12 DIMENSIONING OF DOMESTIC HOT WATER SYSTEMS

In this chapter mainly the dimensioning guidelines for domestic hot water systems are given. For the detailed dimensioning of complex systems several simulation programs are available.

12.1 Hot Water Demand

The hot water demand in a household is decisive for the dimensioning of a domestic hot water (DHW) solar system. However, this depends on the users' habits. For example, if a family is used to have a shower rather than a bath, the daily hot water demand is significantly lower than if a bath is frequently taken. The daily hot water demand can be estimated as shown in the table below.

Table 10: Hot water demand for different users at a hot water temperature of 50 °C.

		Low demand (litres)	Medium demand (litres)	High demand (litres)
Residential buildings	per person and day	30	50	60
Sport facilities	per shower	20	30	50
Accommodation	per bed	20	40	60

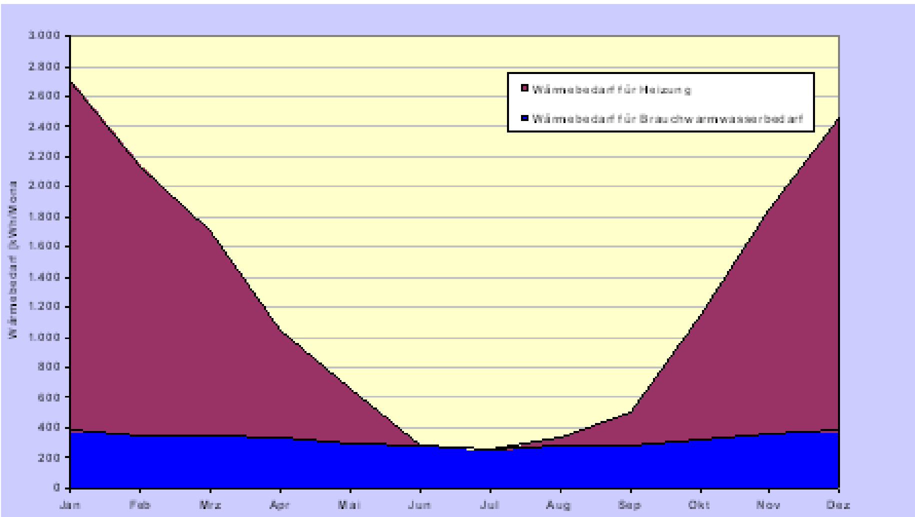


Figure 117: Typical annual hot water (blue) and space heating demand (red) of a single family house in central Europe

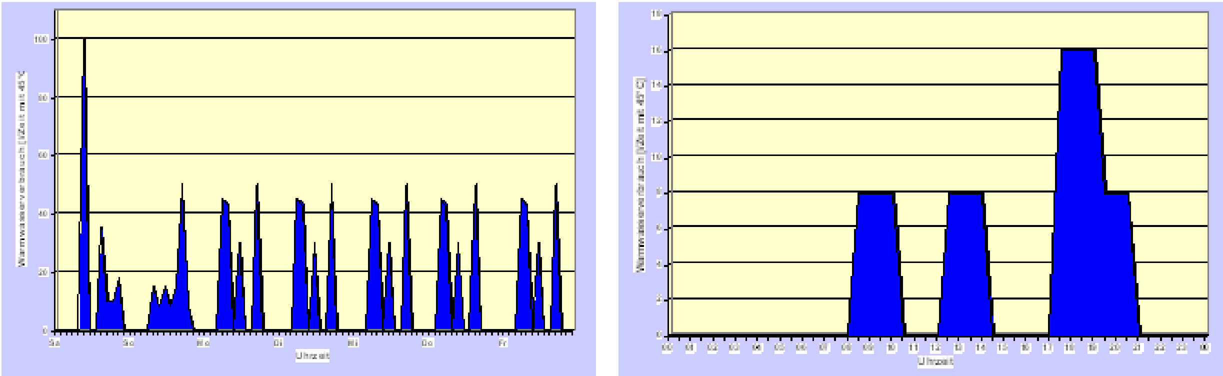


Figure 118: Typical hot water demand profile for a week (left) and for a day (right); single-family house.

12.2 The Hot Water Storage Tank Capacity

When the daily hot water demand has been determined, the volume of the storage tank can be specified. It should be some 0.8 to 1.2 fold the daily demand for regions with high solar radiation and 2 to 2.5 fold the daily demand for regions with lower solar radiation (central and northern Europe) so that consumption peaks can be met well and cloudy days can be compensated.

Examples

For an average hot water demand (HWD) of 50 litres per person (P), the daily demand (DD) for a four-person household is 200 litres. The volume of the storage tank (V_{St}) is thus calculated as follows:

A) For central European climatic conditions:

$$V_{St} = HWD \times P \times 2 = 50 \times 4 \times 2 = 400 \text{ litres}$$

B) For regions with high solar radiation and low hot water demand (Central America, Africa)

$$V_{St} = HWD \times P \times 1.2 = 30 \times 4 \times 1 = 120 \text{ litres}$$

As the manufacturers do not offer tanks in every possible size, the choice has to be made among those generally available on the market. However, it is recommended that the storage tank capacity is not less than 90% and not more than 120% of the calculated volume.

12.3 Collector Area

When the daily hot water demand is known the collector area can be determined. The required collector area depends on several factors such as:

- collector type
- size of the solar storage tank
- location, tilt, and orientation of the collectors
- local climatic conditions

12.3.1 Location, Tilt, and Orientation of the Collectors

The most usual place to install collectors is the roof area. If it is not possible to mount the collectors on the roof, they can also be mounted on a suitable frame near the house, they can be integrated into an earth bank, or mounted on a flat roof. However, in each case attention should be paid to keeping the pipes to and from the tank as short as possible.

As a general rule, the collector should be aligned to the equator. That means in the southern hemisphere facing north and in the northern hemisphere facing south. A deviation of 40° to the east or west is nevertheless possible, as it does not reduce the yield significantly.
In addition, care should be taken that the collectors are not shaded at any time of the year, either by trees or buildings, if possible.

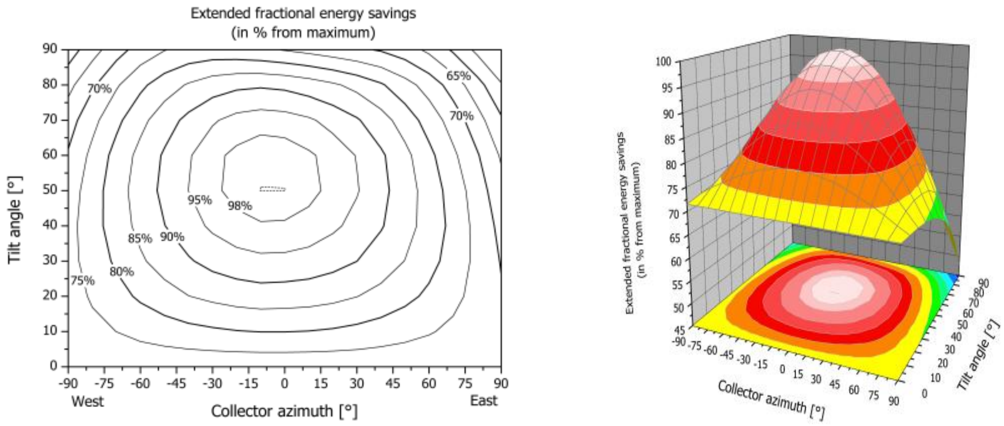


Figure 119: Dependency of the extended fractional energy savings on tilt angle and azimuth of the collector (climate: central Europe) (Heimrath 2002)

Collector orientation can vary ±30° from south and from 30° to 75° in slope with less than a 10% reduction in energy savings for a middle European climate. Within this range it is generally easy to compensate with a slightly larger collector area.

Angle of tilt

Apart from the effect of the characteristics of the collector itself, the output of the solar system is strongly dependent on the inclination angle of the collector to the sun. The largest yield is obtained when the collector is always orientated perpendicular to the sun. However, the optimal tilt angle for the collectors varies according to the season, as the sun is higher in the sky in summer than in winter. As a general rule, *the optimum angle of tilt is equal to the degree of latitude of the site* . But the minimum angle of the collector should be 15 degree to assist the thermosyphon effect. The following table shows optimum tilt angles of different latitudes and seasons.

Table 11: Tilt angle for different latitudes and seasons

Latitude [degree]	Best collector tilt in:					
	June	Orientation	**Sept./March**	Orientation	**December**	Orientation
50 N	26.5	S	50	S	73.5	S
40 N	16.5	S	40	S	63.5	S
30 N	6.5	S	30	S	53.5	S
20 N	3.5	N	20	S	43.5	S
15 N	8.5	N	15	S	38.5	S
10 N	13.5	N	10	S	33.5	S
Equator = 0	**23.5**	**N**	**0**	**-**	**23.5**	**S**
10 S	33.5	N	10	N	13.5	S
15 S	38.5	N	15	N	8.5	S
20 S	43.5	N	20	N	3.5	S
30 S	53.5	N	30	N	6.5	N
40 S	63.5	N	40	N	16.5	N
50 S	73.5	N	50	N	26.5	N

Optimum tilt angle ï Example 1: Vienna, Austria

Location: Vienna, Austria
Latitude: 48 degree north (see table: latitude = 50 degree)

For a south-orientated surface, the optimum tilt angle in December is 73.5°. In June, the most favourable angle would be 26.5°. An angle of 45 - 50° is ideal for use throughout the year.

Optimum tilt angle ï Example 2: Johannesburg, South Africa

Location: Johannesburg, South Africa
Latitude: 26.12 degree South (see table: latitude = 30 degree)

For a north-orientated surface, the energy gain in June is largest for a tilt angle of 53.5°. In December, the most favourable angle would be 6.5° north facing. An angle of 26° is ideal for use throughout the year.

Note: To ensure a good performance of a thermosyphon system a minimum tilt angle of 15° is recommended.

12.3.2 Dimensioning Guidelines

The dimensioning indicated in the tables below is to be understood as guidelines for central European (Table 12) and Central American (Table 13) conditions. In order to gain exact information, a calculation based on the system site characteristics in question is recommended. Such calculations can be performed with the help of simulation programs. These give exact predictions of the solar fraction and the system efficiency for the planned system as well as information on the additional energy needed during the rainy season.

Table 12. Dimensioning of domestic hot water solar systems for central European conditions

Daily hot water demand [litres]	Solar storage capacity [litres]	Collector area* SC [m^2]
- 100	200	4
- 200	400	6
- 300	500 ï 750	8 - 12
- 500	750 - 1000	12 - 16

Table 13: Dimensioning of domestic hot water solar systems for southern African conditions

Daily hot water demand [litres]	Solar storage capacity [litres]	Collector area* SV [m^2]	Collector area* SC [m^2]
50	50 ï 75	1.0 ï 1.5	0.9 ï 1.3
100	100 ï 150	2.0 ï 3.0	1.5 ï 2.5
200	200 ï 300	3.5 ï 4.5	3.0 ï 4.0
300	300 ï 450	4.5 ï 6.0	4.0 ï 5.0
500	500 - 750	7.5 - 10	6.0 ï 8.5

*) depending on the required solar fraction
SV ... coating of solar varnish
SC ... selective coating

13 DIMENSIONING - SOLAR COMBISYSTEMS

Solar combisystems differ from purely solar domestic hot water systems in several key aspects, which means that the dimensioning of them differs in several ways. The main difference is the extra space-heating load, resulting in a total heat demand that varies considerably during the year, and the fact that the thermal energy is not usually stored as hot water used for showers etc. Consequently, solar combisystems tend to be more complex and larger than solar domestic hot water systems, and they have excess capacity during the summer.

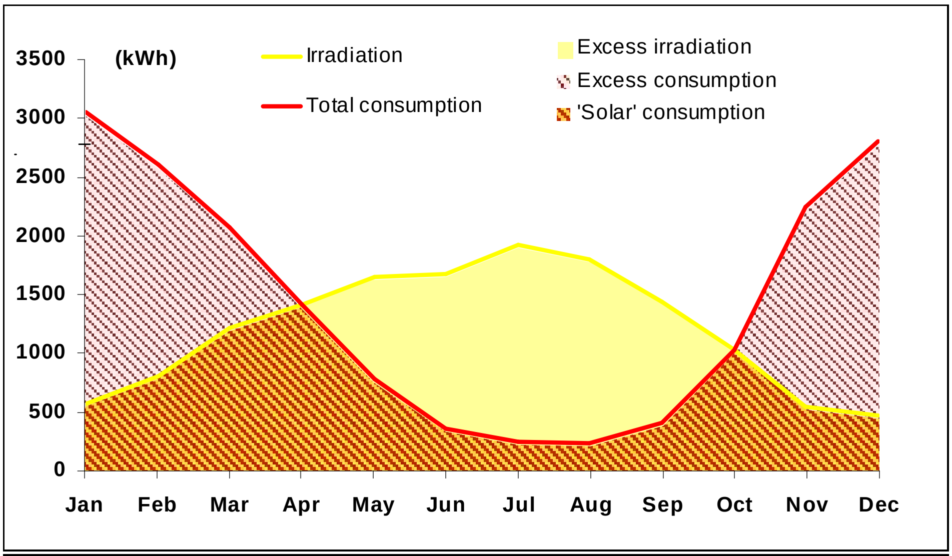


Figure 120: Example of solar energy surplus in summer and shortage in winter

Characteristic for the heat load of solar combisystems is the combination of rather constant energy draw-off distribution over the year for hot water and energy demand for space heating which varies over the year, mainly related to ambient temperature. Generally, solar energy gain in summer season is higher than the demand, i.e. mainly for hot water. In winter, energy consumption both for hot water and space heating is higher than solar gain. Hence, auxiliary heating is needed.

The figure above shows an example of solar gain and energy consumption over the year.
Further characteristic for the hot water part is the daily draw-off in peaks. Distribution of heat load both over the year and over the day determines optimum dimensions for the solar combisystem.

Requirements for the hydraulic layout of solar combisystems can be summarized as follows:

- deliver solar energy to heat store(s) with as low heat loss as possible;
- distribute all the heat needed to hot water and space heating demand;
- reserve sufficient store volume for auxiliary heating taking into account minimum running time for the specific heater;
- low investment costs;
- low space demand;
- easy and failure safe installation.

Furthermore, specific properties of components influence the operation of the other components.
As mentioned before, heat demand and annual and daily load distribution are also of major importance for system dimensions.
Generally, the heat store is the heart of a solar combisystem. Solar heat is stored in the lower part of the store and, if applicable, auxiliary heat in the upper part. The type of collector influences the height of the collector loop outlet to the store. For high flow collectors, this connection can be quite low. On the other hand, this connection should be higher for low flow collectors and the heat store should be prepared to enhance thermal stratification.

For combisystems with indirect integrated auxiliary heating, the inlet pipe from the heater is connected at the top. The height of the outlet depends on the peak hot water demand, the outlet pipes to the heat distribution system and the volume needed for solar energy. The distance Minimum operation time for the heater also determines the auxiliary volume. Requirements are stricter for wood burners than for gas. Another influence is the type of heat distribution system, e.g. connection from high temperature radiators to the store should be higher than from a low temperature heat distribution system.
This indicates that system design largely depends on national building traditions, auxiliary energy source and user behavior.

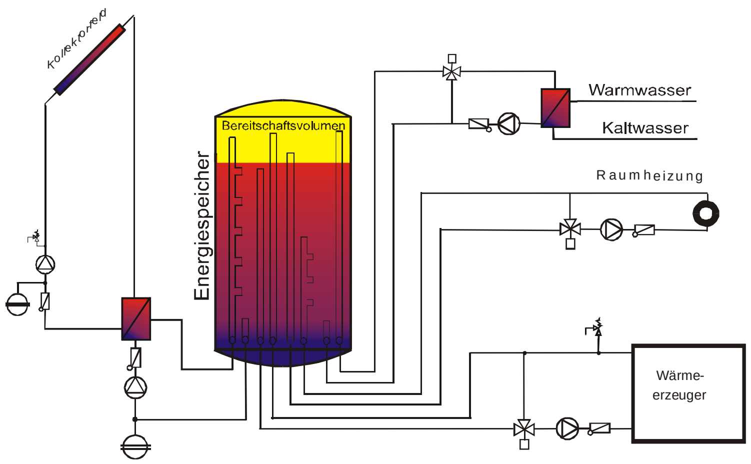


Figure 121: Hydraulic scheme of a solar combisystem (single storage system)

Since the dimensioning of a solar combisystem depends on several conditions, in the following a dimensioning nomogram is shown, which is based on the following hydraulic scheme and the data presented in the table below. It is also based on Austrian climatic conditions:

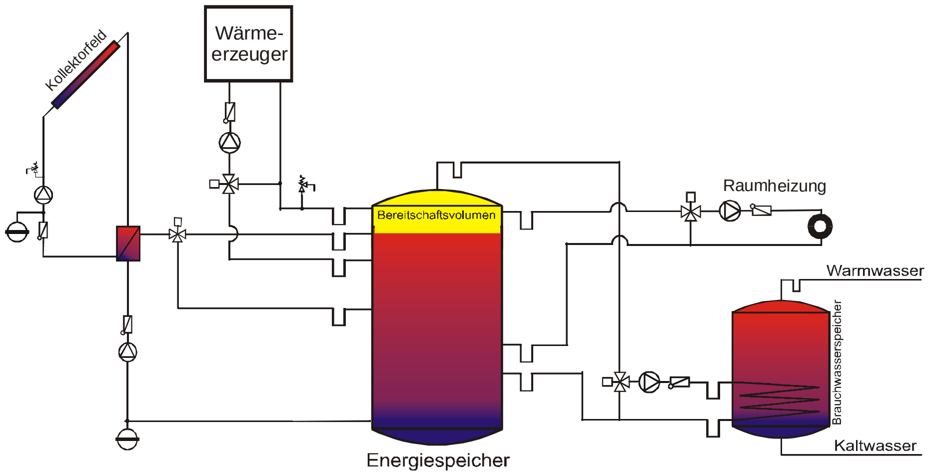


Figure 122: Hydraulic scheme of a solar combisystem (two storage system)

Single family house	
Location	Graz, Austria
Heat load	8 kW
Flow- and return temperature of the space heating system	40/30 °C
Hot water demand	
Dayly demand	200 l at 45 °C
Solar Collector	
Collector area	30 m² flat-plate collector
Orientation	South
Tilt angle	45°
Hot water Storage	
Volume	500 Litre
Space heating storage	
Volume	2.000 Litre
Storage insulation	150 mm (Lambda = 0,05) W/mK)

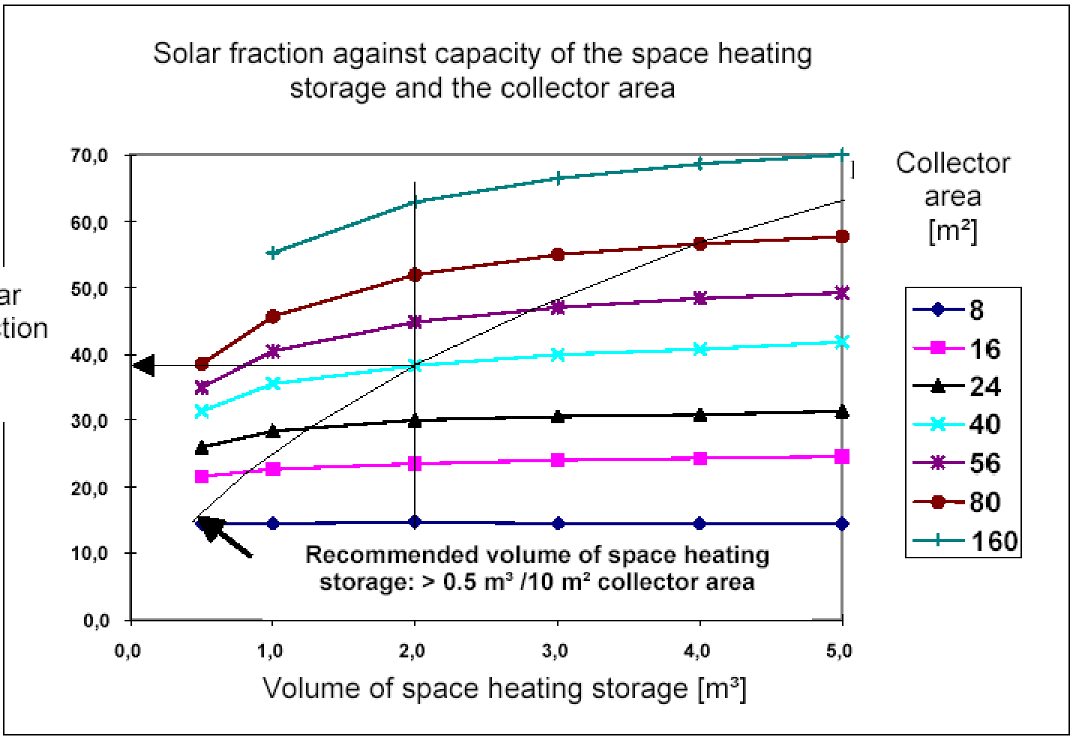


Figure 123: Dimensioning nomogram for a solar comb system for the conditions given in the table above.

For solar combisystems with a relatively small collector area and 2 ï 5 kW heat load of the house, optimum heat store volume appears to be 50 ï 200 liters per kW heat load. The optimum tilt angle is between 30° and 75°. Orientation is best between 30° east and 45° west.

Simulation programs

There are several computer programs on the market for the thermal performance calculation of solar combisystems: Polysun, TSOL and SHWwin. All are transient simulation programs with time steps of a few minutes and feature database support for components and systems. Heat loads can also be defined in great detail. Possible system layouts are, however, restricted and differ from one program to the other.

A more general computer program is TRNSYS., Solar combi systems can be composed from TRNSYS modules.

Further information on solar combi systems:

Weiss, W. (Ed.): Solar Heating Systems for Houses. A design handbook for solar combisystems, James & James, London 2003

14 LAY OUT OF SOLAR THERMAL SYSTEMS

14.1 Mode of Operation

The specific mass flow of the heat transfer fluid (kg per m² collector area per hour) differs a lot depending on the type of operation. This is caused by the different operation strategies together with various heat transfer concepts between the collector loop and the space heating storage tank. In general solar thermal systems can be operated with a ñhigh-flowò, ñlow-flowò or ñmatched-flowòrate of the heat transfer fluid.
The type of operation depends oné

- The size of the systems
- The temperature level
- The strategy of operation and heat stratification

14.1.1 High-Flow Systems

High-flow systems are typically operated with a mass flow of about 21 to 70 kg/m²h. Due to the high specific mass flow the rise of temperature (10-15 K) is very low in one collector pass of the heat transfer fluid (at an irradiance of 800 W/m²). With each pass of the heat transfer fluid trough the collector the temperature in the energy storage tank raises only a little (see figure below). Therefore it takes quite a long time until the energy storage tank reaches the necessary temperature level.
High flow systems are utilized at plants with a collector area up to 25 m² at the maximum. The primary use is for the production of domestic hot water (DHW) at one family houses.
The high rate of the mass flow leads to a high pressure drop in the heat exchanger. It further demands bigger dimensions of the pipes (flow and return pipe of the collector loop) and therefore leads to higher energy losses at the pipes.

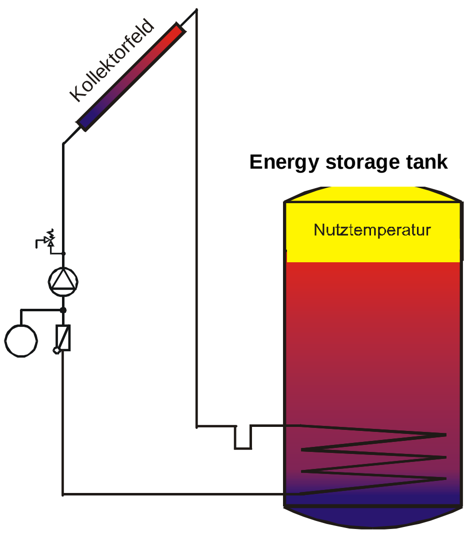


Figure 124: Hydraulic scheme of a high flow system.

14.1.2 Low-Flow Systems

Large solar thermal systems (more than 10 to 15 m²) should be operated according to the "low-flow principle". This means a specific mass flow in the collector of 5-20 kg/m²h. Due to this low mass flow - compared to high-flow systems - the rise of temperature (delta T) in one collector pass is much higher. A temperature level of e.g. 65°C can be reached in one collector pass already. The energy storage tank must be loaded orientated to the temperature in order to be able to provide this high temperature level immediately to the user (see figures below). Stratified energy storage tanks are necessary for efficient low-flow systems.

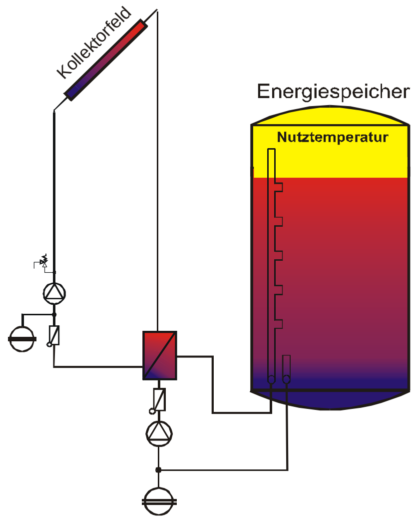
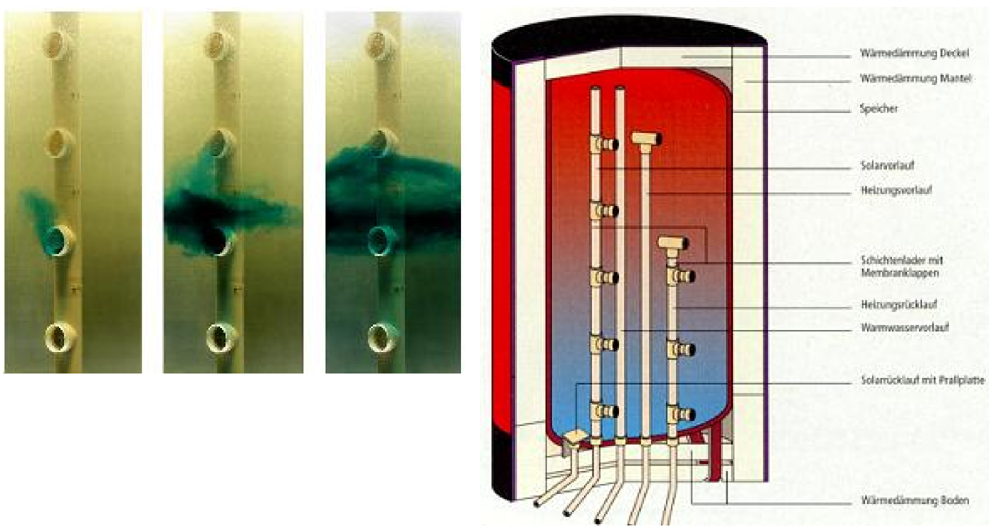


Figure 125: Hydraulic scheme of a low flow system with stratified charging of the energy storage tank.



Insulation of the lid
Insulation of casing

Tank

Solar flow pipe
Space heating flow pipe

Stratifier with membrane flaps

Space heating return pipe
Hot water flow pipe

Solar return pipe (with impact panel)

Insulation bottom

Figure 126: Stratified charging of the storage tank (Source: SOLVIS)

14.1.3 Low-Flow Systems versus High-Flow Systems

In the following advantages and disadvantages of the two described types of operation of solar thermal systems are now discussed:

- The low-flow operation of a system leads to smaller dimensions of the tubes. This causes lower investment costs for the whole solar thermal system.

- Low-flow systems demand (and enable) a big thermal length in the collector (this means a long serial connection of the pipes). Therefore a collector area of 80 to 100 m^2, which is connected in series, can be realised depending on the geometry of the absorber and the resulting pressure drop. This leads to a significant reduction of the piping, as there is only one flow and return tube necessary for the whole collector field. For high-flow systems the maximum collector area, which can be connected in serial is 25 m^2 (depending on the geometry of the absorber and the resulting pressure drop). This advantage of low-flow systems reduces the investment costs (tubing, insulation material, man power) significantly.

- Due to the reduction of the tubing on the one hand and the smaller tube diameter on the other hand the heat loss at a low-flow system can be reduced and the annual efficiency of the system can be risen significantly compared to a high-flow system.

- At low-flow systems the reduced mass flow leads to lower hydraulic performances and to a lower demand of electrical energy for the pumps.

- The demand for auxiliary heating is reduced significantly at low-flow systems because a high temperature level can be provided for the user very quick.

The table below shows the spectrum of the specific mass flow of the different types of operation. Further it shows the difference in the total mass flow at a collector area of 50 m^2. This example shows that at high-flow conditions of big solar thermal systems the requested pipe diameters are significantly higher than at low-flow conditions. This leads to high costs in investment and operation of these systems.

Table 14: Comparison of the mass flow for high-flow, low-flow and matched-flow systems

Type of operation	Specific mass flow	Example: mass flow at 50 m^2 collector area
Low-Flow	5 - 20 kg/m^2h	12 kg/m^2h => 600 kg/h
High-Flow	(21) 40 - 70 kg/m^2h	45 kg/m^2h => 2,250 kg/h
Low-Flow ï r.p.m. controlled	5 - 20 kg/m^2h	250 to 1,000 kg/h

The following equation shows the calculation of the mass flow for the primary circuit of the solar thermal system:

$$\dot{m}_{primary} = A_{collector} \cdot \dot{m}_{specific} \quad \text{[kg/h]}$$

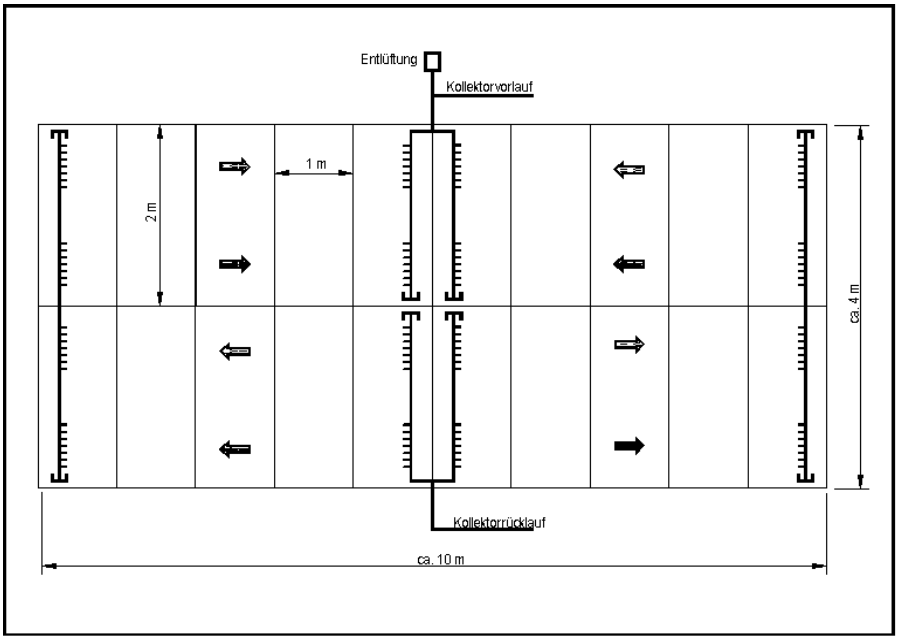
$\dot{m}_{primary}$	mass flow of the primary circuit of the solar thermal system	[kg/h]
$A_{collector}$	collector area (aperture area)	[m²]
$\dot{m}_{specific}$	specific mass flow for the primary circuit of the solar thermal system	[kg/m²h]

Comparison of a typical low-flow and high-flow system by the means of:

- Collector hydraulics
- Efficiency of the collector
- Pressure drop of the collector and the system
- Hydraulic efficiency and electrical pump efficiency

The boundary conditions for the comparison are:

- Gross collector area: 40 m² (10 m x 4 m)
- Collector values: c_0=0.77, c_1=3.33 W/m²K, c_2=0.012 W/m²K²
- Inner diameter of the absorber pipe: 8.25 mm
- Ambient temperature: 20 °C
- Irradiation on the collector area: 800 W/m²
- Average collector temperature: 46.5°C (for both systems)



$\dot{m}_{primary}$ = 2037 kg/h
$\dot{m}_{specific}$ = 50 kg/m²h
L_{Therm} = 9.6 m

Figure 127: Hydraulic collector scheme for a high-flow operation mode

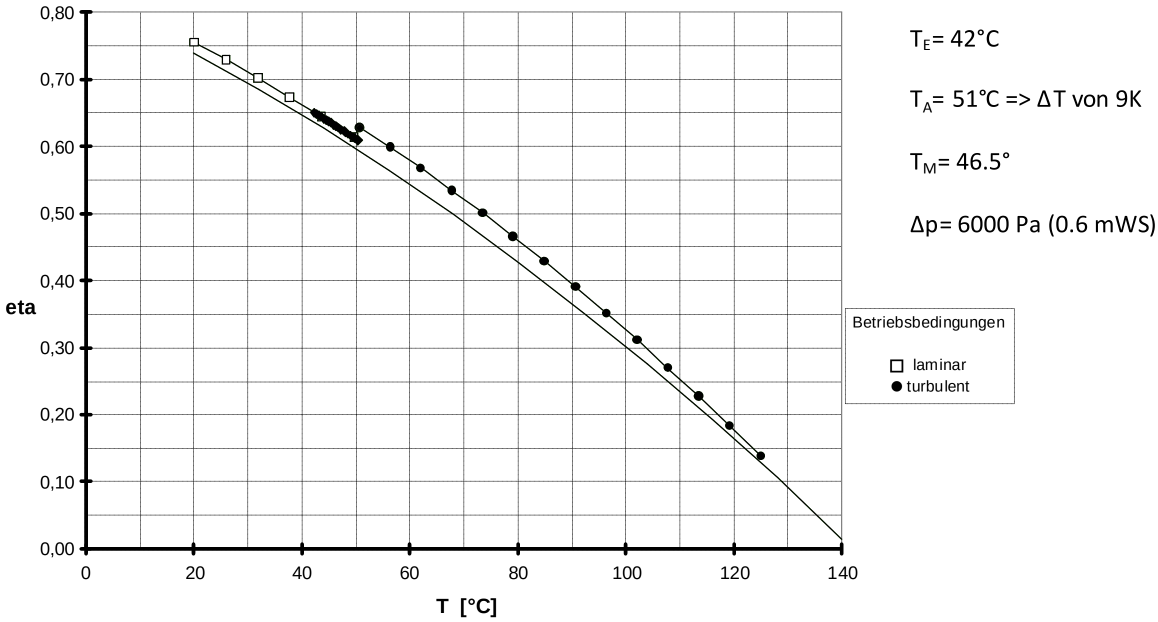
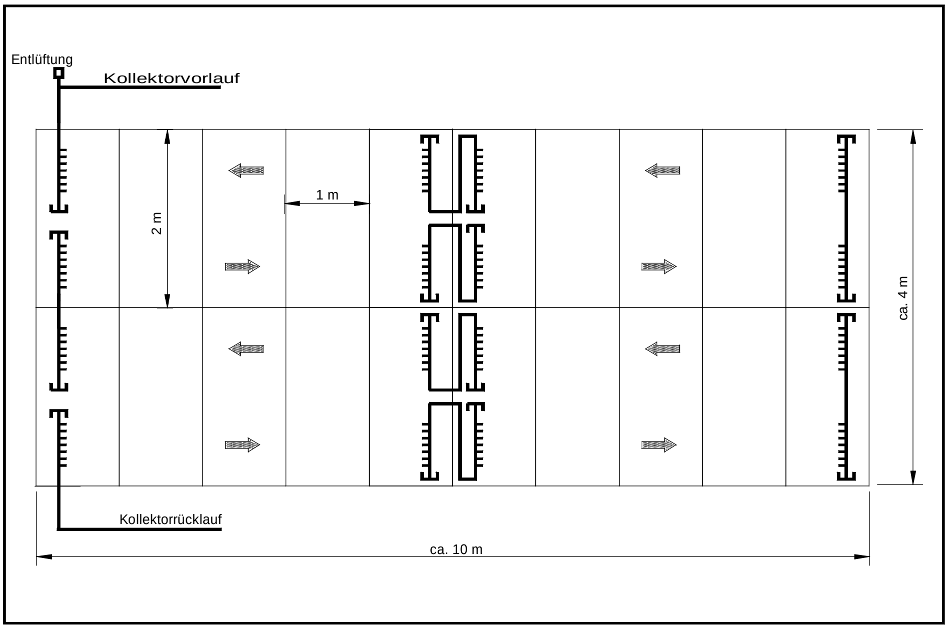


Figure 128: Efficiency curve under defined conditions (high-flow operation)



$\dot{m}_{primary}$ = 555 kg/h

$\dot{m}_{specific}$ = 14 kg/m²h

L_{Therm} = 38.4 m

Figure 129: Hydraulic collector scheme for a low-flow operation mode

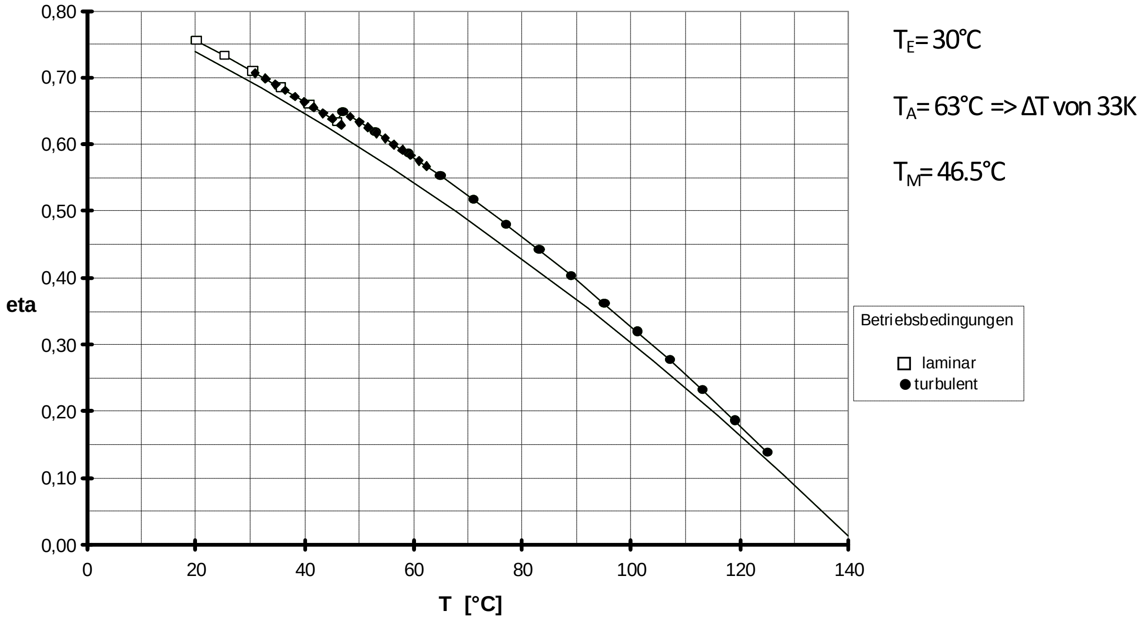


Figure 130: Efficiency curve under defined conditions (low-flow operation)

Pressure drop of the whole High-Flow System (2000 kg/h):

Component	**Pressure drop [Pa]**
37.03 net absorber area, high flow connected	6,000
Flat plate heat exchanger SWEP B25-30	16,700
Pipes ï collector loop 5/4ò	6,080
Other components of the system (flap trap, fittings, etc.)	4,000
Total	**32,780**

Pressure drop of the whole Low-Flow System (560 kg/h):

Component	**Pressure drop [Pa]**
37.03 net absorption area, low flow connected	17,200
Flat plate heat exchanger 2 x SWEP B15-20 in series	12,200
Pipes ï collector loop 3/4ò	6,000
Other components of the system (flap trap, fittings, etc.)	4,000
Total	**39,400**

Calculation of the hydraulic efficiency of the two systems:

$$P_{system} = \frac{\dot{m}\,\Delta p_{system}}{\rho \cdot 3600}$$

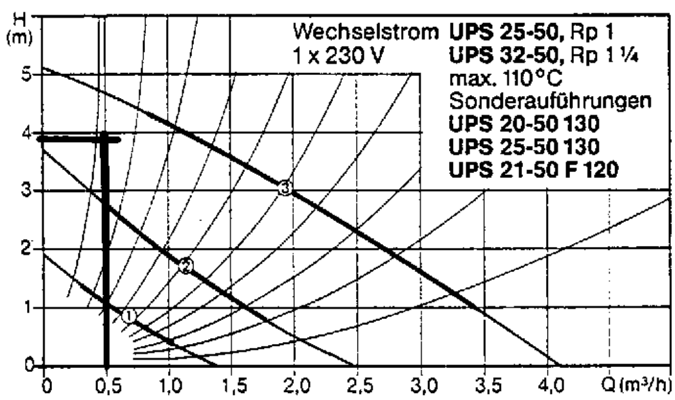
P_{system}	hydraulic efficiency	W
$\dot{m}$	mass flow	kg/h
Δp_{system}	pressure drop of the system	Pa
ρ	average density of the medium	kg/m³

Hydraulic efficiency high-flow system:

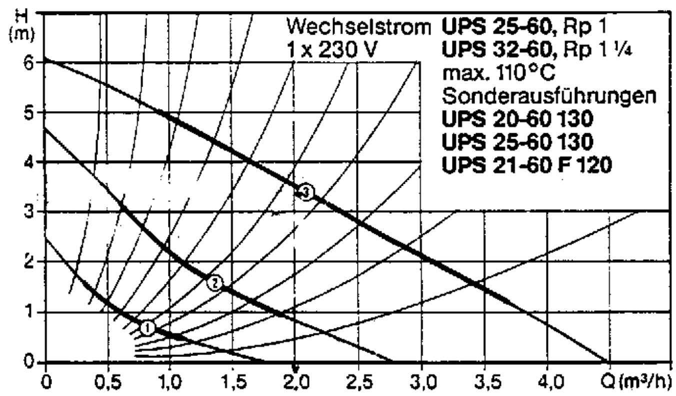
$$P_{system\_HF} = \frac{2037 \cdot 32780}{1039 \cdot 3600} = 18W$$

Hydraulic efficiency of the low-flow system:

$$P_{system\_LF} = \frac{555 \cdot 39400}{1039 \cdot 3600} = 6W$$



Low-flow system: UPS 25-50, Level 3



High-flow system: UPS 25-60, Level 3

Elektrische Daten

Typ	Stufe Drehzahl n [min⁻¹]	Leistungsaufn. P_1 [W]
UPS 25-20 UPS 32-20	3-2500 2-2050 1-1450	65 40 25
UPS 25-40 UPS 32-40	3-1850 2-1200 1- 750	75 50 30
UPS 25-50 UPS 32-50	3-1700 2-1050 1- 650	85 60 35
UPS 25-60 UPS 32-60	3-1800 2-1100 1- 700	100 65 40

Figure 131: Choice of pump

Conclusion:

=> pump efficiencies <20%
=> Required power of the pump of the low-flow system is about 15% lower than for high-flow system

For the heat exchanger applies the same as for the collector!

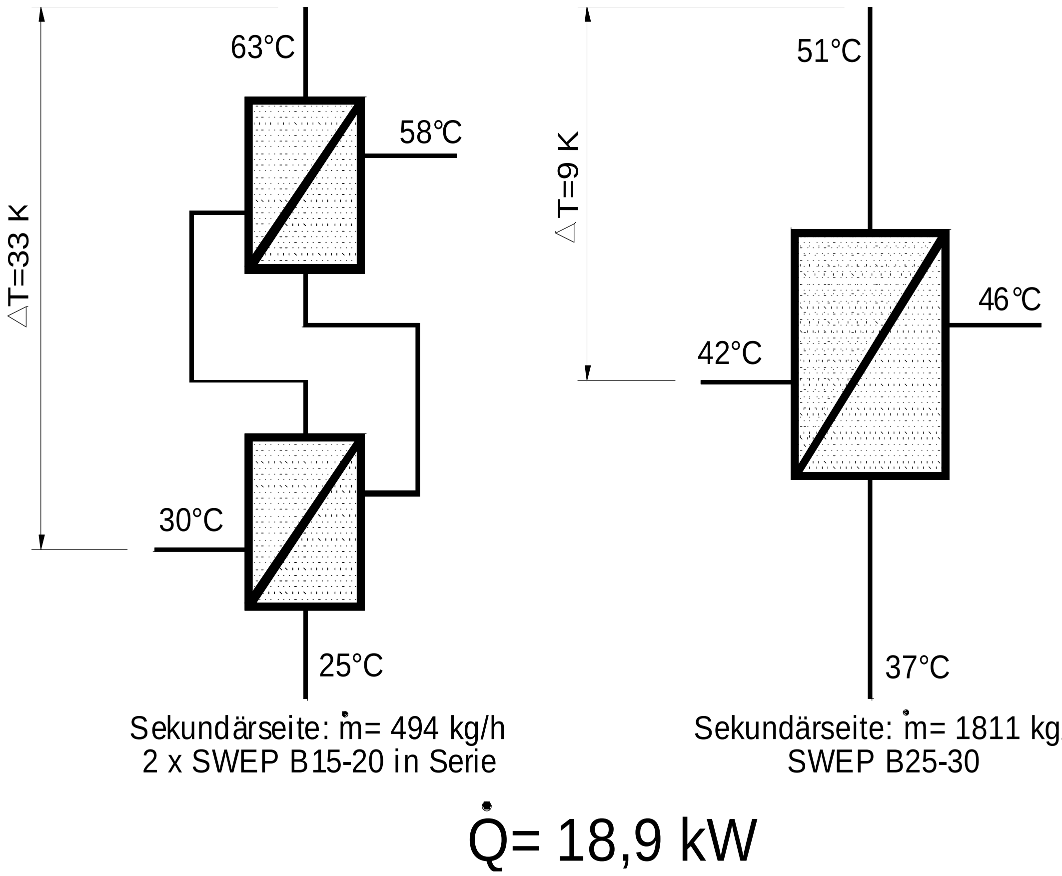


Figure 132: Scheme of both states

14.2 Calculation of the Expansion Vessel

14.2.1 Primary Fluid in the Vessel

The MEV must contain a certain amount of fluid at all states of operation of the system. That assures there is always enough fluid in the system.
When taking the system into operation, the pressure in the system (pressure of the fluid) is adjusted slightly higher (approx. 0.5 bar) than the pressure in the MEV. It is important that this adjustment is done as long as the whole system is cold and the pump is off. That guarantees the establishment of the primary fluid in the vessel, which is absolutely necessary.
In the state of stagnation of solar thermal collectors the heat transfer fluid vaporises. The primary fluid in the MEV must be able to cool down the hot fluid that comes from the collector (with temperatures up to 130 °C) to the maximum permissible temperature of the membrane (90 °C). For that reason there must be enough fluid in the expansion vessel already [20].

14.2.2 Primary Pressure in the MEV

In order to push back the expanded volume into the system and to make sure not too much fluid enters the expansion vessel a primary pressure is necessary.
If there is too little pressure, a lot of heat transfer fluid enters the expansion vessel at low temperatures in the system. The consequences would be that at higher temperatures no more fluid could enter the expansion vessel. At closed systems the content of the collector vaporises at stagnation and the expansion vessel must be able to take in the whole volume of the collector. Other wise the pressure of the system would excel the pressure that is necessary to release the safety valve, which would lead to a loss of fluid.
It is very important to check the primary pressure when installing the expansion vessel. It is further recommended to have periodical checks every one or two years.

14.2.3 Permeability

Inside the MEV the entrance of gas in the water is almost not possible. This is caused by the low permeability of the installed membrane. The use of expansion vessels that are open to the atmosphere is not recommended.

14.2.4 Design of the solar membrane expansion vessel

In general it must be said that it is better to choose the expansion vessel rather too big than too small!
The results of simulations of expansion vessels are often too optimistic. Certain processes in the solar thermal system like the stagnation have not (or not enough) been taken into account. Please see in the following a calculation method that considers the influence of the stagnation to the state of the art.

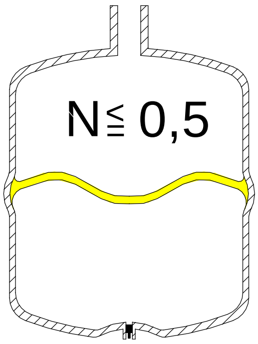


Figure 133: Actual use of the MEV (20)

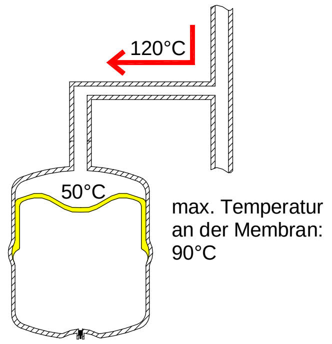


Figure 134: Primary fluid in the MEV (21)

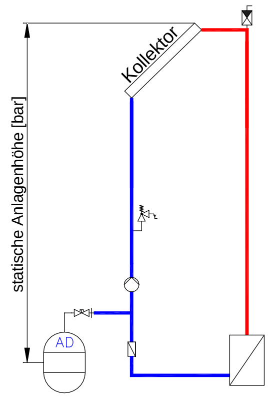


Figure 135: Minimal primary pressure in the MEV (20)

The following important parameters must be well-known for simulation and installation:
The efficiency is given by the manufacturer. It specifies the volume of the expansion vessel that can actually be used without expanding the membrane too much. This would lead to a damage on the membrane and to a lower life time of the expansion vessel. Usually the efficiency is lower than 0.5 .

The primary pressure in the expansion vessel should at least correspond to the static hydraulic head of the system. Because of the primary pressure the pressure in the system is high enough also in the upper parts of the system. It also prevents air from coming into the system when the heat transfer fluid cools down. After aerating the systems several times there must be enough fluid left. The primary fluid in the expansion vessel takes care of this. Additionally the primary fluid must protect the membrane against high temperatures at stagnation.
The nominal volume V_N of the expansion vessel (see equation) is calculated in the well known way from the heat expansion of the total content of the heat exchanging fluid V_G*n, the primary fluid V_V, the volume of the vapour V_D and the efficiency N. Compared to previous calculations a bigger volume of the vapour has to be taken into account. Corresponding to the given details above the volume of all pipes and components reached by the vapour has to be calculated.
Up to now the efficiency has only been calculated from the pressure of the system P_e and the primary pressure of the expansion vessel P_0. In the new method of calculation the difference of height H_{diff} between the expansion vessel and the safety valve is considered (equation 4). These components are maybe installed in different stores of the building and led to the pressure difference P_{diff}. The rise of temperature of the gas filling during operation is also considered (differences of 30 K have been measured). This leads to the quotient 0.9. These changes result from the application of the general gas law to the conditions at issue:

$$V_N > \frac{V_G \cdot n + V_V + V_D}{N}$$ (equation 1)

$$n = \frac{\rho_{cold}}{\rho_{hot}} - 1 \approx (0.09)$$ (equation 2)

$$N = \frac{P_e + P_{diff} + 1 - \frac{(P_0 + 1)}{0.9}}{P_e + P_{diff} + 1}$$ (equation 3)

$$P_{diff} = \frac{-H_{diff} \cdot \rho_{cold} \cdot 9.81}{100,000}$$ (equation 4)

Hereby means:

V_N	nominal volume of the expansion vessel	litre
V_G	total volume of the heat exchanging fluid	litre
V_V	primary fluid	litre
V_D	maximum volume of the vapour	litre
n	coefficient of expansion (~ 0.09 for expansion at ~120 °C for 40 % propylene glycol)	
N	efficiency of the expansion vessel, according to manufacturer Ò0,5	
ɟ	density of heat transfer fluid	kg/m³

P_e	pressure of system at safety valve = nominal pressure safety valve ï 20 %	bar
P_0	primary pressure, bar. The factor 0.9 in $(P_0+1)/0.9$ stands for a change of temperature in the gas containing space because of the hot fluid	
H_{diff}	difference of height between the expansion vessel and the Safety valve	
H_{diff}	= height of expansion vessel ï height of safety valve	m
P_{diff}	difference of pressure according to H_{diff}	bar

As mentioned above the primary fluid in the expansion vessel must be able to cool down the hot heat exchange fluid coming from the collector. The maximum permissible temperature in expansion vessel according to the manufacturer is 90 °C. The dimensioning of the minimum of primary fluid in the expansion vessel V_V is shown in equation 6:

maximum permissible temperature in expansion vessel T_{max} = 90 °C,
average temperature in the primary circuit 90 °C,
origin temperature of the primary fluid T_V = 50 °C (according to measures),
In the worst case the expansion vessel must be able to take in the whole volume of the collector V_K at a temperature of T_K = 130 °C.

$$V_V \geq V_K \cdot \frac{T_K - T_{max}}{T_{max} - T_V} \qquad \text{(equation 5)}$$

V_V	primary fluid	litre
V_K	volume inside the collector	litre
T_K	temperature of the fluid at entering the expansion vessel	°C
T_{max}	maximum permissible temperature in expansion vessel	°C

From this assumption it follows that the volume of the primary fluid must be equivalent to the volume of the collector.

Example of calculation:

Starting conditions:	Single family house
Collector area:	10 m² (collector that empties well)
Flow pipe V_L:	15 m Cu pipe 18x1
Return pipe V_L:	15 m Cu pipe 18x1
Safety valve:	6 bar
Pressure of the system:	2.5 bar
Primary pressure in the expansion vessel:	2.0 bar

We are looking for the volume of the expansion vessel [litre]

A) Formula for calculation

$$V_N > \frac{V_G \cdot n + V_V + V_D}{N}$$

MEV	nominal volume	V_N	litre
V_D	maximum vapour volume		litre
V_G	total volume of the heat transfer fluid		litre
V_V	primary fluid		litre
n	coefficient of expansion of the heat transfer fluid		

B) Calculation of the MEV efficiency:

$$N = \frac{P_e + P_{diff} + 1 - \frac{(P_0 + 1)}{0.9}}{P_e + P_{diff} + 1}$$

N	MEV efficiency	
P_e	nominal pressure of safety valve	bar
P_0	primary pressure	bar

$$P_{diff} = \frac{-H_{diff} \cdot \rho_{cold} \cdot 9.81}{100{,}000}$$

P_{diff}	pressure difference	bar	
H_{diff}	H_{MEV}ï H_{SV}	m	
r	density of the heat transfer fluid	kg/m³	~1051 kg/m³

$$P_{diff} = \frac{0.5 \cdot 1051 \cdot 9.81}{100{,}000} = 0.052 bar$$

P_e = nominal pressure of safety valve ï tolerance of respond (20 %)
P_e = 6 bar ï 20 % = 4.8 bar
Pressure difference P_{diff} = 0.052 bar
Nominal pressure of safety valve P_e = 4.8 bar
P_o = 2.0 bar

$$N = \frac{P_e + P_{diff} + 1 - \frac{(P_0 + 1)}{0.9}}{P_e + P_{diff} + 1} = \frac{4.8 + 0.052 + 1 - \frac{(2.0 + 1)}{0.9}}{4.8 + 0.052 + 1} = 0.43$$

C) Calculation of the volume of the heat transfer fluid:

$V_G = V_{pipe} + V_{coll} + V_{heat\ exchanger}$ litre

Factor of expansion n

$$n = \frac{\rho_{cold}}{\rho_{hot}} - 1 \approx 0.99$$

$$V_G = \frac{0.16^2 \cdot \pi}{4} \cdot 300 + 4.5 + 2.0 = 12.5 \text{ litre}$$

Calculation of the primary fluid V_V:
The volume of the primary fluid is more or less equivalent to the volume of the collector.
->$V_V = V_{coll}$ = 4.5 litre

D) Calculation of the volume of vapour V_D:

$V_D = V_{coll} + V_{pipe\text{-}vapor}$ litre

V_{coll} = 4.5 litre

Calculation of the volume of vapour in the pipes
Maximum vapor power= 10 m² * 50 W/m² = 500 W

Calculation of the reach of the vapour in the solar pipes $V_{pipe\text{-}vapor}$ (calculation through the thermal power loss of the pipes):
thermal power loss of the pipe: 25 W/m
(reach of vapour in the pipe) = (max. vapour power)/(thermal power loss of the pipe per meter of pipe)
(reach of vapour in the pipe) = 500/25 = 20 meter 16er Cu pipe

V_D = 4.5 + 4.0 = 8.5 litre

$$V_{pipe-vapour} = \frac{0.16^2 \cdot \pi}{4} \cdot 200 = 4.0 \text{ litre}$$

E) Calculation of the volume of the expansion vessel:

$$V_N > \frac{V_G \cdot n + V_V + V_D}{N} = \frac{12.5 \cdot 0.09 + 4.5 + 8.5}{0.43} = 32.8\, litre$$

Selection of the expansion vessel -> 35 litre

14.2.5 Installation

In principle the expansion vessel should be installed in a suspended way. The stationary mounting leads to the following effect:
If hot water streams by the expansion vessel it also enters the expansion vessel, because the density of hot water is lower than the density of the cold water inside the vessel

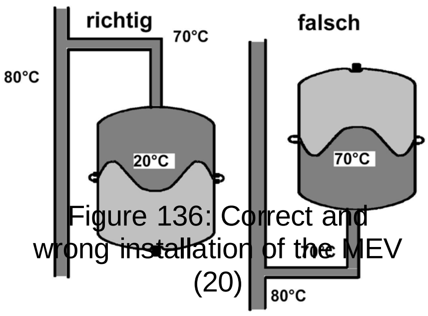


Figure 136: Correct and wrong installation of the MEV (20)

(compare figure 5, right). Usually the expansion vessel is not insulated. Therefore it looses a lot of heat. On the other side when bringing the system into operation hot fluid from the (stagnating) collector could enter the expansion vessel. The membrane is not able to stand high temperatures and will be destroyed in the long run. Therefore the correct installation is in a suspended way, and the hot water will always stream by the expansion vessel.
The suspended installation also allows the expansion vessel to aerate by itself: Air that is possibly existing in the expansion vessel can leave the vessel by rising in the pipes by itself and leave the system via the deaerator. Additionally no more air can enter the suspended vessel.
The expansion vessel is to be mounted without the possibility to lock it from the system. Never the less it is necessary to install lockable fittings in order to revise the system. Those fittings must be protected from closing them unintentional.

14.2.6 Maintenance

Maintenance work on the expansion vessel must be done at least every second year. It is preferable to do this when the solar thermal system is little used. E.g. in wintertime in the case of domestic hot water systems or at the beginning of the heating season at solar systems for space heating.
The vessel is locked from the system and pressure is taken from the water containing part. With a pressure-check-device for tyres the existing primary pressure of the vessel can be tested and if necessary added. The valve should be checked for leakage (e.g. with soapy water).

14.3 Calculation of Heat Exchangers

14.3.1 Introduction

Heat exchanger are necessary in order to transfer energy between two medias without mixing them. The performance of a heat exchanger should be as high as possible and the pressure drop as low as possible. They should be user friendly and of low maintenance. At solar thermal systems the heat exchanger is a very important component. The influence of the heat exchanging temperature is very high on the operation of the collectors.
In the primary circuit heat is generated in the collectors and a heat transfer fluid (mixture of water an anti freeze) is circulating. In the secondary circuit water is circulating (drinking water or water for space heating). The heat from the primary circuit should be transferred with a very low difference of temperature to the secondary circuit.
The temperature difference of a heat exchanger describes difference between the temperature at the entrance of the one circuit and the exit of the other circuit. The lower the temperature difference, the larger the area of the heat exchanger must be. Usually the temperature difference is about 5 K, lower temperature differences are not economically.
The covered distance of the media to transfer the heat (e.g. in a pipe) is called the thermal length. A very low temperature loss is aspired.
According to the type of heat exchanger the circulation in the secondary circuit takes place: free convection because of gravity in internal heat exchangers (finned tubes heat exchanger, smooth tubes heat exchanger) and forced convection in external heat exchangers (plate heat exchangers, coil heat exchangers).
In the following plate heat exchangers and internal heat exchangers are described in detail, as they are used the most.

14.3.2 Plate heat exchanger

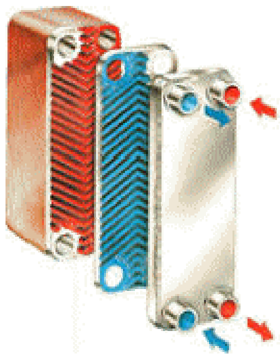


Figure 137: Current in a flat plate heat exchanger (12)

Plate heat exchangers are used at solar collector area of 15 m² and more. They are made of lined up plates. In between the plates there is a counter current of the heat transfer fluid. Due to the special character of the plates a turbulent current is generated. This raises the heat transfer. Plate heat exchangers can be soldered or screwed. At screwed plate heat exchangers the number of the plates can be changed and it is also possible to exchange single plates. At applications up to 300 kW soldered plate heat exchangers are used because of a number of advantages:

- They are very compact compared with ordinary coil heat exchangers. They save about 85 to 90 % in volume and weight.
- Maximum exploitation of the material: the capacity is 25 % higher as the capacity of screwed plate heat exchangers. The capacity is 10 times higher as the capacity of coil heat exchangers.
- Less use of energy because of a better heat transfer coefficient and subsequently a better temperature difference
- Heat transfer still at a temperature difference of 1 K
- Possibility of high pressures at operation
- Principle of counter current
- The high grades of turbulence leads to a self cleaning effect and subsequently to a minimization of costs and a longer lifetime
- One plate heat exchanger can be used to load more than one heat storage tanks

Disadvantages:
Like at all other external heat exchangers an additional pump is necessary on the secondary side of the heat exchanger.

Design
The thermal length of a plate heat exchanger can be varied by means of the installation length, the profile of the plates and also the numbers of plates. According to the design the pressure drop of the heat exchanger varies, too. A plate heat exchanger is always designed precisely to the task (heat transfer fluid, temperature difference, maximum pressure drop). It cannot be replaced by any other heat exchanger without another calculation.
In principle it is designed according to the required heat (kW). The collector area and the mass flow in the collector leads to this value. That means, the heat exchanger must be able to transfer the generated heat at full power (at maximum irradiance) of the collector to the heat storage.

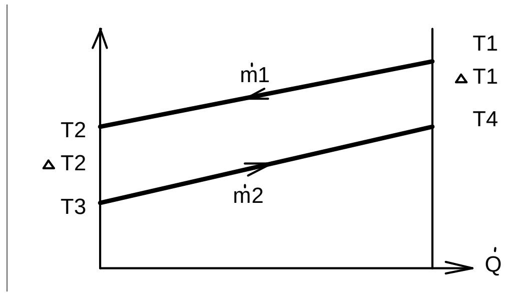


Figure 138: Temperature difference of a heat exchanger shown in a $\dot{Q}$-T-diagram (20)

The transferred power **Q** can be calculated by means of the temperature differences of the media, the mass flows of the media, the specific heat capacities, the U-value of the heat exchanger and the logarithmic temperature difference:

$$\dot{Q} = \dot{m}_1 \cdot cp_1 \cdot (T_1 - T_2) = \dot{m}_2 \cdot cp_2 \cdot (T_4 - T_3) \qquad \text{(equation 6)}$$

$$\dot{Q} = U \cdot A \cdot \Delta T_{log} \quad \text{with} \quad \Delta T_{log} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \qquad \text{(equation 7)}$$

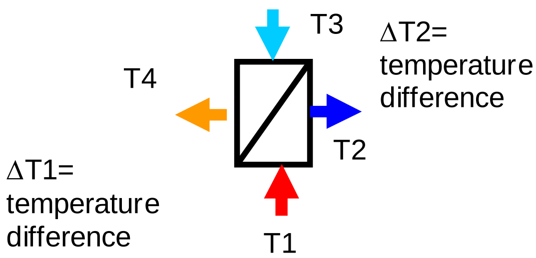


Figure 139: Temperature difference of a heat exchanger shown in a flow scheme (20)

$\dot{Q}$	transferred power	W
$\dot{m}_1, \dot{m}_2$	mass flow per second	kg/s
c_{p1}, c_{p2}	specific heat capacity	kJ/kgK
T_i	temperature of the media (see fig. 8)	°C
U	heat transmission coefficient of the heat exchanger	W/m²K
A	heat transfer area	m²
ΔT_{log}	logarithmic temperature difference	K

Plate hat exchangers reach u-values between 1000 and 2000 W/m²K (high power, low volume). Table 1 gives the necessary values in order to design a heat exchanger.

Table 15: Data for design of heat exchangers

		Primary circuit	Secondary circuit
Media		propylenglycol/water	water
Concentration of the fluid	%	40	0
Temperature at entrance	°C	65	25
Temperature at exit	°C	32	60
Mass flow	kg/s	0.25	ï *

*results from assumed data

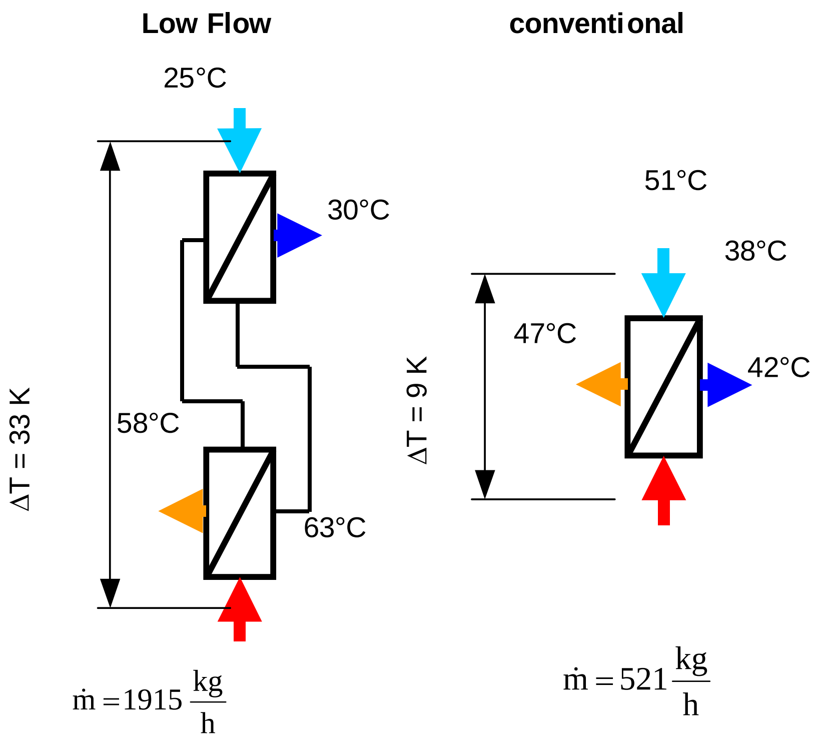


Figure 140: Comparison of mass flows and temperatures of a Low Flow (left) and conventional (right) operated plant (20)

With these data and with the pressure drop and the temperature differences the heat exchanger can be designed. For plate heat exchangers a temperature difference of 5 K can be reached economically. The calculated pressure drop should not go beyond 0.2 bar to keep the size of the pumps low. The temperatures on the side of the collector depend on the type of operation of the collector. Figure 9 shows typical values for the two types of operation, high-flow and low-flow.
The heat exchanger must be able to transfer the maximum appearing heat load. The maximum power of the collector is calculated in the following equation:

$P = \dot{Q} \cdot \eta_{coll} \cdot A_{coll}$ = 0.8·0.63·40 = 20.2 kW		kW	(equation 8)

P	power of the collector	kW
$\dot{Q}$	maximum irradiance, assumed to 0.8 kW/m²	kW/m²
η_{coll}	collector efficiency, assumed to 0.63	
A_{coll}	gross collector area, 40 m²	m²

Insulation

The insulation of the heat exchanger is usually offered by the manufacturer fitting to his type of heat exchanger. A self made insulation made of flexible foam can be used, too. Never the less it is recommended to use the insulation offered by the manufacturer, because there is only a little gap left to insulate with a foam.

At an average logarithmic temperature difference of 10 K the following rules of thumb are good for the dimensioning of an internal heat exchanger:

M	Smooth tube heat exchanger: approx. 0.2 m² heat exchanger surface per m² collector area Corded tube heat exchanger: approx. 0.3 ï 0.4 m² heat exchanger surface per m² collector area

The given numbers represent minimum values. The actual values should nit go below these minimum values!

References:

/1/ Duffie John A., Beckman William A.: Solar Engineering of Thermal Processes, Second Edition, New York, 1991

/2/ Suter J.M., Letz T., Weiss W., Inäbnit J. Solar Combisystems in Austria, Denmark, Germany, Sweden, Switzerland, the Netherlands and the USA, Overview 2000, Bern 2000

Picture Sources

(1) Franz Ketterer Feinmechanik, Sölden bei Freiburg (D)
(2) Feingerätebau K.Fischer GmbH, Drebach (D)
(3) SolarCosa, Berlin (D)
(4) Hessisches Landesamt für Umwelt und Geologie (D)
(5) Fachhochschule Mannheim, Institut für naturwissenschaftliche Grundlagen (D)
(6) Consolar GmbH, Frankfurt/Main (D)
(7) Streicher, W.: Sonnenenergienutzung, Graz, 2003 (A)
(8) Volker Quaschning (D), Consolar GmbH, Frankfurt/Main (D)
(9) Solution Solartechnik GmbH, Kirchdorf (A)
(10) AEE INTEC, 2002 (A)
(11) Anton Eder GmbH, Bramberg (A)
(12) Hauser Automatic ï Energietechnik/Regeltechnik, Wallisellen (CH)
(13) Martin Alberstötter, Gas-Wasser-Sanitär, Baindlkirch (D)
(14) Gestra GmbH, Bremen (D)
(15) Giacomini S.p.A, San Maurizio dŐOpaglio (I)
(16) VOSS Holding GmbH + Co. KG (D)
(17) PKP Prozesstechnik GmbH, Wiesbaden(D)
(18) Hanazeder Electronic GmbH, Ried im Innkreis(A)
(19) Deutsche Gesellschaf für Sonnenenergie: Solarthermische Anlagen, Berlin 2000 (D)
(20) Fink, et al.: Heizen mit der Sonne, AEE INTEC, Gleisdorf 1997 (A)
(21) AUTOTESTGERÄTE Leitenberger GmbH, Kirchentellinsfurt (D)
(22) Universität Regensburg (D)