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# The Reference Heating System, the Template Solar System of Task 32

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**A Report of IEA Solar Heating and Cooling programme - Task 32**

**“Advanced storage concepts for solar and low energy buildings”**

**Report A2 of Subtask A**

**May 2007**

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# The Reference Heating System, the Template Solar System of Task 32

by

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A technical report of Subtask A

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## Executive Summary

This work defines reference conditions and a reference simulation environment for the simulation of solar combisystems for domestic hot water (DHW) preparation and space heating (SH) for various European climates and a range of building loads. These will be used for simulation studies within the IEA-SHC Task 32 by all participants who intend to compare their new products or ideas with the current state of the art or with other products and system approaches.

The reference climates represent a moderate Northern European climate (Stockholm), a moderate, Central European climate (Zurich), a Mediterranean coastal climate (Barcelona) and a Mediterranean continental climate (Madrid). The heat loads of the reference buildings are 15, 30, 60 and 100 kWh/m<sup>2</sup>a for the climatic conditions of Zurich. All four buildings have the same architectural design with 140 m<sup>2</sup> of floor area, but different insulation thicknesses and/or different ventilation systems. The domestic hot water load is assumed to be 200 liters per day at 45 °C.

The reference simulation environment (RefSol) has been programmed with the simulation software TRNSYS. It includes meteorological data, building description files and other load-files together with the main file that contains the description of the system, the so called “deck”. The “deck” also contains the description of a standard solar system, solar store, auxiliary heater, pipes, mixing valves and radiators for space heating and the settings for the control of the system. The RefSol files have been continuously tested and improved and the final version described in this report is V45c. Test results concerning stability under changing simulation timesteps or different convergence tolerances are included in section 7 of this report.

Simulation results are written to ASCII text files. An Excel spreadsheet has been worked out that will load the simulation results with the help of Macros and calculate basic indicators for the performance of solar systems that are described in section 6 of this report. The spreadsheet also allows for a fast checking and detecting of energy balances and calculation errors.

This report may be much more detailed than other IEA-SHC Task 32 reports. The reason for this is that all participants who will perform comparative simulations within this Task will work with the reference conditions and the reference simulation environment and therefore depend on an accurate and precise description of these.

Whenever results from simulations based on the reference simulation environment are published, only the choices (climate, building, etc.) and changes made within the reference simulation environment will have to be reported by the author, together with a reference to this report.



## IEA Solar Heating and Cooling Programme

The *International Energy Agency* (IEA) is an autonomous body within the framework of the Organization for Economic Co-operation and Development (OECD) based in Paris. Established in 1974 after the first “oil shock,” the IEA is committed to carrying out a comprehensive program of energy cooperation among its members and the Commission of the European Communities.

The IEA provides a legal framework, through IEA Implementing Agreements such as the *Solar Heating and Cooling Agreement*, for international collaboration in energy technology research and development (R&D) and deployment. This IEA experience has proved that such collaboration contributes significantly to faster technological progress, while reducing costs; to eliminating technological risks and duplication of efforts; and to creating numerous other benefits, such as swifter expansion of the knowledge base and easier harmonization of standards.

The *Solar Heating and Cooling Programme* was one of the first IEA Implementing Agreements to be established. Since 1977, its members have been collaborating to advance active solar and passive solar and their application in buildings and other areas, such as agriculture and industry. Current members are:

Australia	Finland	Portugal
Austria	France	Spain
Belgium	Italy	Sweden
Canada	Mexico	Switzerland
Denmark	Netherlands	United States
European Commission	New Zealand	
Germany	Norway	

A total of 39 Tasks have been initiated, 30 of which have been completed. Each Task is managed by an Operating Agent from one of the participating countries. Overall control of the program rests with an Executive Committee comprised of one representative from each contracting party to the Implementing Agreement. In addition to the Task work, a number of special activities—Memorandum of Understanding with solar thermal trade organizations, statistics collection and analysis, conferences and workshops—have been undertaken.

The Tasks of the IEA Solar Heating and Cooling Programme, both underway and completed are as follows:

**Current Tasks:**

- Task 32 *Advanced Storage Concepts for Solar and Low Energy Buildings*
- Task 33 *Solar Heat for Industrial Processes*
- Task 34 *Testing and Validation of Building Energy Simulation Tools*
- Task 35 *PV/Thermal Solar Systems*
- Task 36 *Solar Resource Knowledge Management*
- Task 37 *Advanced Housing Renovation with Solar & Conservation*
- Task 38 *Solar Assisted Cooling Systems*
- Task 39 *Polymeric Materials for Solar Thermal Applications*

**Completed Tasks:**

- Task 1 *Investigation of the Performance of Solar Heating and Cooling Systems*
- Task 2 *Coordination of Solar Heating and Cooling R&D*
- Task 3 *Performance Testing of Solar Collectors*
- Task 4 *Development of an Insolation Handbook and Instrument Package*
- Task 5 *Use of Existing Meteorological Information for Solar Energy Application*
- Task 6 *Performance of Solar Systems Using Evacuated Collectors*
- Task 7 *Central Solar Heating Plants with Seasonal Storage*
- Task 8 *Passive and Hybrid Solar Low Energy Buildings*
- Task 9 *Solar Radiation and Pyranometry Studies*
- Task 10 *Solar Materials R&D*
- Task 11 *Passive and Hybrid Solar Commercial Buildings*
- Task 12 *Building Energy Analysis and Design Tools for Solar Applications*
- Task 13 *Advance Solar Low Energy Buildings*
- Task 14 *Advance Active Solar Energy Systems*
- Task 16 *Photovoltaics in Buildings*
- Task 17 *Measuring and Modeling Spectral Radiation*
- Task 18 *Advanced Glazing and Associated Materials for Solar and Building Applications*
- Task 19 *Solar Air Systems*
- Task 20 *Solar Energy in Building Renovation*
- Task 21 *Daylight in Buildings*
- Task 23 *Optimization of Solar Energy Use in Large Buildings*
- Task 22 *Building Energy Analysis Tools*
- Task 24 *Solar Procurement*
- Task 25 *Solar Assisted Air Conditioning of Buildings*
- Task 26 *Solar Combisystems*
- Task 28 *Solar Sustainable Housing*
- Task 27 *Performance of Solar Facade Components*
- Task 29 *Solar Crop Drying*
- Task 31 *Daylighting Buildings in the 21<sup>st</sup> Century*

**Completed Working Groups:**

*CSHPSS, ISOLDE, Materials in Solar Thermal Collectors, and the Evaluation of Task 13 Houses*

To find Solar Heating and Cooling Programme publications and learn more about the Programme visit **[www.iea-shc.org](http://www.iea-shc.org)** or contact the SHC Executive Secretary, Pamela Murphy, e-mail: [pmurphy@MorseAssociatesInc.com](mailto:pmurphy@MorseAssociatesInc.com)

*September 2007*

## IEA SHC Task 32 Subtask A

### “Evaluation and dissemination”

This report is part of Subtask A of the Task 32 of the Solar Heating and Cooling Programme of the International Energy Agency dealing with evaluation of new storage concepts.

The work in this report is based on previous works of the Task 26 “Solar combisystems” of IEA SHC, where a common reference case of building and solar combisystem was defined by all participants in order to compare their design to a “reference”.

However for the purpose of Task 32, the reference case has been much extended during the Task 32 period and can be considered as a strong basis for any comparison purposes of solar active systems for heating and cooling and seasonal storage for a one family house.

The Operating Agent would like to thank the authors of this document for their constant work on the reference case and reference system during Task 32 period (2003-2007). They have managed to take into account the numerous wishes and suggestions from all experts within Task 32, and to bring not only to IEA participants but to the whole solar energy community, a common framework for analyzing any solar active system.

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#### NOTICE:

The Solar Heating and Cooling Programme, also known as the Programme to Develop and Test Solar Heating and Cooling Systems, functions within a framework created by the International Energy Agency (IEA). Views, findings and publications of the Solar Heating and Cooling Programme do not necessarily represent the views or policies of the IEA Secretariat or of all its individual member countries.

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# 1 Approach and intention of this report

The intention of this work was to define reference conditions and a reference simulation environment for the simulation of solar combisystems for domestic hot water (dhw) preparation and room heating for various European climates and a range of building loads. This system, called “RefSol”, will be used for simulation-studies within the IEA-SHC Task 32 by all participants who intend to compare their new products or ideas with the current state of the art or with other products and system-approaches.

The used reference building, as well as other parameters, are based on a further development of the reference conditions of the IEA-SHC Task 26. All relevant information from IEA-SHC Task 26 is included in this report, and no particular knowledge of Task 26 reports is needed.

## 2 Reference climates

Data from four reference locations have been selected which cover the whole range from the tropics to temperate-cold climates; these data sets were used to perform the calculations and generate the results presented in this handbook. The four locations are:

Stockholm, Sweden: represents a moderate, Northern European climate.

Zurich, Switzerland: represents a moderate, Central European climate.

Barcelona, Spain: represents Mediterranean coastal climates with high humidity and temperature during summer and a moderate climate in winter. The typical cooling season is between April and October.

Madrid, Spain: represents a Mediterranean, continental climate with high temperatures during summer but moderate air humidity. The heating season is longer than on the coast.

The hourly climatic data profiles correspond to the profiles delivered with the TRNSYS simulation software [TRNSYS 16] in the standard format '.tm2'. These data profiles are based on the commercial database Meteonorm® [Meteotest 2003]. Tables 1 to 2 show the characteristics of the used climatic data sets.

City	Lat. [°]	Long. [°]	Alt. [m]	T <sub>AVE</sub> [°C]	T <sub>MIN</sub> [°C]	T <sub>MAX</sub> [°C]	I <sub>G</sub> [kWh/m <sup>2</sup> ]	I <sub>B</sub> [kWh/m <sup>2</sup> ]	I <sub>D</sub> [kWh/m <sup>2</sup> ]	RH [%]	Wind [m/s]
Stockholm / Arlanda	59.65	-17.95	61	5.31	-19.9	28.3	980	476	504	82.5	4.1
Zurich	47.38	-8.57	556	9.07	-9.7	29.2	1087	456	630	75.3	2.2
Barcelona City	41.42	-2.13	175	16.27	0.4	31.3	1444	731	713	72.7	3.6
Madrid / Barajas	40.45	3.55	582	13.91	-5.9	36.8	1662	1008	654	66.2	2.6

Table 1: Climate of the four reference locations of IEA SHC Task 32: general weather data

City	Lat. [°]	Long. [°]	Alt. [m]	T <sub>AMBD</sub> [°C]	T <sub>AMPL</sub> [°C]	d <sub>OFF</sub> [d]	HDD <sub>20/12</sub> [Kd]	HD [d]	T <sub>CW, AV</sub> [°C]	dT <sub>CW, AMP</sub> [°C]	dt <sub>CW, Shift</sub> [d]
Stockholm / Arlanda	59.65	-17.95	61	-17	10.5	24.2	4940	271	8.5	6.4	80
Zurich	47.38	-8.57	556	-10	9.3	20.3	3526	225	9.7	6.3	60
Barcelona City	41.42	-2.13	175	-1	7.2	28.1	980	94	13.5	4.5	19
Madrid / Barajas	40.45	3.55	582	-7	9.3	23.1	2139	170	13.5	4.5	19

Table 2: Climate of the four reference locations of IEA SHC Task 32: parameters related to building heating and dhw preparation

Alt. [m]	Geographical Altitude
d <sub>OFF</sub> [d]	Shift of day with minimum temperature with respect to January first
dT <sub>CW, AMP</sub> [K]	Amplitude of sine-curve describing the dependency of cold water temperature over the year
dt <sub>CW, Shift</sub> [d]	Shift of day with minimum cold water temperature with respect to January first
HD [d]	Heating Days / Accumulated length of heating periods in a year
HDD <sub>20/12</sub> [Kd]	Heating Degree Days for a room temperature of 20 °C and a balance point temperature (ambient temperature above which no heating is needed due to internal gains) of 12 °C
I <sub>B</sub> [kWh/m <sup>2</sup> a]	Beam incident irradiation on the horizontal plane
I <sub>D</sub> [kWh/m <sup>2</sup> a]	Diffuse incident irradiation on the horizontal plane
I <sub>G</sub> [kWh/m <sup>2</sup> a]	Global (Total) incident irradiation on the horizontal plane
Lat. [°]	Geographical Latitude
Long. [°]	Geographical Longitude

RH [%]	Relative humidity
$T_{AMB,D}$ [°C]	Design temperature of the building heating system
$T_{AMPL}$ [K]	Amplitude of sine-curve that describes the time dependence of a temperature over the year
$T_{AVE}$ [°C]	Average yearly ambient air temperature
$T_{CW,AV}$ [°C]	Average yearly cold water temperature
$T_{MAX}$ [°C]	Maximum ambient air temperature
$T_{MIN}$ [°C]	Minimum ambient air temperature

The climate Zurich is the basic climate for the definition of the building energy consumption. Cold water temperatures are calculated as dependent on the time of the year (Eq. 2.1).

$$Eq. 2.1 \quad T_{CW} = T_{CW,AV} + dT_{CW,AMP} \cdot SIN(360 \cdot (TIME + 24 \cdot (273.75 - dt_{CW,Shift})) / 8760)$$

$TIME$  [h]            time of the year

$T_{CW}$  [°C]            time of the year dependent temperature of cold water supply

Figures 1 to 3 show the annual distributions of monthly averages of ambient air temperatures, relative humidity and global horizontal radiation totals, respectively, for each of the selected sites.

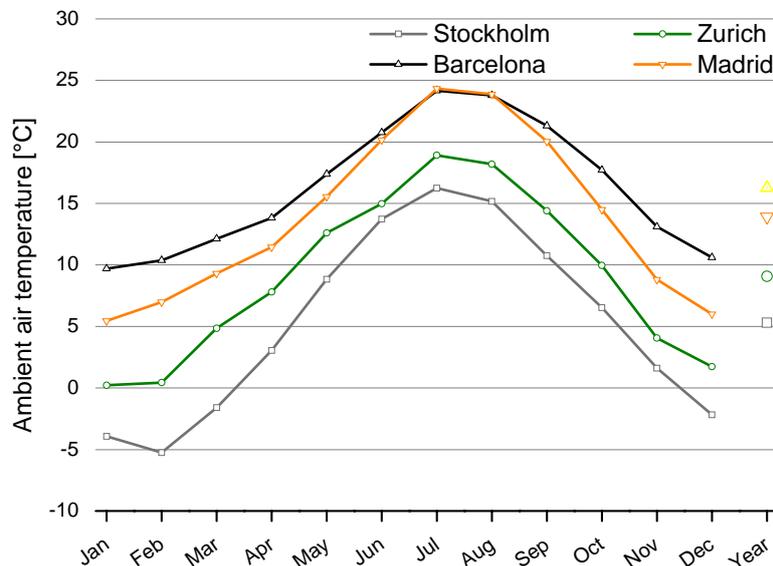


Figure 2.1: Annual distribution of average monthly ambient air temperatures for the four locations used in this task.

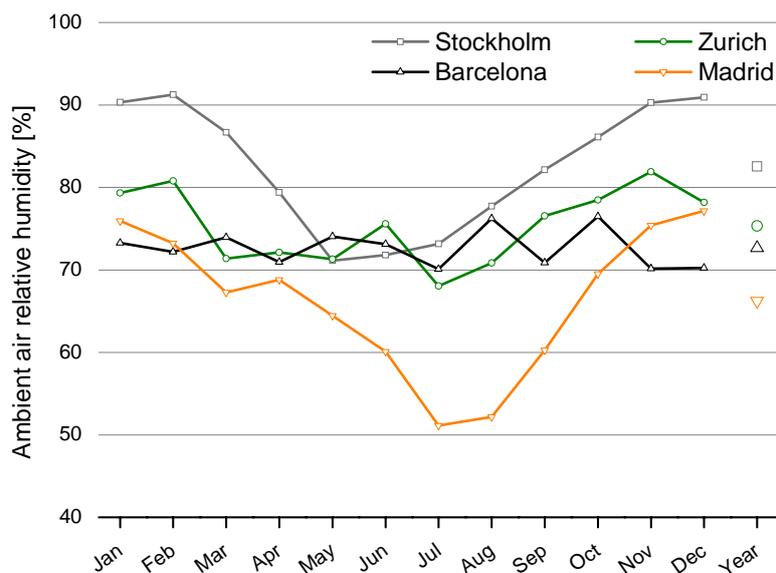


Figure 2.2: Annual distribution of average monthly ambient air relative humidity for the four locations used in this task.

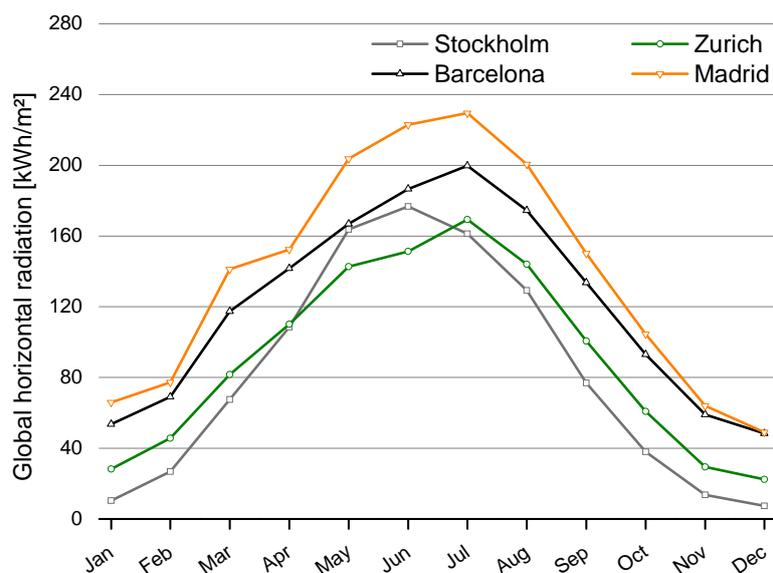


Figure 2.3: Annual distribution of average monthly global horizontal radiation totals for the four locations used in this task.

### 3 Reference building(s)

In this part the reference buildings are defined. The definitions include architectural design and orientation as well as the constructive descriptions of all building elements (walls, floors, windows and roof). Four single family houses with different heating loads are defined. The buildings are named after their heating loads (15, 30, 60 and 100 kWh/m<sup>2</sup>a) for the reference climatic conditions of Zurich with SFH 15, SFH 30, SFH 60 and SFH 100. All four buildings have the same architectural design, but different insulation thicknesses and/or different ventilation systems (see 3.2). In addition to the four buildings classified by heat loads, there is a fifth building that corresponds to SFH 100, but does not have the external venetian blinds shading implemented in the standard SFH 100, and therefore has an increased cooling demand in summer. This building will be referred to as the SFH 100n (no venetian blinds shading).

#### 3.1 Architectural design

The reference object is a free-standing, two storey single family building (see Figure 3.1). The effective floor area on one storey, including the areas covered by the internal walls and the access areas, is 70.0 m<sup>2</sup>. The glazed area on the south façades amounts to 25 % of the façade area, 10 % on the east and west façades and 6 % on the north façade. Both storeys are simulated as one common thermal zone.

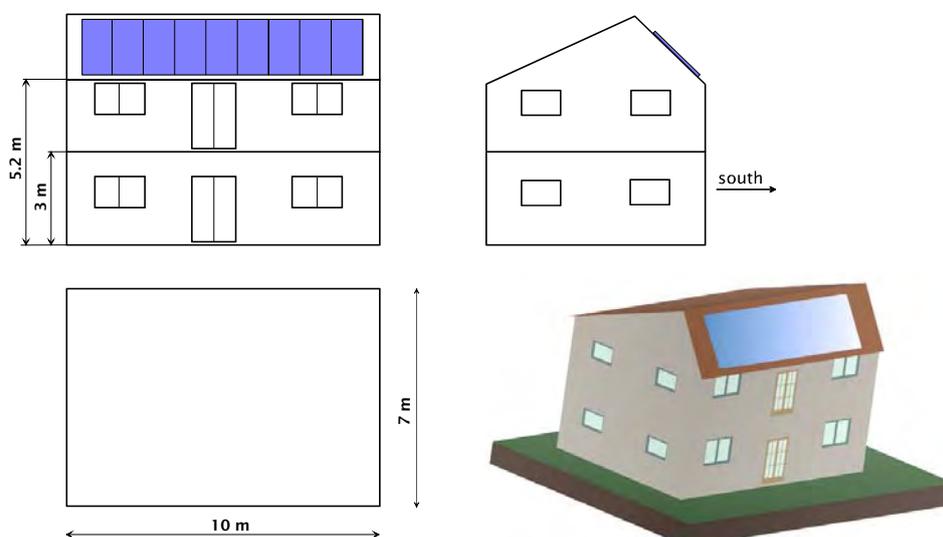


Figure 3.1: Sketch of the single-family house (SFH) used in IEA SHC Task 32.

## 3.2 Façade structures

### 3.2.1 Wall construction

The construction of the building envelope (external walls, roof and floor slab between ground floor and basement) is more or less reduced to the base slab (storage layer) and the insulation layer. The various materials, layer thicknesses and energy performance data describing the reference building(s) are listed in Table 3.

Figure 3.1 displays the geometry and dimensions of the office building. The resulting areas and volumes are summarized in Table 3.

assembly	layer	layer thickness				density	conduct.	capacity	U - Value construction			
		SFH 15	SFH 30	SFH 60	SFH 100				SFH 15	SFH 30	SFH 60	SFH 100
		[m]	[m]	[m]	[m]	[kg/m <sup>3</sup> ]	[W/mK]	[kJ/kgK]	[W/m <sup>2</sup> K]	[W/m <sup>2</sup> K]	[W/m <sup>2</sup> K]	[W/m <sup>2</sup> K]
external wall	plaster inside	0.015	--	--	--	1200	0.600	1.00				
	Viertl brick	0.210	--	--	--	1380	0.700	1.00	0.154	0.154	0.283	0.491
	EPS	0.240	--	0.120	0.060	17	0.040	0.70				
	plaster outside	0.003	--	--	--	1800	0.700	1.00				
	Σ	0.468	0.468	0.348	0.288							
ground floor	wood	0.015	--	--	--	600	0.150	2.50				
	plaster floor	0.060	--	--	--	2000	1.400	1.00	0.157	0.157	0.270	0.561
	XPS	0.220	--	0.120	0.060	38	0.037	1.45				
	concrete	0.150	--	--	--	2000	1.330	1.08				
	Σ	0.445	0.445	0.345	0.285							
roof ceiling	gypsumboard	0.025	--	--	--	900	0.211	1.00				
	plywood	0.015	--	--	--	300	0.081	2.50	0.119	0.119	0.275	0.380
	rockwool	0.280	--	0.120	0.060	60	0.036	1.03				
	plywood	0.015	--	--	--	300	0.081	2.50				
	Σ	0.335	0.335	0.175	0.115							
internal wall	clinker	0.200	--	--	--	650	0.230	0.92	0.962	0.962	0.962	0.962

Table 2: Construction of building elements for the 15, 30, 60 and 100 kWh/m<sup>2</sup>a single-family house.

	net values						gross values						
	dimensions			floor area	facade area (north/south)	volume	dimension			floor area	facade area (north/south)	facade area (east/west)	volume
	L	W	H	L x W	L x H	L x W x H	L	W	H	L x W	L x H	W x H	L x W x H
	[m]	[m]	[m]	[m <sup>2</sup> ]	[m <sup>2</sup> ]	[m <sup>3</sup> ]	[m]	[m]	[m]	[m <sup>2</sup> ]	[m <sup>2</sup> ]	[m <sup>2</sup> ]	[m <sup>3</sup> ]
EG	10.0	7.0	2.6	70.0	26.0	182.0	10.5	7.5	3.0	78.8	31.5	22.5	236.3
OG	10.0	7.0	3.0	70.0	30.0	210.0	10.5	7.5	3.4	78.8	35.7	25.5	267.8
Summary				140.0	56.0	392.0				157.5	67.2		504.0

Table 3: Areas and volumes of the reference single-family house (SFH). Gross values are calculated with a hypothetical wall-thickness of 0.25 m and a floor-thickness of 0.4 m. Please adjust for the reference building you are interested in if you need values that are more precise.

### 3.2.2 Windows

The window areas are the same for all four building types (15, 30, 60 and 100 kWh/m<sup>2</sup>a) and are shown in detail in the Table 4.

	window area [m <sup>2</sup> ]	window quotient [%]	total area [m <sup>2</sup> ]
South	12.0	24.0	50.0
East	4.0	9.9	40.5
West	4.0	9.9	40.5
North	3.0	6.0	50.0
<b>Summary</b>	<b>23.0</b>	<b>12.7</b>	<b>181.0</b>

Table 4: Window areas in m<sup>2</sup> for the 15, 30, 60 and 100 kWh/m<sup>2</sup>a single-family house

Depending on the thermal quality of the building envelope, different window types are chosen. The relevant data are for the four buildings are summarized in the Table 5.

building	U <sub>WINDOW</sub> [W/m <sup>2</sup> K]	g-Value [-]	U <sub>FRAME</sub> [W/m <sup>2</sup> K]	construction [mm]	window ID [-]
SFH15	0.52	0.585	1.6	4/16/4/16/4	13006
SFH30	0.52	0.585	1.6	4/16/4/16/4	13006
SFH60	1.40	0.622	2.3	4/16/4	2004
SFH100	2.83	0.755	2.3	4/16/4	1202

Table 5: thermal properties of windows for the 15, 30, 60 and 100 kWh/m<sup>2</sup>a single-family house

### 3.2.3 Heating system parameters

building	$Q_{d,ZH}$ [kJ/h]	$T_{d,in}$ [°C]	$T_{d,out}$ [°C]
SFH100	23573	60	50
SFH60	16578	40	35
SFH30	11160	35	30
SFH15	6606	35	30

Table 6: building dependent values used for heating system simulation

$Q_{d,ZH}$  [kJ/h] standard heat load at design outdoor temperature for Zurich

$T_{d,in}$  [°C] design inlet temperature of the heating system

$T_{d,out}$  [°C] design outlet temperature of the heating system

### 3.3 Internal gains

#### 3.3.1 Inhabitants

In the case of full occupancy 3.3 people are present simultaneously in the reference building. This value is based on a mean value from several building-objects evaluated during an Austrian study published in 2004 [HdZ15/2004]. No distinction is made between weekdays and weekend. Numerical values for the overall occupancy profile are displayed in Figure 3.2.

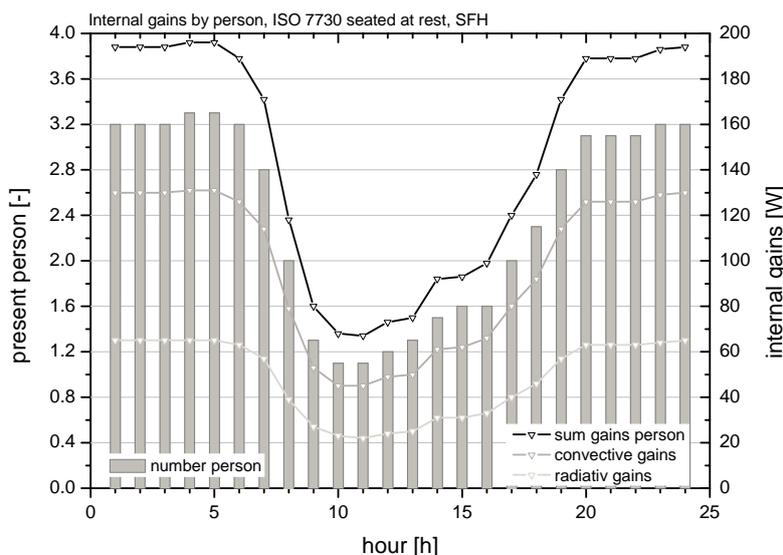


Figure 3.2: Occupancy profile for each day of the week within the reference building.

The heat dissipation to the surroundings is assumed to be 100 Watt per person. This value is in accordance with the ISO 7730 [ISO 7730] standard and is based on the following scenario:

Degree of activity – seated at rest

<b>sensible heat:</b>	<b>60 W/person</b>
<i>convective gains:</i>	40 W/person
<i>radiative gains:</i>	20 W/person
<b>latent heat:</b>	<b>40 W/person</b>
<i>humidity:</i>	0.059 kg/hr person

The average internal gains through inhabitants are about 145 W or 1.04 W per square meter net area of the building. This results in a yearly heat gain of 9.03 kWh/m<sup>2</sup> by the inhabitants of the building.

### 3.3.2 Electricity

The total electricity consumption of an average household amounts up to 3043 kWh/year [Feist 1998]. We assumed that 58% of this electric energy (12.6 kWh/m<sup>2</sup>a or 1.44 W/m<sup>2</sup> in our case), remain inside the building as thermal energy and are classified as internal electrical gains.

The rate of heat dissipation from electric equipment depends on its operational status and is thus correlated to the occupancy profile in the building. The assumed electrical heat dissipation is assumed to be dependent on the hour of the day and on the day of the week and is shown in Figure 3.3. This weekly profile is based on outcomes from an Austrian study [HdZ15/2004], where about 60 flats are measured in detail.

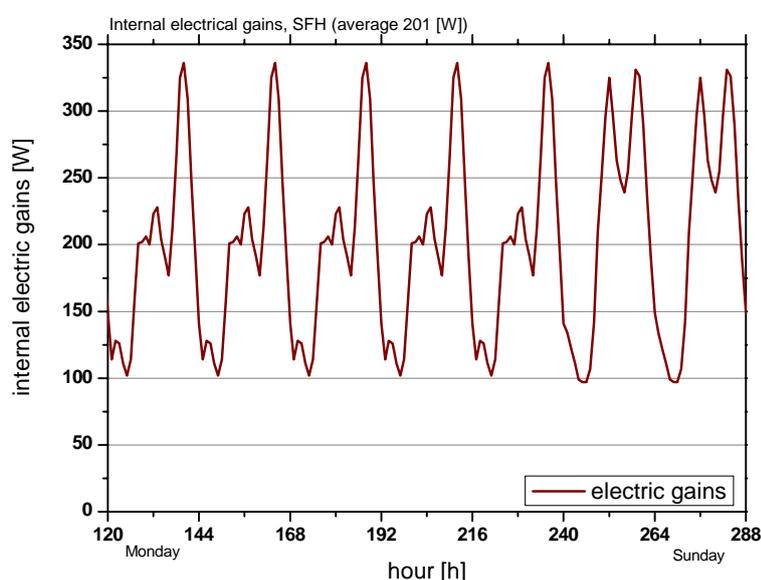


Figure 3.3: Weekly internal electricity gains profile for the reference building.

### 3.4 Thermal comfort, PMV, PPD and clothing level

Creating a thermally comfortable environment is one of the most important parameters to be considered when designing buildings. Thermal comfort is defined in the ISO 7730 standard as: "That condition of mind which expresses satisfaction with the thermal environment" [ISO 7730]. A definition most people can agree on, but also a definition which is not easily converted into physical parameters. In this reference simulation environment, the PMV<sup>1</sup> and

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<sup>1</sup> PMV represents the 'predicted mean vote' (on the thermal sensation scale) of a large population of people exposed to a certain environment. PMV is derived from the physics of heat transfer combined with an empirical fit to sensation. PMV establishes a thermal strain based on steady-state heat transfer between the body and the environment and assigns a comfort vote to that amount of strain.

PPD<sup>2</sup> are calculated and printed out in a separate printer to be evaluated by those who want to work with these parameters. In addition to this, penalty values are calculated if the room temperatures are outside a limited “comfort” temperature range and this penalty is used in the calculation of performance indicators described in section 6.1 and 6.4. If the penalty is too large, then it is recommended to discard the simulation results (see section 7.1).

The Thermal Comfort Equation (PMV, PPD) derived by P.O. Fanger [ISO 7730] combines the effect of six parameters:

- Metabolism (met)<sup>3</sup>
- Clothing level (clo)<sup>1</sup>
- Air Velocity<sup>1</sup>
- Air Temperature<sup>4</sup>
- Mean Radiant Temperature<sup>2</sup>
- Humidity<sup>2</sup>

### ***Metabolic rate estimation***

The metabolism is the energy released by oxidation processes in the human body and depends on the muscular activity. Normally, all muscle activity is converted to heat in the body, but during hard physical work this ration may drop to 75%. For instance when walking up-hill, part of the energy is stored in potential energy of the body.

Traditionally, metabolism is measured in met (1 met=58 W/m<sup>2</sup> of body surface). A normal adult has a surface of 1.7 m<sup>2</sup>, and a person in thermal comfort with an activity level of 1 met will thus have a heat loss of approximately 100 W.

When evaluating the metabolic rate, it is important to use an average of the activity the person has performed within the last hour. Met values for various activity levels can be found in EN ISO 7730.

In our case the metabolic rate is fixed for all persons through the year with an average value of

$$\text{Eq. 3.1} \quad \text{met} = 1.2$$

which corresponds to the degree of activity “seated, light work”.

### ***Clothing level estimation***

Clothing reduces the body's heat loss. Therefore, clothing is classified according to its insulation value. The unit normally used for measuring clothing's insulation is the clo unit. The more technical unit m<sup>2</sup>C/W is also used frequently (1 clo = 0.155 m<sup>2</sup>C/W).

The clo value can be calculated by adding the clo value of each individual garment. The insulation value for individual garments can be found in EN ISO 7730. In our case the

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<sup>2</sup> PPD is the predicted percent of dissatisfied people at each PMV. As PMV changes away from zero in either the positive or negative direction, PPD increases.

<sup>3</sup> Defined value within the building simulation

<sup>4</sup> Calculated value within the building simulation

clothing level is fixed for all persons through the year with a seasonal varying value. The value range is from 1.0 “typical business suit” to 0.5 “light summer clothing” (see Figure 3.4).

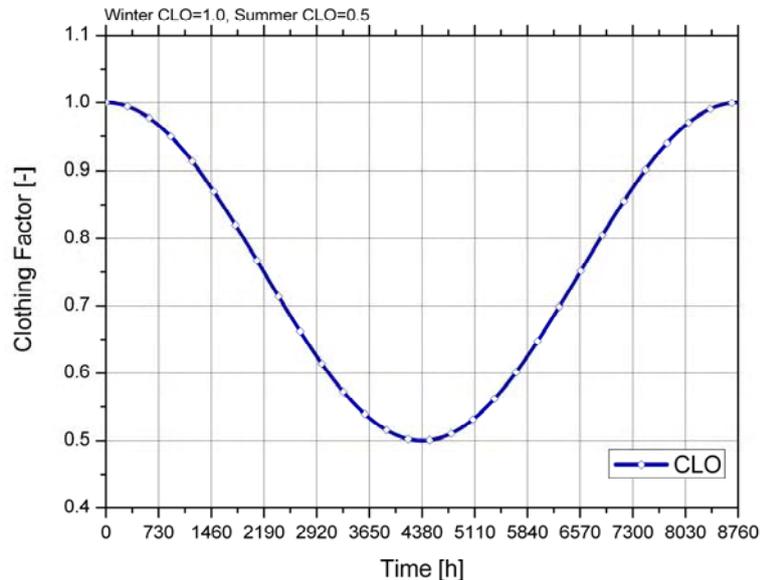


Figure 3.4: Seasonal varying clothing level [clo] used within the reference building.

### **Air velocity estimation**

Air movement in ventilated rooms is a combination of forced flow from the ventilation system and free-convection flows caused by temperature differences in the room. Recording air velocity reveals strongly fluctuating patterns of a stochastic nature.

The feeling of comfort in an air-conditioned office is, among other physical parameters, a combination of air velocity and temperature fields. The most common complaint in relation to indoor climate is draught, defined as an undesirable local cooling of the body caused by air movement.

In our case the air velocity is fixed through the year with a preset-value of

$$\text{Eq. 3.2} \quad v_{AIR} = 0.1 \text{ [m/s]}$$

## **3.5 Wing wall shading**

Type 34 is used in the upstream component mode 0 for the calculation of a wing wall. The parameters are the following:

- Receiver height            4 m
- Receiver width            4 m
- overhang projection        1.5 m
- overhang gap                1 m
- overhang left extension    2 m
- overhang right extension   2 m

### 3.6 External venetian blinds shading

Overheating due to the solar gains through the glazed areas is reduced by external shading devices. Their transmission is shown in Figure 3.5.

Control of the external shading is modeled as a function of the incident solar irradiation on the horizontal and depending on the room temperature. This shading is not used in the building **SFH 100n** (no venetian blinds shading)

**Control strategy (all points must be fulfilled, off-values for hysteresis in parenthesis):**

- Total global irradiation on the horizontal is above 300 (200) W/m<sup>2</sup>
- Room temperature must be above 23.8 (22.8) °C
- 24 hours average ambient temperature must be above 12 °C

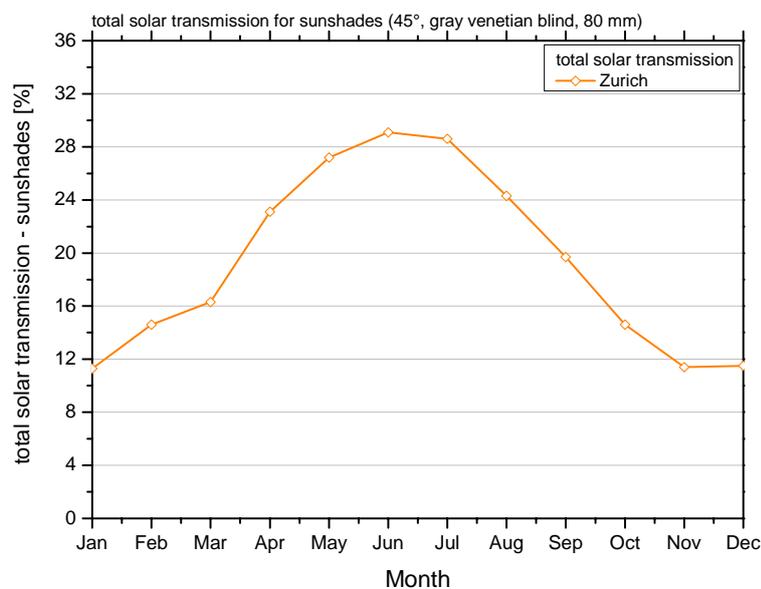


Figure 3.5: Month-dependent solar transmission for the used sunshades (grey venetian blinds, 80 mm slats, tilt-angle 45°, calculated with the program Parasol<sup>5</sup>).

<sup>5</sup> <http://www.parasol.se/>

### 3.7 Ventilation

A natural air change rate of 0.4 is assumed due to leakages in the building envelope during wintertime. In summertime a free driven night ventilation mode is activated if the following conditions are met:

- Time between 21:00 and 8:00
- Daily average temperature is above 12 °C
- Room temperature is above 24 °C
- Ambient temperature is 2 K below the actually room temperature

This free driven tilted window ventilation is simulated with Type 358 based on the method from A. Weber [WEBER 1997] using a simple but CFD validated Bernoulli – approach. The air exchange rate through the window is calculated depending on the temperature difference between room and ambient temperature, the geometry of the window and the window opening tilt angle (see section 10).

Three windows with tilt angles of 10° have been defined. Two windows are 1.04m x 1.04m (H x W) and one window is 2.00m x 0.94m (H x W).

Figure 3.6 displays the function of the combination between free and fixed ventilation. The dark blue line shows the used air change rate and the light blue one is calculated free driven air change rate.

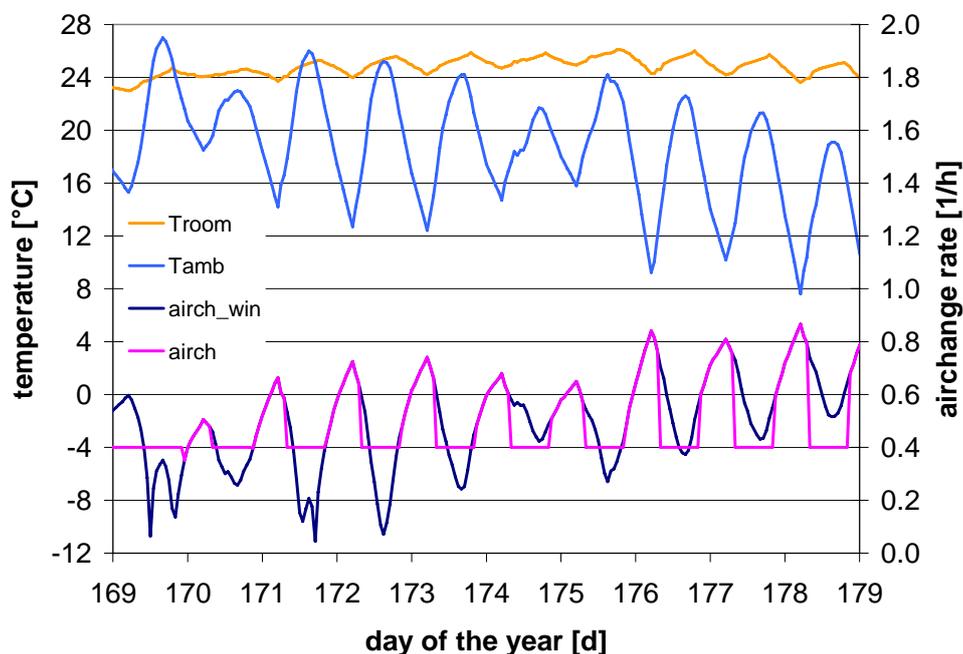


Figure 3.6: Temperature and the calculated free ventilation curve over ten days of the year for Zurich SFH 15. *Troom*: room temperature, *Tamb*: ambient temperature, *airch\_win*: natural airchange rate as a result of open windows simulation (day and night), *airch*: natural airchange rate assuming open windows only at night and with a minimum of 0.4 h<sup>-1</sup> during day and night (used for simulation). Room temperatures are relatively high because of a large influence of internal loads (passive house) and effect of increased night ventilation is relatively small because of high thermal masses.

### 3.8 Heating and cooling set points

#### Heating season

The controls of the heating system were set in a way, that the room temperature is kept around  $T_{\text{ROOM}} = 20 \pm 0.5 \text{ } ^\circ\text{C}$  and never drops below  $19.5 \text{ } ^\circ\text{C}$  during heating season.

#### Cooling season

The necessary cooling set temperature of the reference system is defined according to the upper cooling line of DIN 1946/2 with two straight lines depending on the ambient temperature (see Figure 3.7). However, penalties are given for not cooling to the lower cooling line that is set to  $25 \text{ } ^\circ\text{C}$  up to an ambient temperature of  $26 \text{ } ^\circ\text{C}$  both, for the reference system without solar assisted cooling and for solar systems that provide cooling (see section 6.4).

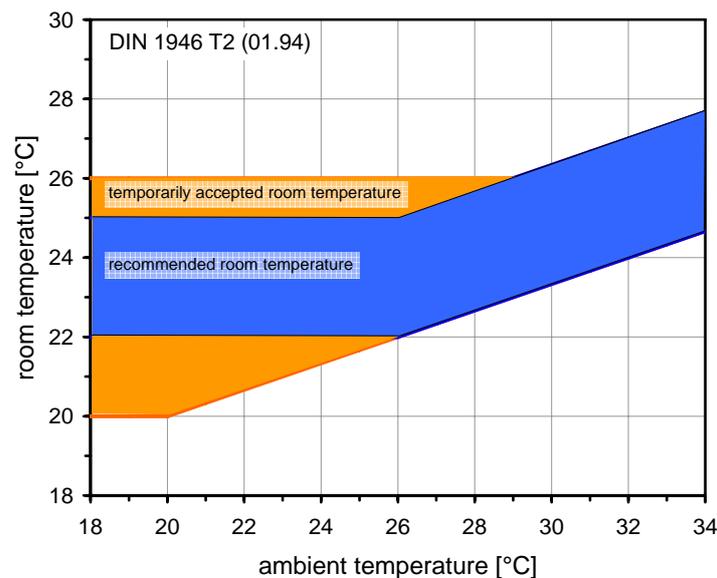


Figure 3.7: Boundary set temperatures for heating and cooling according to the DIN 1946/2.

## 4 DHW-load profiles

Domestic hot water load profiles are stored in two data-files calculated with the program form Jordan and Vajen [Jordan 2000, Jordan 2001] which can be downloaded free of charge from the homepage of the Solar Energy Laboratory of the University of Wisconsin, Madison.<sup>6</sup>

<sup>6</sup> <http://sel.me.wisc.edu/trnsys/trnlib/library15.htm#IEA26Load>

The load files are based on a timestep of 6 minutes and a daily hot water consumption of 200lt (06Dhw002.txt). The default profile for this setting was chosen from the download.

The dhw-load in kWh/a is different for different climates, as the cold water temperature is calculated dependent on the climate and the time of the year as described in Eq. 2.1. The set temperature for hot water preparation is 45 °C for all cases.

## 5 Simulation environment

For simulation studies, a TRNSYS-deck have been established that includes the reference building(s) and gives users the option of choosing one of the above mentioned climate data-sets. The TRNSYS-deck also includes a solar system, a water storage tank and all utilities needed for the simulation of a fully operating solar combisystem. The schematic layout of the system can be seen in Figure 5.1.

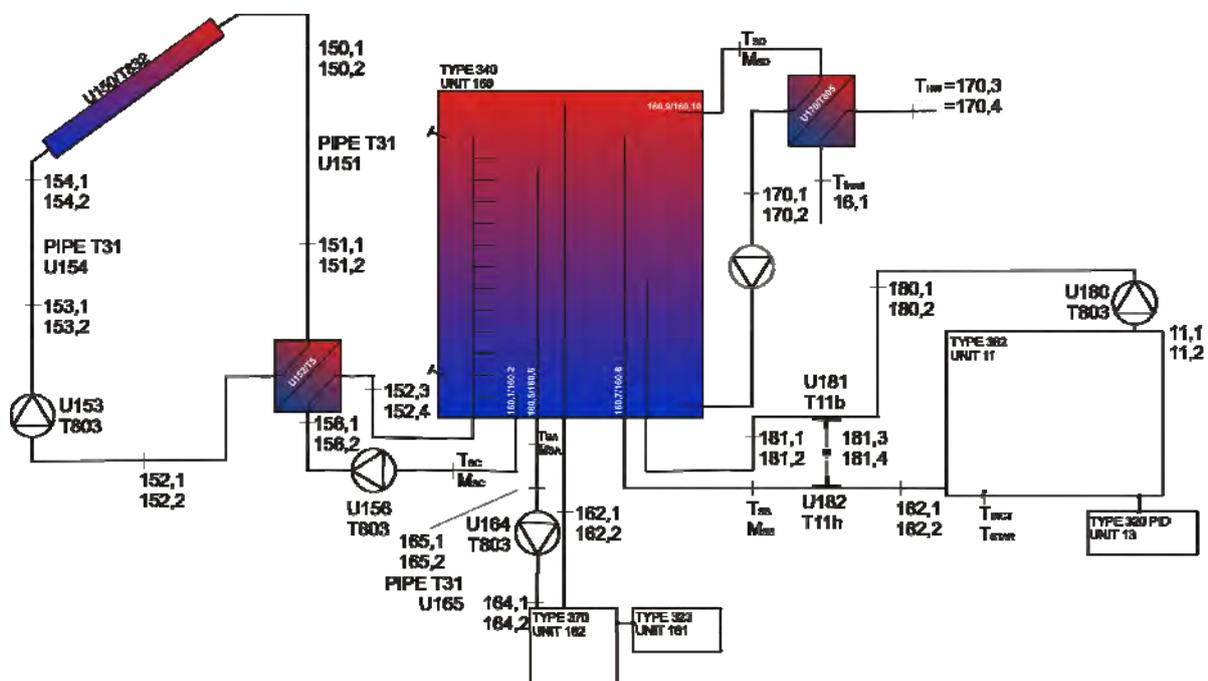


Figure 5.1: Total system layout – template solar system

TRNSYS Types and versions used for the simulation of the components are listed in section 9.

### 5.1 General simulation parameters

Tolerances for integration errors and convergence are both set to 0.003. With tolerances set to 0.001, convergence problems may occur in the auxiliary loop. Therefore, the lower limit of this value has been set to 0.002. The default values are used for the TRNSYS differential

equations solver (DFQ 1), equations solver (EQSOLVER 0)<sup>7</sup> and solver (SOLVER 0). Values for thermal capacity and density of the fluids used in the simulations are listed in Table 7.

Fluids	Cp [kJ/kgK]	Density [kg/m <sup>3</sup> ]	Used for
Brine	3.186	1.016	Collector Loop
Water	4.19	0.998	Secondary collector loop, store, loops for auxiliary heating, dhw preparation and space heating

Table 7: Properties of fluids used

## 5.2 Collector loop

### 5.2.1 Collector

For collector simulation, Type 832 from Bengt Perers and Chris Bales is used (Type 832, Version Number 2.06) [Perers 2002]. All reference collectors are calculated in Mode 1, without wind or long-wave dependency and with the sky radiation factor set to zero.

Ref. Collector	$\eta_0$ [-]	$a_1$ [W/m <sup>2</sup> K]	$a_2$ [W/m <sup>2</sup> K <sup>2</sup> ]	Cp [J/m <sup>2</sup> K]	$K_D$	$b_0$	Omode
flat-plate selective	0.80	3.50	0.015	7000	0.90	0.18	1
flat-plate non-selective	0.75	5.46	0.021	7000	0.88	0.18	1
evacuated tube	0.773	1.09	0.0094	44400	0.95	--	4

Table 8: Reference Collector Data for Type 832. Omode = optical mode. In Optical Mode = 1, the incidence angle modifier for beam radiation is calculated with the parameter  $b_0$ , for Optical Mode = 4, the incident angle modifier is calculated with a data table in an external file (see Table 9).  $K_D$  is the incident angle modifier for diffuse radiation.

Incident angle [°]	0	10	20	30	40	50	60	70	80	90
IAL	1.000	0.9999	0.9989	0.9943	0.9811	0.9500	0.8843	0.7532	0.4978	0.0000
IAT	1.000	1.038	1.075	1.106	1.121	1.091	0.970	0.737	0.390	0.000

Table 9: Incident angle modifier-table for the reference evacuated tube collector. IAL = Incident angle modifier longitudinal, IAT = incident angle modifier transversal, see section 11.1 for equations used to calculate time-dependent transversal and longitudinal incidence angles on the collector plane.

<sup>7</sup> Note that this does not match the documentation of TRNSYS 16 [Klein 2006], since there is a mistype in the documentation and the values for EQSOLVER do not go from 1 to 3 but from 0 to 2 [Arias 2007].

## 5.2.2 Collector heat exchanger

An external heat-exchanger (Type 5) is used in counter flow mode (mode 2) to separate the brine in the collector loop from the water in the storage tank. The UA value is calculated dependent on the collector field area chosen for the simulation (Eq. 5.1) using an equation from [Heimrath 2004]. The heat exchanger is connected to the collector pipes on the primary side and directly to the store doubleports on the secondary side.

$$\text{Eq. 5.1} \quad UaHX_c = (88.561 \cdot A_c + 328.19) \cdot 3.6$$

$UaHX_c$  [kJ/hK] Overall heat transfer coefficient of the heat exchanger

$A_c$  [m<sup>2</sup>] Collector aperture area

The heat exchanger Type 5 does not calculate heat losses of the heat exchanger. Heat losses between collector and store are accounted for by the pipes between the heat exchanger and the collector.

## 5.2.3 Collector loop flow rate

The mass flow rate on the primary side of the heat-exchanger (collector-side) is calculated dependent on the collector area and the specific mass flow chosen by the user. The mass flow of the secondary side of the heat exchanger is calculated to give the same capacitance flow rate as the primary side.

$$\text{Eq. 5.2} \quad \dot{m}_c = \dot{m}_{c\text{spec}} \cdot A_c$$

$$\text{Eq. 5.3} \quad \dot{m}_{c\text{sec}} = \dot{m}_c \cdot Cp_{\text{Bri}} / Cp_{\text{Wat}}$$

$\dot{m}_c$  [kg/h] mass flow on the primary side of the collector loop

$\dot{m}_{c\text{spec}}$  [kg/hm<sup>2</sup>] specific mass flow of the primary collector loop

$A_c$  [m<sup>2</sup>] Collector aperture area

$\dot{m}_{c\text{sec}}$  [kg/h] mass flow on the secondary side of the heat exchanger

$Cp_{\text{Bri}}$  [kJ/kgK] Thermal capacity of brine (primary side)

$Cp_{\text{Wat}}$  [kJ/kgK] Thermal capacity of water (secondary side)

## 5.2.4 Collector pipes

Two pipes (Type 31) have been used to connect the collector to the collector-loop heat-exchanger. Base Properties of these copper pipes are listed in Table 10.

	$l$ [m]	$\lambda_w$ [W/mK]	$\lambda_{iso}$ [W/mK]	$\alpha_i$ [W/m <sup>2</sup> K]	$\alpha_a$ [W/m <sup>2</sup> K]
Collector pipes	15	372	0.042	1000	10

Table 10: Length of collector pipes and heat transfer relevant properties of material, insulation and surfaces. For variable descriptions see Annex B

Pipe diameter and insulation are calculated based on the flow rate of the fluid in the pipe such that the velocity of the fluid is about 0.6 m/s (Eq. 5.4). The lower limit of the pipe inside diameter is DN 8. Pipe wall thickness is 1 mm, insulation thickness is defined with a simple function that represents closely [EN12976-2] (see Eq. 5.6).<sup>8</sup>

$$\text{Eq. 5.4} \quad d_i = \frac{0.8 \cdot \sqrt{\dot{m}_C}}{1000}$$

$$\text{Eq. 5.5} \quad d_a = d_i + 0.002$$

$$\text{Eq. 5.6} \quad d_{iso} = \text{MAX}(3 \cdot d_i; d_i + 0.04)$$

With a collector field aperture area of 15 m<sup>2</sup> and a specific flow rate of 15 kg/hm<sup>2</sup>, the collector pipe will be DN12. Calculated results for this pipe are shown in Table 11. Heat losses are calculated towards an ambient temperature of 15 °C.

	$\dot{m}_C$ [kg/h]	$d_i$ [m]	$d_a$ [m]	$d_{iso}$ [m]	$U_{pipe}$ [kJ/hm <sup>2</sup> K]
Collector pipes	225	0.012	0.014	0.092	12.7

Table 11: Example calculation for collector pipe U Value. Variable descriptions can be found in Annex B.

## 5.2.5 Control of the collector loop

The collector loop flow is controlled by a Type 2 controller. Flow is turned on when collector temperature is higher than the temperature of the collector control temperature sensor in the store plus the upper dead band Temperature, and turned off again when collector temperature drops below the Temperature of the sensor plus the lower dead band temperature. Dead band temperatures can be set by the user in the TRNEDIT interface.

Independent of the output of this controller, collector operation can be suspended for overheat protection reasons as shown in Table 12.

<b>Reason</b>	<b>Sensor</b>	<b>Off-Criteria</b>	<b>Hyst.</b>
Collector stagnation	Collector Temperature	Boiling Temp. of fluid as defined by user (TRNEDIT)	15 K
Storage tank protection	Temperature in the uppermost Node of the store	Cut-off Temperature T <sub>in</sub> as defined by user (TRNEDIT)	5 K

Table 12: Suspension of collector operation for safety / overheat reasons

<sup>8</sup> Note that  $d_{iso}$  is the diameter of the insulated pipe, NOT the thickness of the insulation

## 5.3 Storage tank

Type 340 is used to simulate the storage tank. Exactness for the calculation of temperatures (Parameter 121) is set to 0.001 K, heat exchangers are not used, and precision of the mixing process in the store was set to 1000. Refer to the documentation of Type 340 [Drück 2006]. if you need more information about these parameters.

### 5.3.1 Height and heat losses of the storage tank

The height of the storage tank is calculated as a function of the store volume chosen by the user. Different functions are applied for large ( $>14 \text{ m}^3$ ) stores as the height does not increase with the same proportion for seasonal stores as it does for short-time stores. The functions are a simplified version of the ones derived by Heimrath [Heimrath 2004] which are based on data of solar stores sold on the market (Eq. 5.7, Eq. 5.8, Eq. 5.9).

$$\text{Eq. 5.7} \quad H_{S1} = 0.32 \cdot V_S + 1.65$$

$$\text{Eq. 5.8} \quad H_{S2} = 0.09302 \cdot V_S + 4.698$$

$$\text{Eq. 5.9} \quad H_S = \text{MIN}(H_{S1}, H_{S2})$$

$H_{S1}, H_{S2}$  [m]      Height of store for small volumes and for large volumes respectively

$V_S$  [ $\text{m}^3$ ]              Volume of store

$H_S$  [m]                Height of store used for simulation

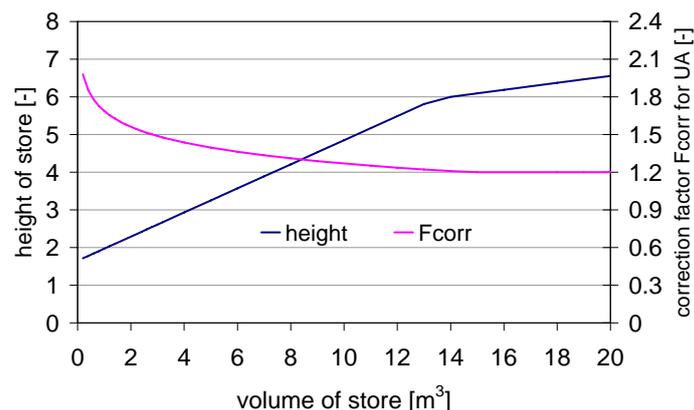


Figure 5.2: height of store and UA correction factor  $F_{corr,UAstore,A}$  as a function of the store volume.

Heat loss and heat transfer relevant properties of the store are listed in Table 13. Store heat losses are calculated as a lumped parameter UA [kJ/hK] accounting for all heat losses of the store, based on physical properties and correction factors to account for imperfect insulation

based on an evaluation of stores on the market performed by Heimrath [Heimrath 2004] (see Eq. 5.10).

store	$\lambda_{iso}$ [W/mK]	$\alpha_i$ [W/m <sup>2</sup> K]	$\alpha_a$ [W/m <sup>2</sup> K]	$\lambda_S$ [W/mK]	$T_{S,amb}$ [°C]
	0.042	300	10	2	15

Table 13: heat transfer relevant properties of store, store insulation and surfaces.  $\lambda_S$ = effective thermal conductivity in the store.  $T_{S,amb}$  = ambient temperature at the location of the store. For other parameter descriptions see Annex B

$$\text{Eq. 5.10} \quad UA_{S,corr} = F_{corr_{UAstore,A}} \cdot F_{corr_{UAstore,B}} \cdot (UA_{S,side} + UA_{S,caps})$$

$$\text{Eq. 5.11} \quad UA_{S,caps} = 3.6 \cdot \frac{2 \cdot A_{S,cap}}{R_{S,caps}}$$

$$\text{Eq. 5.12} \quad R_{S,caps} = \frac{1}{\alpha_a} + \frac{d_{iso}}{\lambda_{iso}} + \frac{1}{\alpha_i}$$

$$\text{Eq. 5.13} \quad F_{corr_{UAstore,A}} = \text{MAX}(1.2; -0.1815 \text{LN}(V_S) + 1.6875)$$

$UA_{S,corr}$ [kJ/hK]	total heat transfer rate from store to environment calculated based on physical properties and corrected for effects of imperfect insulation and heat-bridges
$F_{corr_{UAstore,A}}$ [-]	factor for correction of heat losses from store accounting for imperfect insulation and heat bridges (source?).
$F_{corr_{UAstore,B}}$ [-]	additional factor set by the user (TRNEDIT) for the correction of the heat loss coefficient. This factor has been introduced for the simulation of less/more perfect insulated stores or more/less insulation thickness.
$UA_{S,side}$ [kJ/hK]	heat transfer rate from store sidewalls to environment calculated considering $\lambda_{iso}$ , $\alpha_i$ , $\alpha_a$ , calculation according to section 11.2
$UA_{S,caps}$ [kJ/hK]	heat transfer rate from store top and bottom caps to environment
$A_{S,cap}$ [m <sup>2</sup> ]	area of cap (top or bottom)
$R_{S,caps}$ [m <sup>2</sup> K/W]	heat transfer resistance of caps

For a store volume of 1000 liter, an insulation thickness of 0.15 m and  $F_{corr_{UAstore,B}} = 1.4$ , the resulting overall UA value of the store is 18.0 kJ/hK.

### 5.3.2 Auxiliary heated volume and number of nodes

The number of nodes and the volume for auxiliary heated DHW ( $V_{aux}$ ) are a function of the store volume chosen by the user in the TRNEDIT interface.  $V_{aux}$  is 0.2 m<sup>3</sup> per default. The number of nodes of the store is calculated such that  $V_{aux}$  covers exactly 10 nodes. As the maximum number of nodes is reached,  $V_{aux}$  is increased in order to still cover 10 nodes of the store.

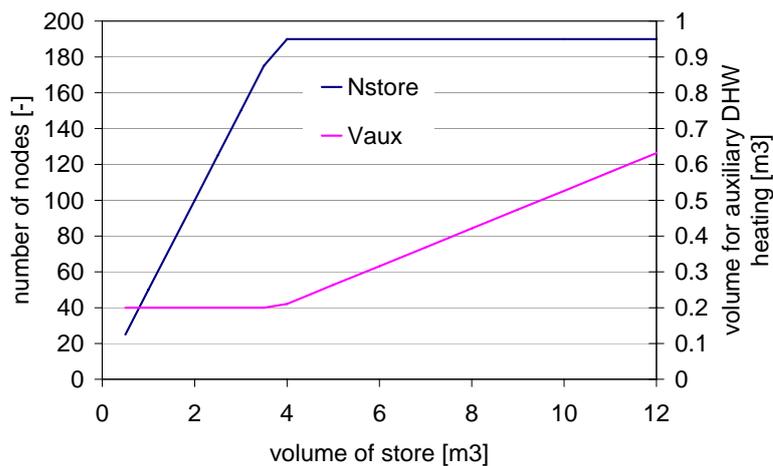


Figure 5.3: Number of nodes and volume of auxiliary heated DHW as a function of store volume.

### 5.3.3 Relative heights of store doubleports and temperature sensors

Relative heights of store connections are shown in Table 14, relative heights of temperature sensors are shown in Table 14.

<b>Doubleport description</b>	<b>relative height</b>	<b>Dp Nr.</b>
inlet of collector loop (stratified)	$= 1 - \frac{1.5}{N_{\max}}$	1
outlet of collector loop	0.05	1
inlet of auxiliary heating	$= 1 - \frac{1.5}{N_{\max}}$	3
outlet of auxiliary heating	$= 1 - \frac{V_{aux}}{V_S}$	3
inlet of DHW loop	0.05	5
outlet of DHW loop	$= 1 - \frac{1.5}{N_{\max}}$	5
inlet of space heating loop	0.15	4
outlet of space heating loop	$= 1 - 0.5 \frac{V_{aux}}{V_S} + \frac{1.5}{N_{\max}}$	4

Table 14: Heights of double port connections to the store.  $V_{aux}$  is auxiliary heated volume,  $V_S$  is total store volume, Dp Nr. is the number of the double port in the store model

<b>Sensor description</b>	<b>relative height</b>
Collector control temperature	user defined (TRNEDIT)
Storage protection temperature	$= 1 - \frac{0.5}{N_{\max}}$
First auxiliary On/Off temperature	$= 1 - 0.5 \frac{V_{aux}}{V_S}$
Second auxiliary On/Off temperature	$= 1 - \frac{V_{aux}}{V_S} + \frac{2.5}{N_{store}}$

Table 15: Heights of doubleport connections to the store.  $V_{aux}$  is auxiliary heated volume,  $V_S$  is total store volume,  $N_{store}$  is number of nodes for store simulation

## 5.4 Auxiliary loop

### 5.4.1 Auxiliary controller

Type 323 is used as auxiliary controller.

<b>Nr.</b>	<b>Description</b>	<b>Value(s)</b>
1,2,5,8,11	not used from this type	-11, -12, -30, 0, 0
3	ambient design temperature	taken from dataset for location chosen by the user
4	room set temperature	20 [°C]
6	set temperature for auxiliary heat supplied to store for DHW preparation	$T_{aux,set}$ [°C] set by the user (TRNEDIT)
7	nominal mass flow rate for DHW preparation	$=P_{aux}/(dT_{aux}*Cp_{Wat})$ [kg/hr]
9	radiator exponent n	0.2
10	radiator exponent m	0.3
12,13	minimum time off, minimum time on	MAX(dtSim*60;1) [min]

Table 16: Parameters of Type 323

$P_{aux}$ [kJ/h]	nominal power of auxiliary boiler (set by user)
$dT_{aux}$ [K]	temperature rise of auxiliary boiler (set by user)
$Cp_{Wat}$ [kJ/kgK]	heat capacity of water (see 5.1)

<b>Nr.</b>	<b>Description</b>	<b>Value(s)</b>
1,2,3	not used for this type	ambient design temp, 15, 0.0
4	temperature of DHW in store	first aux. on/off temperature (see 5.3.3)
5,6	return temperature of DHW preparation and room heating respectively	outlet temperature of space heating pump
7	controller for DHW preparation	output 1 of unit 169 storage control type 890
8	status of auxiliary burner from last timestep	output 18 of boiler type 370
9	temperature of boiler outlet	output 28 of boiler type 370

Table 17: Inputs of Type 323

The storage control Type 890 is used for the control signal for auxiliary heating of the DHW section of the store. The control signal is on (DHW preparation) whenever the first auxiliary On/Off sensor temperature drops below  $T_{aux,set} - 8$  [°C], and once it is on it will stay on until the second auxiliary On/Off sensor temperature is above  $T_{aux,set} - 2$  [°C]. Auxiliary heating of the DHW section of the store is possible any time of the day (not restricted through time windows).

## 5.4.2 Auxiliary boiler

Type 370 is used for the simulation of the auxiliary gas boiler

<b>Nr.</b>	<b>Description</b>	<b>Value(s)</b>
1	temperature setpoint for auxiliary DHW to store	$T_{aux,set}$ [°C] set by the user (TRNEDIT)
2	Fueltype	2 (natural gas high)
3	ambient temperature at location of boiler	15 [°C]
4	standby temperature	35 [°C]
5	hysteresis for standby temperature	5 [K]
6	maximum water temperature	90 [°C]
7	nominal power	set by the user (TRNEDIT)
8	air surplus (lambda) value	1.2
9	lowest modulation factor	0.25
10	mass of the boiler water	17.5 [kg]
11	temperature difference between flue gas and return temperature of water	10 [K]
12	radiation losses	3.5 [%]
13	standby losses as percent of nominal power	1.5 [%]
14	simulation mode	0 (original)
15	number of nodes in heat exchanger	10
16	exhaust gas temperature at entrance of heat exchanger	1000
17	minimum flow on water side	$=P_{aux}/(dT_{aux} * Cp_{Wat})$ [kg/hr] (see 5.4.1)

Table 18: Parameters of Type 370

<b>Nr.</b>	<b>Description</b>	<b>Value(s)</b>
1	actual return temperature	temperature from pump space heating
2	mass flow of heating medium	mass flow of pump space heating
3	temperature of air flow to boiler	ambient outdoor temperature from weather data-file
4	set supply temperature	set by the user (TRNEDIT)
5	set return temperature from controller	temperature from pump space heating
6	operation mode	0 (domestic hot water)
7	heat capacity of fluid	$Cp_{Wat}$ [kJ/kgK]
8	water content of biomass fuel	30 [%]
9	CO content of flue gas related to C content of flue gas	0.02
10	control signal	output 1 from heating control type 323

Table 19: Inputs of Type 370

## 5.4.3 Auxiliary pipe

Type 31 is used for the simulation of a pipe which connects the store to the inlet of the auxiliary heater. The purpose of this pipe is to introduce a buffer element which helps to avoid problems with the auxiliary heater type. Heat losses are calculated towards an ambient (room) temperature of 15 °C.

	$l$ [m]	$d_i$ [m]	$U_{pipe}$ [kJ/hm <sup>2</sup> K]
Collector pipes	10	0.051	3.01

Table 20: Length, diameter and heat loss coefficient of auxiliary pipes. For parameter descriptions see Annex B

## 5.5 Draw-Off loop

In the Draw-Off loop, a heat exchanger of Type 805 controls the heat transfer from the storage water to the tap water. The overall heat transfer coefficient of the heat exchanger has been set to a value which results in a return temperature of 15 °C to the store in the case of 10 °C cold water temperature, 60 °C temperature from store and a secondary mass flow rate (DHW) of 1200 kg/h.

<b>Nr.</b>	<b>Description</b>	<b>Value(s)</b>
1,2	specific heat capacity of primary and secondary side fluid respectively	$C_{p_{Wat}}$ , $C_{p_{Wat}}$
3	maximum allowed flow rate on primary (hot) side	1400 [kg/h]

Table 21: Parameters of Type 805

<b>Nr.</b>	<b>Description</b>	<b>Value(s)</b>
1	inlet temperature of primary (hot) side	store DHW doubleport outlet temperature
2	inlet temperature of secondary (cold) side	$T_{CW}$ (see Eq. 2.1)
3	flow rate on secondary (cold) side	$\dot{m}_{D,sec}$ mass flow from file (draw-off profile)
4	temperature setpoint for secondary side outlet	45 [°C]
5	overall heat transfer coefficient UA of heat exchanger	19200 [kJ/hK]

Table 22: Inputs of Type 805

## 5.6 Space heating loop

### 5.6.1 Air to air heat exchanger (for SFH15 only)

Type 91 is used to simulate an air-to-air heat exchanger for heat recovery in the case of SFH15. All other cases (SFH30 to SFH100) do not use this unit.

<b>Nr.</b>	<b>Description</b>	<b>Value(s)</b>
1	heat exchanger effectiveness	0.6
2, 3	specific heat of hot side fluid and cold side fluid respectively	1.0048 [kJ/kgK]

Table 23: Parameters of Type 91

<b>Nr.</b>	<b>Description</b>	<b>Value(s)</b>
1	hot side inlet temperature	air temperature of building
2, 4	hot side flow rate and cold side flow rate respectively	354(n_L), (see 3.7 for explanation of n_L)
3	cold side inlet temperature	outdoor temperature from weather data file

Table 24: Inputs of Type 91

## 5.6.2 Radiator

Type 362 is used to simulate radiators for room heating.

<b>Nr.</b>	<b>Description</b>	<b>Value(s)</b>
1-5	length of supply pipe and exhaust pipe respectively	not used
6	specific heat of fluid	$Cp_{Wat}$ (see 5.1)
7	max. flow rate of fluid	$\dot{m}_{B,max}$
8	radiative fraction of total emitted power	0.35
9	nominal power of radiator	$Q_{Rd,n}$
10	radiator exponent	$n_{Rd} = 1.3$
11	thermal capacitance of radiator	1150 [kJ/K]
12	initial temperature	55 [°C]

Table 25: Parameters of Type 362

<b>Nr.</b>	<b>Description</b>	<b>Value(s)</b>
1	supply temperature	output from Tee-piece (mixer) unit [182,1]
2	building star node temperature	output from building type 56
3	control signal for mass flow	output from PID controller unit 13

Table 26: Inputs of Type 362

$$\text{Eq. 5.14} \quad \dot{m}_{B,max} = \frac{Q_{d,loc}}{Cp_{Wat} (T_{d,in} - T_{d,out})}$$

$$\text{Eq. 5.15} \quad Q_{d,loc} = Q_{d,ZH} \frac{(Tset_{room} - T_{AMBD})}{(Tset_{room} - T_{AMBD,ZH})}$$

$$\text{Eq. 5.16} \quad Q_{Rd,n} = Q_{d,loc} \left[ \frac{60}{(T_{d,in} + T_{d,out}) / 2 - Tset_{room}} \right]^{n_{Rd}}$$

$\dot{m}_{B,max}$ [kg/h]	maximum mass flow through radiator(s) of the building
$Q_{d,loc}$ [kJ/h]	standard heat load at design outdoor temperature for chosen location
$T_{d,in}$ [°C]	design inlet temperature of the heating system (from building-file)
$T_{d,out}$ [°C]	design inlet temperature of the heating system (from building-file)
$Q_{d,ZH}$ [kJ/h]	standard heat load at design outdoor temperature for Zurich (from building file)
$Tset_{room}$ [°C]	room setpoint temperature (20 °C)
$T_{AMBD}$ [°C]	ambient design temperature of heating system at chosen location
$T_{AMBD,ZH}$ [°C]	ambient design temperature of heating system at Zurich
$n_{Rd}$ [-]	radiator exponent (=1.3 in this case)

### 5.6.3 PID-controller

The mass flow in the radiators is determined by simulation of a thermostatic valve with the PID controller Type 320.

Nr.	Description	Value(s)
1	temperature width of PID band	3 [K]
2	proportional gain PID band	0.8 [1/K]
3	integral gain PID band	0.05 [1/Kh]
4	differential gain PID band	0 [h/K]
5	proportional gain P band	0.5 [1/K]
6,7	not used	0, 0

Table 27: Parameters of Type 320

Nr.	Description	Value(s)
1	set temperature	room temperature setpoint
2	feedback temperature	room temperature of last timestep
3	control inversion option (direction of action)	2 (decreasing action)

Table 28: Inputs of Type 320

### 5.6.4 Mixing valve

A mixing valve is simulated with a Type 11 used as a tempering valve and another Type 11 used as a Tee-piece. Heat source temperature for the tempering valve is taken directly from the store outlet to the building loop; the cold fluid is taken from the pump in the heating loop. The temperature setpoint for the mixing valve is assumed to be calculated by a controller (see Eq. 5.17).

$$\text{Eq. 5.17} \quad T_{in,Rd,Ct} = T_{Rd,Av,Ct} + \frac{Q_{Rd,Ct}}{2\dot{m}_{B,max} \cdot C_{p,Wat}}$$

$$\text{Eq. 5.18} \quad T_{Rd,Av,Ct} = T_{set,room} + n_{Rd} \sqrt{\frac{Q_{Rd,Ct}}{Q_{d,loc}}} \cdot dT_d$$

$$\text{Eq. 5.19} \quad Q_{Rd,Ct} = \text{MAX} \left( 0; Q_{d,loc} \frac{T_{set,room} - T_{amb}}{T_{set,room} - T_{AMBD}} \right)$$

$$\text{Eq. 5.20} \quad dT_d = \frac{T_{in,d} + T_{out,d}}{2} - T_{set,room}$$

$T_{in,Rd,Ct}$ [°C]	inlet temperature setpoint calculated by controller
$T_{Rd,Av,Ct}$ [°C]	mean radiator temperature for room setpoint calculated by controller
$n_{Rd}$ [-]	radiator exponent (=1.3 in this case)
$dT_d$ [K]	design temperature difference
$Q_{Rd,Ct}$ [kJ/h]	heating rate assumed by the controller

## 5.7 Electricity consumption

The consumption of electric energy of the system is calculated according to the agreements made during IEA SHC Task 26 [IEA-SHC 2002] and published in the Task 26 handbook [Weiss 2003].

$$\text{Eq. 5.21} \quad W_{\text{system}} = W_{\text{burner}} + W_{\text{pump,SH}} + W_{\text{pump,DHW}} + W_{\text{pump,sol}} + W_{\text{control}}$$

$$\text{Eq. 5.22} \quad W_{\text{burner,ON}} = 22.3 + 0.8349P_{\text{burner,nom}}$$

$$\text{Eq. 5.23} \quad W_{\text{burner,stby}} = 9$$

$$\text{Eq. 5.24} \quad W_{\text{pump,SH}} = 90.476 + 0.203P_{\text{burner,nom}}$$

$$\text{Eq. 5.25} \quad W_{\text{pump,DHW}} = 49.355e^{0.0083P_{\text{burner,nom}}}$$

$$\text{Eq. 5.26} \quad W_{\text{pump,sol,sm}} = 78.3e^{0.0156A_C}$$

$$\text{Eq. 5.27} \quad W_{\text{pump,sol,gt}} = 196e^{0.0046A_C}$$

$$\text{Eq. 5.28} \quad W_{\text{control}} = 1$$

$W_{\text{system}}$ [W]	electricity (el.) consumption of the whole solar system
$W_{\text{burner,ON}}$ [W]	el. consumption of the burner during operation
$W_{\text{burner,stby}}$ [W]	el. consumption of the burner during standby
$P_{\text{burner,nom}}$ [kW]	nominal power of burner
$W_{\text{pump,SH}}$ [W]	el. consumption of pump for space heating during heating season
$W_{\text{pump,DHW}}$ [W]	el. consumption of pump for DHW heat exchanger in case of draw off
$W_{\text{pump,sol,sm}}$ [W]	el. consumption of pumps in primary and secondary solar loop together, small system ( $A_C < 75 \text{ m}^2$ )
$W_{\text{pump,sol,gt}}$ [W]	el. consumption of pumps in primary and secondary solar loop together, large system ( $A_C \geq 75 \text{ m}^2$ )
$A_C$ [m <sup>2</sup> ]	collector aperture area
$W_{\text{control}}$ [W]	el. consumption of the controller

## 6 Performance indicators

To compare results obtained from the simulation of solar heating and/or cooling systems with the Task 32 reference TRNSYS deck, it is recommended to use fractional energy savings and the FSC method developed in the IEA-SHC Task 26 and described in the Task 26 handbook [Weiss 2003]. The indicators and their calculation are shown in this section. And a new indicator for solar heating and cooling is introduced in section 6.2. Please note that the calculation of penalties for high room temperatures differs from the one proposed in the IEA SHC Task 26.

### 6.1 fractional savings indicators from IEA SHC Task 26

The **fractional thermal energy savings**  $f_{sav,therm}$  are a measure of the percentage the auxiliary (non-solar) energy input for heating can be reduced with the solar system, this term does not account for electricity use unless it is used directly for heating. Unlike in Task 26, no difference is made in Task 32 for the electricity coming from renewable sources and non renewable sources. The efficiency of electricity production and distribution  $\eta_{el}$  is 0.4 in all cases. This value reflects not only the average UCPT efficiency of conversion and transport, but also the thermodynamic value of the energy and the higher market price of electricity. Electricity will always be more valuable and have a higher price than heat, independent of the question whether the electricity comes from renewable sources or not.

$$Eq. 6.1 \quad f_{sav,therm} = 1 - \frac{\frac{Q_{boiler} + Q_{el,heater}}{\eta_{boiler}} + \frac{Q_{el,heater}}{\eta_{el}}}{\frac{Q_{boiler,ref}}{\eta_{boiler,ref}}}$$

The **extended fractional energy savings**  $f_{sav,ext}$  are defined in a similar way, but they also include electricity use for pumps, valves, controllers etc.

$$Eq. 6.2 \quad f_{sav,ext} = 1 - \frac{\frac{Q_{boiler} + Q_{el,heater} + W_{par}}{\eta_{boiler}} + \frac{Q_{el,heater}}{\eta_{el}}}{\frac{Q_{boiler,ref}}{\eta_{boiler,ref}} + \frac{W_{par,ref}}{\eta_{el}}}$$

Finally, it is theoretically possible to achieve a high  $f_{sav,therm}$  and  $f_{sav,ext}$  and at the same time not meeting the comfort criteria for space heating or warm-water production. Therefore, the **fractional solar savings indicator**  $f_{si}$  is including a penalty term that compensates and even punishes for not meeting comfort criteria. For a complete discussion of these values refer to the design handbook of IEA SHC Task 26 [Weiss 2003].

$$\text{Eq. 6.3} \quad f_{si} = 1 - \frac{\frac{Q_{boiler}}{\eta_{boiler}} + \frac{Q_{el,heater} + W_{par}}{\eta_{el}} + Q_{penalty} - Q_{penalty,ref}}{\frac{Q_{boiler,ref}}{\eta_{boiler,ref}} + \frac{W_{par,ref}}{\eta_{el}}}$$

$Q_{boiler}$ [kWh]	Energy delivered to the system by the auxiliary boiler (energy balance on the water-side of the boiler)
$Q_{boiler,ref}$ [kWh]	Energy delivered to the reference system (system without solar) by the boiler in the reference system (energy balance on the water-side of the boiler)
$Q_{el,heater}$ kWh	Energy delivered to the system by an electrical heater
$\eta_{boiler}$ [-]	Efficiency of auxiliary boiler of the solar system (dependent on the overall system)
$\eta_{boiler,ref}$ [-]	Efficiency of auxiliary boiler of the reference system (= 0.85)
$W_{el}$ [kWh]	Electricity used for pumps, valves, etc. as well as for direct heating or driving a heat pump
$W_{el,ref}$ [kWh]	Electricity used for pumps, valves, etc. of the reference system
$\eta_{el}$ [-]	Efficiency of electricity production and transport to the site of use (= 0.4 independent of the source)
$Q_{penalty}$ [kWh]	Penalty for the solar system for not meeting comfort criteria
$Q_{penalty,ref}$ [kWh]	Penalty for the reference system for not meeting comfort criteria

## 6.2 Fractional savings for solar heating and cooling

For solar heating and cooling, the fractional solar thermal savings and extended solar thermal savings introduced in IEA SHC Task 26 are not applicable. Instead, **fractional solar heating & cooling savings**  $f_{sav,shc}$  are defined:

$$\text{Eq. 6.4} \quad f_{sav,shc} = 1 - \frac{\frac{Q_{boiler}}{\eta_{boiler}} + \frac{W_{el}}{\eta_{el}} + \frac{Q_{cooling,missed}}{SPF_{ref} \cdot \eta_{el}}}{\frac{Q_{boiler,ref}}{\eta_{boiler,ref}} + \frac{W_{el,ref}}{\eta_{el}} + \frac{Q_{cooling,ref}}{SPF_{ref} \cdot \eta_{el}}}$$

$Q_{cooling,missed}$ [kWh]	Cooling demand that has not been met by the solar cooling system
$Q_{cooling,ref}$ [kWh]	Energy withdrawn from the system by the reference cooling system
$SPF_{ref}$ [-]	Seasonal performance factor of the reference cooling system (= 2.8)

The calculation of  $f_{sav,shc}$  is similar to the calculation of the extended solar thermal savings of IEA SHC Task 26, with the following differences:

- 1) The cooling energy load is introduced in the denominator, divided by the seasonal performance factor (of a standard heat pump) and the efficiency of electricity production and distribution.
- 2) The cooling energy not provided by the solar cooling system is not covered by an auxiliary cooling system in reality, but it must be accounted for in the fractional solar heating & cooling savings. This is done by introducing  $Q_{cooling,missed}$ .
- 3) Electricity consumption is not split up into parasitic electric energy and electric energy used for direct heating and/or cooling. Instead, all electricity consumptions of the solar heating and cooling system are included in  $W_{el}$ .

### 6.3 Penalty functions for DHW temperature

Penalty functions for not meeting the required tapping water temperature of 45 °C are calculated according to the IEA-SHC Task 26 handbook [Weiss 2003] and the IEA-SHC Task 26 minutes [IEA-SHC 2002b] as shown in Eq. 6.6 and Eq. 6.5.

$$\text{Eq. 6.5} \quad Q_{penalty,DHW,\Delta t} = \Delta t \cdot \dot{C} \left[ \Delta T_{DHW} + (\Delta T_{DHW} + 1)^{x_{DHW}} - 1 \right]$$

$$\text{Eq. 6.6} \quad \Delta T_{DHW} = \text{MAX}(0; 45^\circ\text{C} - T_{DHW})$$

$Q_{penalty,DHW,\Delta t}$ [Wh]	penalty for not meeting required DHW-temperature of 45 °C
$\Delta t$ [h]	timestep of simulation
$\dot{C}$ [W/K]	capacity flow rate of tapped water
$\Delta T_{DHW}$ [K]	missing degrees for meeting required temperature of 45 °C
$x_{DHW}$ [-]	penalty exponent $\geq 1$ (4 is used per default according to minutes of IEA-SHC Task 26 [IEA-SHC 2002b])

Values of  $Q_{penalty,DHW,\Delta t}$  are calculated within the Task 32 reference deck, integrated over each month and the results are printed out in the Task 32 monthly results printer.

### 6.4 Penalty functions for uncomfortable room temperatures

Penalty functions are calculated for not meeting comfort criteria as agreed upon in IEA-SHC Task 32 minutes. The lowest accepted room temperature is 19.5 °C always. The upper limit follows the recommended line of DIN 1946-2 (compare section 3.8).

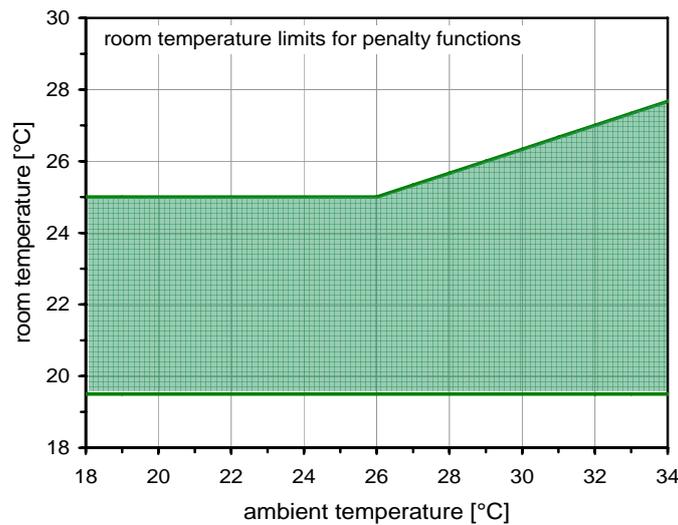


Figure 6.1: room temperature limits for penalty functions for not meeting comfort criteria of IEA-SHC Task 32

The penalty function for not meeting room heating demand (room temperatures below 19.5 °C is calculated according to the IEA-SHC Task 26 handbook [Weiss 2003] as shown in Eq. 6.9 - Eq. 6.8). The penalty function for not meeting cooling demand (room temperatures above 24 °C) from the IEA SHC Task 26 handbook had to be adapted to match the one for room heating demand and to follow the maximum allowed room temperatures defined in DIN 1946-2 (see Eq. 6.9 - Eq. 6.10).

$$\text{Eq. 6.7} \quad \Delta T_{SH} = \text{MAX} \left( 0; 19.5 - T_{room} \right)$$

$$\text{Eq. 6.8} \quad Q_{pen,SH,\Delta t} = UA_{building} \cdot \Delta t \cdot \left[ \Delta T_{SH} + (\Delta T_{SH} + 1)^{x_{SH}} - 1 \right]$$

$$\text{Eq. 6.9} \quad \Delta T_{SC} = \text{MAX} \left[ 0; T_{room} - \text{MAX} \left\{ 25; 26 + \frac{T_{amb} - 29}{3} \right\} \right]$$

$$\text{Eq. 6.10} \quad Q_{pen,SC,\Delta t} = UA_{building} \cdot \Delta t \cdot \left[ \Delta T_{SC} + (\Delta T_{SC} + 1)^{x_{SC}} - 1 \right]$$

$\Delta T_{SH}$ [K]	degrees below minimum room temperature of 19.5°C
$T_{room}$ [°C]	actual room temperature
$Q_{pen,SH,\Delta t}$ [Wh]	penalty for not meeting heating load
$UA_{building}$ [W/K]	overall heat loss coefficient of building
$\Delta t$ [h]	timestep of simulation
$x_{SH}$ [-]	penalty exponent $\geq 1$ ( <b>2</b> is used per default according to IEA-SHC Task 26 [IEA-SHC 2002b])
$\Delta T_{SC}$ [K]	degrees above maximum room temperature according to DIN 1946-2
$T_{amb}$ [°C]	ambient (outdoor) temperature
$Q_{pen,SC,\Delta t}$ [Wh]	penalty for not meeting cooling load
$x_{SC}$ [-]	penalty exponent $\geq 1$ ( <b>2</b> is used per default according to IEA-SHC Task 26 [IEA-SHC 2002b])

Values of  $Q_{pen,SH,\Delta t}$  and  $Q_{pen,SC,\Delta t}$  are calculated within the Task 32 reference deck, integrated over each month and the results are printed out in the Task 32 monthly results printer.

## 7 Interpretation of simulation results

In order to simulate your own system in the IEA-SHC Task 32 simulation environment, it is recommended to first set all parameters in the TRNEDIT environment accordingly, and then open the file *RefSol\_XX.trd* in a text editor to insert your own TRNSYS Types and equations and connect them to the rest of the simulation system. If you are using the printer labeled “Task 32 monthly simulation results”, please refer to the short list of the outputs in section 12.

### 7.1 General advices

- When simulating a system with a large store (seasonal storage), it is necessary to run a two year simulation and only use the values from the second year. Special attention has to be paid that there is no significant difference in energy stored in the store at the beginning and at the end of the evaluation period.
- The standard TRNSYS-deck *RefSol\_XX.trd* does not include a cooling device, and therefore the values for energy used for cooling will be zero. The user should insert his/her own cooling device and connect it to the building in order to calculate primary energy use of his/her cooling unit. The building is set up to calculate cooling demand (see section 12).
- Refer to section 7.3 for the choice of tolerance settings for convergence and integration errors and for the choice of simulation timestep.
- Check the energy balance of each loop and of the major Units. TRNSYS does not do this for you.
- Do not use your simulation results if penalties for dhw or space heating are above 5% of the energy demand of dhw or space heating respectively.

### 7.2 Reference systems (without solar)

Reference systems without solar collectors have been defined and simulated. The reference system calculations followed closely the descriptions in the Task 26 handbook [Weiss 2003] and can be summarized as follows:

- auxiliary energy measured on the water-side of the auxiliary boiler and divided by an efficiency factor of 0.85.
- store volume of 200 liters and a store loss of 644 kWh/a
- electric energy demand of space heating pump: 93 W
- electric energy demand of dhw pump: 55 W
- electric energy demand of burner: 35 W (on) and 9 W (standby)
- electric energy demand of controller: 1 W

Figure 7.1 - Figure 7.4 give an overview over the simulation results of the reference systems without solar heating or solar driven cooling for all buildings in all climates included in the reference conditions of this Task. The simulation timestep chosen for the simulation of the reference systems was 1/20 h.

**The variables displayed are:**

QSH	space heating demand of an “ideal” heating system (convective and radiation energy from radiator)
QBURN	demand of energy from burner to cover space heating, storage losses and DHW consumption
Qcool	space cooling demand of an “ideal” cooling system
Qpen	penalty for the system if no cooling is applied; since room temperatures below 19.5 °C do not occur in these reference systems and DHW demand is always met, the penalty is a result of not meeting cooling demand (room temperatures too high) only
WTOT	total electricity consumption of the system

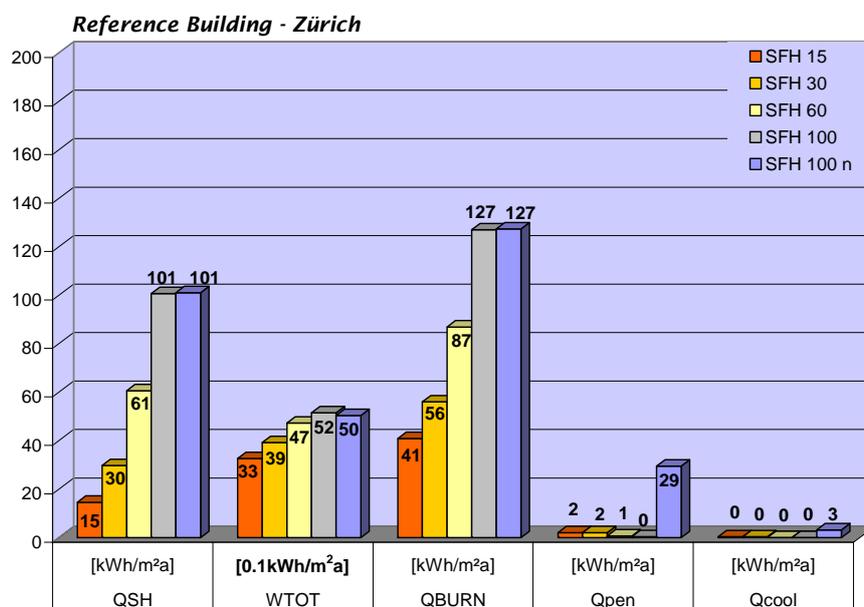


Figure 7.1: Simulation results for reference systems without solar collectors, city of Zurich, all reference buildings

The simulated buildings are named SFH (single family house) followed by their heating load for the climate of Zurich (15-100) and an optional “n” that stands for “no venetian blinds shading”. See section 3 for detailed building descriptions.

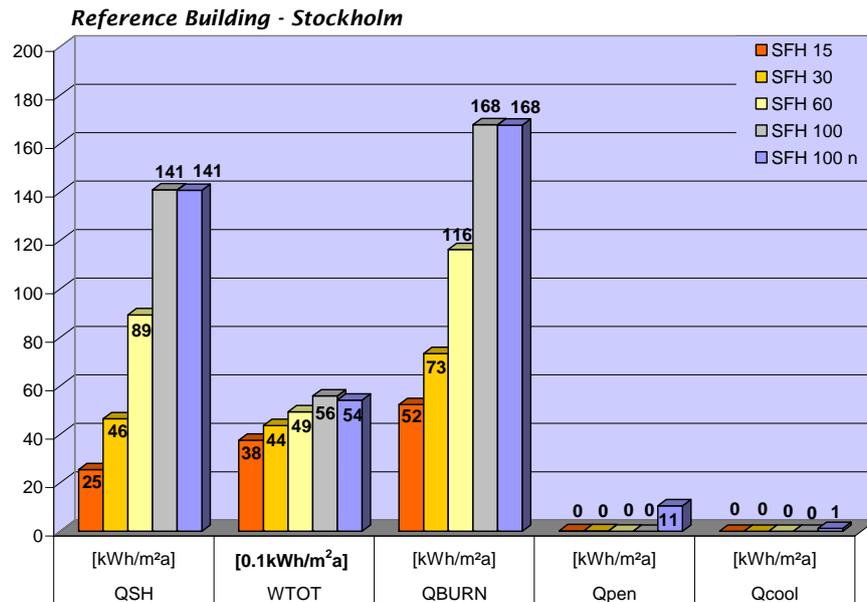


Figure 7.2: Simulation results for reference systems without solar collectors, city of Stockholm, all reference buildings

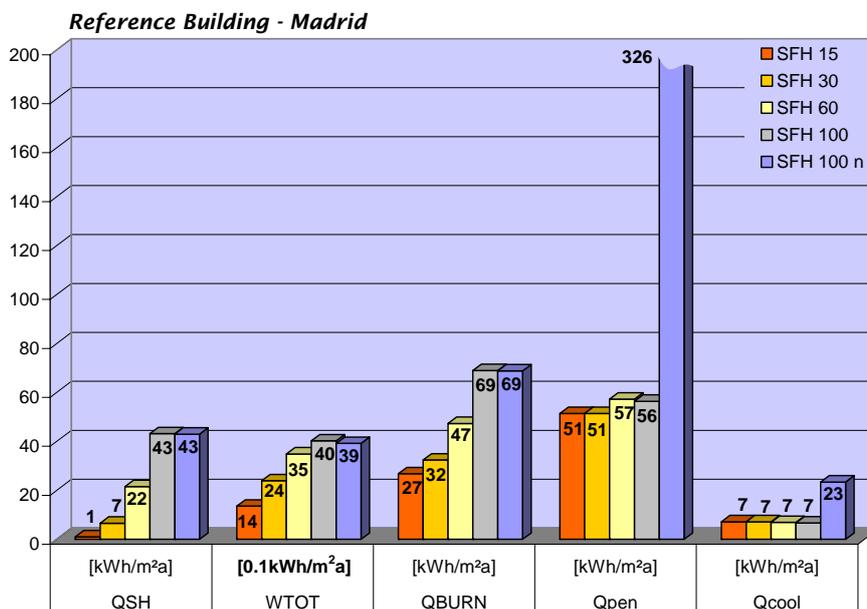


Figure 7.3: Simulation results for reference systems without solar collectors, city of Madrid, all reference buildings

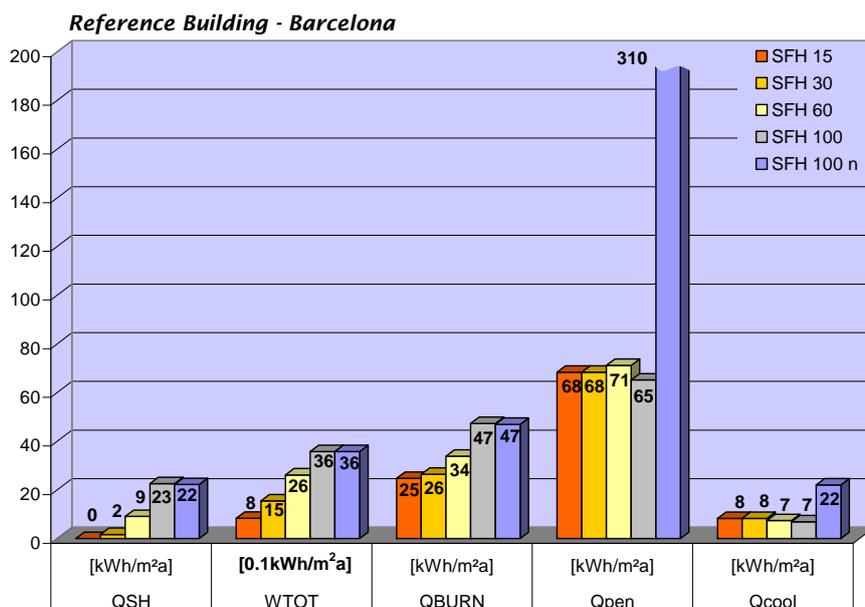


Figure 7.4: Simulation results for reference systems without solar collectors, city of Barcelona, all reference buildings

The difference between QSH and QBURN equals storage losses of 664 kWh/a plus DHW-demand which is dependent on the city as shown in Figure 7.5. As a result, the energy that must be provided by the burner (QBURN) is 24.9 – 27 kWh/m<sup>2</sup>a higher than the energy needed for space heating alone (QSH) for all cities and all buildings.

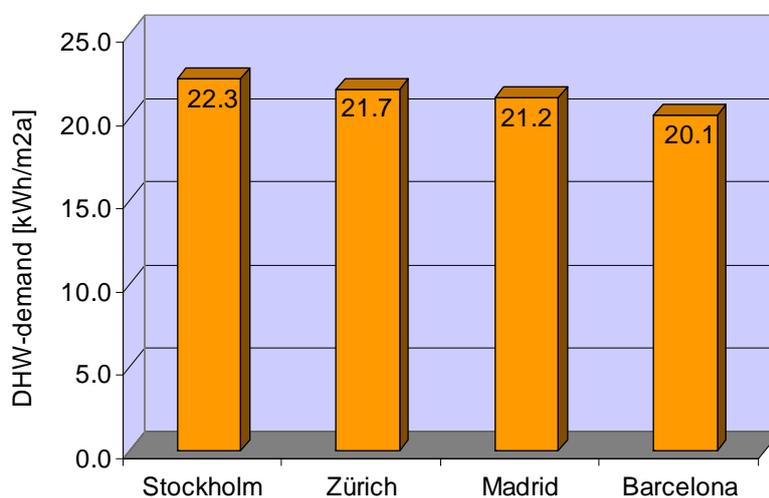


Figure 7.5: DHW-demand differences between the cities as a result of different cold water temperatures

Detailed results of the reference system simulations are listed in Table 29.

Location / BUI	QSH [kwh/a]	QDHW [kwh/a]	QL [kwh/a]	QBURN [kwh/a]	Qcool [kwh/a]	Qpen [kwh/a]	TiHeat_h [h]	W <sub>BUR</sub> [kwh/a]	W <sub>DHW</sub> [kwh/a]	W <sub>SH</sub> [kwh/a]	W <sub>TOT</sub> [kwh/a]	IRRAD [kwh/m <sup>2</sup> a]	
Zürich	15	2041		5725	44	285	3811	89		354	457		
	30	4159		7843	44	285	4760	92		443	549		
	60	8482	3040	644	12165	6	93	5891	100	14	548	661	1212
	100	14093			17776	2	52	6430	110		598	721	
	100 n	14146			17830	446	4129	6258	110		582	705	
Stockholm	15	3559		7325	1	30	4524	92		421	526		
	30	6507		10273	1	30	5383	97		501	611		
	60	12511	3122	644	16277	0	9	6125	107	14	570	690	1232
	100	19733			23499	0	6	6973	120		649	782	
	100 n	19706			23472	170	1474	6716	120		625	758	
Madrid	15	147		3761	1009	7206	987	85		92	190		
	30	926		4540	1008	7203	2536	87		236	336		
	60	3019	2970	644	6633	971	8027	4142	90	13	385	489	1864
	100	6046			9660	940	7902	4889	96		455	564	
	100 n	6026			9640	3278	45659	4732	96		440	549	
Barcelona	15	4		3467	1159	9571	210	85		20	117		
	30	236		3700	1159	9565	1278	85		119	217		
	60	1274	2819	644	4738	1036	9967	2862	87	13	266	366	1624
	100	3158			6621	966	9125	4277	90		398	501	
	100 n	3107			6570	3090	43460	4275	90		398	500	

Table 29: Detailed results of the reference system simulations for each location and building. QSH = energy input from radiator to building (convective and radiative); QDHW = domestic hot water draw off; QL = store losses; QBURN = amount of fuel burnt \* net calorific value; Qcool = cooling demand (energy withdrawn from building for ideal cooling); Qpen = total penalties (dhw + cooling + heating); TiHeat = total hours of room heating; W<sub>BUR</sub> = electricity for burner, W<sub>DHW</sub> = electricity for dhw preparation; W<sub>SH</sub> = electricity for space heating pump, W<sub>TOT</sub> = total electricity; IRRAD = irradiation on collector plane (slope 45°, south)

### 7.3 Simulation results with solar systems

For the SFH 60 building and the location of Zurich, fractional energy savings and the time needed for the TRNSYS simulation are shown in Figure 7.6 and Figure 7.7. The different values obtained for different tolerances for convergence and integrated output (Tol) and different simulation timesteps lead to the following conclusions:

- consistent results are obtained for tolerance settings (convergence and integrated outputs) of 0.003 or lower and simulation timesteps of 2 minutes or 3 minutes
- especially the penalty-values ( $f_{si}$ ) show a high uncertainty when the tolerance settings for convergence and integration are too high
- lowering the tolerance settings from 0.01 to 0.002 does not affect the time needed for calculations, whereas lowering the simulation timestep does.

Energy balances of the total system and/or of parts of the system showed errors that were larger than  $\pm 1\%$  for tolerance settings  $\geq 0.006$  (not shown in graphs).

Therefore, it is recommended to use a **convergence tolerance setting of 0.002 or 0.003**, and a **simulation timestep of 2 minutes or 3 minutes**. When different simulations are compared, care has to be taken not to compare results from simulations that were performed with different convergence tolerance settings and/or simulation timestep.

The general system settings used for these simulations are shown in Table 1. The simulations were performed on a FSC Amilo Pro V3205/T2250 mobile computer with 1.73 GHz and 504 MB RAM under Windows XP Professional, SP2.

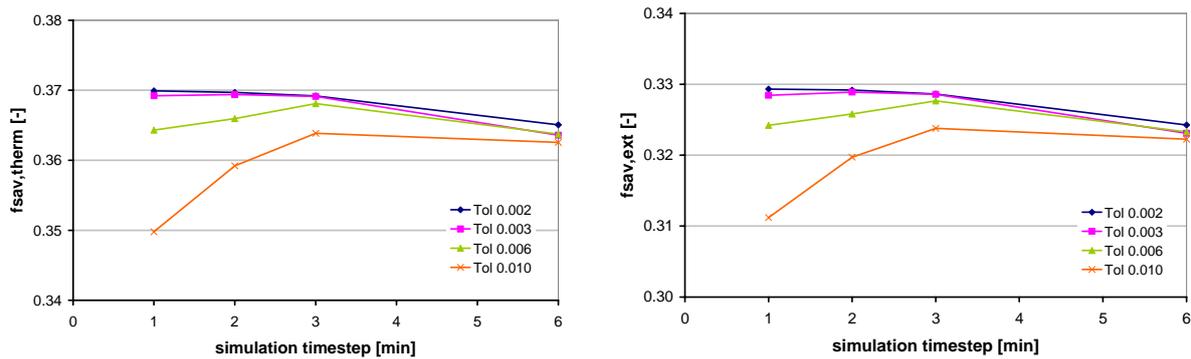


Figure 7.6: fractional thermal energy savings (left) and extended fractional energy savings (right) calculated with different simulation timesteps and different tolerance settings for convergence and integrated outputs. SFH60 Zurich, 20 m<sup>2</sup> selective flat plate collector, 1 m<sup>3</sup> store volume. Calculated with version V45 of the reference deck.

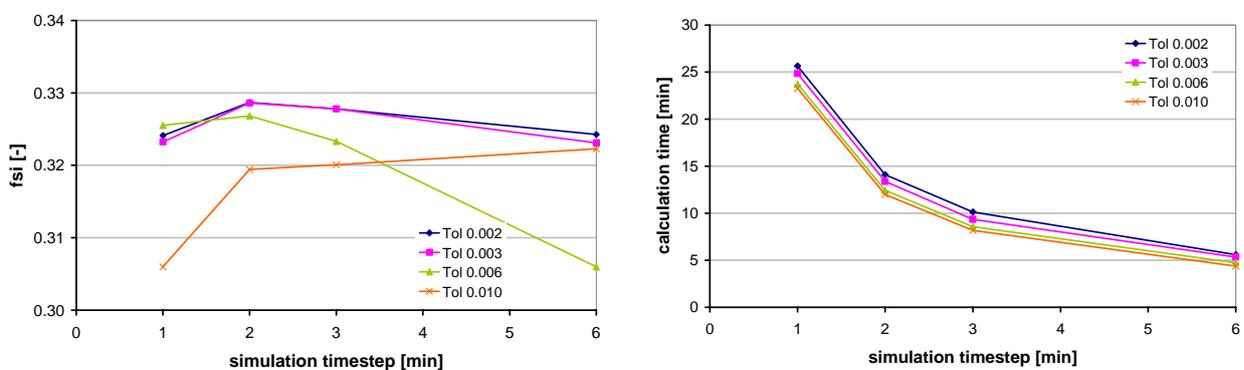


Figure 7.7: fractional savings indicator (left) and time needed to run a simulation (right) calculated with different simulation timesteps and different tolerance settings for convergence and integrated outputs. SFH60 Zurich, 20 m<sup>2</sup> selective flat plate collector, 1 m<sup>3</sup> store volume. Calculated with version V45 of the reference deck.

Table 30: Settings in TRNEDIT for the simulations of this section

<b>Main</b>		<b>Collector</b>	
Length of Simulation	13 months	Collector	Flat plate selective
City	Zurich	Aperture Area	20m <sup>2</sup>
Building	SFH 60	tilt angle	45°
		azimuth	0°
<b>Auxiliary</b>		mass flow	15kg/hm <sup>2</sup>
Aux Power	10 kW	dT upper	7K
Aux Set T	63 °C	dT lower	4K
Aux dT	10 K	rel height of Sensor	0.1
<b>Store</b>		Tmax Store	90°C
Store volume	1 m <sup>3</sup>	Tboil coll	110°C
insul. thickness	0.15 m	Tmax Store	90°C
corr. factor	1.40	Tboil coll	110°C

## 8 Known problems

- With **convergence tolerance** set to **0.0001**, there are convergence problems in the auxiliary loop. Use higher values.
- According to the simulation results shown in section 7.3, it is recommended to set tolerances for convergence and integration error to 0.002 - 0.003, and run the simulations with timesteps of 2 or 3 minutes.
- Very large timesteps (e.g. > 10 minutes) should not be used at all, since in some cases values of previous timesteps are used for controller calculations.
- For **large stores**, the store-section heated by the auxiliary boiler for domestic hot water may be too large. This is due to the minimum number of nodes for the control of auxiliary heating of the **domestic hot water volume in the store**. If this could be a problem for your simulation studies, change the deck accordingly and report your changes thoroughly.

## 9 Annex A: Types used for the reference system

### Standard Types used for the Task32 reference system

Type	Description	Remarks
2	Differential Controller	standard TRNSYS 16.0.0038
5	Heat Exchanger	standard TRNSYS 16.0.0038
9	Weather Data Reader	standard TRNSYS 16.0.0038
11	Flow Mixer / Flow Diverter / Tempering Valve	standard TRNSYS 16.0.0038
14	Forcing Function	standard TRNSYS 16.0.0038
24	Quantity Integrator	standard TRNSYS 16.0.0038
25	Printer	standard TRNSYS 16.0.0038
28	Simulation Summary	standard TRNSYS 16.0.0038
31	Pipe / Duct	standard TRNSYS 16.0.0038
33	Psychrometrics	standard TRNSYS 16.0.0038
34	Overhang and Wingwall Shading	standard TRNSYS 16.0.0038
56	Multizone Building	standard TRNSYS 16.0.0038
65	Online Plotter	standard TRNSYS 16.0.0038
69	Sky Temperature	standard TRNSYS 16.0.0038
91	Constant Effectiveness Heat Exchanger	standard TRNSYS 16.0.0038
93	Input Value Recall	standard TRNSYS 16.0.0038
109	Weather Data Reading and Processing	standard TRNSYS 16.0.0038

### Other Types used for the Task32 reference system (*int.* = compiled into *Trndll.dll*)

Type	Version	Description	Authors / Remarks	<i>int.</i>	<i>doc.</i>
320	2004-07-00 : AK	PID-controller			yes
323	??	heating control	Author: ??	yes	yes
340	1.99EHD 24.01.2002	multiport store model	Author: Harald Drück		yes
358	Version 1.1 RH 03.06.2005	tilted window ventilation	Authors: R. / A. Weber, R. Heimrath		yes
362	2004-05-03 : AK	radiator			yes
370	06-Nov-2000 :CB 3.00	gas, oil and automatic wood boilers	Authors: C. Beurer, A. Knirsch, T. Fiedler, A. Koschack, C. Bales	yes	yes
803	2006-11 RH	variable flow pump	Author: R. Heimrath		yes
805	2006-12 MH	DHW heat exchanger	Author: M. Haller		yes
816	2006-11 MH	transient value averaging	Author: M. Haller		yes
832	2.06 RH 20/07/06	CEN solar collector	Authors: B. Hellström / B. Perers / C. Bales / S. Fischer	yes	yes
890	April 05, 2006	storage control	Author: R. Heimrath		yes

*int.* = compiled into *Trndll.dll*, *doc.* = documentation available

## 10 Annex B: particular non standard types

### 10.1 TYPE 803: variable speed pump

RH, 2006/11

This type is identical with Type 3 of TRNSYS 16, except that the FORTRAN JFIX-function performed on the control signal has been removed, such that it is possible again to vary the mass flow between 0 and maximum flow by using control-function values from 0 to 1.

### 10.2 TYPE 890: storage control

Version: 2006/04, RH

Type 890 is used to control the auxiliary heating of the store by two sensors (one for starting to heat, and one for the stop criteria) AND a hysteresis / two different temperatures.

**Parameters: None**

#### Inputs

Nr.	Description	Unit
1	Tup	°C
2	Tlow	°C
3	TupSet	°C
4	TlowSet	°C

Table 31: Inputs of Type 890

#### Outputs

Nr.	Description	Unit
1	gamma	--

Table 32: Outputs of Type 890

**The output-control function gamma will adopt the following values at timestep n:**

$Tup < TupSet$	$Tlow < TlowSet$	$gamma_{n-1}$		$gamma_n$
YES	NO	1 or 0	→	1
YES	YES	1 or 0	→	1
NO	YES	1	→	1
NO	NO	1 or 0	→	0
NO	YES	0	→	0

Table 33: output-values of Type 890, dependent on inputs and gamma of last timestep

### 10.3 TYPE 805: DHW heat exchanger

Version 1.0, MH 2006/12

#### Parameters

Nr.	Description	Unit
1	specific heat of primary (hot) side fluid	[kJ/hK]
2	specific heat of secondary (cold) side fluid	[kJ/hK]
3	maximum allowed flow rate on primary (hot) side fluid	[kg/h]

Table 34: Parameters of Type 805

#### Inputs

Nr.	Description	Unit
1	inlet temperature of primary (hot) side	[°C]
2	inlet temperature of secondary (cold) side	[°C]
3	flow rate on secondary (cold) side	[kg/h]
4	temperature setpoint for secondary side outlet	[°C]
5	overall heat transfer coefficient UA of heat exchanger	[kJ/hK]

Table 35: Inputs of Type 805

#### Outputs

Nr.	Description	Unit
1	primary side outlet temperature	[°C]
2	primary side flow rate	[kg/h]
3	secondary side outlet temperature	[°C]
4	secondary side flow rate	[kg/h]
5	heat transfer rate	[kJ/h]
5	effectiveness $\varepsilon$	[-]

Table 36: Outputs of Type 805

The equations are based on the effectiveness-NTU calculation for a standard counterflow heat exchanger [Duffie 1991] which are also used in the standard TRNSYS Type 5. The primary side flow rate is calculated iteratively until the setpoint temperature on the secondary side or maximum flow rate on the primary side is reached.

$$\text{Eq. 10.1} \quad \varepsilon = \frac{1 - e^{-UA/(\dot{m}C_p)_{\min}(1-C^*)}}{1 - C^* \cdot e^{-UA/(\dot{m}C_p)_{\min}(1-C^*)}} \quad \text{if } |1 - C^*| \geq 0.01$$

$$\text{Eq. 10.2} \quad \varepsilon = \frac{UA/(\dot{m}C_p)_{\min}}{1 + UA/(\dot{m}C_p)_{\min}} \quad \text{if } |1 - C^*| < 0.01$$

$$\text{Eq. 10.3} \quad C^* = \frac{(\dot{m}C_p)_{\min}}{(\dot{m}C_p)_{\max}}$$

$\varepsilon$  [-] effectiveness of heat exchange

$(\dot{m}C_p)_{\min}$  [kJ/hK] minimum capacity flow rate from both sides of heat exchanger

$(\dot{m}C_p)_{\max}$  [kJ/hK] maximum capacity flow rate from both sides of heat exchanger

## 10.4 TYPE 358: free driven window ventilation

Version 1.1, AW, RH 2005/06

This free driven tilted window ventilation type is based on the method from A. Weber [WEBER 1997] using a simple but CFD validated Bernoulli – approach. The air exchange rate through the window is calculated depending on the temperature difference between room and ambient temperature, the geometry of the window and the window opening tilt angle (Eq. 10.4, Eq. 10.5, Eq. 10.6).

### Parameters

Nr.	Description	Unit
1	mode (not used)	
2	number of windows (max. 20)	[-]
3	height of window 1	[m]
4	width of window 1	[m]
5,6...	height and width of window 2, 3, etc.	[m],[m]

Table 37: Parameters of Type 358

### Inputs

Nr.	Description	Unit
1	ambient air temperature	[°C]
2	room temperature for window 1	[°C]
3	aperture tilt angle for window 1	[°]
4,5...	room temperature and aperture angle for window 2,3, etc.	[°C], [°]

Table 38: Inputs of Type 358

### Outputs

Nr.	Description	Unit
1	air exchange rate through window 1	[m <sup>3</sup> /h]
2,3...	air exchange rate through window 2,3, etc.	[m <sup>3</sup> /h]

Table 39: Outputs of Type 358

$$\text{Eq. 10.4} \quad \dot{V}(\alpha) = C_k(\alpha) \cdot C_d \cdot H \cdot W \cdot \frac{1}{3} \sqrt{\frac{\Delta T}{T}} gH$$

$$\text{Eq. 10.5} \quad C_d = 0.0147\alpha - 0.0928 \cdot \frac{H}{W} + 0.4116$$

$$\text{Eq. 10.6} \quad C_k(\alpha) = 2.6E-7 \cdot \alpha^3 - 1.19E-4 \cdot \alpha^2 + 1.86E-2 \cdot \alpha$$

$\dot{V}(\alpha)$ [m <sup>3</sup> /s]	volume flow rate at opening tilt angle $\alpha$
$\alpha$ [°]	opening tilt angle
$C_d$ [-]	discharge coefficient
$H$ [m]	height of the window
$W$ [m]	width of the window
$\Delta T$ [K]	difference of room and ambient temperatures
$T$ [K]	average of room and ambient air temperatures
$g$ [m/s <sup>2</sup> ]	acceleration of earth's gravity
$C_k(\alpha)$ [-]	relative volume flow rate at opening tilt angle $\alpha$

## 10.5 TYPE 816: transient value averaging

Version: 2006-11, MH

Type 816 delivers the average of a time-dependent value. At the beginning of the simulation, the average will be taken of the available values. As soon as the time of averaging is reached, only the values from current time minus time of averaging will be considered.

### Parameters

Nr.	Description	Unit
1	number of values	[-]
2	time of averaging	[h]
3	not in use	[-]

Table 40: Parameters of Type 816

### Inputs

Nr.	Description	Unit
1	value 1	any
2,3...	value 2, value 3, ...	any, any, any, ...

Table 41: Inputs of Type 816

### Outputs

Nr.	Description	Unit
1	average of value 1 over time	any
2,3...	average of value 2,3, etc.	any, any, any, ...

Table 42: Outputs of Type 816

In order to calculate the average, this Type must store all values back to the current time minus the time of averaging. The number of storage spots this Type is allowed to cover is limited to 2000, independently of the number of values to be averaged. Type 816 does not simply add all values together each timestep and divide the result by the number of timesteps in the averaging period, but performs the calculations shown in Eq. 10.7 and Eq. 10.8. These calculations are a lot faster when dealing with long averaging periods over many timesteps.

$$\text{Eq. 10.7} \quad \text{Average}_N = \text{Average}_{N-1} + \frac{\text{Value}_N - \text{Value}_{N-X}}{X}$$

$$\text{Eq. 10.8} \quad X = \frac{dt\text{Average}}{dt\text{Sim}}$$

$\text{Average}_N$  [any] average of the variable calculated for timestep N

$\text{Value}_N$  [any] instant value of the variable at timestep N

$dt\text{Average}$  [h] time of averaging

$dt\text{Sim}$  [h] simulation timestep

## 11 Annex C: General calculations

### 11.1 Transversal and longitudinal incidence angles

The calculation of the transversal and longitudinal incident angles is done based on equations from [Klein 1996], which had to be slightly adapted / corrected to produce the correct results:

$$\theta_L = \left| ATAN \left( TAN(\theta_z) \cdot COS(\gamma - \gamma_s) \right) - \beta \right|$$

$$\theta_T = ATAN \left( \frac{SIN(\theta_z) \cdot SIN(|\gamma - \gamma_s|)}{COS(\theta)} \right)$$

$\theta_L$  Angle of incidence on collector plane in longitudinal direction

$\theta_T$  Angle of incidence on collector plane in transversal direction

$\theta_z$  Solar zenith angle

$\gamma$  Collector azimuth angle

$\gamma_s$  Solar azimuth angle

$\theta$  Angle of incidence on collector plane

$\beta$  Collector slope

### 11.2 U value for cylindrical elements (pipes, store)

$$U_{pipe} = \frac{3.6}{R}$$

$$R = \frac{d_i \cdot \ln\left(\frac{d_a}{d_i}\right)}{2 \cdot \lambda_w} + \frac{d_i \cdot \ln\left(\frac{d_{iso}}{d_a}\right)}{2 \cdot \lambda_{iso}} + \frac{d_i}{\alpha_a \cdot d_{iso}} + \frac{1}{\alpha_i}$$

$U_{pipe}$  [kJ/hm<sup>2</sup>K] heat loss coefficient of the pipe wall (based on inner diameter/surface)

$R$  [m<sup>2</sup>K/W] thermal resistance of pipe wall (based on inner diameter/surface)

$d_i$ ,  $d_a$ ,  $d_{iso}$  [m] diameter of pipe inside, pipe outside and pipe outside including insulation

$\lambda_w$ ,  $\lambda_{iso}$  [W/mK] thermal conductivity of pipe wall and pipe insulation respectively

$\alpha_a$  [W/m<sup>2</sup>K] outer heat transfer coefficient (from pipe insulation to ambient)

$\alpha_i$  [W/m<sup>2</sup>K] inner heat transfer coefficient (from pipe wall to fluid inside pipe)

## 12 Annex D: Outputs of Task 32 printer

QBurn_kWh	energy of fuel burnt by the auxiliary burner based on the lower heating value
QSH_kWh	energy delivered to building by the heating system (convective and radiative energy)
QCool_kWh	energy used for the cooling device used for space cooling (make sure to handle conversion factors for electricity appropriately)
QSC_kWh	energy drawn from the building by the cooling system (cooling load)
QDHW_kWh	energy drawn by domestic hot water tapping
Qpen45_kWh	penalties for dhw-temperatures below 45 °C
Qpen25_kWh	penalties for room temperatures too high (see Figure 6.1)
Qpen20_kWh	penalties for room temperatures below 19.5 °C
Wsolar_kWh	electricity demand of solar system
IRRAD_kWh	irradiation on total collector field
refID	reference ID used to identify city and building chosen for simulation
Vstore_m3	volume of heat store simulated
QStColl_kWh	energy balance of store doubleport used for collector loop
QStAux_kWh	energy balance of store doubleport used for auxiliary loop
QStBuil_kWh	energy balance of store doubleport used for building loop
QStDraw_kWh	energy balance of store doubleport used for draw-off loop
QBurnWat_kWh	energy balance of water-side of auxiliary boiler
Wpumpsol_kWh	electricity demand of solar pumps
Wburner_kWh	electricity demand of burner
Wcontr_kWh	electricity demand of controller
Wpumpheat_kWh	electricity demand of space heating pump
Wpumpdhw_kWh	electricity demand of dhw pump
QSt_inp_kWh	energy delivered to the store
QSt_out_kWh	energy drawn from the store
QSt_loss_kWh	energy losses from store
QCpipe_loss_Wh	collector pipe losses
Qst_int_kWh	energy stored in store
QC_kWh	energy delivered by the collector (balance at collector-connections..)
Aux_starts	number of starts of the auxiliary burner

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