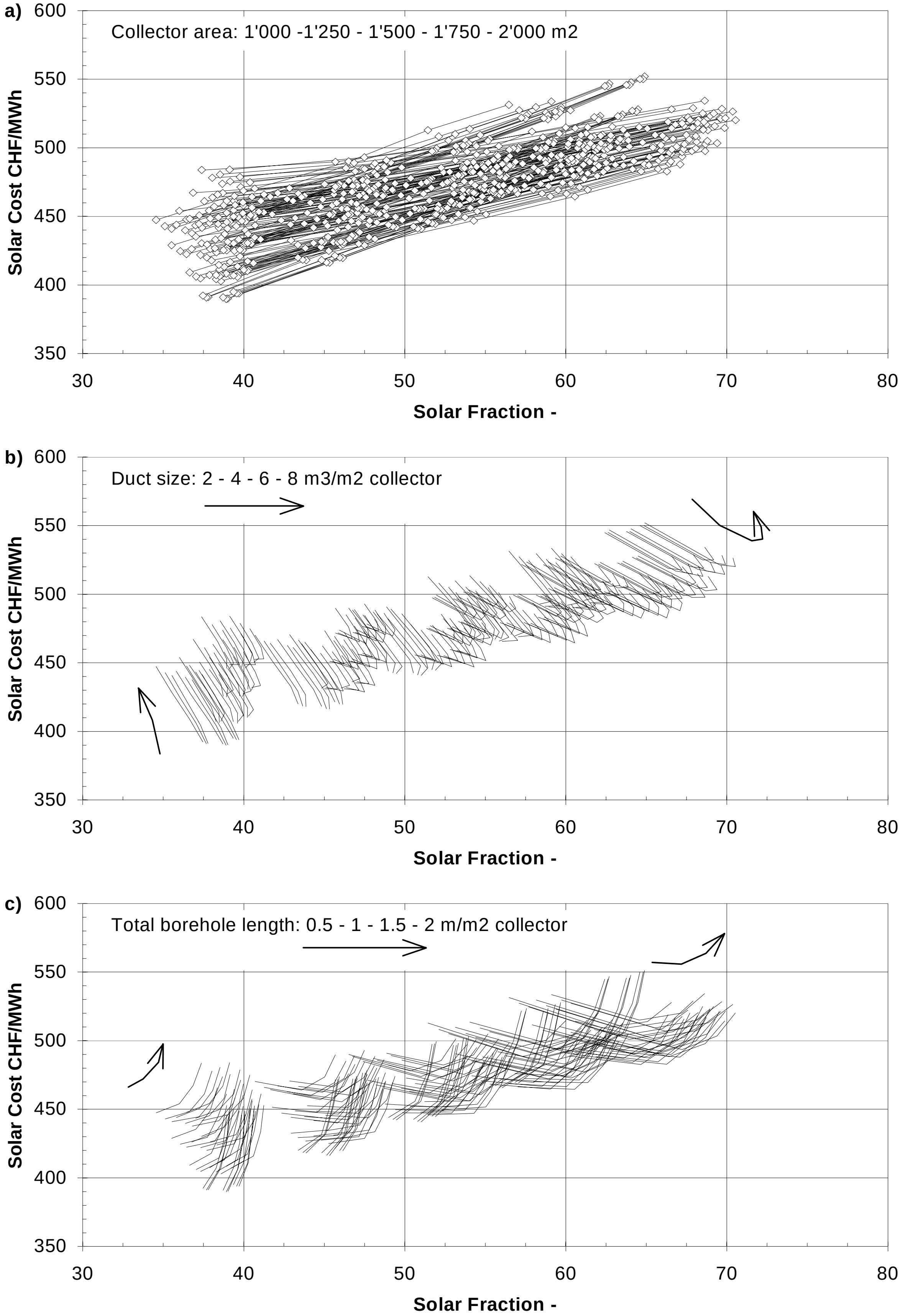



Fig. 8.1a-c: Small heat load, 500 MWh/y (75% space heating and 25% domestic hot water); moderate-temperature heat distribution.

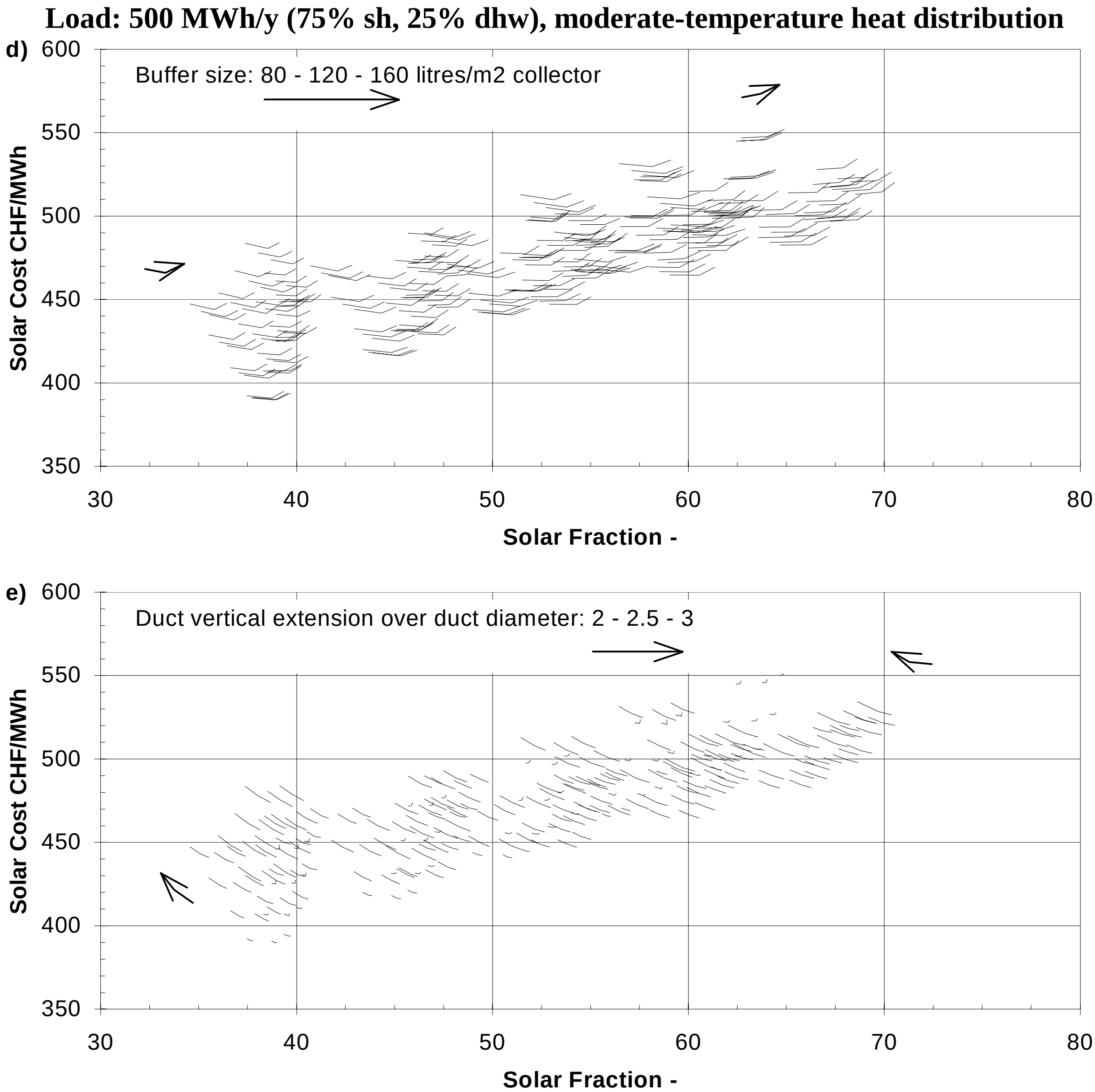



Fig. 8.1d-e: Small heat load, 500 MWh/y (75% space heating and 25% domestic hot water); moderate-temperature heat distribution.

The arrows in the graphs indicate the direction of the variation when the varied parameter is increased. The duct store volume remains very small, below 10'000 $m^3$ for solar fractions smaller than 70%. The duct storage efficiency, defined as the ratio of the recovered energy over the injected energy during a cycle (1 year), does not exceed 30%. With solar fractions lower than 50%, the optimum duct volume tends to be zero. The solar heat is mainly covered by the buffer tank, and the duct store is used only to cool down the buffer tank during the summer. With this system, a duct store might be considered only if a large solar fraction is desired, i. e., if seasonal storage of the solar heat is needed. Nevertheless, the large heat losses of the duct store have to be paid for with solar heat.

For solar fractions of 60 to 70%, the cheapest systems require 3.5 to 4 $m^2$ of collector per annual MWh load, a duct store volume of 4 to 6 $m^3$ per $m^2$ of collector, about 1 m of borehole per $m^2$ of collector and 120 litres per $m^2$ of collector. The ratio duct vertical extension over duct diameter, with the top of the store insulated, is below 2. The solar cost amounts to 460 - 490 CHF/MWh.

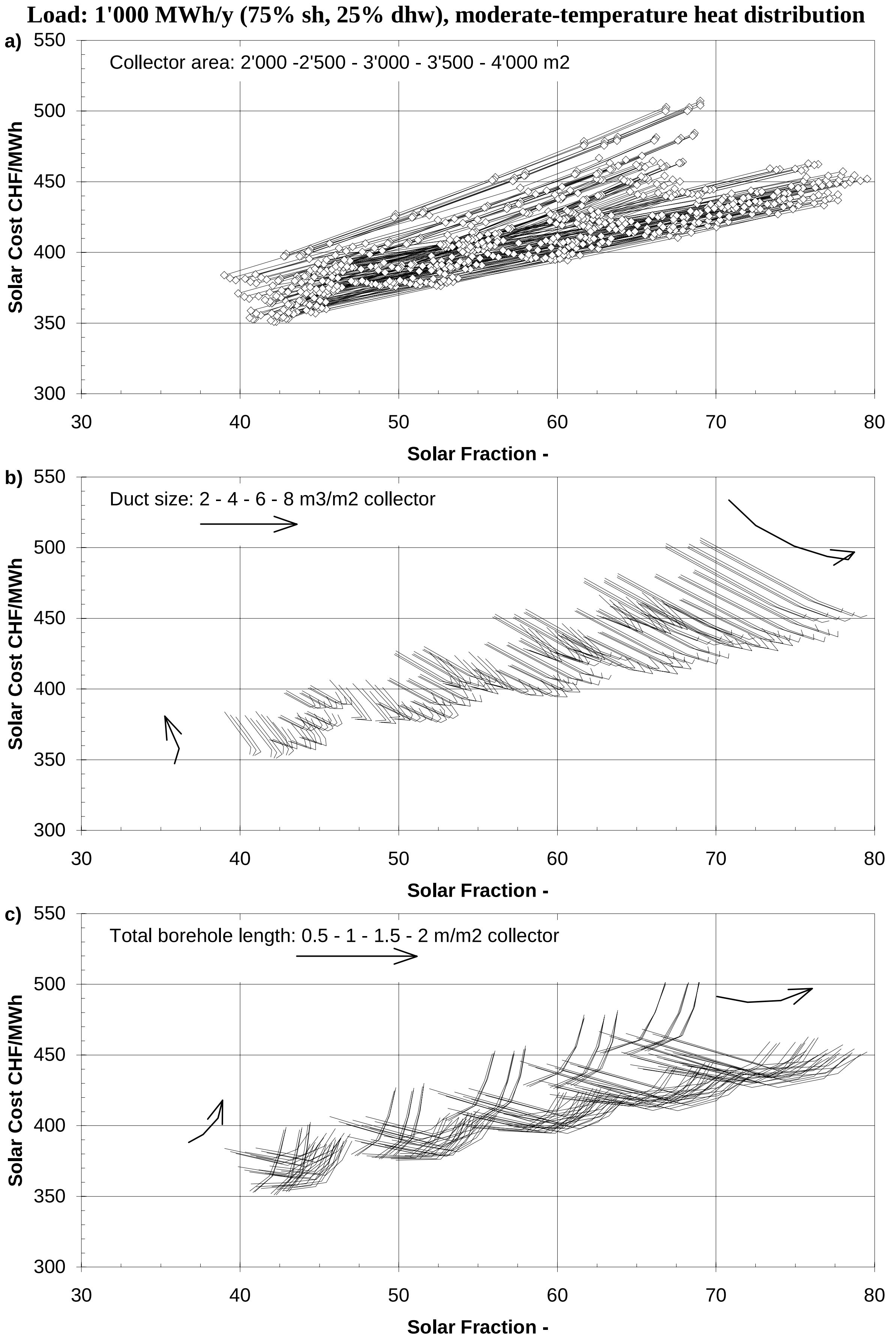



Fig. 8.2a-c: Medium heat load, 1'000 MWh/y (75% space heating and 25% domestic hot water); moderate-temperature heat distribution.

**Load: 1'000 MWh/y (75% sh, 25% dhw), moderate-temperature heat distribution**

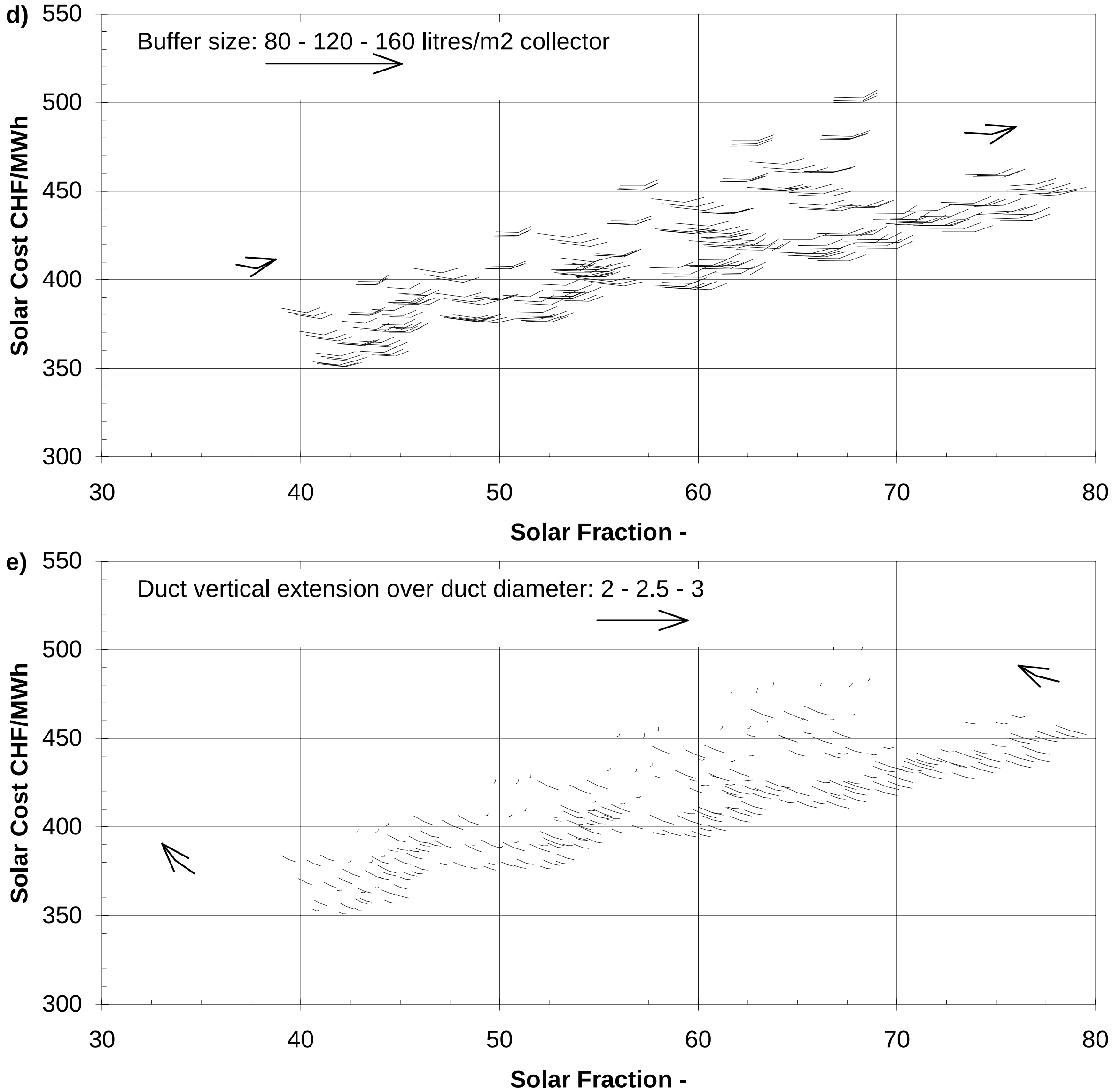



Fig. 8.2d-e: Medium heat load, 1'000 MWh/y (75% space heating and 25% domestic hot water); moderate-temperature heat distribution.

For solar fractions of 60 to 70%, the cheapest systems require 3 to 3.5 $m^2$ of collector per annual MWh load, a duct store volume of about 6 $m^3$ per $m^2$ of collector, 1 to 1.5 m of borehole per $m^2$ of collector and about 120 litres per $m^2$ of collector. The ratio duct vertical extension over duct diameter, with the top of the store insulated, remains below 2. The solar cost amounts to 390 - 420 CHF/MWh.

**Load: 5'000 MWh/y (75% sh, 25% dhw), moderate-temperature heat distribution**

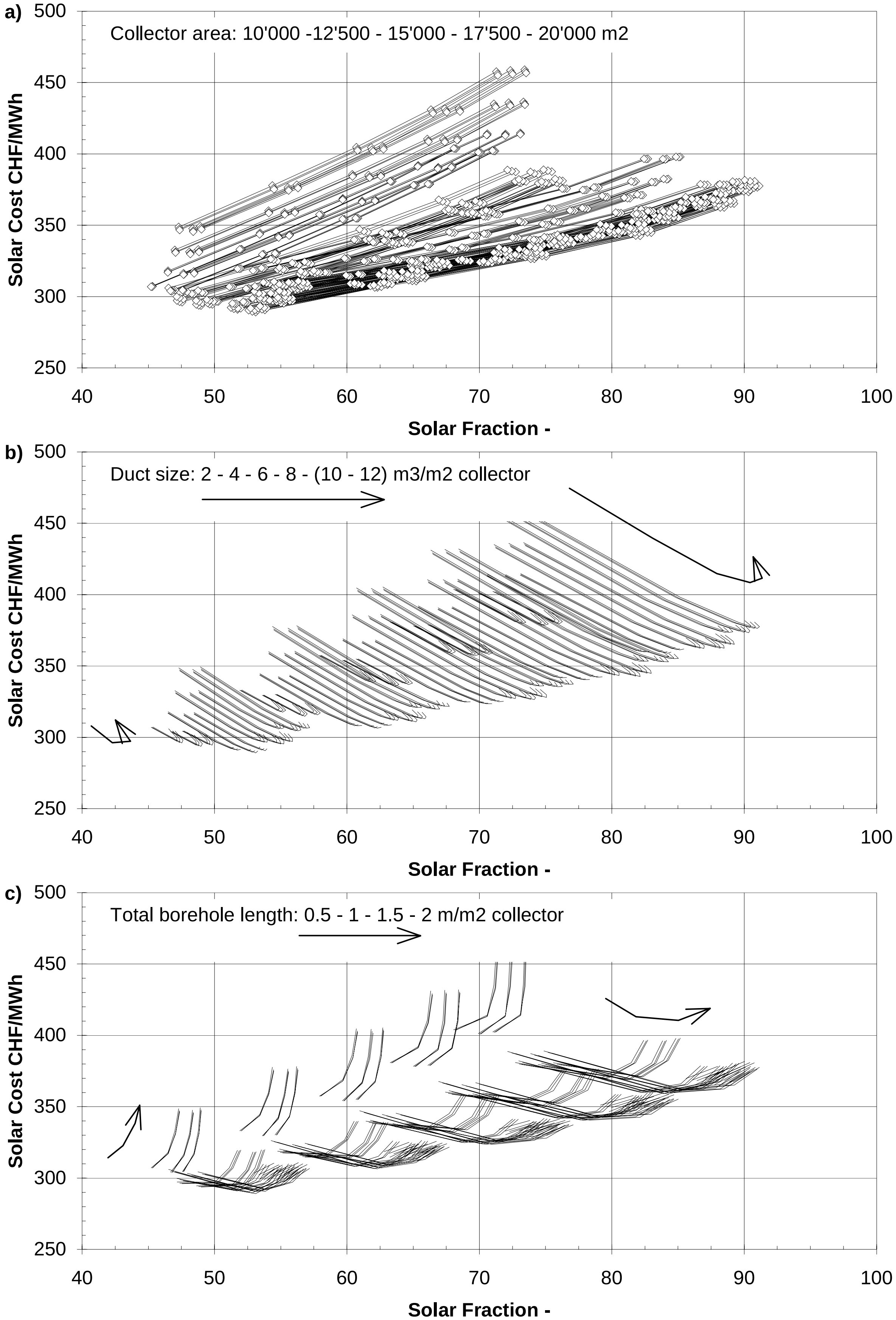



Fig. 8.3a-c: Large heat load, 5'000 MWh/y (75% space heating and 25% domestic hot water); moderate-temperature heat distribution.

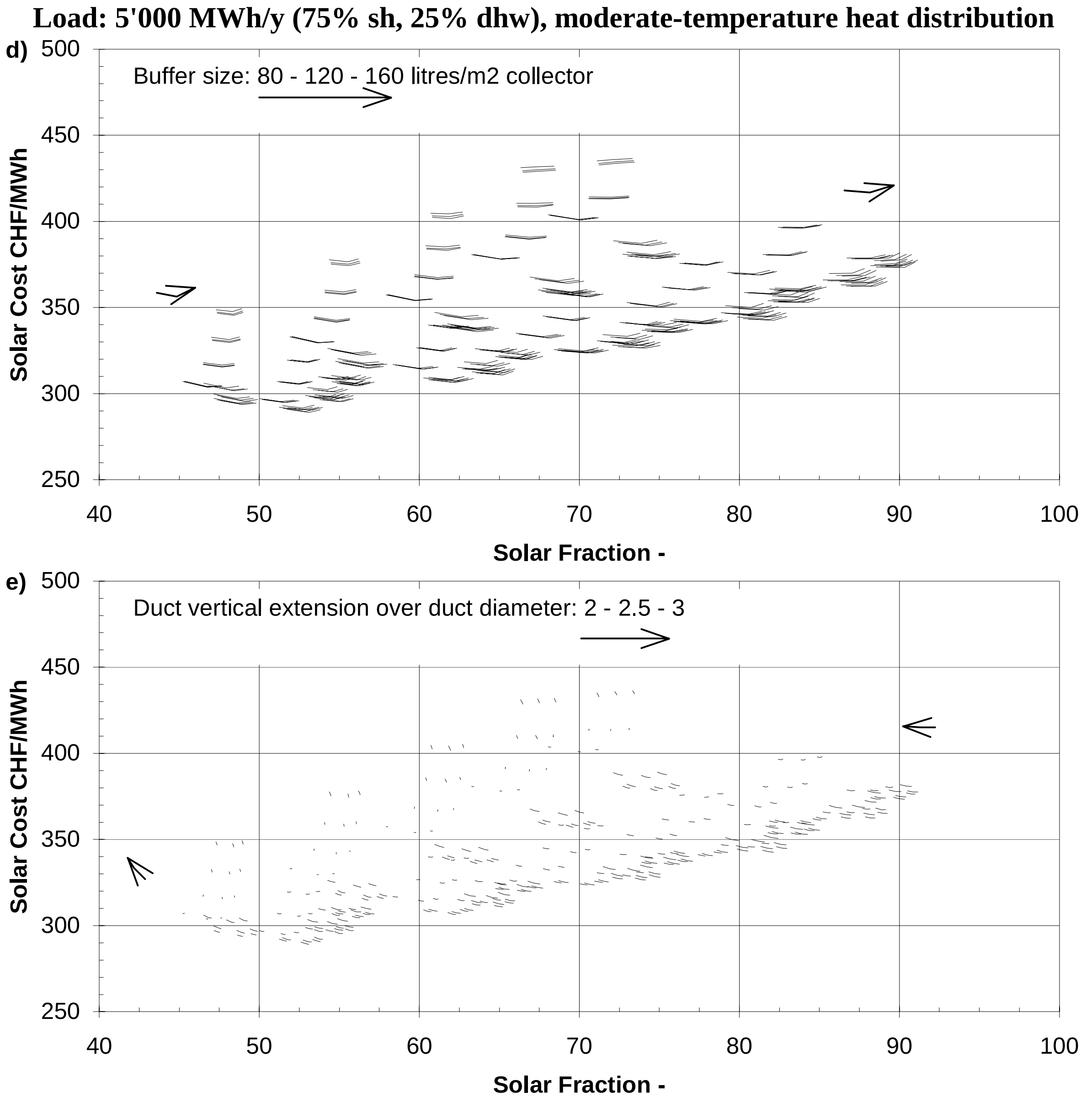



Fig. 8.3d-e: Large heat load, 5'000 MWh/y (75% space heating and 25% domestic hot water); moderate-temperature heat distribution.

With larger duct stores, the relative importance of the heat losses diminishes, and the storage volume can be enlarged to increase the storage capacity. Nevertheless, the temperature level of the heat to be recovered from the storage is important, as the temperature of a larger store will not rise as much as that of a smaller one, and thus penalises larger store. With this large system, the duct volume can be increased up to 8 $m^3$ per $m^2$ of collector. A too-large duct volume has a smaller influence on the solar cost than a too-small duct volume.

For a solar fraction of 60 - 70%, the cheapest systems require less than 2.5 - 3 $m^2$ of collector per annual MWh load, a duct store volume of about 8 $m^3$ per $m^2$ of collector, 1 to 1.5 m of borehole per $m^2$ of collector and about 120 litres per $m^2$ of collector. The ratio duct vertical extension over duct diameter, with the top of the store insulated, remains below 2. It has less influence on the solar cost than a smaller system. The solar cost amounts to 300 - 320 CHF/MWh. The increase of the solar cost with the solar fraction is smaller than the increase observed with the small and medium heat loads, and with a moderate-temperature heat distribution.

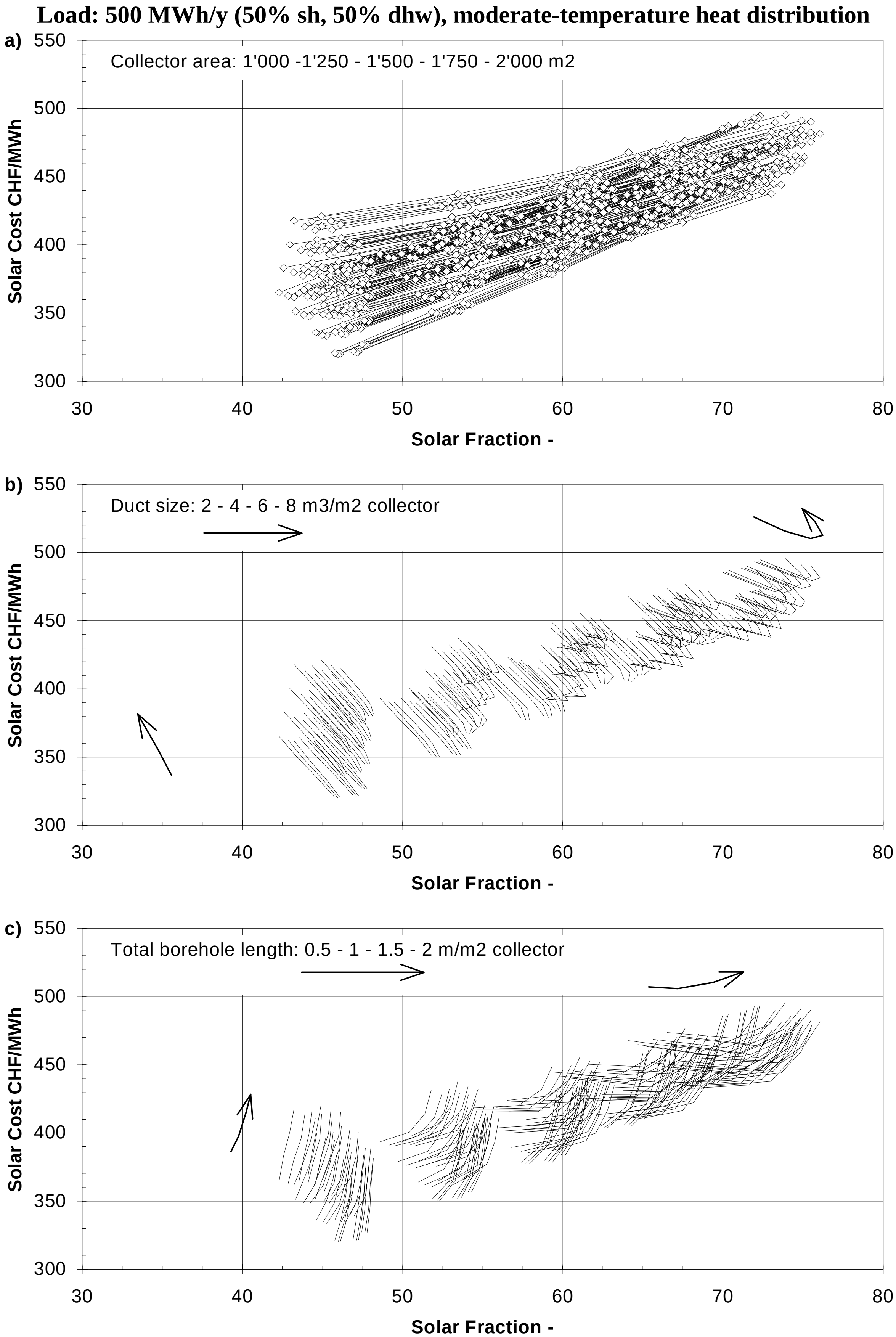



Fig. 8.4a-c: Small heat load, 500 MWh/y (50% space heating and 50% domestic hot water); moderate-temperature heat distribution.

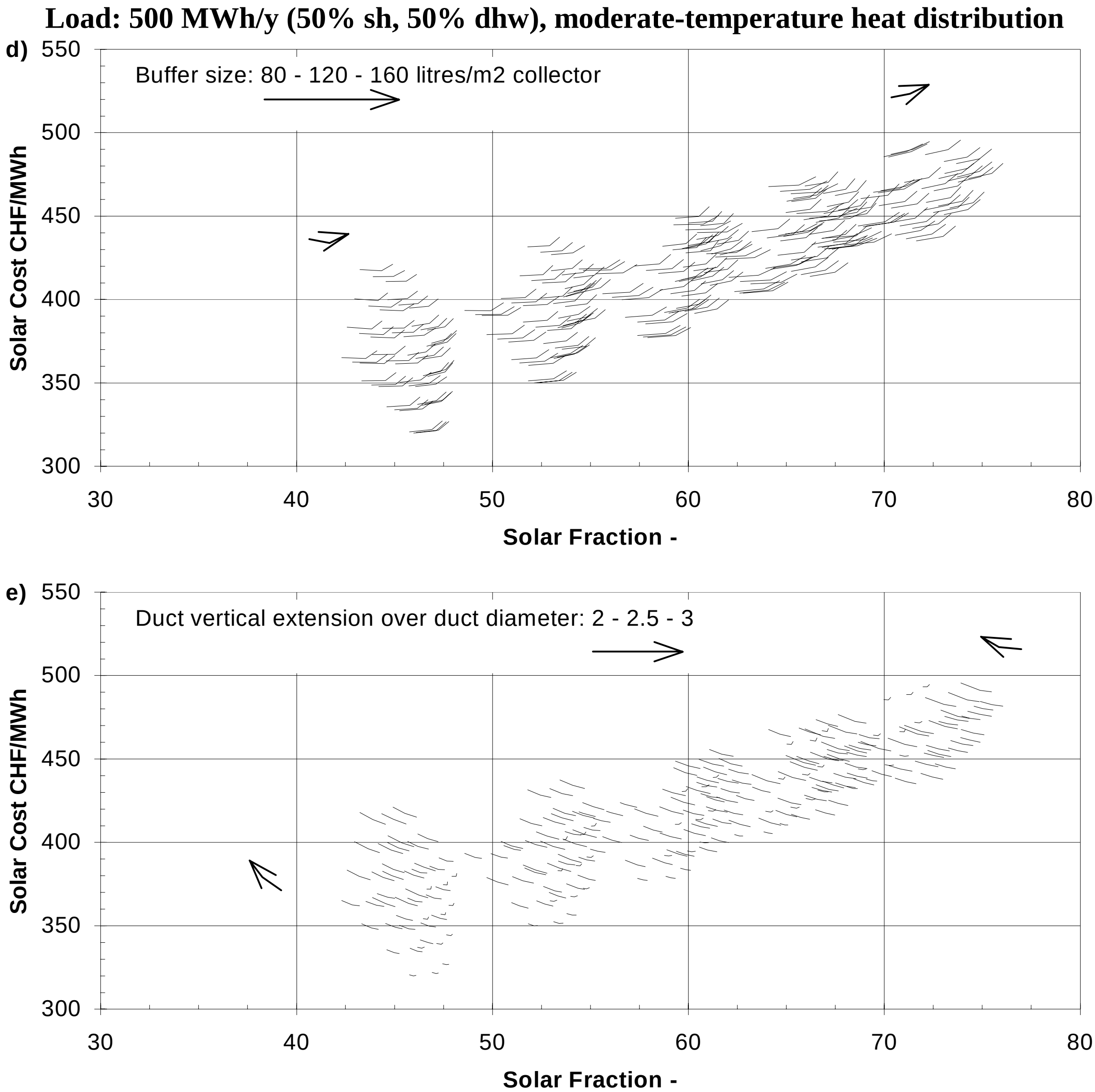



Fig. 8.4d-e: Small heat load, 500 MWh/y (50% space heating and 50% domestic hot water); moderate-temperature heat distribution.

With a larger proportion of domestic hot water demand (50% instead of 25% of the total annual heat demand), the short-term heat requirement is more important, and results in a more efficient use of the collected heat. For a solar fraction of 60 - 70%, the solar cost is 60 - 80 CHF/MWh lower than the system with only 25% of domestic hot water. Nevertheless, the use of a duct store might be considered only for solar fractions larger than 70%.

For a solar fraction of 70%, the cheapest systems require less than 4 $m^2$ of collector per annual MWh load, a duct store volume of about 4 $m^3$ per $m^2$ of collector, about 1 m of borehole per $m^2$ of collector and about 120 litres per $m^2$ of collector. The ratio duct vertical extension over duct diameter, with the top of the store insulated, is below 2. The solar cost amounts to 430 CHF/MWh. It should be noted that the optimum buffer size is determined relatively to the slope of the expansion path (shortest distance). In consequence, it can be assessed as the minimum solar cost only if the slope of the expansion path is zero, i. e., parallel to the "solar fraction" axis.

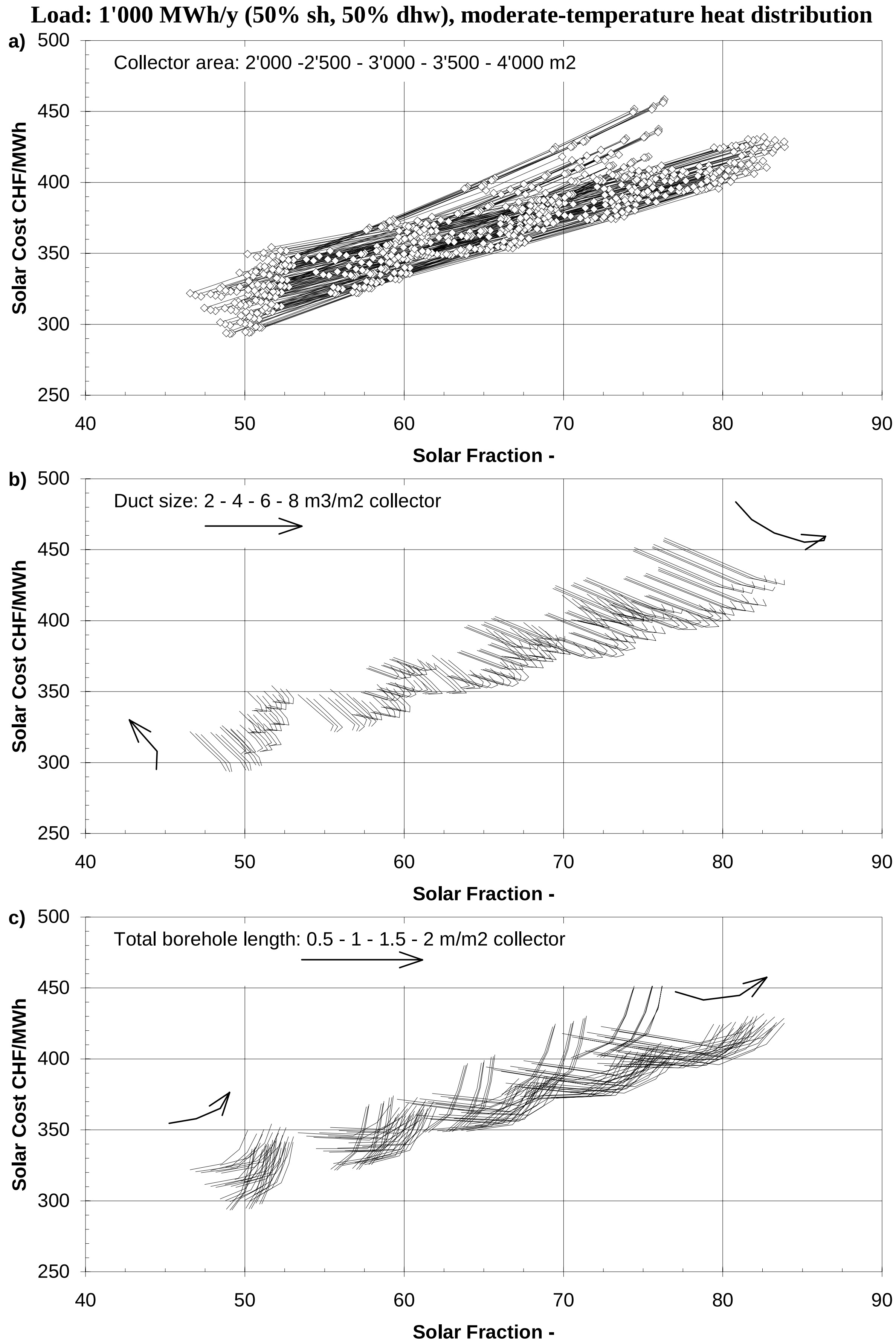
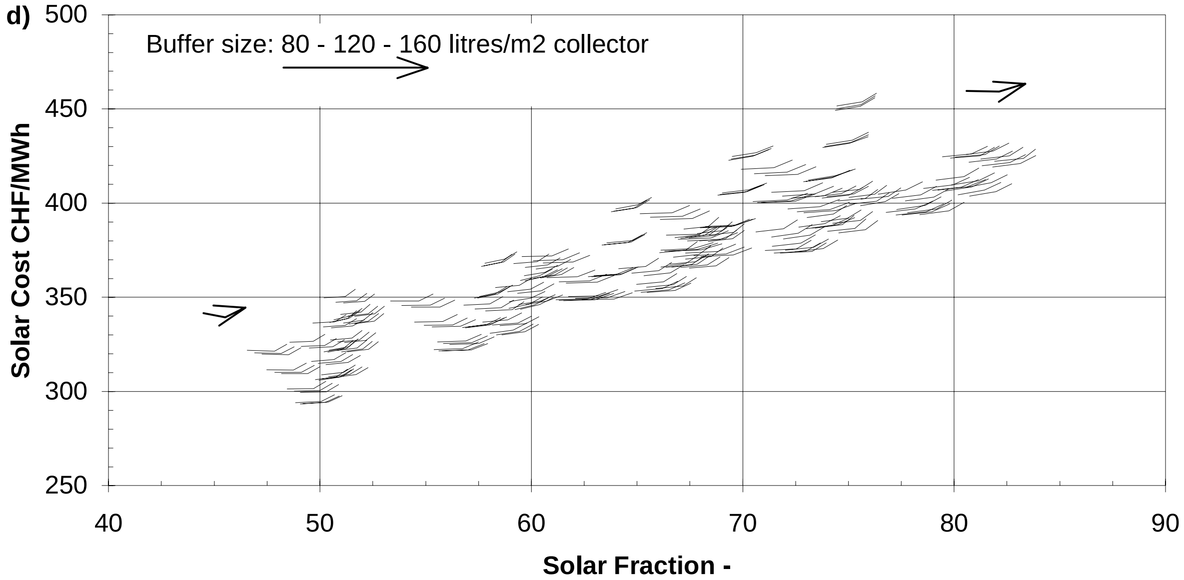



Fig. 8.5a-c: Medium heat load, 1'000 MWh/y (50% space heating and 50% domestic hot water); moderate-temperature heat distribution.

**Load: 1'000 MWh/y (50% sh, 50% dhw), moderate-temperature heat distribution**



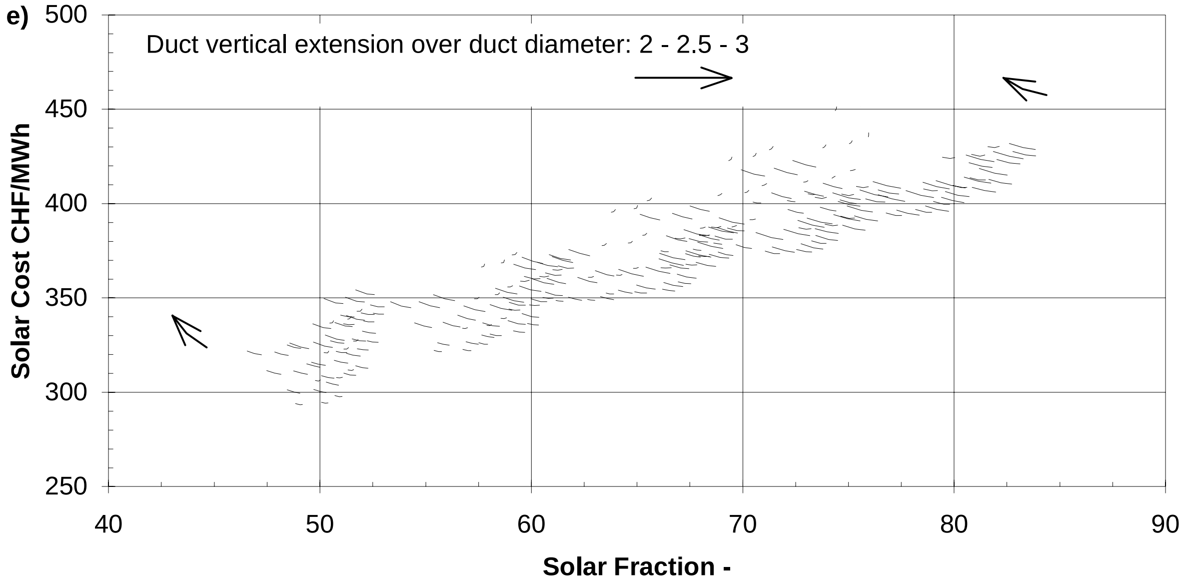




Fig. 8.5d-e: Medium heat load, 1'000 MWh/y (50% space heating and 50% domestic hot water); moderate-temperature heat distribution.

For solar fractions of 60 to 70%, the cheapest systems require 2.5 to 3 $m^2$ of collector per annual MWh load, a duct store volume of about 4 to 6 $m^3$ per $m^2$ of collector, about 1 m of borehole per $m^2$ of collector and 120 litres per $m^2$ of collector. The ratio duct vertical extension over duct diameter, with the top of the store insulated, is below 2. The solar cost amounts to 330 - 360 CHF/MWh.

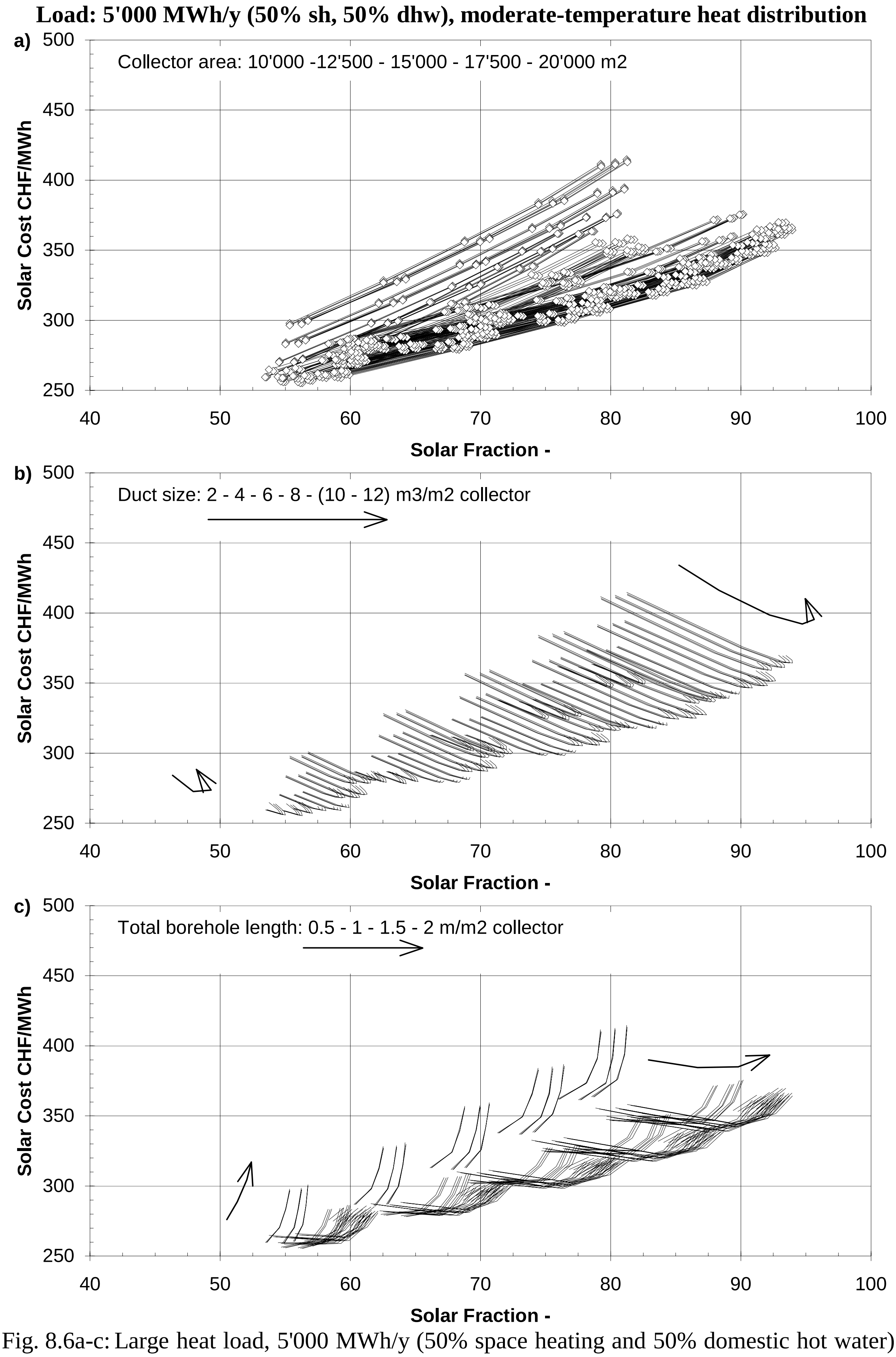



Fig. 8.6a-c: Large heat load, 5'000 MWh/y (50% space heating and 50% domestic hot water); moderate-temperature heat distribution.

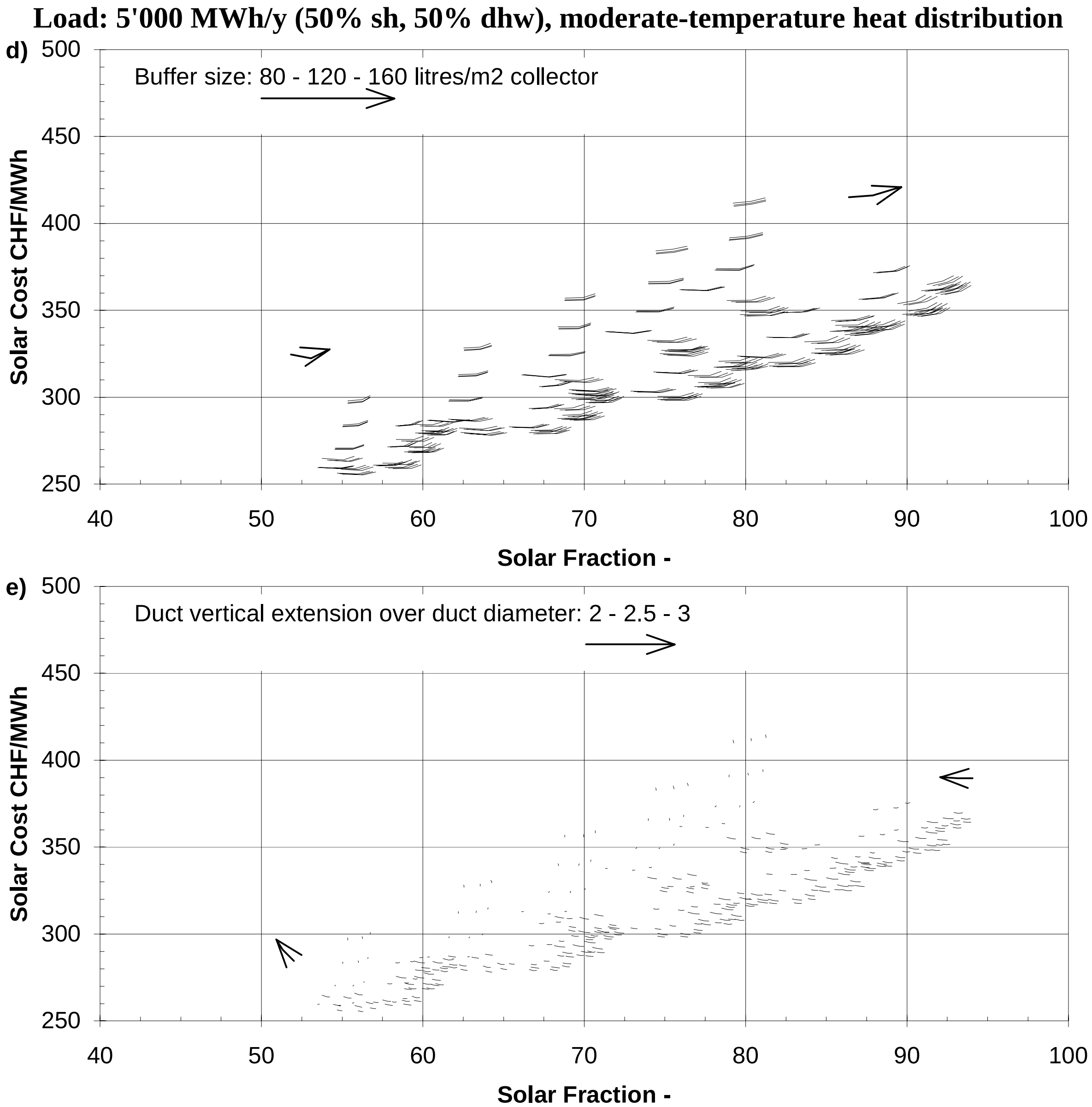



Fig. 8.6d-e: Large heat load, 5'000 MWh/y (50% space heating and 50% domestic hot water); moderate-temperature heat distribution.

For solar fractions of 60 to 70%, the cheapest systems require 2.0 to 2.5 $m^2$ of collector per annual MWh load, a duct store volume of 6 to 8 $m^3$ per $m^2$ of collector, 1 to 1.5 m of borehole per $m^2$ of collector and about 120 litres per $m^2$ of collector. The ratio duct vertical extension over duct diameter, with the top of the store insulated, is around 2. The solar cost amounts to 260 - 280 CHF/MWh.

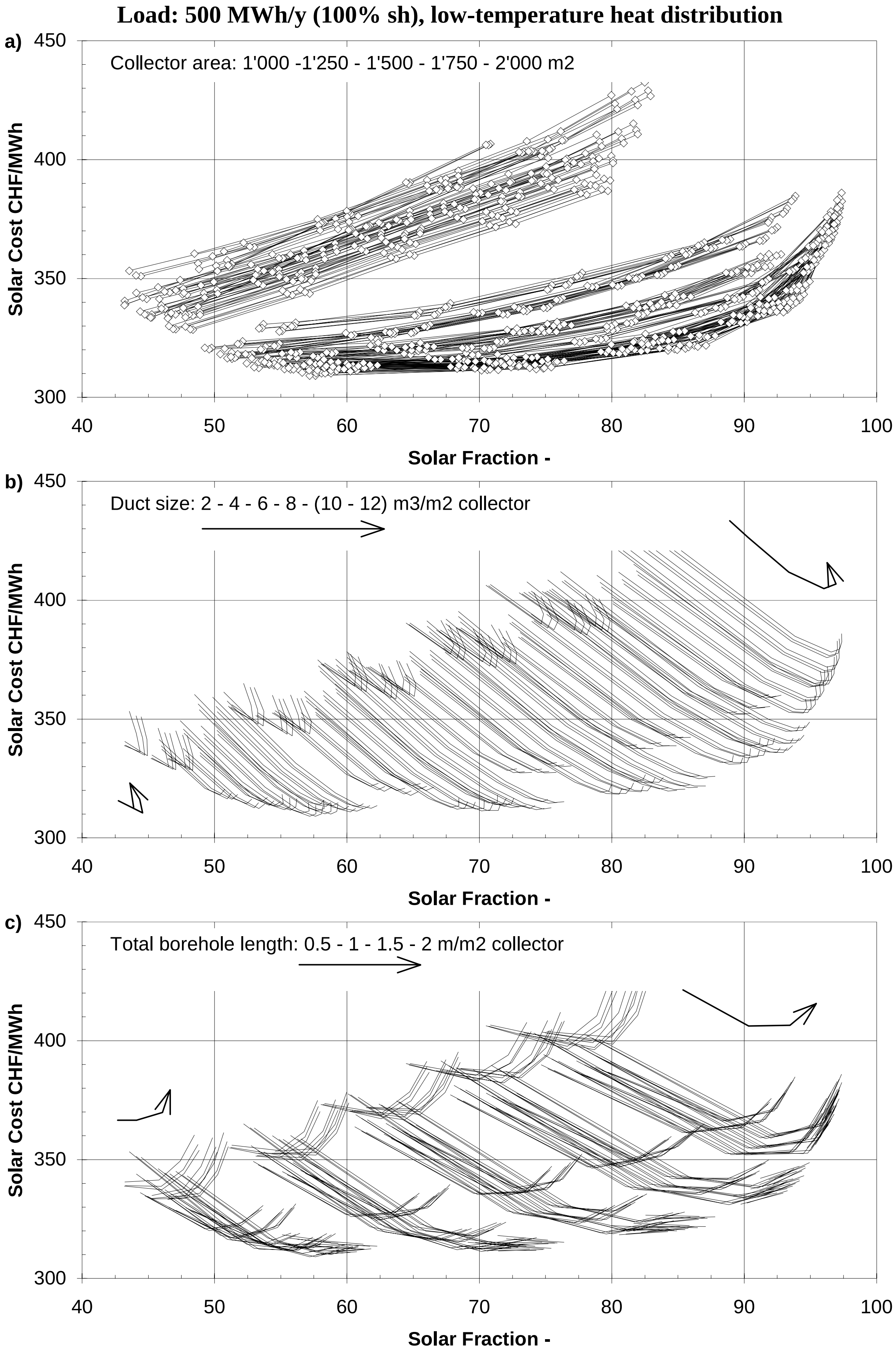



Fig. 8.7a-c: Small heat load, 500 MWh/y (100% space heating); low-temperature heat distribution.

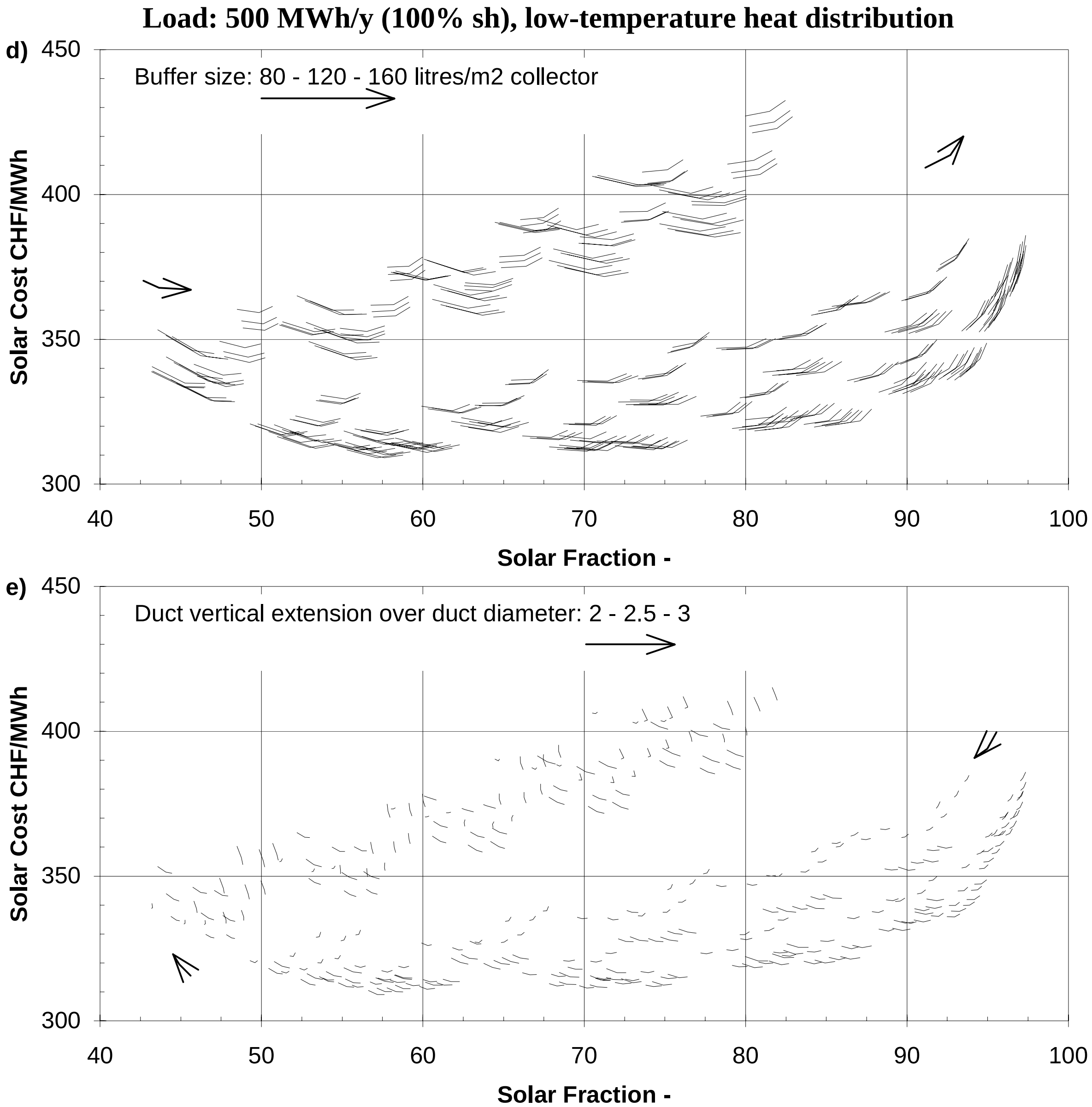



Fig. 8.7d-e: Small heat load, 500 MWh/y (100% space heating); low-temperature heat distribution.

This example shows the good potential of a duct store if heat can be recovered at a low temperature level. This is not only beneficial for the reduction of the heat losses, but also for the possibility of having a larger temperature difference between the heat carrier fluid and the mean storage temperature, which conditions the heat rate that can be recovered from the store. The graphs shows a net increase of the solar cost as the solar fraction approaches 100%. This increase would probably be larger if a real succession of 25 years of weather data had been used, as heat storage from one year to another is needed to account for the variations of the weather from year to year, especially is the overall solar fraction is close to 100%.

For a solar fractions of 60 to 70%, the cheapest systems require 2 to 2.5 m$^2$ of collector per annual MWh load, a duct store volume of 8 to 10 m$^3$ per m$^2$ of collector, 1.5 to 2 m of borehole per m$^2$ of collector and about 120 litres per m$^2$ of collector. The ratio duct vertical extension over duct diameter, with the top of the store insulated, is about 2.5. The solar cost is mostly constant and amounts to 310 CHF/MWh.

Among the five varied parameters, two have to be varied together: the duct store volume and the total length of borehole. The other three, the collector area, the buffer store volume (per square meter collector) and the ratio duct vertical extension over duct diameter, have a weak mutual influence; they can be varied independently.

The solar cost increases with an increase of the solar fraction, unless heat can be distributed at a very low temperature level. In this case, the solar cost is mostly constant up to a solar fraction of 80%. With higher temperature levels in the heat distribution, the increase is more important with a smaller heat load as well as with a larger proportion of domestic hot water. A larger proportion of this latter makes the solar cost lower, but also makes the optimum size of a duct store smaller, making a ground duct store attractive for larger solar fractions. The size of the heat load and the temperature levels of the fluid in the distribution network have a capital influence on the thermal performance of the solar heating system, and in consequence on the solar cost. A heat load of 500 MWh/year is a critical size if the temperature levels of the heat distribution correspond to those of the moderate-temperature heat distribution; (forward temperature: 50 -55 °C; return temperature: 35 - 40 °C). Relative to a larger heat load, a parameter deviation from the calculated optimum design has more influence on the solar cost. In other words, the risk of having unexpected thermal performances is greater with a smaller heat load. This is particularly important with the ground storage heat losses. If the annual heat losses are large in comparison to the annual recovered energy, as is usually the case with a small duct store, a small error in the assessment of the heat losses may cause a significant difference to the calculated recovered energy. This is a crucial point as heat losses are often larger than expected.

Comparison of the solar cost with a system without ground duct store is performed with the small load (500 MWh/year, 75% space heating and 25% domestic hot water), and the moderate-temperature heat distribution. The tank volume is varied from 100 to 2'500 litres/m$^2$ of collector, by steps of 100 litres/m$^2$. In order to take into account more accurately the effect of a vertical stratification of the fluid temperatures with large volumes, the number of nodes in the tank model is increased from 3 to 15. In Fig. 8.8, the solar cost is shown for the system with and without ground duct store. The lines connect systems with identical parameters, apart from the collector area which is varied from 1'000 m$^2$ to 2'000 m$^2$, and the parameters that depend on it. This example shows that a ground duct store is more attractive than a tank alone for large solar fractions (about 60%). In Fig. 8.1b, it corresponds approximately to the solar fraction when the volume of the duct store has an optimum value which is larger than 2 m$^3$ per square meter collector. For lower solar fractions, the main function of the duct store is to cool down the buffer tank during the summer. In this situation, a system without duct store is cheaper, but the fluid temperature in the collector array reaches higher values, and may cause overheating problems.

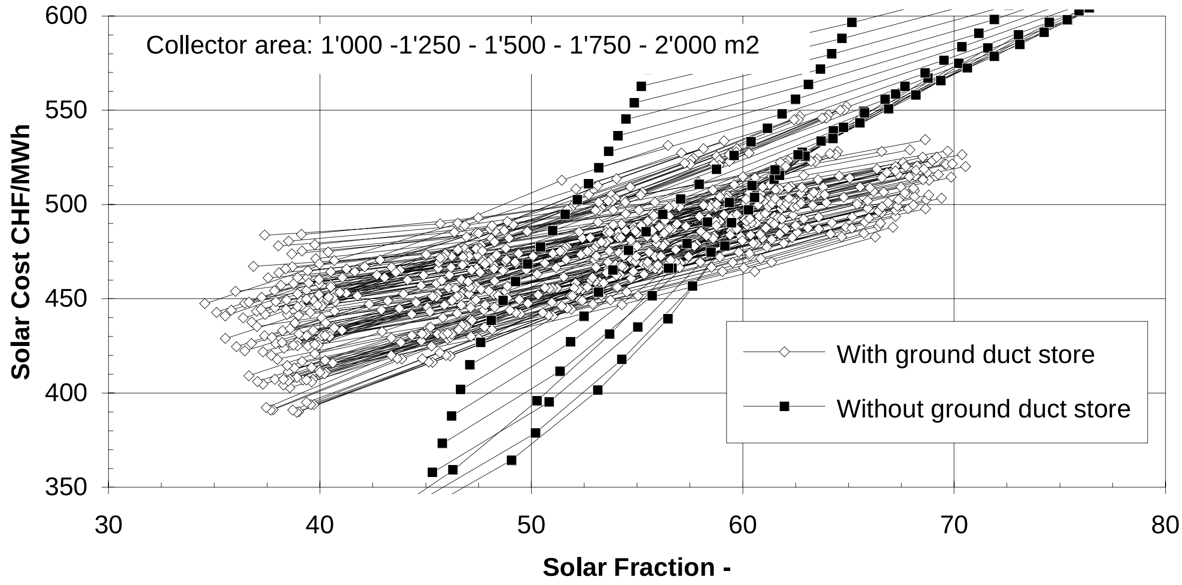



Fig. 8.8: Small heat load, 500 MWh/y (75% space heating and 25% domestic hot water); moderate-temperature heat distribution. Comparison of the solar cost for systems with or without ground duct store, in addition to a water store.

## 8.2 Minimum Solar Cost for a Solar Fraction of 70%

The five varied parameters in the previous section are adjusted so that the solar fraction of the 7 different systems corresponds to 70% within 1%. The slope of the expansion path at a solar fraction of 70% must be known for a proper determination of the optimum parameters. It can be evaluated from the slope of the lowest lines in the graphs *a)* of Fig. 8.1 to 8.7. With a low-temperature heat distribution system, the slope is approximately zero. Optimum parameters for annual heat loads of 1'000 and 5'000 MWh are also determined for a low-temperature heat distribution, assuming a zero-slope for the expansion path.

The parameters are roughly estimated based on the graphs of the previous section. The optimum buffer size and ratio duct vertical extension over duct diameter are then determined independently. Steps of 100 litres per square meter of collector are used for the determination of the optimum buffer volume. The optimal ratio duct vertical extension over duct diameter is calculated within 0.25 m/m. The optimum duct volume is then determined together with the optimum length of boreholes, set this time by imposing the borehole spacing. The resolution of these two parameters is fixed to 0.5 $m^3$ per square meter collector and 0.2 m respectively. The vertical extension of the duct store is adjusted to obtain an integer number of boreholes. Finally, the collector area is varied if necessary, so that the solar fraction lies in the specified range.

In Table 8.1 to 8.5, some characteristics of the 9 optimum systems are given. In table 8.4, the **duct store fraction** is defined as the ratio of the annual recovered energy from the duct store over the annual load. The **buffer store fraction** is defined as the difference between the solar fraction and the duct store fraction. It corresponds to nearly all the solar heat delivered to the heat load which has not transited through the duct store. This would be equal if the recovered energy from the duct store were be integrally delivered to the heat load, without heat losses through the buffer store. The denomination of the systems is set as following:

S75MOD: small heat load, 500 MWh/y, 75% space heating and 25% domestic hot water, moderate-temperature heat distribution.

M75MOD: medium heat load, 1'000 MWh/y, 75% space heating and 25% domestic hot water, moderate-temperature heat distribution.

L75MOD: large heat load, 5'000 MWh/y, 75% space heating and 25% domestic hot water, moderate-temperature heat distribution.

S50MOD: small heat load, 500 MWh/y, 50% space heating and 50% domestic hot water, moderate-temperature heat distribution.

M50MOD: medium heat load, 1'000 MWh/y, 50% space heating and 50% domestic hot water, moderate-temperature heat distribution.

L50MOD: large heat load, 5'000 MWh/y, 50% space heating and 50% domestic hot water, moderate-temperature heat distribution.

S100LOW: small heat load, 500 MWh/y, 100% space heating, low-temperature heat distribution.

M100LOW: medium heat load, 1'000 MWh/y, 100% space heating, low-temperature heat distribution.

L100LOW: large heat load, 5'000 MWh/y, 100% space heating, low-temperature heat distribution.

| Load type | Collector area $m^2$ | Buffer volume $m^3$ | Duct volume $m^3$ | Borehole number - | Duct vert. extension m |
|---|---|---|---|---|---|
| S75MOD | 2'130 | 230 | 10'700 | 70 | 35 |
| M75MOD | 3'650 | 400 | 22'000 | 94 | 44 |
| L75MOD | 13'900 | 1'670 | 111'000 | 235 | 76 |
| S50MOD | 1'860 | 220 | 6'500 | 50 | 30 |
| M50MOD | 3'250 | 420 | 14'600 | 72 | 38 |
| L50MOD | 12'700 | 1'650 | 83'000 | 176 | 75 |
| S100LOW | 1'200 | 130 | 12'600 | 47 | 43 |
| M100LOW | 2'150 | 240 | 24'000 | 71 | 53 |
| L100LOW | 9'200 | 1'200 | 120'000 | 157 | 105 |

Table 8.1 Optimum values of the five varied parameters for a minimum solar cost at a solar fraction of 70%. The parameters are given for the 9 different load types.

| Load type | Collector area per annual MWh load $m^2$/MWh | Buffer volume per $m^2$ collector litre/$m^2$ | Duct volume per $m^2$ collector $m^3/m^2$ | Duct volume and annual MWh $m^3$/MWh | Borehole spacing m | Borehole length per $m^2$ collector m/$m^2$ | Ratio duct vert. ext. over diam. m/m |
|---|---|---|---|---|---|---|---|
| S75MOD | 4.3 | 110 | 5.0 | 21 | 2.1 | 1.1 | 1.8 |
| M75MOD | 3.7 | 110 | 6.0 | 22 | 2.3 | 1.1 | 1.8 |
| L75MOD | 2.8 | 120 | 8.0 | 22 | 2.5 | 1.3 | 1.8 |
| S50MOD | 3.7 | 120 | 3.5 | 13 | 2.1 | 0.8 | 1.8 |
| M50MOD | 3.3 | 130 | 4.5 | 15 | 2.3 | 0.9 | 1.8 |
| L50MOD | 2.5 | 130 | 6.5 | 17 | 2.5 | 1.0 | 2.0 |
| S100LOW | 2.4 | 110 | 10.5 | 25 | 2.5 | 1.7 | 2.3 |
| M100LOW | 2.2 | 110 | 11.0 | 24 | 2.5 | 1.8 | 2.3 |
| L100LOW | 1.8 | 130 | 13.0 | 24 | 2.7 | 1.8 | 2.8 |

Table 8.2 Characteristics of the optimum systems that give a minimum solar cost at a solar fraction of 70%. The borehole spacing corresponds to a quadratic arrangement of the boreholes.

| Load type | Specific cost of the collectors CHF/m$^2$ | Specific cost of the buffer store CHF/m$^3$ | Specific cost of the duct store CHF/m$^3$ | Total investment cost MCHF | Collector array part | Buffer store part | Duct store part |
|---|---|---|---|---|---|---|---|
| S75MOD | 600 | 570 | 30 | 1.7 | 74% | 8% | 18% |
| M75MOD | 600 | 520 | 24 | 2.9 | 75% | 7% | 18% |
| L75MOD | 600 | 420 | 18 | 11.0 | 76% | 6% | 18% |
| S50MOD | 600 | 580 | 32 | 1.5 | 77% | 9% | 14% |
| M50MOD | 600 | 520 | 25 | 2.5 | 77% | 9% | 14% |
| L50MOD | 600 | 420 | 18 | 9.8 | 78% | 7% | 15% |
| S100LOW | 600 | 630 | 22 | 1.1 | 67% | 7% | 26% |
| M100LOW | 600 | 570 | 20 | 1.9 | 68% | 7% | 25% |
| L100LOW | 600 | 440 | 15 | 7.8 | 71% | 7% | 23% |

Table 8.3 Subsystem costs of the optimum systems that give a minimum solar cost at a solar fraction of 70%.

| Load type | Annual collector efficiency | Duct store efficiency | Duct store fraction | Buffer store fraction | Solar fraction (12th year) | Overall solar fraction (25 years) |
|---|---|---|---|---|---|---|
| S75MOD | 24% | 32% | 24.6% | 48.1% | 72.7% | 70.3% |
| M75MOD | 24% | 43% | 27.2% | 45.6% | 72.8% | 70.3% |
| L75MOD | 25% | 66% | 30.5% | 41.4% | 71.9% | 69.4% |
| S50MOD | 24% | 26% | 14.0% | 56.7% | 70.7% | 69.2% |
| M50MOD | 25% | 37% | 16.6% | 55.2% | 71.8% | 70.1% |
| L50MOD | 27% | 62% | 21.1% | 50.4% | 71.5% | 69.6% |
| S100LOW | 36% | 50% | 35.7% | 36.5% | 72.2% | 69.8% |
| M100LOW | 36% | 61% | 38.3% | 33.9% | 72.2% | 69.9% |
| L100LOW | 36% | 78% | 40.6% | 31.9% | 72.5% | 70.2% |

Table 8.4 Thermal performances of the systems for the 12th year of operation.

| Load type | Solar cost CHF/MWh | Expansion path slope (CHF/MWh)/% | Maximum fluid temperature in the collector array °C | Dissipated heat in the environment MWh/year |
|---|---|---|---|---|
| S75MOD | 490 | 3.09 | 105 | 1.9 |
| M75MOD | 420 | 2.36 | 105 | 2.5 |
| L75MOD | 320 | 1.74 | 103 | 2.1 |
| S50MOD | 420 | 3.67 | 106 | 2.5 |
| M50MOD | 360 | 3.18 | 105 | 1.8 |
| L50MOD | 280 | 2.35 | 102 | 0.8 |
| S100LOW | 310 | 0 | 92 | 0.0 |
| M100LOW | 270 | 0 | 92 | 0.0 |
| L100LOW | 220 | 0 | 91 | 0.0 |

Table 8.5 Solar cost for the 9 systems for a solar fraction of 70%. An estimation of the slope of the expansion path is also indicated. The maximum temperature indicated corresponds to the maximum fluid temperature simulated in the collector array. The dissipated heat is released in the environment through the pressure relief valve.

The load characteristics have a significant influence on the optimum size of the collector area and the duct store. Nevertheless, for the three load sizes investigated within a load type, the ratio duct store volume per annual MWh load is mostly constant for a fixed solar fraction. This is also the case for the total borehole length per square meter collector. On the other hand, the collector area per annual MWh load is variable, depending on the importance of the storages' heat losses. The optimum ratio buffer store volume per square meter of collector is mostly constant, which seems to indicate that the optimum buffer size depends principally on the collector area (given the thermal characteristics of the collectors), and the local weather conditions.

A greater proportion of domestic hot water in the heat load means less seasonal heat storage requirements. As the size of a duct store tends to be shrunk with a greater proportion of domestic hot water, this gives another indication that a duct store seems to be more attractive than a water store for seasonal heat storage only.

A small duct store has important annual heat losses relative to the annual heat that can be recovered. In consequence, short spacing between the boreholes and a compact shape is favoured in order to limit the importance of the heat losses. When their relative importance decreases (with a larger storage volume and/or a lower mean annual storage temperature), the size of the duct store, relative to the collector area, can be increased and larger spacing can be used. Nevertheless, apart from the two smallest duct store volumes, the borehole spacing is mostly constant and lies between 2.3 and 2.7 m for the investigated cases; (the soil has a thermal conductivity of 2.5 W/mK and a volumetric heat capacity of 2.3 MJ/m$^3$K). It should be noted that the maximum fluid temperature in the collector array sometimes exceeds 100 °C in the case of a moderate-temperature heat distribution. (The maximum fluid temperature is determined as the maximum outlet fluid temperature from the collector array simulated with a time-step of 0.25 hour). With a low-temperature heat distribution, the risk of overheating in the collector array is reduced to unexpected situations (pump failure, etc.), as the maximum fluid temperature in the collectors remains below 100 °C during normal operation.

In Fig. 8.9, the solar cost is shown in relation to the load size and for the three kinds of heat loads defined. It should be remembered that the specific collector price should decrease with larger collector area, due to the effect of a larger scale installation. Nevertheless, cost reductions should be concentrated on cheaper collectors, as two thirds to three quarters of the solar cost is due to the collector array. The solar cost can be compared to the cost of conventional heating with oil, which is around 120 CHF/MWh.

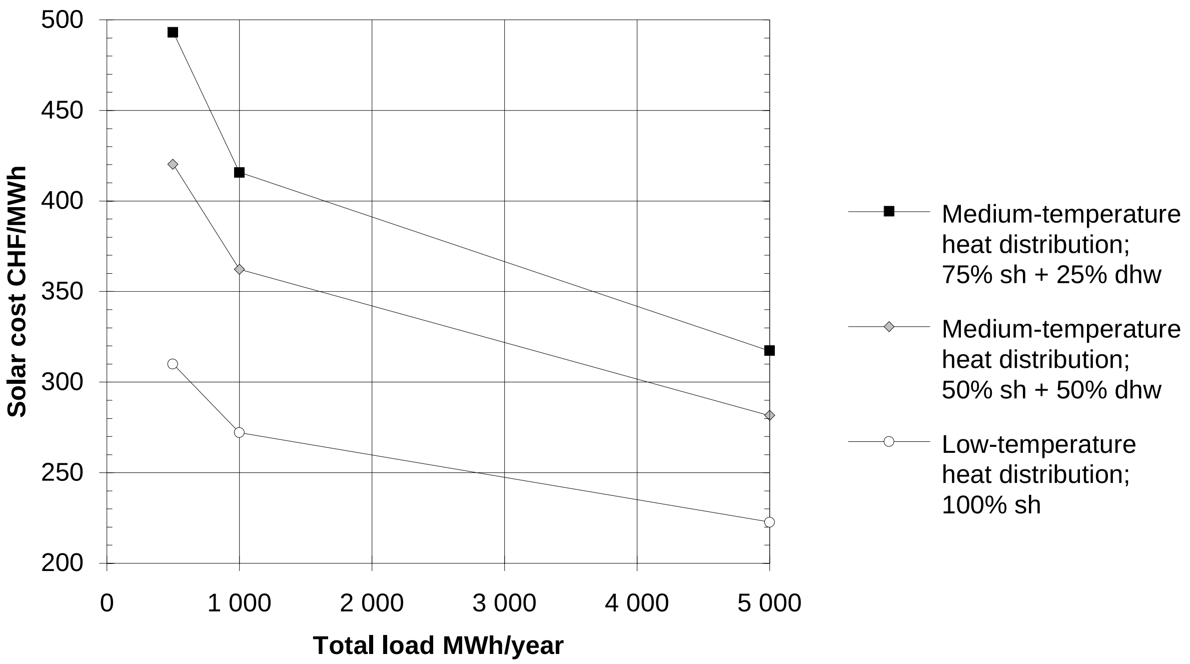



Fig. 8.9: Solar cost of the systems for the three kind of heat loads shown in relation to the load size. An annuity factor of 0.1 is assumed in the calculation of the solar cost.

## 8.3 Monthly heat balance

The monthly heat balances of the systems are shown in Fig. 8.10 to 8.12 for the small heat load (500 MWh/year) and the three kinds of load types. The thermal performances correspond to the 12th year of operation. The positive energy columns represent the collected heat and the negative ones the heat load. The black part on top of the collected heat is the dissipated energy through the expansion valve. It only occurs for the months of July and August in the case of a moderate-temperature heat distribution. The relative importance of the dissipated heat becomes smaller with a larger heat load. The difference between the used collected heat and the stored heat in the duct store, is injected in the buffer store and either immediately delivered to the heat load or after a short-term storage in the buffer store. The stored and recovered heat in and from the duct store are respectively marked in the positive and negative energy columns in grey. The relative importance of the recovered heat from the duct store in relation to the heat load is defined as the duct store fraction. The black part in the negative columns is the auxiliary heat and the rest is the solar heat. The shaded part with vertical lines corresponds to the fraction of the heat load that is met by short-term storage in the buffer tank, if one assumes that the recovered heat from the duct store is integrally delivered to the heat load. This part is defined as the buffer store fraction. The monthly temperature levels in the collector array (white rhomb), the ground heat exchanger during heat injection (grey square), the duct store (white square), as well as the ground heat exchanger during heat extraction (black square), are also shown on the graphs.

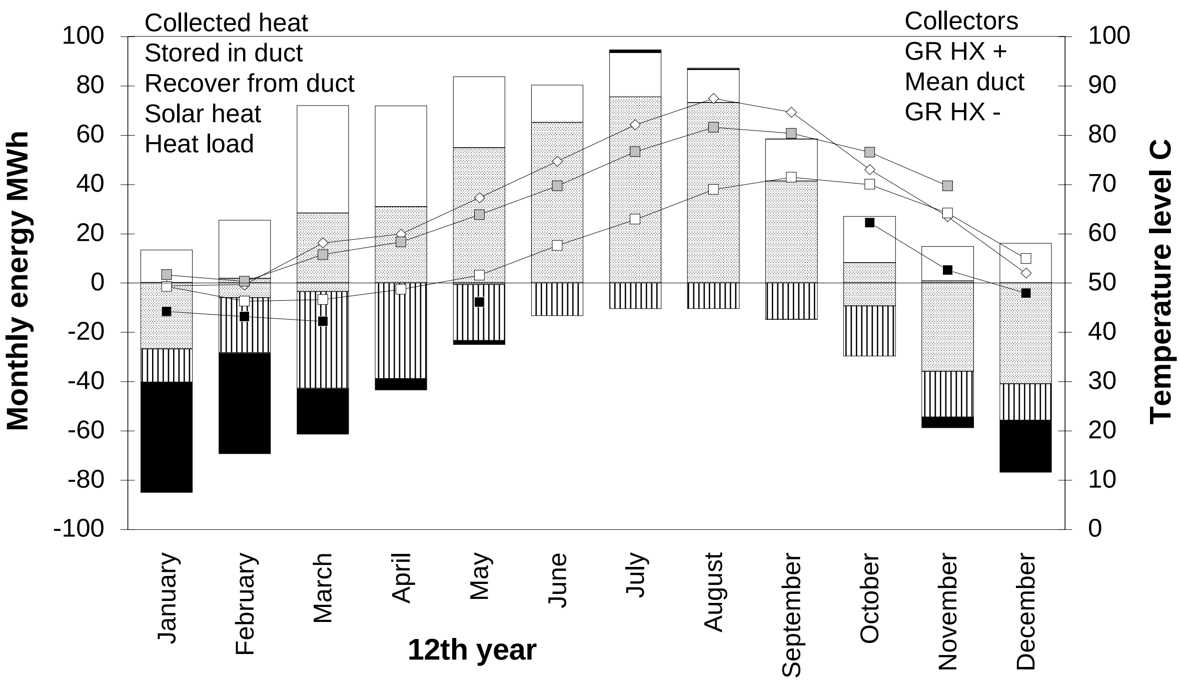



Fig. 8.10: Monthly heat balance of the system designed for the S75MOD load type; (small heat load, 500 MWh/y, 75% space heating and 25% domestic hot water, moderate-temperature heat distribution). The solar fraction corresponds to approximately 70%.

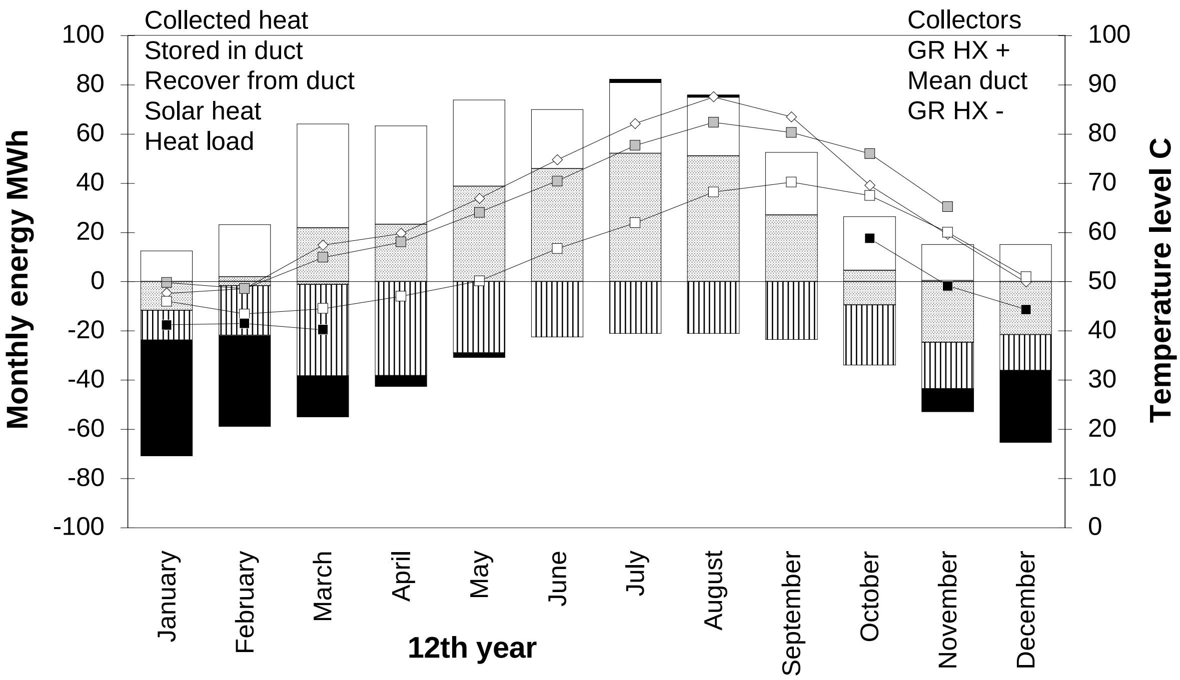



Fig. 8.11: Monthly heat balance of the system designed for the S50MOD load type; (small heat load, 500 MWh/y, 50% space heating and 50% domestic hot water, moderate-temperature heat distribution). The solar fraction corresponds to approximately 70%.

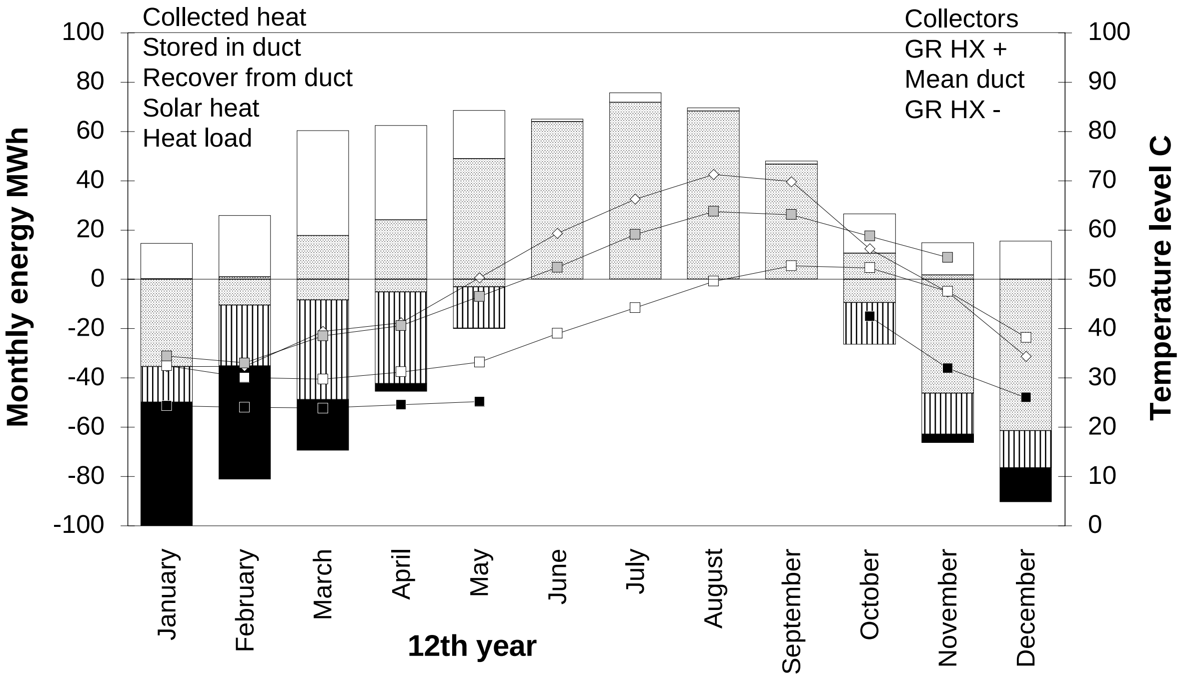



Fig. 8.12: Monthly heat balance of the system designed for the S100LOW load type; (small heat load, 500 MWh/y, 100% space heating, moderate-temperature heat distribution). The solar fraction corresponds to approximately 70%.

The graphs clearly show the task of each store: the short-term heat storage requirements are mainly covered by the buffer store, whereas the duct store is principally used for the seasonal heat storage requirements. In Fig. 8.12, the difference between the collected heat and the heat stored in the duct store during the months without heat demand is partly stored in the buffer store and partly lost through the buffer store heat losses. The temperature loss between the temperature level in the collector array and the mean temperature of the duct store is mostly significant when the duct store is loaded. For the nine simulated systems, a monthly loss of around 20 K is calculated, of which about 15 K is caused by the ground heat exchanger. Another 5 to 10 K is lost when heat is recovered from the duct store.

With a moderate-temperature heat distribution, the fluid temperature in the collector array sometimes exceeds 100 °C during July and August. The collector array should be designed so that the highest expected fluid temperatures should not paralyse the operation of the collectors. Nocturnal cooling of the buffer tank through the collector array can be implemented in the control strategy for the situations when the risk of overheating is high.

The systems with a larger load present figures similar to those shown in Fig. 8.10 to 8.12. The positive columns, which represent the monthly collected energy, are proportionally shorter than those corresponding to the small heat load, due to the smaller storages' heat losses.

## 8.4 System Control Strategy

The evolution of the mean temperatures of the buffer store and the duct store are shown in Fig. 8.13 for the system with the S100LOW load type (small heat load, 500 MWh/y, 100% space heating, low-temperature heat distribution). Similar figures are obtained with the other load types. With a moderate-temperature heat distribution, the curves are shifted by 10 to 20 additional Kelvins.

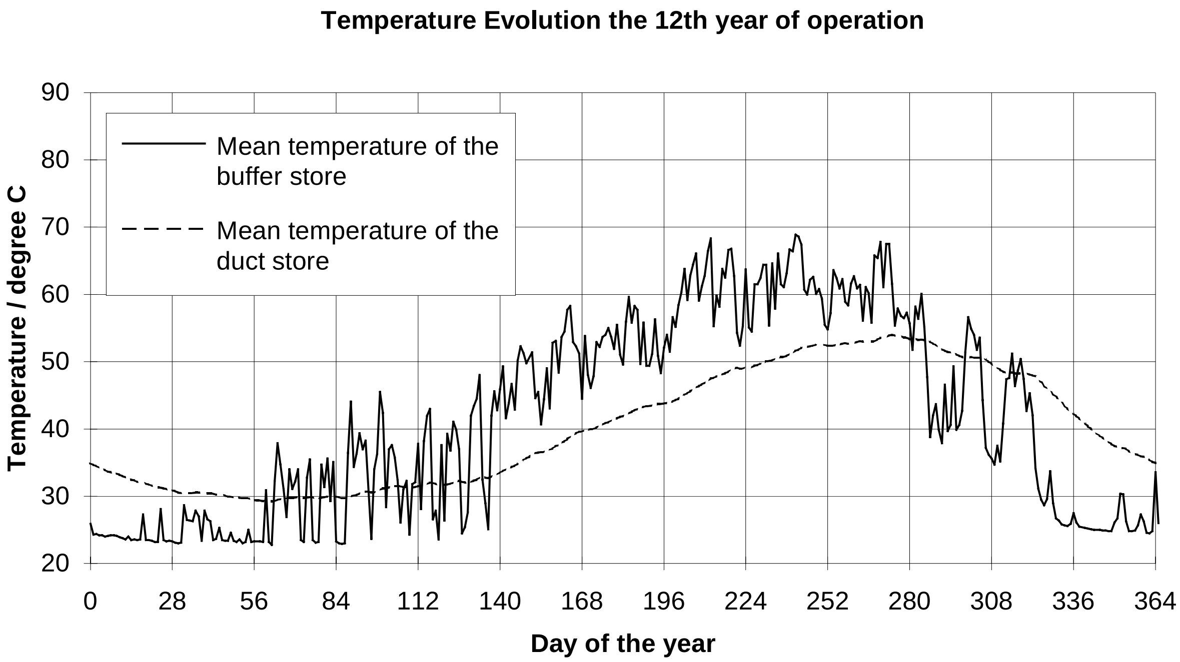



Fig. 8.13: Evolution of the mean temperature of the buffer and duct store. The simulated temperatures correspond to the system designed for the S100LOW load type; (small heat load, 500 MWh/y, 100% space heating, low-temperature heat distribution).

Even if the maximum mean temperature of the buffer does not exceed 70 °C, the maximum fluid temperature in the collector array is simulated to 92 °C, the maximum inlet fluid temperature in the buffer to 86 °C and 81 °C in the ground heat exchanger.

Two distinct operation modes are seen in the graph of Fig. 8.13. A summer mode, defined by the period when the mean buffer temperature is always greater than that of the duct store, lasts from early June until late September. During this summer period, heat is never extracted from the duct store. Similarly, a winter mode is observed from early December until late February. During this winter period, heat is never injected in the duct store. Then there are two transition periods, in the spring from March to May and during the autumn, from October to November, where both modes are present. During these transition periods, the operation strategy of the system may have some influence on the thermal performances of the system. Is it better to keep as much heat as possible in the buffer store, so that the heat load can be met by solar heat as often as possible, or to transfer heat to the duct store as soon as possible, in order to enhance the efficiency of the collector array? Depending on the weather forecast, each alternative can have its advantages. If the next day is sunny, it might be better to load the duct store in order to make "room" in the buffer store for the solar gains to come. On the other hand, if the next day is cloudy, it might be preferable to keep the heat in the buffer store to have it available for the heat load. It should be remembered that once a heat quantity is transferred to the duct store, it probably won't be available to the heat load the next day, due to the large temperature losses caused by the ground heat exchanger.

The optimisation of the system control can be achieved with the help of a new generation of simulation tools applied to solar heating with seasonal heat storage. Numerical optimisation procedures are integrated with the dynamic models describing the system. An optimum system design is directly calculated, given the objectives (for example the solar fraction), the optimisation criteria and the constraints on the variables. A multi-parameter optimisation of a system can be realised in one run. Such a simulation tool can not yet provide a simulation as detailed as TRNSYS can, but it has successfully been used for the simulation of a central solar heating plant with a water tank in Särö, Sweden (Rüdiger, 1996a). The methodology has been further developed to simulate a CSHPSS system with a ground heat storage, including a buffer store in the system design (Rüdiger, 1996b). Preliminary simulations have shown that relative to a simple system control that would transfer heat between the buffer and the duct store as soon as it is possible, an optimum system control would increase the annual solar heat of a typical solar heating system by about 10% (Rüdiger, 1996c). This optimum control is established on the knowledge of the weather in the near future, so that the best decision can be anticipated at the right time. So far, it has turned out difficult to reproduce the optimum system control with a simple control criterion. However, the optimum system control suggests that the buffer store should cover the short-term heat storage requirement "as much as possible", while the mean temperature level of the buffer store is kept "as low as possible".

If the size of the buffer store tends towards zero, the system looks and behaves like the system design without buffer store, which has a rigid control strategy (see section 3.1 and 6.2). With a larger relative size of the buffer store, there is more freedom with the control decisions, which, for example, determine in which store the collected heat should be stored. Thus the influence of the control strategy increases with an increase of the buffer store size. Larger buffer stores may also enhance the beneficial effect of a vertical temperature stratification in the buffer. Nevertheless, with a buffer size of 120 litres per square meter of collector, the volume of water moved during less than 5 hours of collector operation equals the volume of the buffer; (flow rate of 25 litres/hour/$m^2$ or 0.007 kg/s/$m^2$). Stratification effects can be improved with smaller flow rates through the buffer store. In the collector array, the effect of a smaller flow rate should be analysed together with overheating problems, as the fluid temperature will reach higher values. In the duct store, the flow rate can not be reduced without penalising the heat transfer in the ground heat exchanger. Finer analyses can determine an optimal hydraulic coupling between the boreholes and an optimum flow rate. However, the temperature difference between the inlet and outlet fluid in the ground heat exchanger is usually below 10 K, and tends to destroy a temperature stratification in the buffer. An alternative would be to connect the inlet and outlet closer to each other in the buffer store, instead of one at the top and the other at the bottom. A variable flow control in the heat load subsystem ensures the lowest possible fluid temperature and flow rate into the buffer store. All these specific problems can be assessed with the design tools developed with TRNSYS.

With the systems designed for a low-temperature heat distribution (space heating only), the mean temperature of the buffer store exceeds 50 °C from May to September. During these five months, the temperature level of the solar heat is high enough for the needs of the domestic hot water. The rest of the year, preheating of the domestic hot water could be also realised. This would make the solar cost more attractive. However, special care should be taken so that the return fluid temperature to the solar plant should remain as low as with space heating only.

# 9. CONCLUSION

Simulation tools which use the TRNSYS programme have been developed for the simulation of CSHPSS systems with a ground duct store. The version of the duct heat storage model (DST) for TRNSYS has been improved and other components were created for the needs of the simulation tools. They provide different levels of complexity in the way such systems can be simulated. Fine effects, related to the final design of CSHPSS systems, may also be assessed. In this study, only the influence of the main system parameters were investigated, in order to determine optimal ratios between the main parameters.

A system has been defined to perform a case study applied to typical Swiss conditions. Several load types were defined to assess the influence of the load size, the proportion of domestic hot water in the heat load and the temperature levels of the heat delivered in the distribution network.

The load characteristics have a significant influence on the relative size of the different subsystems. Nevertheless, for the three load sizes investigated within a load type (500, 1'000 and 5'000 MWh/year), the ratio duct store volume per annual MWh load is mostly constant for a fixed solar fraction. Ratio of about 15, 22 and 25 $m^3$/MWh were found for, respectively; the 50% domestic hot water and moderate-temperature heat distribution load, the 25% domestic hot water and moderate-temperature heat distribution load and the low-temperature heat distribution load without domestic hot water, and for a solar fraction of 70%. The borehole spacing tends to increase with larger ratio duct store volume over collector area. However, apart from the two smallest duct store volumes which have a spacing of 2.1 m, the borehole spacing is mostly constant and lies between 2.3 and 2.7 m. (The soil has a thermal conductivity of 2.5 W/mK and a volumetric heat capacity of 2.3 MJ/$m^3$K.) The shape of the duct store, defined with the ratio duct vertical extension over duct store diameter, has an optimum value that lies between 1.8 and 2.8, depending on the duct store volume and the mean annual temperature level of the store. (These values are obtained for a duct store at the ground surface and covered by 20 cm of insulation.) The optimum ratio buffer store volume per square meter of collector is mostly constant for the 9 load types and any solar fraction (> 50%). It seems to indicate that the optimum buffer size depends principally on the collector area (given the thermal characteristics of the collectors), and the local weather conditions. For the studied cases, the optimum value is about 120 litres/$m^2$. The optimal collector area, for a solar fraction of 70%, strongly depends on the load characteristics. The ratio collector area over annual heat load varies from 1.8 to 4.3 $m^2$/MWh. With a small system and a moderate-temperature heat distribution, the relatively large heat losses of the duct store have to be paid for with solar heat.

A larger proportion of domestic hot water makes the solar cost lower, but also makes the optimum size of a duct store smaller, making a ground duct store attractive for larger solar fractions. A heat load of 500 MWh/year is a critical size if the temperature levels of the heat distribution correspond to those of the moderate-temperature heat distribution; (forward temperature: 50 - 55 °C; return temperature: 35 - 40 °C). Relative to a larger heat load, a parameter deviation from the calculated optimum design has more influence on the solar cost. The risk of having unexpected thermal performances is greater with a smaller heat load. For a solar fraction of 70%, the solar cost is calculated at 490 CHF/MWh for 25% of domestic hot water and at 420 CHF/MWh for 50%. With a low-temperature heat distribution (for space heating only), the solar cost drops to 310 CHF/MWh. A load size of 500 MWh/year is very small for a CSHPSS system; a heat load ten times larger would drop the cost to respectively 320, 280 and 220 CHF/MWh, without taking into account the effect of a larger scale installation of the collector area, which would decrease its specific price, set to 600 CHF/$m^2$. Even these last prices are expensive, if compared to conventional heating with oil, which is around 120 CHF/MWh. A smaller annuity factor than 0.1 could be expected. Nevertheless, cost reductions should be concentrated on cheaper collectors, as two thirds to three quarters of the solar cost is due to the collector array. Present Swedish costs in Switzerland would drop the solar cost to the price level of conventional heating!

A duct store seems to be more attractive than a water store for seasonal storage only, i. e. for large values of the solar fraction. For low solar fractions (typically lower than about 50%), the main function of the duct store is to cool down the buffer store during the summer. In this situation, a system without duct store is cheaper, but the fluid temperature in the collector array reaches higher values, and may cause overheating problems.

Even with a low-temperature heat distribution, the fluid temperatures sometimes reach high values during normal operation, which are close to 100 °C in the collector array. Even the ground heat exchanger has to withstand high temperatures. A low-temperature application is not only a prerequisite for the efficient use of a duct store in the range of systems investigated, but also to reduce the risk of overheating in the collector array to unexpected situations (pump failure, etc.). Domestic hot water could be partly covered by a system designed for a low-temperature heat distribution, as the temperature level of the solar heat during the summer is high enough for the domestic hot water needs. This would make the economy of such systems more attractive. However, care should be taken in order to avoid designs that would increase the return fluid temperature in the solar heating plant or make the system operate at higher temperatures (undersized heat exchangers, the way the auxiliary heat is delivered to the heating plant, fluid mixing, etc.).

The influence of the control strategy is expected to increase with a larger buffer store in a system. An optimum strategy would aim to cover most of the short-term heat requirements with the buffer store, whereas the ground duct store would primarily be used to cover the seasonal heat requirements.

The thermal performances of a CSHPSS system with a seasonal duct storage in the ground are very sensitive to the temperature levels of the fluid in the distribution network, especially if the load size tends to be small for such a plant (below 1'000 MWh/year). The load characteristics, including the temperature levels of the forward and return fluid temperatures from the heating plant, need to be determined with the best possible accuracy. A more accurate load model than that used in the simulation tools should be used to characterise a known consumer, once a final design has to be assessed and optimised.

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# A1. PRESSURE DROP COMPONENT

The pressure drop component TYPE 64 calculates the pressure drop and head loss caused by the fluid circulation in a ground heat exchanger. The heat rate dissipated by the fluid circulation and the electric power consumed by the pump, assuming a constant overall efficiency of the "pump + electric motor", are also assessed. The collecting and connecting pipes on ground surface may be included in the calculations.

## A1.1 Basic Relations

A complete description of the pressure drop caused by fluid friction is given in ASHRAE (1989). Basic relations and some constant values are summarised below.

Pressure drop caused by fluid friction in fully developed flows of all "well behaved" (Newtonian) fluids is described by the Darcy-Weisbach equation:

$$\Delta p = f \frac{L}{D} \frac{\rho V^2}{2} \qquad \text{(A1.1)}$$

where
$\Delta p$ = pressure drop [Pa];
f = friction factor, dimensionless [-];
L = length of pipe [m];
D = internal diameter of pipe [m];
$\rho$ = fluid density at mean temperature [kg/m$^3$];
V = average velocity [m/s].

This equation is often presented in head or specific energy form as:

$$\Delta h = \frac{\Delta p}{\rho g} = f \frac{L}{D} \frac{V^2}{2g} \qquad \text{(A1.2)}$$

where
$\Delta h$ = head loss [m];
g = acceleration of gravity [m/s$^2$].

In this form, the density of the fluid does not appear explicitly (although it is in the Reynolds number Re, which influences f). The Reynolds number is given by:

$$Re = \frac{DV\rho}{\mu} \qquad \text{(A1.3)}$$

where
$\mu$ = dynamic viscosity of the fluid [kg/ms].

Note: for an annular pipe, the diameter D is replaced with the difference between the outer and inner diameter of the annular space.

For fully developed laminar-viscous flow in a pipe, the friction factor is given by:

$$f = \frac{64}{Re} \qquad \text{if Re < 2'000} \qquad (A1.4)$$

With turbulent flow, the friction factor depends not only on flow conditions, as characterised by the Reynolds number, but also on the nature of the conduit wall surface. With smooth conduit walls, empirical correlations give:

$$f = \frac{0.3164}{Re^{0.25}} \qquad \text{for Re up to 100'000} \qquad (A1.5)$$

$$f = 0.0032 + \frac{0.221}{Re^{0.237}} \qquad \text{for Re > 100'000} \qquad (A1.6)$$

Generally, the friction factor f depends on the wall roughness ε. The mode of variation is complex and best expressed in a Moody chart, giving f as a function of Re with ε/D as a parameter. A useful correlation of smooth and rough pipe data for turbulent flow regime is the Colebrook equation:

$$\frac{1}{\sqrt{f}} = 1.74 - 2 \log\left( \frac{2\,\varepsilon}{D} + \frac{18.7}{Re\,\sqrt{f}} \right) \qquad (A1.7)$$

The equation is implicit in f; f appears on both sides, so a value for f is usually obtained after some iterations. The first try can be calculated for a smooth pipe wall. The roughness height ε, which may increase with conduit use or ageing, must be evaluated from the conduit surface (see Table A1.1)

| Material: | ε [μm] |
|---|---|
| Commercially smooth brass, lead, copper, or plastic pipe | 1.5 |
| Steel and wrought iron | 45 |
| Galvanised iron or steel | 150 |
| Cast iron | 250 |

Table A1.1 Effective roughness of some conduit surfaces.

Pipes and fitting losses cause pressure losses greater than those caused by the pipe alone. One formulation expresses losses as:

$$\Delta h = k\,\frac{V^2}{2g} \qquad (A1.8)$$

where

k = geometry and size dependent loss coefficient [-].

Some k values are given in Table A1.2 and A1.3.

The heat rate dissipated by the circulation of the fluid is:

$$P_{dis} = \rho\ S\ V\ g\ \Delta h = S\ V\ \Delta p \qquad (A1.9)$$

where
$P_{dis}$ = heat rate dissipated by the fluid circulation [W];
S = section through which the fluid circulates [$m^2$].

| Nominal Pipe Diam. [mm] | 90° Elbow Regular | 90° Elbow Long | 45° Elbow | Return Bend (180°) | Tee-Line | Tee-Branch | Gate Valve |
|---|---|---|---|---|---|---|---|
| 10 | 2.5 | - | 0.38 | 2.5 | 0.90 | 2.7 | 0.40 |
| 15 | 2.1 | - | 0.37 | 2.1 | 0.90 | 2.4 | 0.33 |
| 20 | 1.7 | 0.92 | 0.35 | 1.7 | 0.90 | 2.1 | 0.28 |
| 25 | 1.5 | 0.78 | 0.34 | 1.5 | 0.90 | 1.8 | 0.24 |
| 32 | 1.3 | 0.65 | 0.33 | 1.3 | 0.90 | 1.7 | 0.22 |
| 40 | 1.2 | 0.54 | 0.32 | 1.2 | 0.90 | 1.6 | 0.19 |
| 50 | 1.0 | 0.42 | 0.31 | 1.0 | 0.90 | 1.4 | 0.17 |
| 65 | 0.85 | 0.35 | 0.30 | 0.85 | 0.90 | 1.3 | 0.16 |
| 80 | 0.80 | 0.31 | 0.29 | 0.80 | 0.90 | 1.2 | 0.14 |
| 100 | 0.70 | 0.24 | 0.28 | 0.70 | 0.90 | 1.1 | 0.12 |

Table A1.2 k values for screwed pipe fittings.

| Nominal Pipe Diam. [mm] | 90° Elbow Regular | 90° Elbow Long | 45° Elbow Long | Return Bend Regular | Tee-Line | Tee-Branch | Gate Valve |
|---|---|---|---|---|---|---|---|
| 25 | 0.43 | 0.41 | 0.22 | 0.43 | 0.26 | 1.00 | - |
| 32 | 0.41 | 0.37 | 0.22 | 0.41 | 0.25 | 0.95 | - |
| 40 | 0.40 | 0.35 | 0.21 | 0.40 | 0.23 | 0.90 | - |
| 50 | 0.38 | 0.30 | 0.20 | 0.38 | 0.20 | 0.84 | 0.34 |
| 65 | 0.35 | 0.28 | 0.19 | 0.35 | 0.18 | 0.79 | 0.27 |
| 80 | 0.34 | 0.25 | 0.18 | 0.34 | 0.17 | 0.76 | 0.22 |
| 100 | 0.31 | 0.22 | 0.18 | 0.31 | 0.15 | 0.70 | 0.16 |
| 150 | 0.29 | 0.18 | 0.17 | 0.29 | 0.12 | 0.62 | 0.10 |
| 200 | 0.27 | 0.16 | 0.17 | 0.27 | 0.10 | 0.58 | 0.08 |
| 250 | 0.25 | 0.14 | 0.16 | 0.25 | 0.09 | 0.53 | 0.06 |
| 300 | 0.24 | 0.13 | 0.16 | 0.24 | 0.08 | 0.50 | 0.05 |

Table A1.3 k values for flanged welded pipe fittings.

## A1.2 Parameters, Inputs and Outputs of the Pressure Drop Component

The ground heat exchanger is formed by RBore boreholes with NSerie boreholes connected in series. They are thus RBore/NSerie branches of boreholes coupled in parallel. The flow rate is supposed to be equally divided into each branch. If water has to be lifted in an open system, this extra head loss has to be added to those calculated by this subroutine. The pipes on the ground surface can be included in the calculations. Up to 10 different pipe sizes can be defined. The electric power consumed by the pump is calculated by assuming a constant overall efficiency of the "pump + electric motor".

**PARAMETERS (14 to 64):**

1. Depth: total length of one borehole [m]
2. RBore: total number of boreholes [-]
3. NSerie: number of boreholes connected in series (≤ RBore) [-]
4. EtaPum: overall efficiency of the "pump+electric motor" (constant) [-]
5. FlType: FlType = 0: plain WATER; the fluid properties are calculated and depend on the mean fluid temperature.
   FlType > 0: FlType is the fluid density (constant value) [kg/m$^3$]
   FlType < 0: the fluid properties depend on the mean fluid temperature (the fluid density and dynamic viscosity).
   –FlType is the logical unit through which the fluid properties are provided. An assign statement should link the file name to this logical unit in the TRNSYS deck.
6. FlNumb: if FlType = 0; FlNumb is not used;
   if FlType > 0; FlNumb is the dynamic viscosity (constant value) [kg/ms]
   if FlType < 0; FlNumb is the number of different fluid temperatures (≤ 10), for which the fluid properties are given.
7. BorIns: borehole installation (0: coaxial pipe, 1: single U-pipe, 2: double U-pipe, etc.) [-]
8. Dinint: internal diameter, U-pipe or central pipe if coaxial [m]
9. Epsin: effective roughness of pipe wall [m]
10. Dinout: external diameter of central pipe (not used if U-pipe) [m]
11. Doutin: internal diameter of outer pipe (not used if U-pipe) [m]
12. Epsout: effective roughness of annulus wall (not used if U-pipe) [m]
13. Skpipe: sum of loss values (k) per borehole (bend, fitting, etc.) [-]
14. NTypip: number of different pipe types on ground surface (0 - 10), that form a flow loop through a branch of boreholes chosen as representative [-]

For i = 1 to NTypip (if NTypip=0, the pipes on ground surface are not considered and the five next parameters are not required)

15+(i-1)·5 LsurfI: total length of pipe i on surface in the chosen flow loop [m]
16+(i-1)·5 FlowiI: fraction of total flow rate in pipe i [-]
17+(i-1)·5 DiamI: internal diameter of pipe i [m]
18+(i-1)·5 EpsinI: effective roughness of the pipe wall i [m]
19+(i-1)·5 SkpipI: sum of loss values (k) associated to pipe i (bend, fitting, etc.) [-]

**INPUTS (4):**

1. Tin: inlet fluid temperature in the ground heat exchanger [degree C]
2. Tout: outlet fluid temperature from the ground heat exchanger [degree C]
3. $m_{fl}$: total mass flow rate through the ground heat exchanger [kg/h]
4. Reset: reset output values number 5 to 8 to zero [-]
   Reset > 0: reset; Reset ≤ 0: no reset.

**OUTPUTS (8):**

1. deltaP: total pressure drop in the ground heat exchanger [kPa]
2. deltaH: total head loss in the ground heat exchanger [m]
3. Qdiss: heat rate dissipated by the circulation of the fluid [kJ/h]
4. Qel: electric power consumed by the pump [kJ/h]
5. dPmax: maximum pressure drop in the ground heat exchanger [kPa]
6. dHmax: maximum head loss in the ground heat exchanger [m]
7. Qdmax: maximum heat rate dissipated by fluid circulation [kJ/h]
8. Qemax: maximum electric power consumed by the pump [kJ/h]

# A2. DUCT STORE CONTROLLER COMPONENT

A duct store controller component (TYPE 61) is developed to control the operation of the duct store. Its main purpose is to combine the different control signals related to the duct store operation with a criterion based on the electric power consumed. This component may also be used to implement a more complex strategy of the duct store operation.

## A2.1 Parameters, Inputs and Outputs of the Duct Store Controller Component

The control signals of the loading and unloading controllers, which determine the operation mode of the duct store, are each multiplied with an associate control signal. Typically, the control signals of the controllers have a value 1 or 0 (ON or OFF), and their respective associate control signal a value comprised between 1 an 0. In this way, it is possible to unload the duct store with a variable flow rate, by setting the associate control signal of the unloading controller to any variable control signal.

The real purpose of the component is to condition the control signals with a criterion based on the electric power required for the circulation of the heat carrier fluid. If the electric power consumed by the pump, multiplied by a user given coefficient, is larger than the heat rate transferred in the duct store for any iteration at a given TRNSYS simulation time-step, the loading or unloading pump is switched off.

**PARAMETER (1):**

1. Fpump: weight factor which multiplies Qpump, the electric power consumed by the pump [-]. Qsto, the heat transferred in the duct store, is compared to Fpump·Qpump. If Qsto-Fpump·Qpump<0 then GAMMAQ=0

**INPUTS (6):**

1. Gload: control signal which controls the loading mode of the duct store [-]
2. Gloass: control signal associated to Gload [-]
3. Gunlo: control signal which controls the unloading mode of the duct store [-]
4. Gunass: control signal associated to Gunlo [-]
5. Qsto: heat rate transferred in the duct store [kJ/h]
6. Qpump: electric power consumed by the circulation pump [kJ/h]

**OUTPUTS (2):**

1. Gload: control signal which controls the loading mode of the duct store [-]
   Gload(out) = Gload(in) · Gloass · GAMMAQ

2. Gunlo: control signal which controls the unloading mode of the duct store [-]
   Gunlo(out) = Gunlo(in) · Gunass · GAMMAQ

# A3. VARIABLE FLOW RATE COMPONENT

The component simulates a pump and a two-way valve controlled by a temperature, according to the system designs shown in chapter 3. This component is used together with a heat exchanger, as it controls the flow rate of the hot side, so that the outlet fluid temperature of the cold side is adjusted to a desired value.

This component TYPE 67 can only be used with a counter-flow heat exchanger, as the mathematical model used to calculate the heat transfer in the heat exchanger must correspond to that assumed in the variable flow rate. The flow rate in the hot site is calculated to its exact value.

## A3.1 Basic Relations

At a given TRNSYS iteration, the input and output quantities of the heat exchanger are "known" to the variable flow rate component. If the outlet fluid temperature from the heat exchanger (in the cold side) differs from the required fluid temperature, the variable flow rate component adjusts the flow rate (in the hot side), so that the two temperatures would match with the new flow rate; (if a solution exists). In order to save a TRNSYS iteration at a given time-step, the calculations of the heat exchanger should be performed before those of the variable flow rate component; (the unit number of the heat exchanger should be smaller than that of the variable flow rate component).

The following relations describe the heat rate transferred by the heat exchanger, for the actual ($Q_{try}$) and desired case ($Q_{aim}$):

$$Q_{try} = (T_{hi} - T_{ho})\, m_{htry}\, C_{ph} \tag{A3.1}$$

$$Q_{aim} = (T_{hi} - T_{hoaim})\, m_{haim}\, C_{ph} \tag{A3.2}$$

$$Q_{try} = (T_{co} - T_{ci})\, m_c\, C_{pc} \tag{A3.3}$$

$$Q_{aim} = (T_{set} - T_{ci})\, m_c\, C_{pc} \tag{A3.4}$$

$$Q_{try} = UA\, \frac{((T_{hi} - T_{co}) - (T_{ho} - T_{ci}))}{\ln((T_{hi} - T_{co}) / (T_{ho} - T_{ci}))} \tag{A3.5}$$

$$Q_{aim} = UA\, \frac{((T_{hi} - T_{set}) - (T_{hoaim} - T_{ci}))}{\ln((T_{hi} - T_{set}) / (T_{hoaim} - T_{ci}))} \tag{A3.6}$$

where

| | | |
|---|---|---|
| $T_{hi}$ | = | inlet fluid temperature in the heat exchanger, hot side [°C] |
| $T_{ho}$ | = | outlet fluid temperature from the heat exchanger, hot side [°C] |
| $T_{ci}$ | = | inlet fluid temperature in the heat exchanger, cold side [°C] |
| $T_{co}$ | = | outlet fluid temperature from the heat exchanger, cold side [°C] |
| $C_{ph}$ | = | specific heat of the heat carrier fluid, hot side [kJ/kgK] |
| $C_{pc}$ | = | specific heat of the heat carrier fluid, cold side [kJ/kgK] |
| UA | = | overall heat transfer coefficient of the counter-flow heat exchanger [kJ/hrK] |

$m_{htry}$ = mass flow rate in the hot side [kg/hr]
$m_c$ = mass flow rate in the cold side [kg/hr]
$Q_{try}$ = actual heat rate transferred through the heat exchanger [kJ/hr]
$T_{set}$ = desired outlet fluid temperature from the heat exchanger, cold side [°C]
$m_{haim}$ = desired mass flow rate in the hot side [kg/hr]
$T_{hoaim}$ = desired fluid temperature from the heat exchanger, hot side [°C]
$Q_{aim}$ = desired heat rate transferred through the heat exchanger [kJ/hr]

The set of equations can be reduced to three equations with three unknowns: the ratio $Q_{aim}/Q_{try}$, the temperature $T_{hoaim}$ and the sought ratio $m_{haim}/m_{try}$.

$$\frac{Q_{aim}}{Q_{try}} = \frac{m_{haim}}{m_{htry}} \cdot \frac{(T_{hi} - T_{hoaim})}{(T_{hi} - T_{ho})} \tag{A3.7}$$

$$\frac{Q_{aim}}{Q_{try}} = \frac{(T_{set} - T_{ci})}{(T_{co} - T_{ci})} \tag{A3.8}$$

$$\frac{Q_{aim}}{Q_{try}} = \frac{((T_{hi} - T_{set}) - (T_{hoaim} - T_{ci}))}{((T_{hi} - T_{co}) - (T_{ho} - T_{ci}))} \frac{\ln((T_{hi} - T_{co})/(T_{ho} - T_{ci}))}{\ln((T_{hi} - T_{set})/(T_{hoaim} - T_{ci}))} \tag{A3.9}$$

The solution is found by solving a transcendent equation of the type:

$$X = B \exp(A(X-B)) \tag{A3.10}$$

The trivial solution $X = B$ is rejected. A second solution exists only if the coefficients $A$ and $B$ are greater than $0$.

## A3.2 Parameters, Inputs and Outputs of the Variable Flow Rate Component

An input control signal is foreseen in case the operation of the variable flow rate component should be forced to stop. An error output, set to 10 times the value of GAMMA, the fraction of the maximum flow rate, can be connected to the corresponding input, so that TRNSYS will iterate until the error variation between two iterations is below the user given tolerance. This is particularly useful when the tolerance is set as an absolute tolerance. The old value of GAMMA, calculated at the previous iteration or time-step, is used if the flow rate does not need to be changed.

**PARAMETERS (2):**

1. MMAXI maximal mass flow rate of the variable flow rate pump (in hot side of the heat exchanger) [kg/hr]
2. MMIN minimal mass flow rate of the variable flow rate pump in operation [kg/hr]

**INPUTS (9):**

1. Tinlet: inlet fluid temperature in the variable flow rate component [°C]
2. Minlet: inlet mass flow rate in the variable flow rate component [kg/h]
3. Thxci: inlet fluid temperature in the heat exchanger, cold side [°C]
4. Mhxci: mass flow rate in the cold side of the heat exchanger [kg/hr]

5. Thxho: outlet fluid temperature from the heat exchanger, hot side [°C]
6. Tset set point temperature desired for Thxco [°C]
7. Thxco: outlet fluid temperature from the heat exchanger, cold side [°C]
8. CTRL: control signal of an external controller [-]
   CTRL ≤ 0: OFF, the pump of the variable flow rate component is stopped.
   CTRL > 0: ON, the pump of the variable flow rate component is ON whenever possible and the flow rate adjusted to ensure the lowest return temperature possible.
9. Err2: error control, to be connected with output number 4 [-]

**OUTPUTS (6):**

1. Tout: outlet fluid temperature from the component, set to the inlet fluid temperature [°C]
2. Mout: outlet mass flow rate, adjusted by the component [kg/hr]
3. GAMMA: fraction of the maximum flow rate [-]
4. Err2: error control of the outlet mass flow rate; Err2 = 10 x GAMMA [-]

The outputs 5 and 6 are used for an internal use of the component. The two previous values of GAMMA are stored in these outputs.

# A4. HEAT LOAD COMPONENT

## A4.1 Heat Load Component Description

The load component TYPE 59 is based on a simple load model, intended to generate hourly values for the heat demand of a building or group of similar buildings; (see Pahud (1995) for the determination of the space heating parameters and a validation of the model). It is a one-node model based on the steady-state heat losses of a building. A solar effective area may be used to evaluate the passive solar gains. An effective heat capacity is used to store the hourly passive solar gains. The indoor temperature can not rise above a maximum user-given value. If the passive solar gains are not explicitly calculated, they may implicitly be taken into account with an outdoor temperature limit above which the heating is stopped; (similar to the Degree-Day approach). The internal gains, due to the occupancy and activities inside the building, are assumed to be constant over the heating period.

Seven parameters are required for the space heating model; three for the heat losses, one for the internal gains and three for the passive solar gains. They are:

**Tin**: indoor set-point temperature [°C];
**H**: total specific heat losses of the building(s); (transmission and ventilation losses) [kJ/hK];
**Tnh**: outdoor temperature heat cut-out limit.; (if the outdoor temperature exceeds this limit, then the heat requirement is assumed to be negligible) [°C];
**ΔTingains**: correction to the indoor set-point temperature due to the internal gains; if *Pingains* represents the heat rate generated by internal gains, the correction is given by the ratio *Pingains*/*H* [K];
**Aeff**: solar effective area for passive solar gains collection; this area is relative to the insolation falling on a vertical plane (preferably facing south) [$m^2$];
**τ**: time constant of the building(s) defined by the ratio *Ceff*/*H* [h]; *Ceff* is an effective heat capacity of the building(s), for a daily storage of the passive solar gains [kJ/K];
**Tmaxin**: maximum indoor temperature allowed [°C].

The weather data, given with hourly values, should contain **Tout**, the outdoor temperature of the air, and **Ivs**, the global insolation on a plane corresponding to Aeff, the solar effective area, if the passive solar gains are explicitly calculated.

The following three variables are calculated. The latter two, coupled together with the relation A4.2b, need initial values; they may be initialised at zero.

**Psh**: heat rate required for space heating [kJ/h];
**Qcapa**: heat accumulated in the structure of the housing [kJ];
**ΔTin**: elevation of the indoor temperature due to passive solar gains [K].

For each time-step the heat balance of the building gives:

$$Psh = H \cdot [(Tin - \Delta Tingains) + \Delta Tin - Tout] - Aeff \cdot Ivs - Qcapa/\Delta t \qquad (A4.1)$$

**Tout**: utdoor temperature of the air [°C];
**Ivs**: global insolation on a plane corresponding to Aeff, the solar effective area [kJ/m$^2$hr];
**Δt**: duration of the time-step [h].

Heat is stored in the structure of the building only if the heat rate for space heating is negative:

If (Psh < 0) AND (H > 0) AND (τ > 1 hour) Then

$$Qcapa = -Psh \cdot \Delta t \qquad \text{(A4.2a)}$$

$$\Delta Tin = Qcapa / (\tau \cdot H) \qquad \text{(A4.2b)}$$

Else

$$Qcapa = 0 \qquad \text{(A4.2c)}$$

$$\Delta Tin = 0 \qquad \text{(A4.2d)}$$

The heat demand is only considered for positive values and when the outdoor temperature is below the heat cut-out temperature limit:

If (Psh < 0) OR (Tout > Tnh) Then Psh = 0 (A4.3)

The indoor temperature cannot exceed the maximum limit allowed:

If (Tin + ΔTin) > Tmaxin Then

$$\Delta Tin = Tmaxin - Tin \qquad \text{(A4.4a)}$$

$$Qcapa = (\tau \cdot H) \cdot (Tmaxin - Tin) \qquad \text{(A4.4b)}$$

The Degree-Day approach can be reproduced if the passive solar gains and the internal gains are not explicitly computed with Aeff and ΔTingains; (Aeff = 0, τ = 0 and ΔTingains = 0). The use of the hourly rather than daily outdoor temperature in order to determine if heating is required or not, leads to similar results. The annual heat demand does not differ by more than a few percent (with Swiss weather data for Geneva).

Monthly correction factors are foreseen in case the load model should match monthly space heating data. Used together with the Degree-Day approach of the model, this corresponds to a total specific heat losses coefficient H that may vary from month to month. The monthly correction factors may also be used to set the monthly space heating requirement to zero, if the space heating is stopped during a whole month.

The domestic hot water heat demand is given with a daily heat requirement value. Monthly correction factors may also be used to take into account a seasonal variation of the heat requirements for the domestic hot water. The annual heat demand is the sum of the daily heat requirements. A representative hourly profile during the day is used and repeated every day. It is shown as relative values in Figure A4.1.

Distribution losses are calculated, based on the forward and return fluid temperatures in the distribution network (see below), the losses coefficients of the forward and return pipes and a mean annual sink temperature. They do not contribute to the total annual heat demand if the heat distribution network is switched off. This latter is supposed to be off only when the space heating heat demand is null and the domestic hot water system is off; (i.e. if the monthly correction factor is zero or the daily heat requirement is zero).

The forward temperature of the fluid in a distribution network is usually controlled and set to a desired value which depends on the outdoor temperature. The return temperature is difficult to assess, as the heat requirement of the housing, the characteristics of the distribution network and the thermal performances of the heat devices in the housing are involved. One assumes that the return temperature may be described in the same way as the forward temperature, with a similar relation. See Fig. A4.2 for the parameters that define the forward and return fluid temperature in the distribution network. The flow rate in the heat distribution network is entirely determined.

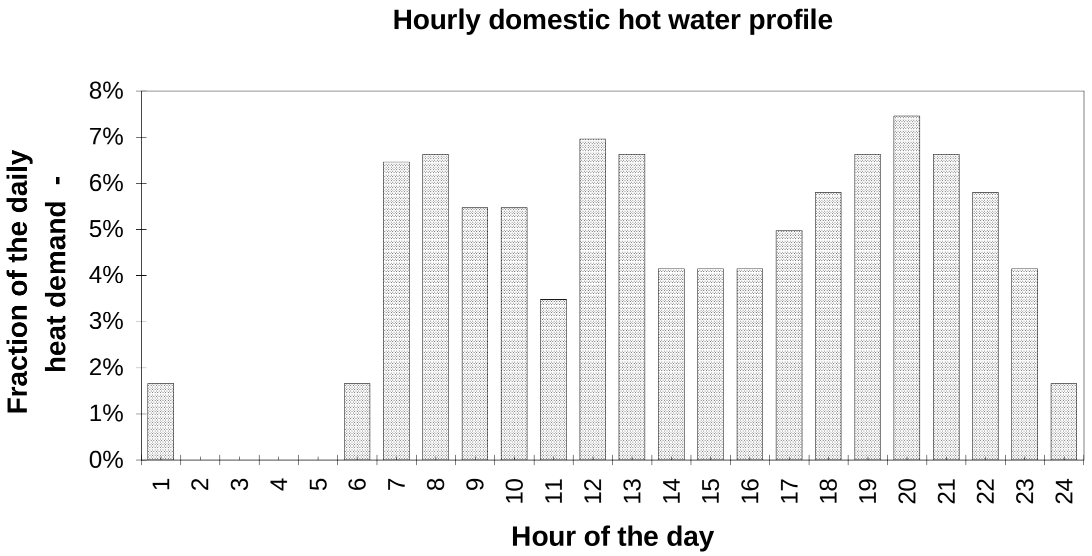


Fig. A4.1 Hourly profile of the domestic hot water heat demand. The hourly values are shown as a fraction of the daily heat requirement.

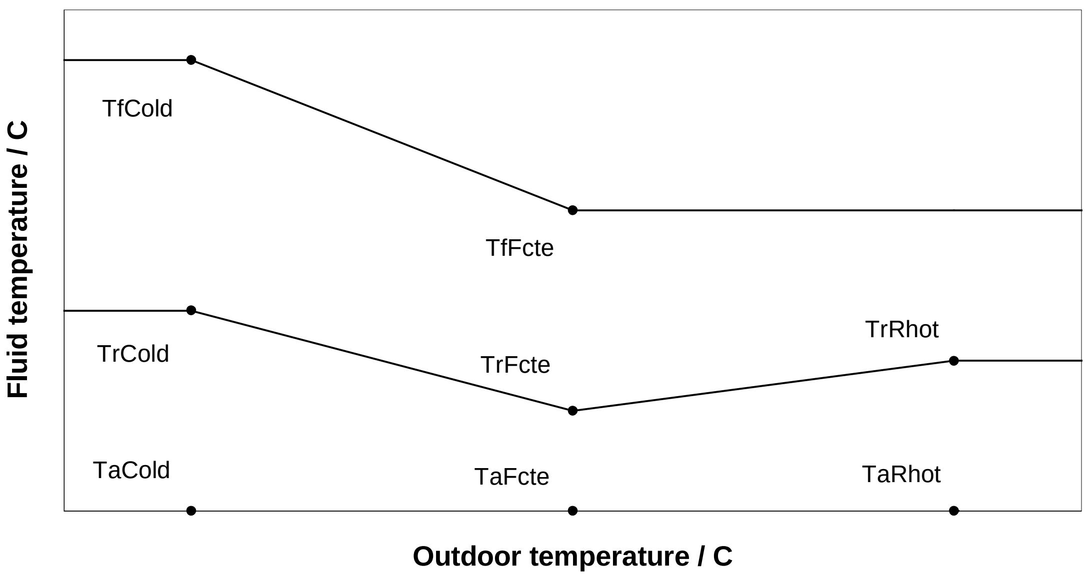


Fig. A4.2 The forward and return fluid temperatures in the distribution network are prescribed and depend on the outdoor air temperature.

## A4.2 Parameters, Inputs and Outputs of the Heat Load Component

**PARAMETERS (47):**

Forward and return fluid temperature in the distribution network:

1. TaCold: outdoor air temperature below which the forward and return fluid temperatures in the distribution network are constant [°C]
2. TfCold: forward fluid temperature corresponding to TaCold [°C]
3. TrCold: return fluid temperature corresponding to TaCold [°C]
4. TaFcte: outdoor air temperature over which the forward fluid temperature is constant[°C]
5. TfFcte: forward fluid temperature corresponding to TaFcte [°C]
6. TrFcte: return fluid temperature corresponding to TaFcte [°C]
7. TaRhot: outdoor air temperature over which the return fluid temperature is constant [°C]
8. TrRhot: return fluid temperature corresponding to TrRhot [°C]
(linear interpolation between the given temperatures)

Space heating requirement:

9. Tin: indoor set-point temperature [°C]
10. Hspec: total specific heat losses of the building(s) (transmission and ventilation losses) [kJ/hK]
11. TaNoSH: outdoor air temperature heat cut-out limit [°C]
12. DTinIG: correction to the indoor set-point temperature due to the internal gains [K]. DTinIG = Pint_gain [kJ/h] / Hspec [kJ/hK]
13. Aeff: solar effective area for the collection of passive solar gains [$m^2$]
14. Tau: time constant of the housing defined by the ratio Ceff/Hspec [h]. Ceff is an effective heat capacity that characterises the ability of the building(s) to store passive solar gains.
15. TinMax: maximum indoor temperature allowed [°C]
16. initial value of DTinSG: elevation of the indoor temperature due to the passive solar gains [K]

17. to 28. SHfact(i): monthly correction factors (Jan. to Dec.) to the hourly space heating values [-]

Domestic hot water requirement:

29. QDDHWR: reference daily hot water consumption, [kJ] per day

30. to 41. DHWfac(i): monthly correction factors (Jan. to Dec.) to the hourly domestic hot water values [-]

Distribution losses:

42. LDist: distribution network length (forward or return length pipes) [m]
43. HDistF: specific heat losses of the forward pipes [kJ/hmK]
44. HDistR: specific heat losses of the return pipes [kJ/hmK]
45. TDiRef: sink temperature for the heat losses calculation [°C]

Miscellaneous:

46. FlRef: reference flow rate for pump control signal [kg/h]
47. CFluid: specific heat of the heat carrier fluid in the distribution network [kJ/kgK]

**INPUTS (2):**

| | | |
|---|---|---|
| 1. | Tair: | outdoor air temperature [°C] |
| 2. | ITvs: | global radiation on a plane that corresponds to the definition of the solar effective area (usually vertical plane facing south) [kJ/m$^2$h] |

**OUTPUTS (8):**

| | | |
|---|---|---|
| 1. | Wload: | total flow rate in the distribution network [kg/h] |
| 2. | Tret: | return fluid temperature at the heating plant [°C] |
| 3. | Tfor: | forward fluid temperature from the heating plant [°C] |
| 4. | Pload: | total heat demand (space heating + domestic hot water + distribution losses) [kJ/h] |
| 5. | Psh: | space heating requirement [kJ/h] |
| 6. | Pdhw: | domestic hot water requirement [kJ/h] |
| 7. | Pdist: | heat losses in the distribution network [kJ/h] |
| 8. | Gpump: | control signal for the pump (relative to the reference flow rate FlRef) [-] |

## A4.3 Use of the Load Component for the Generation of the Load Data

The load model is used in its simplest form in this study. It is similar to the Degree-Day 10/18. The indoor set-point temperature is fixed to 20 °C, and the temperature reduction due to internal gains is set to 2K. The space heating heat demand is assumed to be null if the outdoor air temperature is above 10 °C. The parameters relative to the calculation of the passive solar gains are set to zero (Aeff, Tau and the initial value of DTinSG). The domestic hot water demand, if present, is assumed constant during the year.

The daily hot water heat demand and the total heat losses of the building(s) are adjusted so that the annual load is reproduced with the desired proportion of space heating and domestic hot water energy, given the hourly weather data. A spread-sheet EXCEL is used, which calculates and adds the 8'760 load values. The parameters of the load component can be adjusted in a practical and interactive way.

# A5. MISCELLANEOUS COMPONENTS

## A5.1 Auxiliary Heater Component

The code of the auxiliary heater TYPE 6 is rewritten so that auxiliary heat can be provided at any rate in order to meet exactly the requisite temperature level of the heat carrier fluid at any time. The code of the subroutine is given below:

```
        subroutine type 6  (time,xin,out,t,dtdt,par,info)
        dimension  par(20),xin(20), out(10),info(10)
        real qaux
        IF (INFO(7).GE.0) GO TO 1
C       FIRST CALL OF SIMULATION
        INFO(6)=3
        INFO(9)=1
        CALL TYPECK(1,INFO,3,2 ,0)
C       SET PARAMETER AND INPUT VARIABLES
1       QMAX=PAR(1)
        CP=PAR(2)

        tin=xin(1)
        flow=xin(2)
        tset=xin(3)

        if (tset .gt. tin) then
        Qaux=flow*cp*(tset -tin)
        else
        tset=tin
        qaux=0.
        endif
        out(1)=tset
        out(2)=flow
        out(3)=qaux
        return
        end
```

## A5.2 Temperature Controlled Flow Diverter Component

The code of the temperature controlled flow diverter component TYPE 11 is corrected. The old lines are:

```
        GAM = 1.0
        IF (THOT .GT. TSET) GAM = (TSET-TSOURC)/(THOT-TSOURC)
C***    NOTE CHANGE
        IF (THOT.LT.TSOURC .AND. MODE.EQ.4) GAM = 1.0
        IF (THOT.LT.TSOURC .AND. MODE.EQ.5) GAM = 0.0
C       MODE 4 AND 5 OUTPUTS
```

The new lines are (the correction are highlighted with bold characters):

```
      GAM = 1.0
      IF (THOT.NE.TSOURC) THEN
      IF (THOT .GT. TSET) GAM = (TSET-TSOURC)/(THOT-TSOURC)
      ENDIF
C***  NOTE CHANGE
      IF (THOT.LT.TSOURC .AND. MODE.EQ.4) GAM = 1.0
      IF (THOT.LT.TSOURC .AND. MODE.EQ.5) GAM = 0.0
      IF (TSOURC.GT.TSET) GAM = 0.0
C     MODE 4 AND 5 OUTPUTS
```

## A5.3 Heat Exchanger Component

The code of the heat exchanger component TYPE 5 is slightly modified.
The old lines are:

```
20    CHECK=ABS(1.0-RAT)
      IF(CHECK .LT. .01) GO TO 25
      EFF=(1.0-EXP(-UC*(1.0-RAT)))/(1.0-RAT*EXP(-UC*(1.0-RAT)))
```

The new lines are (the modification is highlighted with bold characters):

```
20    CHECK=ABS(1.0-RAT)
      IF(CHECK .LT. .0001) GO TO 25
      EFF=(1.0-EXP(-UC*(1.0-RAT)))/(1.0-RAT*EXP(-UC*(1.0-RAT)))
```

## A5.4 Algebraic Operations Component

The code of the algebraic operations component (TYPE 15) is slightly modified.
The old lines are:

```
C     GO THROUGH PARAMETER LIST
400   DO 800 P = 1,NPAR
      OP = PAR(P)
      IF (.NOT. CONST) GO TO 500
```

The new lines are (the modification is highlighted with bold characters):

```
C     GO THROUGH PARAMETER LIST
400   DO 800 P = 1,NPAR
      IF (.NOT. CONST) OP = PAR(P)
      IF (.NOT. CONST) GO TO 500
```

## A6. COLLECTOR ARRAY PIPES

The heat capacity of a collector has a negative influence on the overall collected heat, as the collector temperature, which is close to the air temperature at the beginning of a day, has to be raised to the operation temperature before heat can be collected. The heat capacitive effects may be significant for heat capacity values larger than 10 kJ/m$^2$K. A 10% reduction of the collected heat is possible with a collector heat capacity of 30 kJ/m$^2$K, if the collector is to be operated at high temperature (Pahud, 1995). The pipes of the collector array, which connect the collectors together and the collector field to the central heating plant, may significantly increase the heat capacity value per square meter of collector area. Nevertheless, the heat capacity due to the pipes may have less influence if the pipes are well insulated. The fluid in the pipes may still be hot the next day, when the operation of the collector array starts again.

In order to assess the heat capacitive effects of the pipes, the collector array is simulated with and without the main collecting pipes. When the pipes are not simulated, the pipe loss coefficient and heat capacity are included in the corresponding parameters of the collectors. The system design with a short-term buffer tank is used for the investigations.

### A6.1 Simulation of the Collector Subsystem

A large system is simulated, corresponding to a reference system defined in the study of Pahud (1995). The collector area corresponds to 35'000 m$^2$, the buffer volume is set to 5'000 m$^3$ (140 litres/m$^2$ of collector area) and the duct store uses 350'000 m$^3$ of ground. An annual load of 12 GWh is prescribed. Weather data corresponding to Göteborg (latitude: 57.8°) is used. Ten years are simulated from a cold start of the storages.

The collector subsystem is simulated in three different ways. In the first case, two pipes and two controllers are used. The tank pump, in the flow loop on the cold side of the solar heat exchanger, is controlled with its own controller. A delay may occur before the tank pump is switched on after the start of the collector pump. The dead band temperature differences of the collector and tank controllers are respectively set to 2 - 14 K and 0 - 2 K. The tank pump is switched off if the collector pump is stopped. The losses of the pipes are calculated relatively to the outdoor air temperature. The PRESIM representation of the collector subsystem is shown in Fig. A6.1.

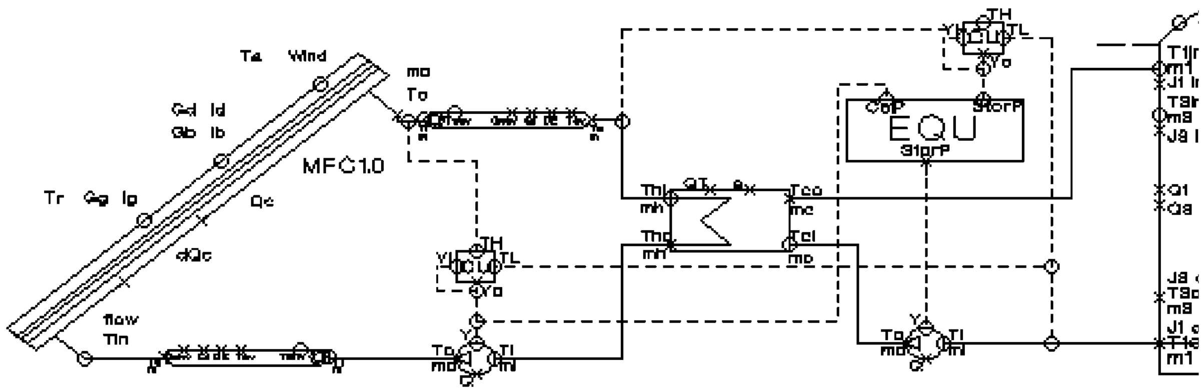


Fig. A6.1 Collector subsystem simulated with two pipes and two controllers.

The second case is simulated as for the first case, but with only the collector controller. The tank pump is controlled as for the collector pump, with the collector controller. The PRESIM representation of the collector subsystem for the second case is shown in Fig. A6.2.

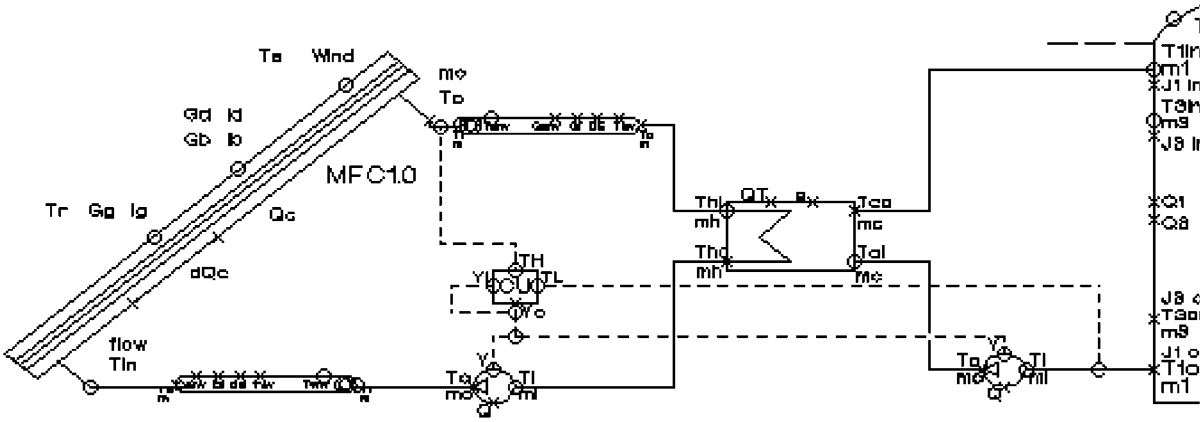


Fig. A6.2 Collector subsystem simulated with two pipes and one controller.

The third case is simulated without pipes. The heat capacity and the loss coefficient of the pipes are included in the corresponding collector parameters. The PRESIM representation of the collector subsystem for the third case is shown in Fig. A6.3.

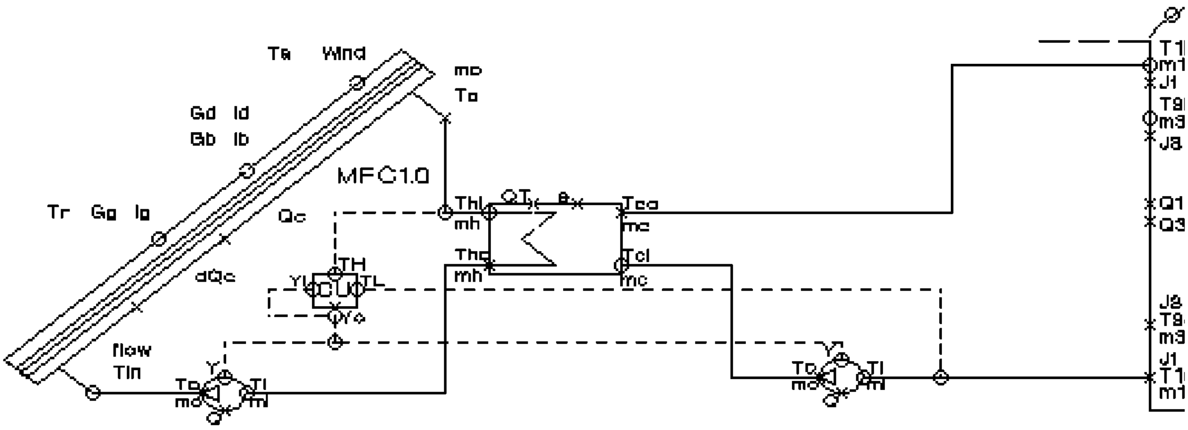


Fig. A6.3 Collector subsystem simulated without pipes.

## A6.2 Simulation Results

For convenience, the total pipe heat capacity and loss factor are expressed per square meter of collector area. The simulations are performed for a total heat capacity of the collectors and pipes of 20 and 30 kJ/m$^2$K, and a total overall loss coefficient of 3.5 W/m$^2$K. The collector heat capacity is set to 10 kJ/m$^2$K for all cases; (except when the pipe heat capacity is added to the heat capacity of the collector). For purposes of comparison, these values are kept constant for the three cases simulated and for different values of pipe loss coefficients. In consequence, larger pipe losses are compensated with smaller collector losses. The collector and pipe parameters are shown in Table A6.1 with their respective time constant for all the cases investigated. The time constant is defined by the ratio of the heat capacity by the loss coefficient. It determines the relaxation of the fluid temperature to the outdoor air temperature, in the absence of solar radiation and flow rate. The larger the time constant, the slower the fluid temperature will reach its asymptotic value. The mean collected heat during the 10 years of simulation is show in Table A6.2 for each case.

| Total collector array (collectors + pipes): **30 kJ/$m^2$K**<br>Collector heat capacity: 10 kJ/$m^2$K | | Total overall loss coefficient (collectors + pipes): **3.5 W/$m^2$K**<br>Pipe heat capacity: 20 kJ/K / $m^2$ collector | |
|---|---|---|---|
| Pipe losses [W/K / $m^2$ coll.] | Pipe time constant [hour] | Collector time constant [hour] | Overall time constant [hour] |
| 1.0 | 5.6 | 1.1 | 2.4 |
| 0.5 | 11.1 | 0.9 | 2.4 |
| 0.1 | 55.6 | 0.8 | 2.4 |
| Total collector array (collectors + pipes): **20 kJ/$m^2$K**<br>Collector heat capacity: 10 kJ/$m^2$K | | Total overall loss coefficient (collectors + pipes): **3.5 W/$m^2$K**<br>Pipe heat capacity: 10 kJ/K / $m^2$ collector | |
| Pipe losses [W/K / $m^2$ coll.] | Pipe time constant [hour] | Collector time constant [hour] | Overall time constant [hour] |
| 1.0 | 2.8 | 1.1 | 1.6 |
| 0.5 | 5.6 | 0.9 | 1.6 |

Table A6.1: Collector and pipe thermal characteristics of the investigated cases.

| Total collector array (collectors + pipes): **30 kJ/$m^2$K**<br>Collector heat capacity: 10 kJ/$m^2$K | | Total overall loss coefficient (collectors + pipes): **3.5 W/$m^2$K**<br>Pipe heat capacity: 20 kJ/K / $m^2$ collector | |
|---|---|---|---|
| Pipe losses [W/K / $m^2$ coll.] | Collected heat with 2 pipes + 2 controllers [kWh/$m^2$ / year] | Collected heat with 2 pipes + 1 controller [kWh/$m^2$ / year] | Collected heat without pipes [kWh/$m^2$ / year] |
| 1.0 | 338 98.7% | 332 96.8% | 343 100% |
| 0.5 | 339 99.0% | 336 97.9% | 343 100% |
| 0.1 | 350 102.3% | 350 102.2% | 343 100% |
| Total collector array (collectors + pipes): **20 kJ/$m^2$K**<br>Collector heat capacity: 10 kJ/$m^2$K | | Total overall loss coefficient (collectors + pipes): **3.5 W/$m^2$K**<br>Pipe heat capacity: 10 kJ/K / $m^2$ collector | |
| Pipe losses [W/K / $m^2$ coll.] | Collected heat with 2 pipes + 2 controllers [kWh/$m^2$ / year] | Collected heat with 2 pipes + 1 controller [kWh/$m^2$ / year] | Collected heat without pipes [kWh/$m^2$ / year] |
| 1.0 | 350 99.3% | 347 98.5% | 353 100% |
| 0.5 | 349 98.8% | 347 98.3% | 353 100% |

Table A6.2: Mean collected heat during the 10 years of simulation for all the investigated cases.

For the three different manners of simulating the collector subsystem, the annual collected heat is approximately the same; the differences do not exceed a few percent of the collected heat simulated with the pipe characteristics included in the collector parameters. If the collector pipes are simulated with two pipe components, the pumps of the two flow loops on both sides of the solar heat exchanger should be operated by two different controllers, but only if the loss coefficient of the pipes is important in relation to the overall loss factor of the collectors. The heat capacitive effects of the pipes are less important than those of the collectors, but only if the time constant of the pipes is larger than one or two days. Nevertheless, the error introduced by including the pipe characteristics in the collector parameter is small for commonly used values (< 2%).

As an example, the thermal characteristics of the Falkenberg array (5'500 $m^2$ of flat plate collectors) were estimated to 70 - 100 kJ/K and 0.33 W/K per linear meter of pipe (Isakson, 1995). The time constant of the pipes corresponds to 60 - 80 hours. With 720 m of forward and return fluid pipes, these values become 20 - 25 kJ/K / $m^2$ and 0.09 W/K / $m^2$, when expressed per square meter of collector area. This example is probably similar to the case simulated with a large pipe time constant. With a heat capacity of 30 kJ/$m^2$K, the simulated collected heat is probably 10% smaller than it would be without heat capacity, if the collector array operates at high temperature (80 - 100 °C). Nevertheless, the large time constant of the pipes would increase the simulated collected energy by around 2%, and the simulated collected heat would be in fact about 8% less than it would be without heat capacity.

# A7. CALCULATED OUTPUT QUANTITIES

The output quantities are produced with the help of the standard output components. They involve the use of simulation summaries, quantity integrators, algebraic operators and printers. The integrated quantities over one month or one year, together with the mean temperature levels and maximum or minimum temperatures during these time intervals, are written in a "∗.SUM" file. The evolution of different storage temperatures, printed every day, is written in a "∗.DAY" file. The names of these files (without extension), are the names of the corresponding deck file.

## A7.1 The "∗.SUM" Output File

*A7.1.1 Output quantities corresponding to the TRNSYS deck with a buffer store*

The quantities in the first part of the file, written with printer components, have no labels. In the first line, initial values of the maximum and minimum quantities are written. The second line contains some of the main parameters of the simulated system. The first value of the line corresponds to the initial time. The next 8 values are defined below, from left to right.

**Second line of the output file, second up to last value:**

| | |
|---|---|
| COAREA: | collector area of the collector field [m]; |
| BUFFV: | short-term buffer store volume [$m^3$]; |
| DUCTV: | duct store volume [$m^3$]; |
| DUCTH: | vertical extension of the duct store, or active length of one borehole [m]; |
| NBORE: | number of boreholes forming the ground heat exchanger of the duct store [-]; |
| DUCISO: | insulation thickness on top of the duct store [m]; |
| DUCFRI: | horizontal extension of the insulation layer from the edge of the store, expressed as a fraction its vertical extension [-]; |
| DUCDHP: | thickness of the ground layer on top of the duct store [m]. |

From the third line, maximum and minimum quantities during regular time intervals are printed. The end of the time interval is the first value of the line, given in hours. The next 8 values are defined below, from left to right.

**From the third line of the output file, second up to last value:**

| | |
|---|---|
| TCMAX: | maximum outlet fluid temperature from the collectors (before the pressure relief valve) [°C]; |
| TXMAX: | maximum inlet fluid temperature in the short-term buffer tank from the solar heat exchanger [°C]; |
| TDMAX: | maximum inlet fluid temperature in the ground heat exchanger [°C]; |
| TDMIN: | minimum inlet fluid temperature in the ground heat exchanger [°C]; |
| TMXMAX: | maximum average temperature of the short-term buffer store [°C]; |
| TMDMAX: | maximum average temperature of the duct store [°C]; |
| TMXMIN: | minimum average temperature of the short-term buffer store [°C]; |
| TMDMIN: | minimum average temperature of the duct store [°C]. |

The other output quantities are produced with simulation summaries and each have a specific label. They are:

**Collector subsystem simulation summary:**

COLIN: total incident insolation in the collector plane [MWh];
QCOMFC: collected heat by the collectors [MWh];
QDISS: wasted heat by the pressure relief valve [MWh];
QCOHX: collected heat transferred through the solar heat exchanger [MWh];
ERRCO%: heat balance of the collector subsystem: [%];
(QCOMFC - QDISS - QCOHX) / QCOHX
EFFCO%: mean efficiency of the collector: [%];
QCOHX / COLIN
QSCOL: collected heat per square meter collector: [kWh/m2];
QCOHX / COAREA
ERRSY%: heat balance of the system: [%];
(QCOMFC - QDISS - QDST + QLOSSX - QEXST + QBOIL - QHXLO) / QHXLO
(see below for the undefined labels)

**Short-term buffer tank simulation summary:**

QCOXST: heat from the collector subsystem to the buffer store [MWh];
QSTORL: heat to the duct store from the buffer store [MWh];
QSTORU: heat from the duct store to the buffer store [MWh];
QLOXST: heat to the load subsystem from the buffer store (called solar heat) [MWh];
QLOSSX: heat losses of the buffer store [MWh];
QEXST: variation of the internal energy of the buffer store [MWh];
ERRXS%: heat balance of the buffer store: [%];
(QCOXST + QSTORL + QSTORU + QLOSSX - QEXST) /
(0.5 · (|QCOXST| + |QSTORL| + |QSTORU| + |QLOSSX| + |QEXST|))
TFXST: mean temperature level of the outlet fluid from the buffer store to the load subsystem (see relation A7.1) [°C];
TRXST: mean temperature level of the inlet fluid to the buffer store from the load subsystem (see relation A7.1) [°C].

$$\mathrm{TFXST} = \frac{\sum_{i=1}^{N} \mathrm{TFXST_i} \cdot \mathrm{QLOXST_i}}{\sum_{i=1}^{N} \mathrm{QLOXST_i}} \quad \text{and} \quad \mathrm{TRXST} = \frac{\sum_{i=1}^{N} \mathrm{TRXST_i} \cdot \mathrm{QLOXST_i}}{\sum_{i=1}^{N} \mathrm{QLOXST_i}} \qquad \text{(A7.1)}$$

The sum is performed during the specified time interval (month, year or else), which contains *N* simulation time-steps. The subscript *i* refers to the calculated quantity at the time-step *i*.

**Duct store and load simulation summary:**

| | |
|---|---|
| QDST: | net heat transferred in the duct store (QSL - QSU, see below for the label definition) [MWh]; |
| QLOSS: | heat losses of the duct store [MWh]; |
| QEDST: | variation of the internal energy of the duct store [MWh]; |
| QELEC: | electric energy consumed for the operation of the duct store pumps [MWh]; |
| ERRDS%: | heat balance of the duct store: [%]; (QDST - QLOSS - QEDST) / (0.5·(\|QDST\| + \|QLOSS\| + \|QEDST\|)) |
| QBOIL: | net auxiliary heat delivered by the boiler [MWh]; |
| QHXLO: | heat transferred through the load heat exchanger to the distribution network (sum of the solar heat, QLOXST, and the auxiliary heat, QBOIL) [MWh]; |
| ERRLO%: | heat balance of the load subsystem: [%]; (QHXLO - QBOIL - QLOXST) / QHXLO |
| QLOAD: | annual heat demand (space heating, domestic hot water and distribution losses) from the load component [MWh]; |
| FLMET%: | fraction of the annual load met by the heating plant: [%]; QHXLO / QLOAD = (QLOXST + QBOIL) / QLOAD |

**Simulation summary for the mean temperature levels in the collector and load subsystems:**

Relation (A7.1) is used to calculate the mean temperature levels, with the corresponding fluid temperatures and associated heat fluxes. The heat flux related to the first 4 fluid temperatures is QCOHXi, the heat flux transferred through the solar heat exchanger. For the last 4 fluid temperatures it is QHXLOi, the heat flux transferred through the load heat exchanger.

| | |
|---|---|
| TCI: | mean temperature level of the outlet fluid from the collectors, after the pressure relief valve [°C]; |
| TCO: | mean temperature level of the inlet fluid in the collectors [°C]; |
| TXI: | mean temperature level of the inlet fluid in the buffer store from the solar heat exchanger [°C]; |
| TXO: | mean temperature level of the outlet fluid from the buffer store to the solar heat exchanger [°C]; |
| QCOLL: | heat transferred through the solar heat exchanger (equal to QCOHX) [MWh] |
| TBI: | mean temperature level of the inlet fluid in the load heat exchanger from the boiler (hot side) [°C]; |
| TBO: | mean temperature level of the outlet fluid from the load heat exchanger to the buffer store (hot side) [°C]; |
| TLI: | mean temperature level of the inlet fluid in the load heat exchanger from the heat distribution network (cold side) [°C]; |
| TLO: | mean temperature level of the outlet fluid from the load heat exchanger to the heat distribution network (cold side) [°C]; |
| QLOAD: | actual heat transferred to the distribution network (equal to QHXLO) [MWh]. |

**Simulation summary for the mean temperature levels in the storage subsystems:**

The heat flux related to the first 3 temperatures is QSLi, the heat flux injected in the duct store through the ground heat exchanger. For the temperatures TOMSU, TIMSU and TMSU it is QSUi, the heat flux extracted from the duct store through the ground heat exchanger.

| | |
|---|---|
| TIMSL: | mean temperature level of the inlet fluid in the ground heat exchanger during heat injection in the duct store (at the centre of the store) [°C]; |
| TOMSL: | mean temperature level of the outlet fluid from ground heat exchanger during heat injection in the duct store (at the border of the store) [°C]; |
| TMSL: | mean temperature level of the duct store during heat injection [°C]; |
| TMXST: | mean temperature of the buffer store [°C]; |
| QSL: | heat injected in the duct store [MWh]; |
| TOMSU: | mean temperature level of the outlet fluid in the ground heat exchanger during heat extraction from the duct store (at the centre of the store) [°C]; |
| TIMSU: | mean temperature level of the inlet fluid in the ground heat exchanger during heat extraction from the duct store (at the border of the store) [°C]; |
| TMSU: | mean temperature level of the duct store during heat extraction [°C]; |
| TMDST: | mean temperature of the duct store [°C]; |
| QSU: | heat extracted from the duct store [MWh]. |

*A7.1.2 Output quantities corresponding to the TRNSYS deck without a buffer store*

The same outputs are produced with the same labels, but without the quantities related to the buffer store. Nevertheless, QLOXST remains, and is calculated with QCOHX-QDST. The quantity QLOXST is the solar heat delivered to the distribution network.

*A7.1.3 Output quantities corresponding to the TRNSYS deck without a duct store*

The same outputs are produced with the same labels, but without the quantities related to the duct store.

## A7.2 The "∗.DAY" Output File

This output file is created with a printer which writes different store temperatures at regular time intervals (set to 1 day). The start and stop times that define the time interval for the printing of the temperatures are to be set by the user. They are the constant TEMPRI and TEMPRO defined at the beginning of the TRNSYS deck. If both the buffer store and the duct store are present in the system, six temperatures are printed. They are:

| | |
|---|---|
| TXSTMA: | fluid temperature at the top of the buffer tank [°C]; |
| TXSTAV: | mean temperature of the buffer store [°C]; |
| TXSTMI: | fluid temperature at the bottom of the buffer tank [°C]; |
| TDSTCE: | average soil temperature near the boreholes in radial subregion 1 (at the centre of the store) [°C]; |
| TDSTAV: | mean temperature of the duct store [°C]; |
| TDSTBO: | average soil temperature near the boreholes in radial subregion N (at the border of the store) [°C]; |

# A8. TRNSYS DECK LISTING

The listing corresponds to the TRNSYS deck of the system simulated with the S100LOW load type: small heat load, 500 MWh/y, 100% space heating, low-temperature heat distribution. The system parameters are set to obtain a minimum solar cost at a solar fraction of 70%; (see section 8.2).

```
ASSIGN optsloa.LST 6
*            Start time  End time   Time step
SIMULATION        0      105120       0.25
*            Integration Convergence
TOLERANCES      -0.05      -0.05
*
*            Max iterations;  Max warnings; Trace limit;
LIMITS           100             20            100
*
*            TRNSYS output file width, number of
characters
WIDTH            72
*
*            TRNSYS numerical integration solver method
DFQ              1
*
*  EQUATIONS in Simulation Control Component
EQUATIONS 31
TIMSTP = 0.25
PRNINT = 8760
MAXPRT = 8760
TEMPRI = 0
TEMPRO = 0
COAREA = 1200.00000
TILT   = 45
AZIM   = 0
COLHX  = 100*COAREA*3.6
FLOCOL = 0.007*COAREA*3600
DFCOL  = 1050
CPFCOL = 3.8
DWA    = 1000
CPW    = 4.19
BUFFV  = 132.00000
BUFFH  = 2.*EXP(LN(BUFFV/6.283185)/3.)
BUFFP  =
2.*EXP(0.5*LN(BUFFV*3.1415927/BUFFH))
CVBUF  = DWA * CPW
BUFTOP = (BUFFV/BUFFH) * (0.05/0.2) * 3.6
BUFSID = (BUFFP*BUFFH) * (0.05/0.2) * 3.6
BUFBOT = (BUFFV/BUFFH) * (0.05/0.2) * 3.6
DUCTV  = 12600.00000
DUCTH  = 42.89362
NBORE  = 47.00000
NSERIE = 3
DUCISO = 0.2
DUCFRI = 0.05
DUCDHP = 1
FLLDST = FLOCOL/2.
FLOMAX = 28500
LOADHX = 45000 * 3.6E15
*
*  EQUATIONS in Equation Components
EQUATIONS 3
TSETBO = [10,3] + 0.01
RESET  = LT(MOD(TIME-1,MAXPRT),0.00001)
IbTCol = [1,3] - [1,4]
*
UNIT 1   TYPE 9  METEON95, standard format
PRESIM TYPE 1109
PARAMS 22
*     Number of values per card to read
       6
*     Data file time interval
       1
*     Interp   Mult factor    Add fact
      -1     3.5997         0
      -2     3.5997         0
      -3     3.5997         0
      -4     3.5997         0
      -5     3.5997         0
       6          1         0
*     File logical unit #
       21
*     Format indicator. (> 0 specifies formatted reading)
       1
(21X,F5.0 ,2X,F5.0,2X,F5.0,2X,F5.0,2X,F5.0,2X,F6.1)
ASSIGN METEON.LST        21
*
UNIT 2   TYPE 16  Sol Rad Proc, I and Idn   PRESIM
TYPE 216
PARAMS 8
*    1  I and Idn are inputs
         7
*    2  Tracking mode:
         1
*    3  Surface Radiation mode
         4
*    4  Day of the year when simulation starts
         1
*    5  Latitude
         46
*    6  Solar constant (normally 4871.1 kJ/h-m2)
      4871.1
*    7  Shift in solar time hour angle (degrees)
         -8
*    8  Treat simulation time as solar time
         1
INPUTS 7
*  1  Radiation on horizontal surface..from..METEON95,
standard format " I "
    1,  1
*  2  Direct normal beam radiation..from..METEON95,
standard format " Ibn "
    1,  5
*  3  Time of last radiation data
reading..from..METEON95, standard format
*        " td1 "
    1, 19
*  4  Time of next radiation data
reading..from..METEON95, standard format
*        " td2 "
    1, 20
*  5  Ground reflectance  Not connected. Fix value
    0,  0
*  6  Slope of surface or tracking axis  Not connected.
Fix value
    0,  0
*  7  Azimuth of surface or tracking axis  Not connected.
Fix value
    0,  0
*   INPUT INITIAL VALUES
*    1
         0
*    2
         0
*    3
         0
*    4
         1
*    5
         0
*    6
       TILT
*    7
       AZIM
*
UNIT 3   TYPE 2     PRESIM TYPE 19
PARAMS 3
*    1  Number of oscillations in a timestep
         3
*    2  Upper dead band temperature difference
         14
*    3  Lower dead band temperature difference
         2
INPUTS 3
*  1  Upper input temperature..from..MFC1.0 2, *, 1, 0, *
" To "
    5,  1
*  2  Lower input temperature..from..Type74/ITW " Td1
"
    9,  1
*  3  Input control function..from.. " Yo "
    3,  1
*   INPUT INITIAL VALUES
*    1
        20
*    2
        20
*    3
         0
*
UNIT 4   TYPE 3  Pump   PRESIM TYPE 20
PARAMS 4
*    1  Maximum flow rate  [ mf(max) ]
      FLOCOL
*    2  Maximum power consumption  [ P(max) ]
         0
*    3  C0.  Constant in relation P = C0 + C1 * mf
         0
*    4  C1.  Constant in relation P = C0 + C1 * mf
         0
INPUTS 3
*  1  Inlet fluid temperature..from.. " Tho "
    7,  1
*  2  Inlet mass flow rate..from.. " mh "
    7,  2
*  3  Control function..from.. " Yo "
    3,  1
*   INPUT INITIAL VALUES
*    1
        20
*    2
         0
*    3
         0
*
UNIT 5   TYPE 52  MFC1.0 2, *, 1, 0, *   PRESIM
TYPE 4521
PARAMS 18
*    1  Collector mode, Cmode=2
         2
*    2  Dynamic mode, Dmode
         0
*    3  Collector apperture area m2
      COAREA
*    4  Effective heat capacity of collect. kJ/m2K
         20
*    5  Fluid volume of collector channels (eq. 5)
    0.001253
*    6  Specific heat of collector fluid kJ/kgK
      CPFCOL
*    7  Density of collector fluid kg/m3
       DFCOL
*    8  Zero loss efficiency, n0
        0.81
*    9  Algebraic constant, a1
      15.479
```

*    10  Algebraic constant, a2  kJ/hK2m2
0.0216
*    11  U-value. Wind dependence, aw
0
*    12  U-value. Long-wave dependence, ar
0
*    13  Heat transfer coeff. fluid to plate (Dm=1)
-3596.1
*    14  Not used, NA
-999
*    15  Solar radiation mode, Smode= 0
0
*    16  Test mode, Tmode
0
*    17  Optical mode, Omode
1
*    18  IAM parameter (b0 / r )
0.11
INPUTS 11
* 1  Collector inlet temperature      ..from..Pump " To "
4,  1
* 2  Inlet flow rate to collector..from..Pump " mo "
4,  2
* 3  Ambient temperature..from..METEON95, standard format " Ta "
1,  6
* 4  Effective long wave radiation temp  Not connected. Fix value
0,  0
* 5  Wind speed  Not connected. Fix value
0,  0
* 6  Incident beam radiation ..from..L  LBL " IbTCOL " IbTCol
* 7  Incident angle of Gb..from..Sol Rad Proc, I and Idn " ai "
2,  9
* 8  Incident diffuse radiation ..from..METEON95, standard format " IdT "
1,  4
* 9  Incident angle of Gd  Not connected. Fix value
0,  0
* 10  Incident ground reflected radiation  Not connected. Fix value
0,  0
* 11  Incident angle of Gg  Not connected. Fix value
0,  0
*   INPUT INITIAL VALUES
*    1
0
*    2
0
*    3
0
*    4
0
*    5
0
*    6
0
*    7
0
*    8
0
*    9
56.9
*    10
0
*    11
0
*
UNIT 6   TYPE 13  Pressure relief valve   PRESIM TYPE 4001
PARAMS 2
*    1  Boiling point of fluid
100
*    2  Specific heat of fluid
CPFCOL
INPUTS 3
* 1  Inlet fluid temperature..from..MFC1.0 2, *, 1, 0, * " To "
5,  1
* 2  Inlet mass flow rate..from..MFC1.0 2, *, 1, 0, * " mo "
5,  2
* 3  Comparison temperature..from..MFC1.0 2, *, 1, 0, * " To "
5,  1
*   INPUT INITIAL VALUES
*    1
0
*    2
0
*    3
0
*
UNIT 7   TYPE 5     PRESIM TYPE 26
PARAMS 4
*    1  Heat exchanger flow mode:
2
*    2  Overall heat transfer coefficient
COLHX
*    3  Specific heat of hot side fluid
CPFCOL
*    4  Specific heat of cold side fluid
CPW
INPUTS 4
* 1  Hot side inlet temperature..from..Pressure relief valve " To "
6,  1
* 2  Hot side mass flow rate..from..Pressure relief valve " mo "
6,  2
* 3  Cold side inlet temperature..from..Pump " To "
8,  1
* 4  Cold side mass flow rate..from..Pump " mo "
8,  2
*   INPUT INITIAL VALUES
*    1
20
*    2
0
*    3
20
*    4
0
*
UNIT 8   TYPE 3  Pump   PRESIM TYPE 20
PARAMS 4
*    1  Maximum flow rate  [ mf(max) ]
FLOCOL
*    2  Maximum power consumption  [ P(max) ]
0
*    3  C0.  Constant in relation P = C0 + C1 * mf
0
*    4  C1.  Constant in relation P = C0 + C1 * mf
0
INPUTS 3
* 1  Inlet fluid temperature..from..Type74/ITW " Td1 "
9,  1
* 2  Inlet mass flow rate..from..Type74/ITW " md1 "
9,  2
* 3  Control function..from.. " Yo "
3,  1
*   INPUT INITIAL VALUES
*    1
20
*    2
0
*    3
0
*
UNIT 9   TYPE 74  Type74/ITW   PRESIM TYPE 4074
PARAMS 82
*    1  storage height
BUFFH
*    2  storage volume
BUFFV
*    3  Specific heat of fluid
CPW
*    4  Density of fluid
DWA
*    5  effective vert. therm cond. in storage
2.1598
*    6  not used
0
*    7  initial temp. of the whole system
10
*    8  heat loss through bottom
BUFBOT
*    9  heat loss through top
BUFTOP
*    10  rel length 1st zone
1
*    11  heat loss through 1st zone
BUFSID
*    12  rel. length 2nd zone
0
*    13  heat loss through 2nd zone
0
*    14  rel. length 3rd zone
0
*    15  heat loss through 3rd zone
0
*    16  heat loss through 4th zone
0
*    17  rel. height inlet 1st doubleport
0.95
*    18  rel. height outlet 1st doubleport
0.05
*    19  stratified charge 1st dp? yes=1
0
*    20  rel. height inlet 2nd dp
0.05
*    21  rel. height outlet 2nd dp
0.95
*    22  strat. charging 2nd dp? yes=1
0
*    23  rel. height inlet 3rd dp
0.05
*    24  rel. height outlet 3rd dp
0.95
*    25  strat. charging 3rd dp? yes=1
0
*    26  rel. height inlet 4th dp
0.95
*    27  rel. height outlet 4th dp
0.05
*    28  strat. charging 4th dp? yes=1
0
*    29  rel. height inlet 5th dp
-1
*    30  rel. height outlet 5th dp
-1
*    31  strat. charging 5th dp? yes=1
0
*    32  rel. position temp. sensor 1
0.05
*    33  rel. pos. temp. sensor 2
0.35
*    34  rel. pos. temp. sensor 3
0.55
*    35  rel. pos. temp. sensor 4
0.75
*    36  rel. pos. temp. sensor 5
0.95
*    37  aux heater mode (0=no,1=Pvar,2=Pconst)
0
*    38  aux heater installed from top? yes=1
0
*    39  absolute length aux heater (if 38=1)
0
*    40  rel. pos. aux heater (if 38 not 1)
0
*    41  rel. pos. of temp. controller to aux heatr
0
*    42  set temp. for aux heater (if 37=2)
0
*    43  dead band temp. of the controller (if38=2)
0
*    44  rel. inlet pos. 1st hx
-1
*    45  rel. outlet pos. 1st hx
-1
*    46  Volume of 1st hx
0
*    47  specific heat of fluid in 1st hx
0
*    48  density of fluid in 1st hx
0
*    49  UA- number from 1st hx to storage
0
*    50  b1.1

0
*   51  b2.1
0
*   52  b3.1
0
*   53  heat loss from 1st hx to ambient
0
*   54  stratified charging with 1st hx? yes=1
0
*   55  rel. inlet pos. 2nd hx
-1
*   56  rel. outlet pos. 2nd hx
-1
*   57  Volume 2nd hx
0
*   58  specific heat of fluid in 2nd hx
0
*   59  density of fluid in 2nd hx
0
*   60  UA- number from 2nd hx to storage
0
*   61  b1.2
0
*   62  b2.2
0
*   63  b3.2
0
*   64  heat loss from 2nd hx to ambient
0
*   65  stratified charging with 2nd hx? yes=1
0
*   66  rel. inlet pos. 3rd hx
-1
*   67  rel outlet pos. 3rd hx
-1
*   68  Volume of 3rd hx
0
*   69  specific heat of fluid in 3rd hx
0
*   70  density of fluid in 3rd hx
0
*   71  UA- number from 3rd hx to storage
0
*   72  b1.3
0
*   73  b2.3
0
*   74  b3.3
0
*   75  heat loss from 3rd hx to ambient
0
*   76  stratified charging with 3rd hx? yes=1
0
*   77  accuracy for calculating temperatures
0.01
*   78  accuracy for UA when using b*.*
1
*   79  precision of mixing process in the tank
1
*   80  flag if temp. dependence timestep control
0
*   81  Number of nodes in the storage
3
*   82  not used
0
INPUTS 18
*  1  Temp. in 1st doubleport..from.. " Tco "
7,  3
*  2  Fluid rate 1st doubleport..from.. " mc "
7,  4
*  3  Temp. in 2nd doubleport..from..Flow diverter, flow&temp " T2 "
21,  3
*  4  Fluid rate 2nd doubleport..from..Flow diverter, flow&temp " m2 "
21,  4
*  5  Temp. in 3rd..from..Temp ctrl flow div,mf&tmp " T1 "
12,  1
*  6  Fluid flow 3rd..from..Temp ctrl flow div,mf&tmp " m1 "
12,  2
*  7  Temp. in 4th..from..Flow diverter, flow&temp " T1 "
21,  1
*  8  Fluid flow 4th..from..Flow diverter, flow&temp " m1 "
21,  2
*  9  Temp. in 5th  Not connected. Fix value
0,  0
*  10  Fluid flow 5th  Not connected. Fix value
0,  0
*  11  Temp. in 1st heat exchanger  Not connected. Fix value
0,  0
*  12  Fluid flow 1st heat exchanger  Not connected. Fix value
0,  0
*  13  Temp. in 2nd hx  Not connected. Fix value
0,  0
*  14  Fluid flow 2nd hx  Not connected. Fix value
0,  0
*  15  Temp. in 3rd hx  Not connected. Fix value
0,  0
*  16  Fluid flow 3rd hx  Not connected. Fix value
0,  0
*  17  Temp. environment  Not connected. Fix value
0,  0
*  18  Aux heater  Not connected. Fix value
0,  0
*   INPUT INITIAL VALUES
*    1
0
*    2
0
*    3
0
*    4
0
*    5
0
*    6
0
*    7
0
*    8
0
*    9
0
*    10
0
*    11
0
*    12
0
*    13
0
*    14
0
*    15
0
*    16
0
*    17
10
*    18
0
*
UNIT 10   TYPE 59  Simple load model   PRESIM TYPE 4059
PARAMS 47
*    1  Outdoor temp. below which cte for&ret temp
-10
*    2  Corresponding forward fluid temperature
30
*    3  Corresponding return fluid temperature
23
*    4  Outdoor temp. over which cte forward temp.
10
*    5  Corresponding forward fluid temperature
25
*    6  Corresponding return fluid temperature
22
*    7  Outdoor temp. over which cte return temp.
20
*    8  Corresponding return fluid temperature
22
*    9  Indoor set point temperature
20
*    10  Total specific heat loss.H(transm.+air ch)
28258
*    11  Outdoor heating-cut-off temperature
10
*    12  Temperature reduction due to internal gain
2
*    13  Solar effective area relatively to ITvs
0
*    14  Housing time constant(Ceff[kJ/K]/H[kJ/hK])
0
*    15  Maximum indoor temperature allowed
25
*    16  Initial elevation of temp. dueto sol. gain
0
*    17  Monthly correction factor for SH, January
1
*    18  Monthly correction factor for SH, February
1
*    19  Monthly correction factor for SH, March
1
*    20  Monthly correction factor for SH, April
1
*    21  Monthly correction factor for SH, May
1
*    22  Monthly correction factor for SH, June
0
*    23  Monthly correction factor for SH, July
0
*    24  Monthly correction factor for SH, August
0
*    25  Monthly correction factor for SH, Sept.
0
*    26  Monthly correction factor for SH, October
1
*    27  Montlly correction factor for SH, November
1
*    28  Monthly correction factor for SH, December
1
*    29  Reference daily hot water requirement
0
*    30  Monthly correction factor for DHW, January
0
*    31  Monthly correction factor for DHW, Feb.
0
*    32  Monthly correction factor for DHW, March
0
*    33  Monthly correction factor for DHW, April
0
*    34  Monthly correction factor for DHW, May
0
*    35  Monthly correction factor for DHW, June
0
*    36  Monthly correction factor for DHW, July
0
*    37  Monthly correction factor for DHW, August
0
*    38  Monthly correction factor for DHW, Sep.
0
*    39  Monthly correction factor for DHW, October
0
*    40  Monthly correction factor for DHW, Nov.
0
*    41  Monthly correction factor for DHW, Dec.
0
*    42  Distribution network length
200
*    43  Specific losses of the forward pipes
1.7999
*    44  Specific losses of the return pipes
1.7999
*    45  Mean sink temp. for distr. losses calcul.
10
*    46  Reference flow rate for pump controlsignal
FLOMAX
*    47  Heat capacity of the fluid in distr. netw.
CPW
INPUTS 2
*  1  Outdoor temperature..from..METEON95, standard format " Ta "
1,  6
*  2  Global radiat. on south vert. plane  Not connected. Fix value
0,  0

```
*   INPUT INITIAL VALUES
*     1
          0
*     2
          0
*
UNIT 11   TYPE 5     PRESIM TYPE 26
PARAMS 4
*     1  Heat exchanger flow mode:
          2
*     2  Overall heat transfer coefficient
      LOADHX
*     3  Specific heat of hot side fluid
        CPW
*     4  Specific heat of cold side fluid
        CPW
INPUTS 4
*  1  Hot side inlet temperature..from.. " Tout "
   15,  1
*  2  Hot side mass flow rate..from.. " Mout "
   15,  2
*  3  Cold side inlet temperature..from..Simple load
model " Tret "
   10,  2
*  4  Cold side mass flow rate..from..Simple load model
" Wload "
   10,  1
*   INPUT INITIAL VALUES
*     1
          0
*     2
          0
*     3
          0
*     4
          0
*
UNIT 12   TYPE 11  Temp ctrl flow div,mf&tmp
PRESIM TYPE 47
PARAMS 2
*     1  Outlet order
          5
*     2  Number of iterations at the same timestep
         15
INPUTS 4
*  1  Inlet temperature..from.. " Tho "
   11,  1
*  2  Inlet mass flow rate..from.. " mh "
   11,  2
*  3  Heat source temperature..from..Type74/ITW " Td3
"
    9,  5
*  4  Desired temperature of mixed fluid  Not connected.
Fix value
    0,  0
*   INPUT INITIAL VALUES
*     1
         30
*     2
       7063
*     3
         47
*     4
        100
*
UNIT 13   TYPE 11  Tee piece mode 1   PRESIM
TYPE 44
PARAMS 1
*     1  Tee Piece, temperature related
          1
INPUTS 4
*  1  Temperature at inlet 1..from..Type74/ITW " Td3 "
    9,  5
*  2  Mass flow rate at inlet 1..from..Type74/ITW " md3
"
    9,  6
*  3  Temperature at inlet 2..from..Temp ctrl flow
div,mf&tmp " T2 "
   12,  3
*  4  Mass flow rate at inlet 2..from..Temp ctrl flow
div,mf&tmp " m2 "
   12,  4
*   INPUT INITIAL VALUES
*     1
         47
*     2
       7063
*     3
         30
*     4
          0
*
UNIT 14   TYPE 6     PRESIM TYPE 25
PARAMS 2
*     1  Max Heating Power
      719.94
*     2  Fluid specific heat
        CPW
INPUTS 3
*  1  Inlet temperature..from..Tee piece mode 1 " To "
   13,  1
*  2  Inlet fluid mass flow rate..from..Tee piece mode 1 "
mo "
   13,  2
*  3  Set point temperature..from..EQUATION "
TSETBO "
TSETBO
*   INPUT INITIAL VALUES
*     1
         60
*     2
      13000
*     3
         60
*
UNIT 15   TYPE 67     PRESIM TYPE 4067
PARAMS 2
*     1  Maximal mass flow rate
      FLOMAX
*     2  Minimal mass flow rate
          0
INPUTS 9
*  1  Inlet temperature in valve..from.. " To "
   14,  1
*  2  Inlet flow rate in valve..from.. " mo "
   14,  2
*  3  Inlet temperature in HX (cold side)..from..Simple
load model " Tret "
   10,  2
*  4  Inlet mass flow rate in HX (cold s)..from..Simple
load model " Wload "
   10,  1
*  5  Outlet temp. from HX (hot side)..from.. " Tho "
   11,  1
*  6  Set temperature controlled..from..Simple load
model " Tfor "
   10,  3
*  7  Outlet controlled temp. (HX cold s)..from.. " Tco "
   11,  3
*  8  External control signal (ON>0)     Not connected.
Fix value
    0,  0
*  9  Error for mass flow rate..from.. " Err "
   15,  4
*   INPUT INITIAL VALUES
*     1
         60
*     2
       5000
*     3
         40
*     4
       5000
*     5
         45
*     6
         50
*     7
         50
*     8
          1
*     9
          0
*
UNIT 16   TYPE 64  PRESSURE DROP GROUND HX
PRESIM TYPE 4064
PARAMS 14
*     1  Length of one borehole
      DUCTH
*     2  Total number of boreholes
      NBORE
*     3  Number of bore. connected in series NSerie
      NSERIE
*     4  Overall efficiency of "pump+electr. motor"
        0.4
*     5  Fluid dens. kg/m3;0:water;<0:LU data file
          0
*     6  Dyn.visc.kg/ms;wat.:not used;LU numb. data
          0
*     7  Bore inst.(0:coaxial;1:1U-p.;2:2U-p.;etc.)
          2
*     8  Internal diameter (U-pipe or central pipe)
      0.026
*     9  Effective roughness of pipe;1.5e-6:plastic
   1.5E-006
*     10  External diam. of central pipe;U-p:notused
      0.063
*     11  Internal diam. of outer pipe;U-p.:not used
      0.115
*     12  Effective roughness of annulus;U-p:notused
     0.0001
*     13  Sum of loss values  per borehole (bend,..)
          3
*     14  Number of pipe types in a branch on surf.
          0
INPUTS 4
*  1  Inlet fluid temp. in ground hx...from..Tee piece
mode 1 " To "
   24,  1
*  2  Outlet fluid temp. from ground hx...from..DST
TRNSYS " Tout "
   17,  1
*  3  Fluid flow rate..from..Tee piece mode 1 " mo "
   24,  2
*  4  Reset to 0 the maximum output val.   Not
connected. Fix value
    0,  0
*   INPUT INITIAL VALUES
*     1
         20
*     2
         20
*     3
          0
*     4
          0
*
UNIT 17   TYPE 60  DST TRNSYS   PRESIM TYPE
4444
PARAMS 46
*     1  Volume of storage
      DUCTV
*     2  Storage height
      DUCTH
*     3  DPH dist. between surface and upper side
     DUCDHP
*     4  number of bore hole : nb
      NBORE
*     5  Outer radius of each borehole
     0.0575
*     6  Number of bore. connected in series NSerie
     NSERIE
*     7  Number of radial subregions NRLoc<=NSerie
          3
*     8  Number of vertical subregions NZLoc
          3
*     9  Thermal conductivity of stor. mat.: ks
     8.9993
*     10  Volumetric heat capacity of stor. mat.: Cs
       2300
*     11  Therm. res. fluid pipe and ground : Rp
    0.02778
*     12  Therm. res. between up. and down. fluid:Ra
      -0.11
*     13  Spec. thermal capacity of pipe fluid : cf
        CPW
*     14  Mass density of fluid : rf
        DWA
*     15  ISO /0/1/2
          2
*     16  Friso : frac. of the height of stock
     DUCFRI
```

*    17  Thickness of insulation : THins
     DUCISO
*    18  Thermal cond. of insulation : kins
    0.17999
*    19  Number of simulation years : sy
       20
*    20  Max temp of storage : Tmax
       90
*    21  Init T of undisturbed ground surf.: Tgs
       10
*    22  Init thermal grad. in undist ground: dtg
       0
*    23  n. of preheating cycles:ipre (nb. of year)
       0
*    24  Max temp during ipre
       60
*    25  Min temp during ipre
       46
*    26  Phase delay in the sin. variation (days)
      183
*    27  Yearly average air Temp. : Tma
       10
*    28  Yearly amplitude of the air temp. : Taa
       10
*    29  Phase delay in the air sin. variat. (days)
      213
*    30  nb of ground layers : Nl
       1
*    31  Thermal conduct. of ground layer : kg
     8.9993
*    32  Vol. therma capacity of the layer : cg
     2300
*    33  Thickness of the layer : THg
     1000
*    34  IPRT 0/1 : print input data (special pr.)
       1
*    35  oprt 0/1 : print output data
       0
*    36  Output logical unit for iprt
       30
*    37  IP1 0/1 : print Tair,Tav.st.,Tout,...
       0
*    38  DT1 : time interval between IP1 print.
      720
*    39  IP2 : print global temp. field
       0
*    40  DT2 : time interval between IP2 print
      720
*    41  IP3 : print heat flow fields
       0
*    42  DT3 : time interval between IP3 print
      720
*    43  IP4 : print local temp. field
       0
*    44  DT4 : time interval between IP4 print
      720
*    45  Ibloss : print losses through sto. bound.
       0
*    46  nspec : nb of stor. locations for T print
       0
INPUTS 5
*  1  Inlet fluid temperature..from..Tee piece mode 1 " To "
   24,  1
*  2  Fluid flow rate..from..Tee piece mode 1 " mo "
   24,  2
*  3  Air temperature on top of store ..from..METEON95, standard format
*       " Ta "
   1,  6
*  4  Air temp. elsewhere on ground surf...from..METEON95, standard format
*       " Ta "
   1,  6
*  5  Fluid circulation:>0 cent. to bord...from.. " Gload "
   20,  1
*   INPUT INITIAL VALUES
*    1
       20
*    2
       0
*    3
       10
*    4
       10
*    5
       1
*
UNIT 18   TYPE 2     PRESIM TYPE 19
PARAMS 3
*    1  Number of oscillations in a timestep
       3
*    2  Upper dead band temperature difference
       5
*    3  Lower dead band temperature difference
       1
INPUTS 3
*  1  Upper input temperature..from..Type74/ITW " Ts5 "
   9, 33
*  2  Lower input temperature..from..DST TRNSYS " Tout "
   17,  1
*  3  Input control function..from.. " Yo "
   18,  1
*   INPUT INITIAL VALUES
*    1
       15
*    2
       60
*    3
       0
*
UNIT 19   TYPE 2     PRESIM TYPE 19
PARAMS 3
*    1  Number of oscillations in a timestep
       3
*    2  Upper dead band temperature difference
       5
*    3  Lower dead band temperature difference
       1
INPUTS 3
*  1  Upper input temperature..from..DST TRNSYS " Tout "
   17,  1
*  2  Lower input temperature..from..Type74/ITW " Ts5 "
   9, 33
*  3  Input control function..from.. " Yo "
   19,  1
*   INPUT INITIAL VALUES
*    1
       15
*    2
       60
*    3
       0
*
UNIT 20   TYPE 61     PRESIM TYPE 4061
PARAMS 1
*    1  Fpump: multiplicative factor for Qpump
       1
INPUTS 6
*  1  Control signal for the load. pump..from.. " Yo "
   18,  1
*  2  Control signal associated to Gload  Not connected. Fix value
   0,  0
*  3  Control signal for the unload. pump..from.. " Yo "
   19,  1
*  4  Control signal associated to Gunlo..from.. " Gamma "
   15,  3
*  5  Heat rate transf. in the duct store..from..DST TRNSYS " Q "
   17,  4
*  6  Heat rate consumed by the pump ..from..PRESSURE DROP GROUND HX
*       " Qel "
   16,  4
*   INPUT INITIAL VALUES
*    1
       0
*    2
       1
*    3
       0
*    4
       1
*    5
       0
*    6
       0
*
UNIT 21   TYPE 11  Flow diverter, flow&temp PRESIM TYPE 45
PARAMS 1
*    1  Mode 2: Flow diverter, temperature related
       2
INPUTS 3
*  1  Entering fluid temperature..from..DST TRNSYS " Tout "
   17,  1
*  2  Entering fluid mass flow rate..from..DST TRNSYS " Mdot "
   17,  2
*  3  Control function..from.. " Gload "
   20,  1
*   INPUT INITIAL VALUES
*    1
       0
*    2
       0
*    3
       0
*
UNIT 22   TYPE 3  Pump   PRESIM TYPE 20
PARAMS 4
*    1  Maximum flow rate  [ mf(max) ]
     FLLDST
*    2  Maximum power consumption  [ P(max) ]
       0
*    3  C0.  Constant in relation P = C0 + C1 * mf
       0
*    4  C1.  Constant in relation P = C0 + C1 * mf
       0
INPUTS 3
*  1  Inlet fluid temperature..from..Type74/ITW " Td2 "
   9,  3
*  2  Inlet mass flow rate..from..Type74/ITW " md2 "
   9,  4
*  3  Control function..from.. " Gload "
   20,  1
*   INPUT INITIAL VALUES
*    1
       0
*    2
       0
*    3
       0
*
UNIT 23   TYPE 3  Pump   PRESIM TYPE 20
PARAMS 4
*    1  Maximum flow rate  [ mf(max) ]
     FLOMAX
*    2  Maximum power consumption  [ P(max) ]
       0
*    3  C0.  Constant in relation P = C0 + C1 * mf
       0
*    4  C1.  Constant in relation P = C0 + C1 * mf
       0
INPUTS 3
*  1  Inlet fluid temperature..from..Type74/ITW " Td4 "
   9,  7
*  2  Inlet mass flow rate..from..Type74/ITW " md4 "
   9,  8
*  3  Control function..from.. " Gunlo "
   20,  2
*   INPUT INITIAL VALUES
*    1
       0
*    2
       0
*    3
       0
*
UNIT 24   TYPE 11  Tee piece mode 1   PRESIM TYPE 44
PARAMS 1
*    1  Tee Piece, temperature related
       1
INPUTS 4

* 1  Temperature at inlet 1..from..Pump " To "
23,  1
* 2  Mass flow rate at inlet 1..from..Pump " mo "
23,  2
* 3  Temperature at inlet 2..from..Pump " To "
22,  1
* 4  Mass flow rate at inlet 2..from..Pump " mo "
22,  2
*   INPUT INITIAL VALUES
*    1
0
*    2
0
*    3
0
*    4
0
*
UNIT 25   TYPE 28  Simulation summary   PRESIM
TYPE 1128
PARAMS 77
*    1  Time interval for summaries
(>0=hours,<0=month)
PRNINT
*    2  Time at which summaries begin       -"-
0
*    3  Time at which summaries to end       -"-
175200
*    4  Logical unit number for output (see manual)
33
*    5  Output mode:
2
*    6  QEXSTnotInt
1
*    7  ColIn kJ/m2
-12
*    8
-1
*    9  ColArea m2
COAREA
*    10  ColIn kJ
1
*    11
-1
*    12  kJ to MWh
3.6E+006
*    13
2
*    14  ColIn MWh
-3
*    15  QcoMFC kJ
-13
*    16
-1
*    17  kJ to MWh
3.6E+006
*    18
2
*    19  QcoMFC MWh
-3
*    20
-21
*    21  Qdiss  kJ
-14
*    22
-1
*    23  kJ to MWh
3.6E+006
*    24
2
*    25  Qdiss  MWh
-3
*    26
-22
*    27  QcoHX  kJ
-15
*    28
-1
*    29  kJ to MWh
3.6E+006
*    30
2
*    31  QcoHX MWh
-3
*    32
-23
*    33  QcoMFC
-31
*    34  Qdiss
-32
*    35
4
*    36  QcoHX
-33
*    37
4
*    38  QcoHX
-33
*    39
2
*    40
-1
*    41    to %
100
*    42
1
*    43  ERRco %
-4
*    44  QcoHX
-33
*    45  CoIn<=>QcHX
-7
*    46
2
*    47
-1
*    48    to %
100
*    49
1
*    50  Eff. col. %
-4
*    51  QcoHX  MWh
-33
*    52
-1
*    53  MWh to kWh
1000
*    54
1
*    55
-1
*    56  Col.area m2
COAREA
*    57
2
*    58  QsCOLkWh/m2
-4
*    59  QcoMFC
-13
*    60  Qdiss
-14
*    61
4
*    62  QDST
-16
*    63
4
*    64  QloXST
-17
*    65
3
*    66  QEXST
-11
*    67
4
*    68  Qboil
-18
*    69
3
*    70  Qhxlo
-19
*    71
4
*    72  Qhxlo
-19
*    73
2
*    74
-1
*    75    to %
100
*    76
1
*    77  BalSYSTEM %
-4
INPUTS 9
* 1  deltaQ in buffer (T74)..from..Type74/ITW " delUs "
9, 51
* 2  Total incident radiation on coll...from..Sol Rad Proc, I and Idn " IT "
2,  6
* 3  Q out MFC..from..MFC1.0 2, *, 1, 0, * " Qc "
5,  3
* 4  Q dissipated by overheating..from..Pressure relief valve " Qboil "
6,  3
* 5  Q transfered by the solar HX..from.. " QT "
7,  5
* 6  net Q transfered in duct store..from..DST TRNSYS " Q "
17,  4
* 7  Q loss from buffer (T74)..from..Type74/ITW " Ql,s "
9, 17
* 8  Q supplied by boiler..from.. " Qaux "
14,  3
* 9  Q transfered by the load HX..from.. " QT "
11,  5
LABELS 8
ColIn QcoMFC Qdiss QcoHX ERRco% Effco% QsCOL ERRSy%
FORMAT
(F7.0,8F8.0)
ASSIGN optsloa.SUM          33
*
UNIT 26   TYPE 15  Algebraic Operator A   PRESIM
TYPE 1
PARAMS 1
*    1  Parameter for operation
1
INPUTS 2
* 1  Value of the first input..from..Type74/ITW " Td3 "
9,  5
* 2  Value of the second input..from..Type74/ITW " Qd3 "
9, 26
*   INPUT INITIAL VALUES
*    1
0
*    2
0
*
UNIT 27   TYPE 15  Algebraic Operator A   PRESIM
TYPE 1
PARAMS 1
*    1  Parameter for operation
1
INPUTS 2
* 1  Value of the first input..from..Temp ctrl flow div,mf&tmp " T1 "
12,  1
* 2  Value of the second input..from..Type74/ITW " Qd3 "
9, 26
*   INPUT INITIAL VALUES
*    1
0
*    2
0
*
UNIT 28   TYPE 28  Simulation summary   PRESIM
TYPE 1128
PARAMS 107
*    1  Time interval for summaries
(>0=hours,<0=month)
PRNINT
*    2  Time at which summaries begin       -"-
0
*    3  Time at which summaries to end       -"-
175200
*    4  Logical unit number for output (see manual)

33
* 5 Output mode:
2
* 6 QEXSTnotInt
1
* 7 QcoXST kJ
-12
* 8
-1
* 9 kJ to MWh
3.6E+006
* 10
2
* 11 QcoXST MWh
-3
* 12
-21
* 13 QstoreL kJ
-13
* 14
-1
* 15 kJ to MWh
3.6E+006
* 16
2
* 17 QstoreL MWh
-3
* 18
-22
* 19 QstoreU kJ
-14
* 20
-1
* 21 kJ to MWh
3.6E+006
* 22
2
* 23 QstoreU MWh
-3
* 24
-23
* 25 QloXST kJ
-15
* 26
-1
* 27 kJ to MWh
3.6E+006
* 28
2
* 29 QloXST MWh
-3
* 30
-24
* 31 QlossXST kJ
-16
* 32
-1
* 33 kJ to MWh
3.6E+006
* 34
2
* 35 QlossX MWh
-3
* 36
-25
* 37 QEXST kJ
-11
* 38
-1
* 39 kJ to MWh
3.6E+006
* 40
2
* 41 QEXST MWh
-3
* 42
-26
* 43 QcoXST
-31
* 44 QstoreLoad
-32
* 45
3
* 46 QstoreUnloa
-33
* 47
3
* 48 QloXST
-34
* 49
3
* 50 QlossXST
-35
* 51
3
* 52 QEXST
-36
* 53
4
* 54 QcoXST
-31
* 55
8
* 56
-31
* 57
7
* 58
8
* 59 ABS(QcoXST)
3
* 60 QstorL
-32
* 61
8
* 62
-32
* 63
7
* 64
8
* 65 ABS(QstorL)
3
* 66
3
* 67 QstorU
-33
* 68
8
* 69
-33
* 70
7
* 71
8
* 72 ABS(QstorU)
3
* 73
3
* 74 QloXST
-34
* 75
8
* 76
-34
* 77
7
* 78
8
* 79 ABS(QloXST)
3
* 80
3
* 81 QlossX
-35
* 82
8
* 83
-35
* 84
7
* 85
8
* 86 ABS(QlossX)
3
* 87
3
* 88 QEXST
-36
* 89
8
* 90
-36
* 91
7
* 92
8
* 93 ABS(QEXST)
3
* 94
3
* 95
-1
* 96 to 1/2 & %
0.005
* 97
1
* 98
2
* 99 ERR XST%
-4
* 100 TfXXST
-17
* 101 QloXST
-15
* 102
2
* 103 TfXST
-4
* 104 TrXST
-18
* 105 QloXST
-15
* 106
2
* 107 TrXST
-4
INPUTS 8
* 1 deltaQ in buffer (T74)..from..Type74/ITW " delUs "
9, 51
* 2 Q to buffer (T74) from collectors..from..Type74/ITW " Qd1 "
9, 24
* 3 Q from buffer (T74) to duct store..from..Type74/ITW " Qd2 "
9, 25
* 4 Q to buffer (T74) from duct store..from..Type74/ITW " Qd4 "
9, 27
* 5 Q from buffer (T74) to load..from..Type74/ITW " Qd3 "
9, 26
* 6 Q loss from buffer (T74)..from..Type74/ITW " Ql,s "
9, 17
* 7 Ponderate temperature to load..from..Algebraic Operator A " TX1 "
26, 1
* 8 Ponderate temperature from load..from..Algebraic Operator A " TX1 "
27, 1
LABELS 9
QcoXST QstorL QstorU QloXST QlossX QEXST ERRXS% TfXST TrXST
FORMAT
(F7.0,9F8.0)
ASSIGN optsloa.SUM 33
*
UNIT 29 TYPE 28 Simulation summary PRESIM TYPE 1128
PARAMS 93
* 1 Time interval for summaries (>0=hours,<0=month)
PRNINT
* 2 Time at which summaries begin -"-
0
* 3 Time at which summaries to end -"-
175200
* 4 Logical unit number for output (see manual)
33
* 5 Output mode:
2

```
*    6  QDST kJ
      -11
*    7
       -1
*    8  kJ to MWh
   3.6E+006
*    9
        2
*    10  QDST  MWh
       -3
*    11  QDST store1
      -21
*    12  QlossT kJ
      -12
*    13  QlossS kJ
      -13
*    14  QlossB kJ
      -14
*    15
        3
*    16
        3
*    17
       -1
*    18  kJ to MWh
   3.6E+006
*    19
        2
*    20  Qloss  MWh
       -3
*    21  Qloss stor2
      -22
*    22  QEDST kJ
      -15
*    23
       -1
*    24  kJ to MWh
   3.6E+006
*    25
        2
*    26  QEDST  MWh
       -3
*    27  QEDST stor3
      -23
*    28  Qelec  kJ
      -16
*    29
       -1
*    30  kJ to MWh
   3.6E+006
*    31
        2
*    32  Qelec MWh
       -4
*    33  QDST
      -31
*    34  Qloss
      -32
*    35
        4
*    36  QEDST
      -33
*    37
        4
*    38  QDST
      -31
*    39
        8
*    40
      -31
*    41
        7
*    42
        8
*    43  ABS(QDST)
        3
*    44  QLOSS
      -32
*    45
        8
*    46
      -32
*    47
        7
*    48
        8
*    49  ABS(QLOSS)
        3
*    50  QEDST
      -33
*    51
        8
*    52
      -33
*    53
        7
*    54
        8
*    55  ABS(QEDST)
        3
*    56
        3
*    57
        3
*    58
       -1
*    59   to 1/2 & %
    0.005
*    60
        1
*    61
        2
*    62  ERR DST %
       -4
*    63  Qboil kJ
      -18
*    64
       -1
*    65  kJ to MWh
   3.6E+006
*    66
        2
*    67  Qboil MWh
       -4
*    68  Qhxlo kJ
      -19
*    69
       -1
*    70  kJ to MWh
   3.6E+006
*    71
        2
*    72  Qhxlo MWh
       -3
*    73  QXSTlo kJ
      -17
*    74  Qboil  kJ
      -18
*    75
        4
*    76  Qhxlo  kJ
      -19
*    77
        3
*    78  Qhxlo  kJ
      -19
*    79
        2
*    80
       -1
*    81     to %
      100
*    82
        1
*    83  ERR Load %
       -4
*    84  Qload
      -20
*    85
       -1
*    86  Qstore
   3.6E+006
*    87
        2
*    88  Qload MWh
       -3
*    89
        2
*    90
       -1
*    91     to %
      100
*    92
        1
*    93  FlMet %
       -4
INPUTS 10
*  1  Q transfered in the duct store..from..DST TRNSYS " Q "
   17,  4
*  2  Q loss duct store TOP..from..DST TRNSYS " Qlosst "
   17,  5
*  3  Q loss duct store SIDE..from..DST TRNSYS " Qlosss "
   17,  6
*  4  Q loss duct store BOTTOM..from..DST TRNSYS " Qlossb "
   17,  7
*  5  delta Q in duct store..from..DST TRNSYS " QDST "
   17,  8
*  6  Qel. consumed by duct store pumps..from..PRESSURE DROP GROUND HX " Qel "
   16,  4
*  7  Q delivered to the load from buffer..from..Type74/ITW " Qd3 "
    9, 26
*  8  Q supplemented by the boiler..from.. " Qaux "
   14,  3
*  9  Q supplied to the load through HX..from.. " QT "
   11,  5
*  10  Q expected to be supplied to load..from..Simple load model " Qload "
   10,  4
LABELS 10
  QDST Qloss QEDST Qelec ERRDS% Qboil Qhxlo ERRLo% QLOAD FlMet%
FORMAT
(F7.0,10F8.0)
ASSIGN optsloa.SUM          33
*
UNIT 30   TYPE 15  Algebraic Operator mode B
PRESIM TYPE 1
PARAMS 32
*    1  TCI
      -11
*    2  QC
      -15
*    3
        1
*    4  TCIX
       -4
*    5  TCO
      -12
*    6  QC
      -15
*    7
        1
*    8  TCOX
       -4
*    9  TXI
      -13
*    10  QC
      -15
*    11
        1
*    12  TXIX
       -4
*    13  TXO
      -14
*    14  QC
      -15
*    15
        1
*    16  TXOX
       -4
*    17  TBI
      -16
*    18  QL
      -20
*    19
```

1
*    20  TBIX
-4
*    21  TBO
-17
*    22  QL
-20
*    23
1
*    24  TBOX
-4
*    25  TLI
-18
*    26  QL
-20
*    27
1
*    28  TLIX
-4
*    29  TLO
-19
*    30  QL
-20
*    31
1
*    32  TLOX
-4
INPUTS 10
*  1  Inlet collectors temp. from coll.HX..from..Pressure relief valve " To "
6,  1
*  2  Outlet collectors temp. in coll.HX..from.. " Tho "
7,  1
*  3  Inlet from XST in coll.HX(2nd side)..from..Type74/ITW " Td1 "
9,  1
*  4  Outlet coll.HX to XST (2nd side)..from.. " Tco "
7,  3
*  5  Heat rate transfered by coll. HX..from.. " QT "
7,  5
*  6  Inlet in load HX from boiler (hot)..from.. " To "
14,  1
*  7  Outlet from load HX to XST (hot s.)..from.. " Tho "
11,  1
*  8  Inlet in load HX from load (cold s)..from..Simple load model " Tret "
10,  2
*  9  Outlet from load HX to load (cold) ..from.. " Tco "
11,  3
*  10  Heat rate transfered by load HX..from.. " QT "
11,  5
*   INPUT INITIAL VALUES
*    1
0
*    2
0
*    3
0
*    4
0
*    5
0
*    6
0
*    7
0
*    8
0
*    9
0
*    10
0
*
UNIT 31   TYPE 28  Simulation summary   PRESIM TYPE 1128
PARAMS 47
*    1  Time interval for summaries (>0=hours,<0=month)
PRNINT
*    2  Time at which summaries begin       -"-
0
*    3  Time at which summaries to end       -"-
175200
*    4  Logical unit number for output (see manual)
33
*    5  Output mode:
2
*    6  TCIX
-11
*    7  QC collect.
-19
*    8
2
*    9  TCI
-4
*    10  TCOX
-12
*    11  QC collect.
-19
*    12
2
*    13  TCO
-4
*    14  TXIX
-13
*    15  QC collect.
-19
*    16
2
*    17  TXI
-4
*    18  TXOX
-14
*    19  QC collect.
-19
*    20
2
*    21  TXO
-4
*    22  QC col. kJ
-19
*    23
-1
*    24    kJ to MWh
3.6E+006
*    25
2
*    26  QCOLL MWh
-4
*    27  TBIX
-15
*    28  QL load
-20
*    29
2
*    30  TBI
-4
*    31  TBOX
-16
*    32  QL load
-20
*    33
2
*    34  TBO
-4
*    35  TLIX
-17
*    36  QL load
-20
*    37
2
*    38  TLI
-4
*    39  TLOX
-18
*    40  QL load
-20
*    41
2
*    42  TLO
-4
*    43  QL load kJ
-20
*    44
-1
*    45    kJ to MWh
3.6E+006
*    46
2
*    47  QLOAD MWh
-4
INPUTS 10
*  1  Sum T*QC, inlet T coll HX hot side..from..Algebraic Operator mode B
*        " TCIX "
30,  1
*  2  Sum T*QC, outl. T coll HX hot side..from..Algebraic Operator mode B
*        " TCOX "
30,  2
*  3  Sum T*QC, inlet T coll HX cold side..from..Algebraic Operator mode B
*        " TXIX "
30,  3
*  4  Sum T*QC, outl. T coll HX cold side..from..Algebraic Operator mode B
*        " TXOX "
30,  4
*  5  Sum T*QL, inlet T load HX hot side ..from..Algebraic Operator mode B
*        " TBIX "
30,  5
*  6  Sum T*QL, outl. T load HX hot side..from..Algebraic Operator mode B
*        " TBOX "
30,  6
*  7  Sum T*QL, inlet T load HX cold side..from..Algebraic Operator mode B
*        " TLIX "
30,  7
*  8  Sum T*QL, outl. T load HX cold side..from..Algebraic Operator mode B
*        " TLOX "
30,  8
*  9  Heat rate transfered in coll HX..from.. " QT "
7,  5
*  10  Heat rate transfered in load HX..from.. " QT "
11,  5
LABELS 10
TCI TCO TXI TXO QCOLL TBI TBO TLI TLO QLOAD
FORMAT
(F7.0,10F8.0)
ASSIGN optsloa.SUM          33
*
UNIT 32   TYPE 15  Algebraic Operator mode B   PRESIM TYPE 1
PARAMS 36
*    1  Q DST
-14
*    2  max. value
8
*    3  QSL (>0)
-21
*    4  Q DST
-14
*    5
7
*    6  min. value
8
*    7
7
*    8  QSU (<0)
-22
*    9  TSI
-11
*    10  QSL
-31
*    11
1
*    12  TIMSLX
-4
*    13  TSO
-12
*    14  QSL
-31
*    15
1
*    16  TOMSLX
-4
*    17  TSI
-11

```
*    18  QSU
       -32
*    19
         1
*    20  TIMSUX
        -4
*    21  TSO
       -12
*    22  QSU
       -32
*    23
         1
*    24  TOMSUX
        -4
*    25  TM DST
       -13
*    26  QSL
       -31
*    27
         1
*    28  TMSLX
        -4
*    29  TM DST
       -13
*    30  QSU
       -32
*    31
         1
*    32  TMSUX
        -4
*    33  QSL
       -31
*    34  QSL
        -4
*    35  QSU
       -32
*    36  QSU
        -4
INPUTS 4
*  1  Fluid temp. in duct GHX at centre  ..from..Tee piece mode 1 " To "
   24,  1
*  2  Fluid temp. in duct GHX at border  ..from..DST TRNSYS " Tout "
   17,  1
*  3  Mean storage temperature DST..from..DST TRNSYS " Tavst "
   17,  3
*  4  Hate rate exchanged in duct store..from..DST TRNSYS " Q "
   17,  4
*   INPUT INITIAL VALUES
*    1
         0
*    2
         0
*    3
         0
*    4
         0
*
UNIT 33   TYPE 28  Simulation summary   PRESIM TYPE 1128
PARAMS 47
*    1  Time interval for summaries (>0=hours,<0=month)
     PRNINT
*    2  Time at which summaries begin        -"-
         0
*    3  Time at which summaries to end       -"-
     175200
*    4  Logical unit number for output (see manual)
        33
*    5  Output mode:
         2
*    6  TIMSLX
       -11
*    7  QSL
       -17
*    8
         2
*    9  TIMSL
        -4
*    10  TOMSLX
       -12
*    11  QSL
       -17
*    12
         2
*    13  TOMSL
        -4
*    14  TMSLX
       -15
*    15  QSL
       -17
*    16
         2
*    17  TMSL
        -4
*    18  TMXST
       -20
*    19
        -2
*    20
         2
*    21  TMXST
        -4
*    22  QSL
       -17
*    23
        -1
*    24    kJ to MWh
    3.6E+006
*    25
         2
*    26  QSL
        -4
*    27  TOMSUX
       -14
*    28  QSU
       -18
*    29
         2
*    30  TOMSU
        -4
*    31  TIMSUX
       -13
*    32  QSU
       -18
*    33
         2
*    34  TIMSU
        -4
*    35  TMSUX
       -16
*    36  QSU
       -18
*    37
         2
*    38  TMSU
        -4
*    39  TMDST
       -19
*    40
        -2
*    41
         2
*    42  TMDST
        -4
*    43  QSU
       -18
*    44
        -1
*    45    kJ to MWh
    3.6E+006
*    46
         2
*    47  QSU
        -4
INPUTS 10
*  1  Sum T*QSL, fluid centre during load..from..Algebraic Operator mode B
*        " TIMSLX "
   32,  1
*  2  Sum T*QSL, fluid border during load..from..Algebraic Operator mode B
*        " TOMSLX "
   32,  2
*  3  Sum T*QSU, fluid border during unl...from..Algebraic Operator mode B
*        " TIMSUX "
   32,  3
*  4  Sum T*QSU, fluid centre during unl...from..Algebraic Operator mode B
*        " TOMSUX "
   32,  4
*  5  Sum T*QSL, mean duct T during load ..from..Algebraic Operator mode B
*        " TMSLX "
   32,  5
*  6  Sum T*QSU, mean duct T during unl...from..Algebraic Operator mode B
*        " TMSUX "
   32,  6
*  7  Q stored in duct store       ..from..Algebraic Operator mode B " QSL "
   32,  7
*  8  Q recovered from duct store..from..Algebraic Operator mode B " QSU "
   32,  8
*  9  Mean duct store temperature (DST)..from..DST TRNSYS " Tavst "
   17,  3
*  10  Mean buffer store temperature (XST)..from..Type74/ITW " Ts,m "
    9, 52
LABELS 10
  TIMSL TOMSL TMSL TMXST QSL TOMSU TIMSU TMSU TMDST QSU
FORMAT
(F7.0,10F8.2)
ASSIGN optsloa.SUM          33
*
UNIT 34   TYPE 24     PRESIM TYPE 1
PARAMS 1
*     1  Time interval for integr. (>0 hour, <0 month)
      TIMSTP
INPUTS 4
*  1  Determine MAXIMUM temperature..from..MFC1.0 2, *, 1, 0, * " To "
    5,  1
*  2  Determine MAXIMUM temperature..from.. " Tco "
    7,  3
*  3  Determine MAXIMUM temperature..from..Tee piece mode 1 " To "
   24,  1
*  4  Determine MINIMUM temperature..from..Tee piece mode 1 " To "
   24,  1
*   INPUT INITIAL VALUES
*     1
     -1E+008
*     2
     -1E+008
*     3
     -1E+008
*     4
      1E+008
*
UNIT 35   TYPE 15  Algebraic Operator mode B PRESIM TYPE 1
PARAMS 49
*     1
          -1
*     2  timestep
      TIMSTP
*     3  Tau
        -23
*     4
          -1
*     5  INITVAL MAX
     -1E+008
*     6
        -21
*     7
          -1
*     8  INITVAL MIN
      1E+008
*     9
        -22
*     10  IN1
        -11
```

*    11  Tau
    -33
*    12
    2
*    13  OLDIN1
    -12
*    14  MAX
    12
*    15  RESET
    -19
*    16  INITVAL MAX
    -31
*    17
    1
*    18
    3
*    19  NEWIN1
    -4
*    20  IN3
    -13
*    21  Tau
    -33
*    22
    2
*    23  OLDIN3
    -14
*    24  MAX
    12
*    25  RESET
    -19
*    26  INITVAL MAX
    -31
*    27
    1
*    28
    3
*    29  NEWIN3
    -4
*    30  IN5
    -15
*    31  Tau
    -33
*    32
    2
*    33  OLDIN5
    -16
*    34  MAX
    12
*    35  RESET
    -19
*    36  INITVAL MAX
    -31
*    37
    1
*    38
    3
*    39  NEWIN5
    -4
*    40  IN7
    -17
*    41  Tau
    -33
*    42
    2
*    43  OLDIN7
    -18
*    44  MIN
    11
*    45  RESET
    -19
*    46  INITVAL MIN
    -32
*    47
    1
*    48
    3
*    49  NEWIN7
    -4
INPUTS 9
*  1  1st input..from.. " Ttau "
   34,  1
*  2  2nd input..from..Algebraic Operator mode B " T1 "
   35,  1
*  3  3rd input..from.. " Ttau "
   34,  2
*  4  4th input..from..Algebraic Operator mode B " T2 "
   35,  2
*  5  5th input..from.. " Ttau "
   34,  3
*  6  6th input..from..Algebraic Operator mode B " T3 "
   35,  3
*  7  7th input..from.. " Ttau "
   34,  4
*  8  8th input..from..Algebraic Operator mode B " T4 "
   35,  4
*  9  9th input..from..L  LBL " RESET "
RESET
*  INPUT INITIAL VALUES
*    1
    0
*    2
    -1E+008
*    3
    0
*    4
    -1E+008
*    5
    0
*    6
    -1E+008
*    7
    0
*    8
    1E+008
*    9
    0
*
UNIT 36   TYPE 24     PRESIM TYPE 1
PARAMS 1
*    1  Time interval for integr. (>0 hour, <0 month)
    TIMSTP
INPUTS 4
*  1  determine MAXIMUM temperature..from..Type74/ITW " Ts,m "
    9, 52
*  2  determine MAXIMUM temperature..from..DST TRNSYS " Tavst "
   17,  3
*  3  determine MINIMUM temperature..from..Type74/ITW " Ts,m "
    9, 52
*  4  determine MINIMUM temperature..from..DST TRNSYS " Tavst "
   17,  3
*  INPUT INITIAL VALUES
*    1
    -1E+008
*    2
    -1E+008
*    3
    1E+008
*    4
    1E+008
*
UNIT 37   TYPE 15  Algebraic Operator mode B     PRESIM TYPE 1
PARAMS 49
*    1
    -1
*    2  timestep
    TIMSTP
*    3  Tau
    -23
*    4
    -1
*    5  INITVAL MAX
    -1E+008
*    6
    -21
*    7
    -1
*    8  INITVAL MIN
    1E+008
*    9
    -22
*    10  IN1
    -11
*    11  Tau
    -33
*    12
    2
*    13  OLDIN1
    -12
*    14  MAX
    12
*    15  RESET
    -19
*    16  INITVAL MAX
    -31
*    17
    1
*    18
    3
*    19  NEWIN1
    -4
*    20  IN3
    -13
*    21  Tau
    -33
*    22
    2
*    23  OLDIN3
    -14
*    24  MAX
    12
*    25  RESET
    -19
*    26  INITVAL MAX
    -31
*    27
    1
*    28
    3
*    29  NEWIN3
    -4
*    30  IN5
    -15
*    31  Tau
    -33
*    32
    2
*    33  OLDIN5
    -16
*    34  MIN
    11
*    35  RESET
    -19
*    36  INITVAL MIN
    -32
*    37
    1
*    38
    3
*    39  NEWIN5
    -4
*    40  IN7
    -17
*    41  Tau
    -33
*    42
    2
*    43  OLDIN7
    -18
*    44  MIN
    11
*    45  RESET
    -19
*    46  INITVAL MIN
    -32
*    47
    1
*    48
    3
*    49  NEWIN7
    -4
INPUTS 9
*  1  1st input..from.. " Ttau "
   36,  1
*  2  2nd input..from..Algebraic Operator mode B " T1h "
   37,  1
*  3  3rd input..from.. " Ttau "

```
   36,  2
*  4  4th input..from..Algebraic Operator mode B " T2h "
   37,  2
*  5  5th input..from.. " Ttau "
   36,  3
*  6  6th input..from..Algebraic Operator mode B " T3l "
   37,  3
*  7  7th input..from.. " Ttau "
   36,  4
*  8  8th input..from..Algebraic Operator mode B " T4l "
   37,  4
*  9  9th input..from..L  LBL " RESET "
RESET
*  INPUT INITIAL VALUES
*    1
       0
*    2
    -1E+008
*    3
       0
*    4
    -1E+008
*    5
       0
*    6
     1E+008
*    7
       0
*    8
     1E+008
*    9
       0
*
UNIT 38   TYPE 25  Printer                    PRESIM
TYPE 1125
PARAMS 4
*    1  Print time interval   (>0=hours <0=months)
     MAXPRT
*    2  Time for start of printer      -"-
       0
*    3  Time for stop of printer       -"-
     175200
*    4  Logical unit No (<=0 for std Line Printer)
       33
INPUTS 8
* 1  Max. fluid temp. outlet collectors..from..Algebraic
Operator mode B
*       " T1 "
   35,  1
* 2  Max. fluid temp. to XST from coll...from..Algebraic
Operator mode B
*       " T2 "
   35,  2
* 3  Max. inlet fluid temp. in DST..from..Algebraic
Operator mode B " T3 "
   35,  3
* 4  Min. inlet fluid temp. in DST..from..Algebraic
Operator mode B " T4 "
   35,  4
* 5  Max. mean XST temperature..from..Algebraic
Operator mode B " T1h "
   37,  1
* 6  Max. mean DST temperature..from..Algebraic
Operator mode B " T2h "
   37,  2
* 7  Min. mean XST temperature..from..Algebraic
Operator mode B " T3l "
   37,  3
* 8  Min. mean DST temperature..from..Algebraic
Operator mode B " T4l "
   37,  4
 TCMAX
 TXMAX
 TDMAX
 TDMIN
 TMXMAX
 TMDMAX
 TMXMIN
 TMDMIN
FORMAT
(F7.0,8F12.1)
ASSIGN optsloa.SUM          33
*
UNIT 39   TYPE 15  Algebraic Operator mode B
PRESIM TYPE 1
PARAMS 24
*    1
       -1
*    2  Coll area
     COAREA
*    3
       -4
*    4
       -1
*    5  Buff volume
      BUFFV
*    6
       -4
*    7
       -1
*    8  Duct volume
      DUCTV
*    9
       -4
*    10
       -1
*    11  Duct height
      DUCTH
*    12
       -4
*    13
       -1
*    14  Bore number
      NBORE
*    15
       -4
*    16
       -1
*    17  Insul.thick
     DUCISO
*    18
       -4
*    19
       -1
*    20  Duct FRISO
     DUCFRI
*    21
       -4
*    22
       -1
*    23  ThickLayTop
     DUCDHP
*    24
       -4
*
UNIT 40   TYPE 25  Printer                    PRESIM
TYPE 1125
PARAMS 4
*    1  Print time interval   (>0=hours <0=months)
  1.752E+006
*    2  Time for start of printer      -"-
       0
*    3  Time for stop of printer       -"-
  1.752E+006
*    4  Logical unit No (<=0 for std Line Printer)
       33
INPUTS 8
*  1  The 1th INPUT to be printed..from..Algebraic
Operator mode B " COAREA "
   39,  1
*  2  The 2nd INPUT to be printed..from..Algebraic
Operator mode B " BUFFV "
   39,  2
*  3  The 3rd INPUT to be printed..from..Algebraic
Operator mode B " DUCTV "
   39,  3
*  4  The 4th INPUT to be printed..from..Algebraic
Operator mode B " DUCTH "
   39,  4
*  5  The 5th..from..Algebraic Operator mode B "
NBORE "
   39,  5
*  6  The 6th..from..Algebraic Operator mode B "
DUCISO "
   39,  6
*  7  The 7th..from..Algebraic Operator mode B "
DUCFRI "
   39,  7
*  8  The 8th..from..Algebraic Operator mode B "
DUCDHP "
   39,  8
 COAREA
 BUFFV
 DUCTV
 DUCTH
 NBORE
 DUCISO
 DUCFRI
 DUCDHP
FORMAT
(F7.0,8F12.2)
ASSIGN optsloa.SUM          33
*
UNIT 41   TYPE 25  Printer                    PRESIM
TYPE 1125
PARAMS 4
*    1  Print time interval   (>0=hours <0=months)
       24
*    2  Time for start of printer      -"-
     TEMPRI
*    3  Time for stop of printer       -"-
     TEMPRO
*    4  Logical unit No (<=0 for std Line Printer)
       34
INPUTS 6
*  1  Max. fluid temp. outlet
collectors..from..Type74/ITW " Ts5 "
    9, 33
*  2  Max. fluid temp. to XST from
coll...from..Type74/ITW " Ts,m "
    9, 52
*  3  Max. fluid temp. to DST from
XST..from..Type74/ITW " Ts1 "
    9, 29
*  4  Max. fluid temp. to XST from DST..from..DST
TRNSYS " Tavbh1 "
   17, 10
*  5  Max. mean XST temperature..from..DST TRNSYS
" Tavst "
   17,  3
*  6  Max. mean DST temperature..from..DST TRNSYS
" TavbhN "
   17, 11
 TXSTMA
 TXSTAV
 TXSTMI
 TDSTCE
 TDSTAV
 TDSTBO
FORMAT
(F7.0,6F12.1)
ASSIGN optsloa.DAY          34
*
END
```