



**Expert-system for an INtelligent Supply of Thermal Energy
in Industry and other large scale applications**

EINSTEIN Software – Tool

Technical Manual

Version 2.5.00

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1 Data input and consistency checking

1.1 General description of the consistency check module

EINSTEIN disposes of an algorithm for consistency and completeness checking (for shortness: consistency check) of all data that are input by the user for describing a system.

The objective of consistency and completeness checking is:

- avoid data that are inconsistent (conflicts): for many data inputs EINSTEIN allows redundant data input (e.g. process mass and volume flows can be specified by both the mass flow rate or the per cycle total volume / mass, etc.)
- assure that all data are defined that are required for a further processing of the data

Consistency checking is done both for input data describing the present state, and for input data that may be added or changed in alternative proposals for the system

1.1.1 Local consistency checks

Local consistency checks are carried out for any user data entry, and for both present state and in alternative proposals.

In local consistency checking, the data set describing a given object (equipment, pipe, process, heat exchanger, etc.) is checked for consistency and for completeness, in some cases also checking for consistency with the direct environment, but without carrying out a global check of the whole company data set.

1.1.2 Global consistency and completeness check at present state

Present state global consistency and completeness checking is activated manually by the user. Global consistency checking includes both the parameters describing the system (equipment dimensions and technical data, etc.) and also the parameters describing the performance (energy flows, etc.).

IMPORTANT: In global consistency checking at present state no system simulation is carried out, but only cumulative or average annual data are considered in the calculations. This means that at this stage only part of the available information is processed!

1.1.2.1 The concept of completeness

In first place completeness means, that all system parameters that are needed in order to do any reasonable further processing have to be defined. E.g. for carrying out any system simulations, nominal powers and efficiencies and other important base data of equipment, pipes, processes have mandatorily to be defined.

But, furthermore, completeness in EINSTEIN means also, that EINSTEIN has not only been able to assign some numeric value to all mandatory parameters, but also that those parameters are defined with sufficient accuracy (→ see section on random errors and absolute limits).

What is “sufficient”, can partially be defined by the user by selecting between the required accuracy levels “quick and dirty”, “standard” and “detailed” (see also EINSTEIN Audit Guide).

1.1.2.2 Top-down vs. bottom-up modelling

A very frequent problem in energy auditing is resolving possible conflicts or filling remaining leaks between top-down data and bottom-up calculation results.

Top-down data are in first place the data on final energy consumption that usually are available from energy bills and nominal powers of equipment, and in second place the data derived from these data (e.g. based on conversion efficiencies of equipment).

Bottom-up data are data calculated from process and building parameters (mass flows, temperatures, etc.).

In between are several parameters that often are unknown in detail, and that, therefore, can be used for adjustment: equipment conversion efficiencies in some cases, pipe and duct losses.

EINSTEIN automatically detects conflicts between top-down and bottom-up data and in some cases can also help to do matching automatically (in cases where the gap between the data is not too large). For bigger conflicts or leaks, the user has to change the original data set manually.

1.1.2.3 Convergence of consistency check algorithm

The measure of convergence used in EINSTEIN consistency checking is the (relative) difference between different calculation results x_i^k for the same parameter x_i .

$$\delta_i = \frac{\max(x_i^k) - \min(x_i^k)}{\max(|x_i^k|)} \quad (1.1.1a)$$

Exceptions are:

- parameters, that are fractions of one, where the reference value is 1:

$$\delta_i = \max(x_i^k) - \min(x_i^k) \quad (1.1.1b)$$

- energy flows, where δ_i is limited in case of large relative but small absolute errors (compared to a critical value ΔE_c (current default: 10% of maximum energy flow in the system))

$$\delta_i = \frac{\max(x_i^k) - \min(x_i^k)}{\max(|x_i^k|, \Delta E_c)} \quad (1.1.1c)$$

From the δ_i - values for the individual parameters, global measures for convergence are derived that are displayed in the EINSTEIN GUI:

- the average relative error

$$\bar{\delta} = \frac{\sum \delta_i}{N} \quad (1.1.2)$$

- the maximum relative error

$$\delta_{max} = \max(\delta_i) \quad (1.1.3)$$

In order to help the user to detect where possible convergence problems may arise from, the “black sheep” is given in the output, which is that parameter with the maximum relative convergence error.

The iterative algorithm is interrupted once the maximum number of iterations is reached or the global error measures are below the critical values: 0,1 % for the average relative error and 1% for the maximum relative error. Furthermore, iteration is also interrupted if after a certain number of iterations no further improvement of the errors is obtained.

1.1.3 *Data estimates*

EINSTEIN offers an automatic tool for data estimation of missing data or data that are not known with sufficient accuracy.

Data estimation is optional and has to be activated manually by the user.

If automatic data estimation is carried out, EINSTEIN changes exclusively those parameters, that in the previous step of consistency and completeness checking could *not* be specified with sufficient accuracy. Data that are already available are not overwritten by the automatic procedure.

The default values and default correlations for data estimation are described below in the sections on the different modules in the sub-section “data estimates”.

1.1.4 *Calibration of simulations*

As stated above, in consistency checking and data estimation, no simulations of the present state system are carried out.

Currently it is optional, but strongly recommended that the user manually carries out a calibration of system simulations.

This is done by generating a copy of the present state, once checked for consistency, as first “alternative” proposal “simulation of present state” or “PSsim”.

As a necessary condition for reliability of simulation results for alternative proposals, it should be expected that the simulation of present state gives similar data that observed in reality (fuel bills, ...).¹

If this is not the case, the user can calibrate simulation results by tuning some of the system parameters.

In current version, manual calibration is necessary for the following aspects:

a) performance of equipment whose technical data depend on instantaneous temperature levels or part load

In the current version V2.2. this equipment may be:

- compression chillers (both air and water cooled) and heat pumps, where the conversion efficiency is considered both depending on temperature levels and on part load.
- all CHP systems, where the utilisation ratio of generated heat depends on the instantaneous temperature profile of heat demands

¹ *Data that should be checked are in first place the total final and primary energy consumption and the totals for USH andn UPH, but also the data for FET and USH for each equipment.*

- heat exchangers: in present state, no calculation of the heat recovery network is carried out, but user specified annual average data are used.

b) groups of equipment, whose relative contribution to heat or cooling demand depends on the control strategy.

The simplest example for this are two boilers, connected in parallel to the same demands (pipes). In present state the user can specify the percentage of the total demand covered by each of both boilers, whereas in system simulations this percentage is a calculation result, based on the simplified control strategy supposed by EINSTEIN, which not necessarily may be in coincidence with the real distribution.

If, furthermore, both boilers have different conversion efficiencies or use different types of fuel, then a difference in the distribution of the total demand on the individual boilers may have an effect on (lead to an error in) total final energy consumption.

Calibration of this type of errors in the current version is not straightforward, as the only direct ways to influence the control strategy supposed in simulations is either change the ordering of the equipment cascade or change the model of the distribution network (e.g. split of some pipe into two parts).

Additional tricks that may have to be used are:

- split-up of some equipment into two or more parts, placed on different positions in the equipment cascade
- artificially limit the contribution of some equipment by entering a nominal power lower than the real value

c) differences between present state and present state simulated due to bad convergence of consistency check

Numeric differences in results may also be due to a not converged data set at consistency check level. In this case the solution is to correct or to complete the original data set in order to obtain a better performance.

1.1.5 Overview: block scheme

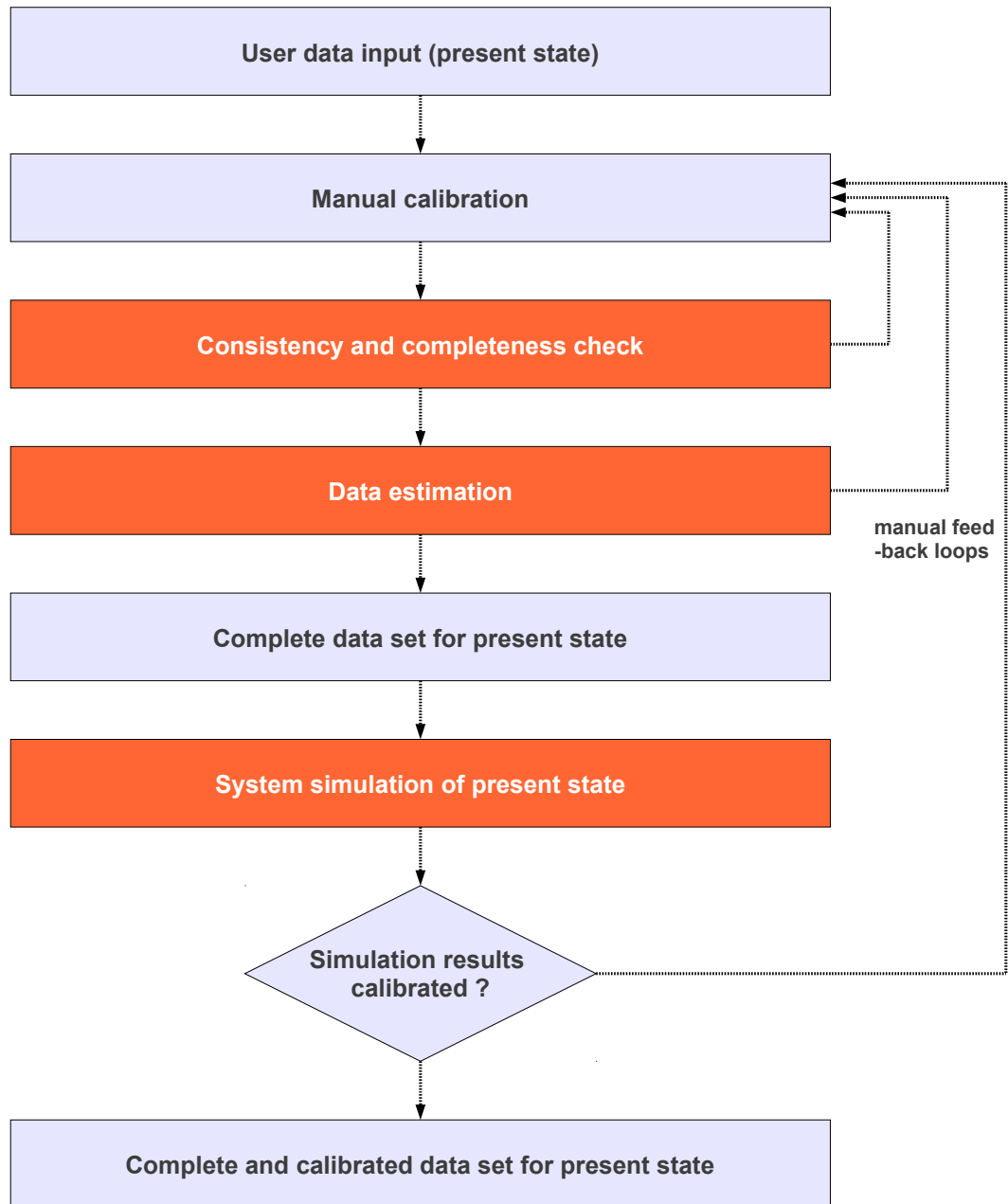


Figure 1. Flow chart of consistency and completeness checking in EINSTEIN.

1.2 Random errors and absolute limits

1.2.1 Basic concept

Each parameter x used by EINSTEIN is considered being associated with a certain degree of uncertainty.

The parameter is described by the following attributes:

x : the most probable value of the parameter. This value is used in further calculations in EINSTEIN.

x_{\min}/x_{\max} : the absolute limits of the parameter. Beyond these limits, no value is possible (probability = 0).

σ : the random error associated with a parameter assuming a Gaussian probability distribution (dev is identical to the standard deviation s of the unconstrained probability distribution)

Figure 2 illustrates the probability distribution for a given parameter.

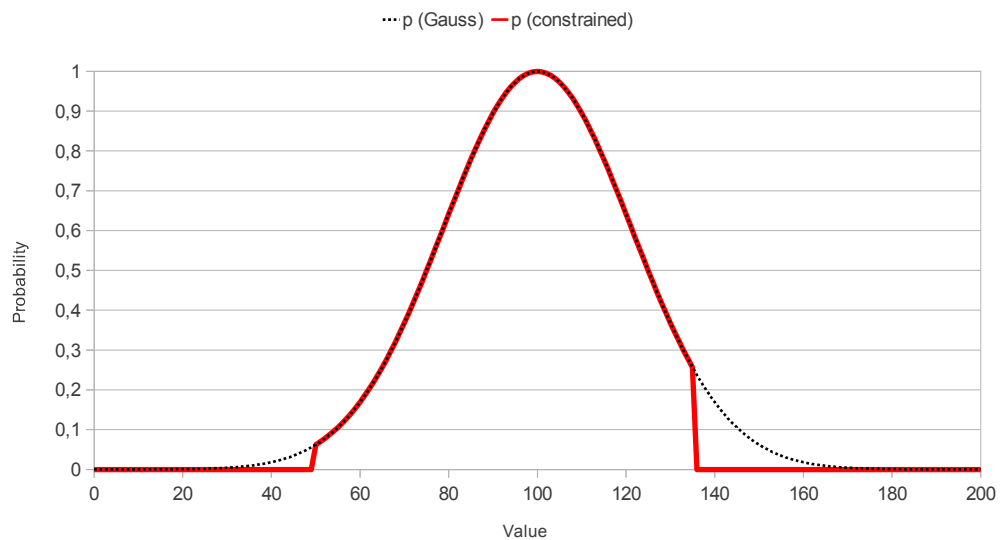


Figure 2. Probability distribution for a parameter with value $x = 100$, standard deviation $\sigma = 30$ and minimum / maximum interval $[50, 135]$.

From the standard deviation σ , the relative error can be calculated:

$$\epsilon = \frac{\sigma}{x} \quad (1.2.1)$$

1.2.2 Default error margins: user input and allowed tolerances

Currently (Version 2.2) EINSTEIN uses a fix critical limit for the relative error of a parameter, above which the parameter is considered as undetermined: $\epsilon_{crit} = 30\%$.²

For user input data, a certain error tolerance is assumed in order to avoid unnecessary conflicts (e.g. due to rounding errors).

In general, user data are considered having a relative error $\epsilon = 0,1\%$. Exceptions are:

- temperatures (but not temperature differences !): are supposed to have an (absolute) standard deviation $\sigma = 0,1$ K.
- zero values are considered as exact 0 ($\epsilon = 0$; except temperatures of 0 °C)
- values $x = 1$ for parameters that are of type fractions of 1 are considered as exact ($\epsilon = 0$), too.

Different estimates of data $x_i^{k_1}$ and $x_i^{k_2}$ are considered to be in conflict if one of the following conditions applies:

- absolute limit ranges ($\mathbf{x}_{min}/\mathbf{x}_{max}$) do not overlap
- $\delta_i > 4\sqrt{(\epsilon_{k_1}^2 + \epsilon_{k_2}^2)}$

where δ_i is the relative error as given by equation (1.1.1a) calculated for the pair of estimates $x_i^{k_1}$ and $x_i^{k_2}$ and the factor 4 describes the margin of tolerance in relation with the standard deviation of the estimates.

1.2.3 Error propagation

The whole system treated by EINSTEIN can be describe by a set of equations correlating the different parameters xi.

The mathematics of error propagation is describe in the simple examples of sums and products. For other types of correlations error propagation is calculated analogous (although mathematics may be a little bit more complex):

a) error propagation in the case of sums

For the equation

$$y = \sum_i x_i \quad (1.2.2)$$

error propagation is obtained as

$$\sigma_y = \sqrt{\sum_i \sigma_{x_i}^2} \quad (1.2.3)$$

$$y_{min/max} = \sum_i x_{i,min/max} \quad (1.2.4)$$

² In some cases, this may lead to unnecessary error messages warning about undetermined parameters, that occur for parameters (e.g. energy flows) with very low absolute value, where a very low absolute error, that does not affect accuracy of global results, may lead to a high value of relative error.

b) error propagation in the case of products

For the equation

$$y = x_1 x_2 \quad (1.2.5)$$

error propagation is obtained as

$$\epsilon_y = \sqrt{\epsilon_{x_1}^2 + \epsilon_{x_2}^2} \quad (1.2.6)$$

$$y_{min/max} = x_{1,min/max} x_{2,min/max} \quad (1.2.7)$$

Note: for simplicity here treatment of special cases where either x_1 or $x_2 = 0$ are not included. The simple correlation for $y_{min/max}$ is valid only for positive parameters. In the general case where x_1 or x_2 also may have negative values, a little bit more complex equations apply.

1.3 Monitoring data and parameter fitting (EINSTEIN plus)

1.3.1 Calculation of the fitting error

Monitoring data for parameter x_i can be defined for instantaneous values of quantities $dx_i^o(t)$ at time t or for subtotals over certain periods $dx_i^o(p)$.

If schedules for some quantities (e.g. process heat demands, production volumes) are known, by mass and energy balances schedules for other quantities can be (approximately) calculated without system simulation, leading to calculated results for the parameters $dx_i(t)$ or $dx(p)$.

The relative error of fitting then is calculated to

$$e_i(t) = \frac{dx_i(t) - dx_i^o(t)}{\overline{dx_i^o}} \quad (1.3.1)$$

If there are several monitoring data, the global error E defined as the normalised sum of the squares of the residuals is minimised:

$$E = \frac{\sum_i \sum_t e_i(t)^2}{N_i N_t} \quad (1.3.2)$$

1.3.2 Determination of the accuracy of a parameter

If the global error depends on a set of N_j independent parameters x_j , then the standard deviation of the fitting estimate for the parameter x_j is calculated from the second derivative of the global error (Hessian matrix) as:

$$\sigma = \sqrt{\frac{2 E_{min}}{\left(\frac{d^2 E}{dx_j^2}\right)}} \quad (1.3.3)$$

2 Process Model

The process model used in EINSTEIN is described in detail in the EINSTEIN Audit Guide , Chapter 2. In the following only some additional details are given referring to the definition of the circulation heat demand and to the time schedules.

2.1 Demand components and temperature levels

Regarding the circulation heat (or cooling) demand of different inflowing streams the following relation between nominal power and energy consumption per cycle defines the effective operating hours per operation cycle.

$$\dot{Q}_{UPH,c}^{nom} = \frac{Q_{UPH,c}^{cycle, nom}}{t_{op}^{cycle}} \quad (1)$$

The nominal power is obtained from user input either via the inflowing volume (V_{inflowCycle}) or by inflowing the nominal mass flow rate.

2.2 Time schedules

The time schedules give a scaling factor for the energy demands and waste heat outputs of a process which gives the ratio of actual to nominal power of the heat (or cooling demand).

f(t) relates the actual power consumption to the nominal power consumption, where x stands for one of the UPH components c, m, s or w:

$$\dot{Q}_{UPH,x} = f(t) \dot{Q}_{UPH,x}^{nom} \quad (2)$$

$f_{av}(t)$ gives the fraction of the yearly energy consumption for a given time interval Δt :

$$\Delta Q_{UPH,x} = f_{av}(t) Q_{UPH,x}^{nom} \quad (3)$$

Both functions are normalized so that

$$\int_{year} f(t) dt = t_{op,eff}^{year} \quad (4)$$

$$\sum_{year} f_{av}(t) = 1 \quad (5)$$

which implies that:

$$f_{av}(t) = \frac{\Delta t}{h_{year}} f(t) \quad (6)$$

2.2.1 Real and effective number of operating hours and part-load factor

$t_{op,eff}^{year}$ is the yearly number of effective operating hours at nominal power.

The actual number of operating hours t_{op}^{year} can be obtained adding up the length of all time intervals where f(t) > 0.

The average part-load factor is then given by

$$f_{partload} = \frac{t_{op,eff}^{year}}{t_{op}^{year}} \quad (7)$$

The number of operating days N_{days} is defined as the number of all days, where at least at some time $f(t) > 0$. Then the average operating hours per operating day can be defined as

$$t_{op}^{day} = \frac{t_{op}^{year}}{N_{days}} \quad (8)$$

and the average effective operating hours per operating day are given by

$$t_{op,eff}^{day} = \frac{t_{op,eff}^{year}}{N_{days}} \quad (9)$$

The (yearly) number of cycles N_{cycles} is defined as the number of all changes from $f(t) = 0$ to $f(t) > 0$ during a day. (important note: 0:00 and 24:00 o'clock are considered as limits of a cycle if one is running. If a process works continuously without breaks during 8760 hours, the total number of cycles would be 365)

Then the average operating hours per operating cycle can be defined as

$$t_{op}^{cycle} = \frac{t_{op}^{year}}{N_{cycles}^{year}} \quad (10)$$

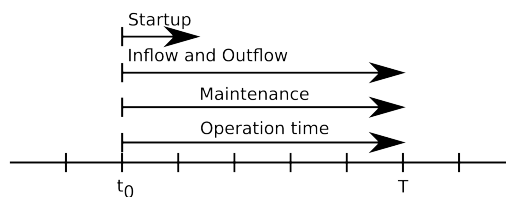
and

$$t_{op,eff}^{cycle} = \frac{t_{op,eff}^{year}}{N_{cycles}^{year}} \quad (11)$$

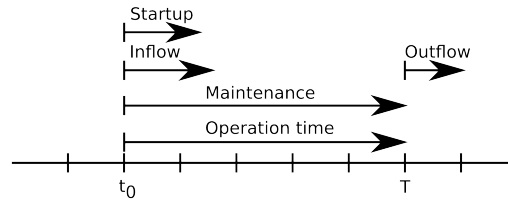
2.2.2 Start-up, Inflow and Outflow times and the profiles

The simpler of the two possible cases is the continuous process where inflow and outflow happen throughout the whole process Operation time, and the two phases of a cycle are only Start-up and Maintenance. Here the times with a scaling factor $f(t) > 0$ are considered as times of activity of the process.

The duration of the Start up defaults to $t_{op}^{cycle} \cdot 0.2$ but can be set explicitly as part of the detailed process schedule.



In the case of a batch process the inflow and outflow happen not continuously during the operation time but at the start and the end of a cycle. The Inflow phase is part of the given operation time just like the Start-up, but the Outflow phase is appended after the end of the Operation time.



The duration of the Inflow and Outflow phases default to $t_{op}^{cycle} \cdot 0.2$ but can be set explicitly for batch processes as part of the detailed process schedule.

2.2.3 Running times of a simple schedule

A simple schedule is defined by the number of *days of operation per year* and by two of the three parameters *Effective operation hours per day*, *Number of cycles per day* and *Duration of 1 cycle* where each one is the direct result of the other two as there is no scaling of the process parameters possible.

The detailed Schedule created from these parameters is build to satisfy

$$N_{days}^{year} = \text{Days of operation per year} ,$$

$$t_{op}^{cycle} = t_{op,eff}^{cycle} = \text{Duration of 1 cycle} ,$$

$$\frac{N_{cycles}^{year}}{\text{Days of operation per year}} = \text{Number of cycles per day} \text{ and}$$

$$t_{op}^{cycle} = t_{op,eff}^{cycle} = \frac{\text{Effective operation hours per day}}{\text{Number of cycles per day}} .$$

This is achieved in the following way:

First a daily profile is created that distributes the cycles evenly over a day as follows. If the *Operation hours per day* are not greater than the *Total hours of operation per working day* from the General data entries then the first cycle starts at $12 - 0.5 \cdot \text{Operation hours per day}$ o'clock. And every next cycle of that days follows $\frac{\text{Total hours of operation per working day}}{\text{Number of cycles per day}}$ hours after the last start.

For example the operation hours of a process with 3 cycles per day and a duration of 2 hours each within a daily operation time of 9 hours would be distributed like shown in Illustration 1.

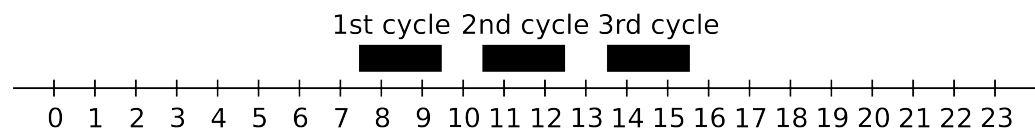


Illustration 1: Distribution of cycles within the daily operation time

If the daily operation time would have been only 5 hours, and the three cycles would therefore not fit into the time frame, then the cycles would start at 0 o'clock and be

spread over the 24 hours of the day, each cycle starting $\frac{24}{\text{Number of cycles per day}}$ hours after the last cycle start.

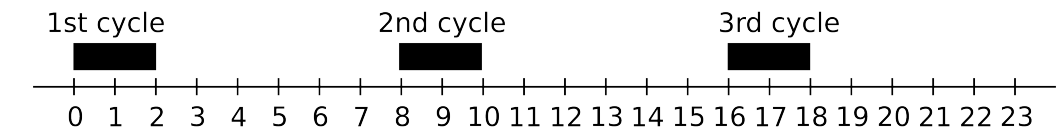


Illustration 2: Distributions of cycles that don't fit within the daily operation time

In the case of a batch process each cycle is then followed by the the Outflow phase of the process. The Outflow phase may actually overlap with the start of the next cycle.

These daily profiles are then used to create a weekly profile of N days per week, filling up weeks starting from Mondays. If the resulting number of days per week is not a full integer, the last day of each week is considered with a part load ratio < 1.

3 System simulation in EINSTEIN

3.1 Overview of system simulation module

3.1.1 General overview

The model of the heat and cooling supply systems treated in EINSTEIN follow the block scheme as outlined in the Audit Guide Chapter 2, dividing the system in the following blocks:

- energy sources used (final energy)
- conversion equipment
- distribution of heat and cooling
- heat and cooling demands (processes)
- heat recovery
- external waste heat sources (waste heat from electrical equipment)

The EINSTEIN system simulation generates a full image of the time- and temperature dependent heat flows in all points of the system under consideration for a full year³ of operation.

3.1.2 Pipe and duct modeling

In the current version of EINSTEIN (V2.2), in system simulations no direct modeling of the heat and cooling distribution is carried out, but energy demand at pipe input is calculated from energy delivered at pipe output based on the following assumptions:

- *constant distribution efficiency*: constant heat or cooling distribution efficiency (ratio between energy output and input), calculated for nominal pipe parameters (temperatures and flow rates) and not for actual ones.
- *minimum required process supply temperature*: determination of required temperature levels at pipe input based on the minimum required process supply temperature (see description of process model) and not on the actual temperatures used in the distribution system.
- *zero temperature drop*: distribution losses are considered as energy losses in the energy balances of the pipe, but not as temperature drop.

Limitations and workarounds:

- possible variations of distribution efficiency due to changes in demand (temperature levels or flow rates) are not taken into account automatically. If desired, the user can consider such changes by manually adjusting nominal pipe parameters.

- differences between actual and simulated system performance may occur due to the fact, that minimum supply temperature is used in simulations and not actual supply temperature. Nevertheless, the user can tune the desired temperature levels by adjusting appropriately the minimum process supply temperature of the connected processes.⁴

³ A representative year of 365 days is used, beginning with a Monday at 1st of January

⁴ Real pipe modeling, i.e. the consideration of real temperature levels and flow rates in system simulation, has not yet been introduced in Version 2.2. mainly due to the fact, that a consideration of real pipe parameters would require that the auto-design of new supply equipment would have to consider also the automatic re-design of distribution (e.g. substitute a steam boiler + steam distribution by a CHP system or solar thermal system + hot water distribution), making the auto-design algorithms more complex. This is work in progress and will be considered in future updates.

3.1.3 Regulation and control: Equipment cascade and master control

3.1.3.1 Equipment cascade

Regulation and control of real systems in the current version of EINSTEIN is approximated by a hierarchy of supply equipments, the equipment cascade.

In the order of their *cascade index*, each equipment contributes as much of heat (or cooling) to the demand of the connected pipes as possible (limits are given e.g. by maximum or nominal power, maximum or minimum temperature level of output, or by availability of required inputs, such as solar energy or waste heat).

The heat demand of the connected pipes is covered from low to high temperatures, without carrying out an exergetic optimisation by matching temperature distributions of supply and demand.

If several pipes are connected to a given equipment, the heat (or cooling) delivered is distributed to the different pipes by the following rules:

- a) priority pipes: those pipes, for which the given equipment is the last in the cascade, have priority for receiving the generated heat (or cooling).
- b) non-priority pipes: the remaining heat (or cooling) is distributed proportionally to the pipes' demands.

3.1.3.2 Master control

Optionally a master control schedule $f(t)$ can be defined for any equipment, with $0 \leq f(t) \leq 1$.

In this case, the maximum power to be supplied within a given time step t is given by

$$\dot{Q}_{max} = f(t) \dot{Q}_{nom} \quad (3.1.1)$$

The actual power supplied is determined by the equipment cascade algorithm as described above.

3.2 The EINSTEIN simulation engine

3.2.1 Iterative solver and flow chart

The solution of the mathematical equations describing a system has to be iterative, in order to consider several possible feed-back loops (e.g. change in supply → change in available waste heat → change in heat recovery → change in demand → change in supply ...).

Nevertheless, the following blocks are considered as constant (in the sense as fully defined by user input) and can therefore be used as fix starting point for simulation:

- processes' energy demands
- waste heat from electrical equipment

The initial waste heat flows from pipes and equipment are estimated at the beginning of the iterative cycle from yearly values (if available from consistency checking or previous simulations) using an approximate time distribution.

Convergence is reached if the relative variation of waste heat flows and required heat inputs of equipments is sufficiently small.

Limitations and workarounds:

- in order to limit calculation time, the maximum number of global iteration cycles is limited to 3 in the current version. this may lead to situations (especially where large fractions of waste heat from pipes or supply equipment is used) where no full convergence is reached. In these cases there may be variations in the results between successive simulations.

Workaround: manual activation of successive iteration cycles

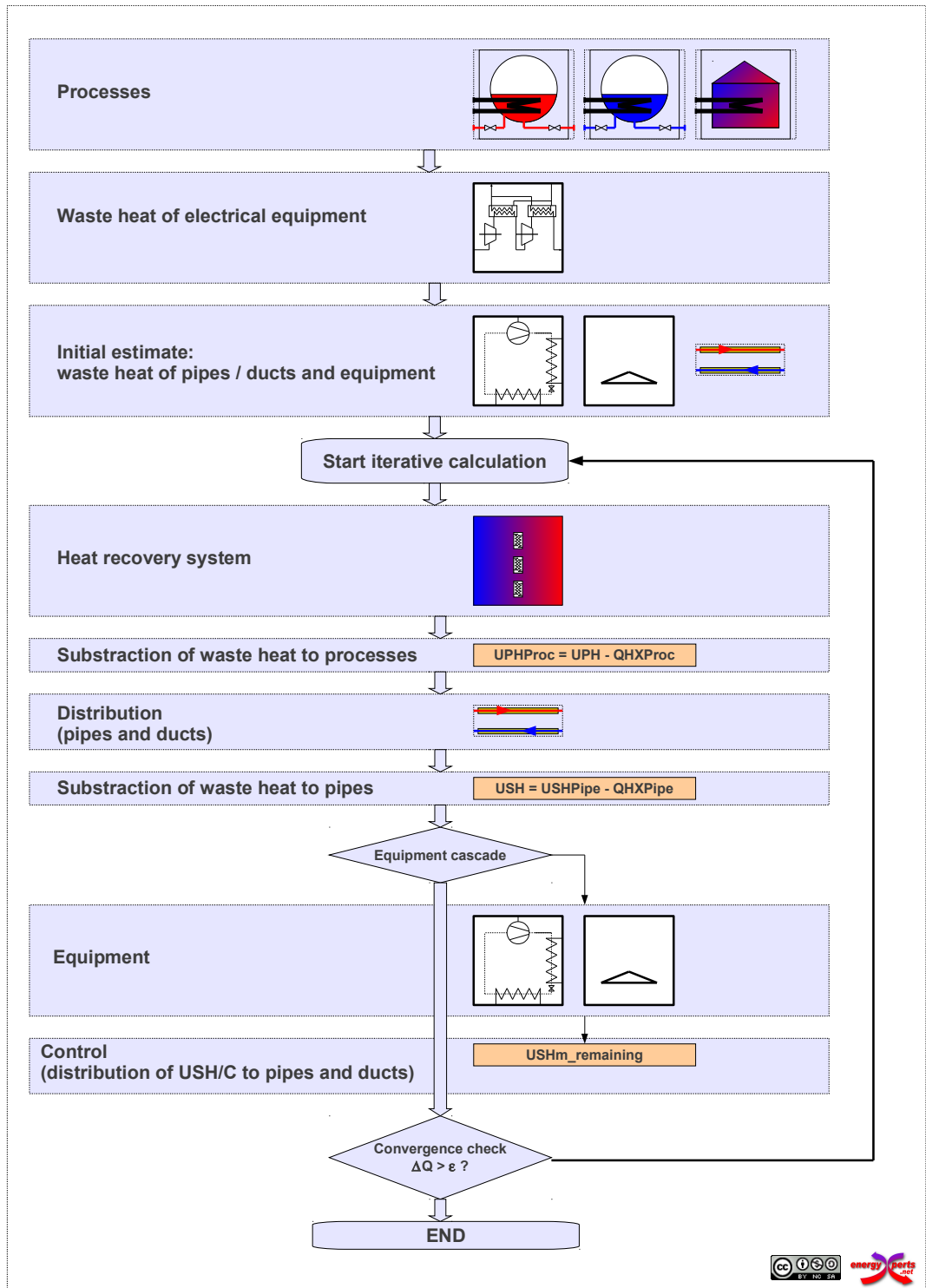


Figure 3. Flow chart of the iterative solver

3.2.2 Discretization and numerical errors

EINSTEIN works with fix time and temperature intervals for numerical solution of the system equations, which may lead to numeric errors.

3.2.2.1 Time intervals

The simulation time step of EINSTEIN is 1 hour. This means that the year is divided into 8760 intervals.

For fluctuations in energy flows on a shorter time scale, calculations are carried out on hourly averages.

Limitations and workarounds:

- simultaneity of energy flows: the hourly averages may lead to erroneous results regarding simultaneity of energy flows. E.g. energy flow 1 from 15:00 to 16:00 and energy flow 2 from 16:00 to 17:00 are correctly detected as non-simultaneous, whereas energy flow 1a from 15:30 to 16:30 and energy flow 1b from 16:30 to 17:30 are considered as partially overlapped within the time interval from 16:00 to 17:00.

Workaround: adjust time schedules in order to fit to full-hour intervals

- underestimation of peak loads: short peak power requirements for durations of less than 1 hour are “averaged out”, e.g. a peak power of 1 MW during 15 minutes is averaged to an hourly average power of 0,25 MW.

Workaround: none.

3.2.2.2 Temperature intervals and temperature scale

For the representation of temperature distributions of heat and cooling demands the following discretization is used:

- heat demands: 5 K temperature intervals from 0°C to 400 °C. Demands of > 400 °C or < 0°C can be considered, but are not differentiated with respect to their temperature.
- cooling demands: 2,5 K temperature intervals from -60 °C to +100 °C. Idem: demands of < -60°C or > +100°C are not differentiated with respect to temperature.

Limitations and workarounds:

- a cooling demand corresponding to a distribution at +7°C/+12°C forward/return temperature is seen by EINSTEIN with part of the energy demand (10%) in the temperature interval from 5 to 7,5 °C (DT = 2,5K). This means, that the minimum temperature assumed in simulation is the lowest temperature of the interval with demand > 0, which in this case is +5 °C (instead of the correct value of +7°C).

Workaround: adjust your temperature levels to the EINSTEIN temperature scale (in this example e.g. to 7,5 / 12,5 instead of 7/12).

3.2.2.3 Yearly periodicity of simulations

The EINSTEIN simulation engine does not carry out a check of periodicity of results, this is that all temperatures and energy flows at 24:00 of 31st of December should be identical to the corresponding values at 0:00 of 1st of January.

For systems with usual storage sizes (energy storage for a maximum of few days) the possible numeric errors are small and can be neglected. Nevertheless, for very large storage sizes (seasonal storage) this may lead to significant errors in energy balances.

Limitations and workarounds:

- errors in energy balances for very large storage sizes (seasonal storage)

Workaround: none

4 Module descriptions

4.1 EINSTEIN – Pipe and Duct Calculation Module

4.1.1 Module parameters

Inputs (system simulation)

| Parameter | Symbol | Description |
|-------------------------------|-----------------|--|
| Required heat demand | UPHproc,m (T,t) | Process heat demand (sum of all processes connected to pipe m) |
| Heat input from heat recovery | QHxpipe,m (T,t) | |

Outputs (system simulation)

| Parameter | Symbol | Description |
|---|-----------------|--|
| Useful heat input to pipe | USHpipe,m (T,t) | Useful heat input required by pipe |
| Useful heat input to pipe from supply equipment | USHm (T,t) | Useful supply heat to pipe m, from supply equipment |
| Generated waste heat | QWHpipe,m(T, t) | Useful waste heat generated (including temperature distribution) |

Parameters

| Parameter (user interface) | Symbol | Description |
|---|-----------------------|---|
| Pipe length | l | One way pipe length |
| Heat loss coefficient | UA | Total heat loss coefficient (forward and return piping) |
| Inlet temperature forward pipe | Tfwd,i | Inlet temperature from heat supply equipment |
| Outlet temperature return pipe | Tret,o | For closed and partially open pipes: Outlet temperature from return pipe to equipment (before mixing with feed-up) |
| Inlet temperature from process to return pipe | Tret,i | Only for fully open pipes |
| Recirculation rate | frec | fraction of mass flow in forward pipe that is recirculated in return pipe |
| Pipe distribution efficiency | η_{dist} | Ratio between generated electricity and fuel consumption ($P_{\text{el}}/\text{fuel}$) |
| Pipe diameter | d_pipe | outer diameter of the pipe |
| Pipe insulation thickness | δ_{ins} | thickness of insulation material |
| Heat conductivity of insulation material | k | Currently a fixed default value of 0,05 W/mK used |

Data files and data bases

None

4.1.2 Mathematical description

4.1.2.1 General aspects

Pipes and ducts in EINSTEIN are based on the following assumptions:

- mass flow is conserved. there is no uncontrolled loss of heat transport medium (e.g. by leaks)
- a detailed calculation is carried out only at stage of parameter consistency checking, for nominal pipe parameters
- during system simulation, thermal losses of pipes are considered as a constant fraction of transported heat. thermal losses are considered as energy losses (loss of quantity) but not as temperature losses (loss of quality).

4.1.2.2 Energy balances

The relation between input and output into pipes is given by

$$\dot{Q}_{USH, pipe}(T, t) = \eta_{dist} \dot{Q}_{UPH, proc}(T, t) \quad (4.1.1)$$

where:

η_{dist} : distribution efficiency of the pipe as obtained from nominal conditions

$\dot{Q}_{USH, pipe}(T, t)$: power input into pipe (from equipment or heat recovery)

$\dot{Q}_{UPH, proc}(T, t)$: power output from pipe to processes

Pipes are calculated from “right” to “left”, this means the sum of heat (or cooling) demands of the connected processes is used to calculate the required energy input to the pipe.

The required output is obtained from the connected processes' heat or cooling demand, reduced by the energy supplied from heat recovery directly to the processes

For heating pipes:

$$\dot{Q}_{UPH, proc}(T, t) = \sum_{connected\ Processes} \dot{Q}_{UPH, k}(T, t) - \sum_{connected\ HX} \dot{Q}_{HX, h}(T, t) \quad (4.1.2a)$$

and for cooling pipes:

$$\dot{Q}_{UPC, proc}(T, t) = \sum_{connected\ Processes} \dot{Q}_{UPC, k}(T, t) - \sum_{connected\ HX} \dot{Q}_{WH, used, h}(T, t) \quad (4.1.2b)$$

The input energy to the pipe is composed by the energy supplied by the equipment and by energy from heat exchangers. For heating pipes:

$$\dot{Q}_{USH, pipe}(T, t) = \dot{Q}_{USH, m}(T, t) + \sum_{connected\ HX} \dot{Q}_{HX, h}(T, t) \quad (4.1.3a)$$

and for cooling pipes:

$$\dot{Q}_{USC, pipe}(T, t) = \dot{Q}_{USC, m}(T, t) + \sum_{\text{connected HX}} \dot{Q}_{WH, used, h}(T, t) \quad (4.1.3b)$$

4.1.3 Detail calculations for nominal parameters during consistency check

At consistency check mass and energy balances of the pipe are calculated:

$$\dot{q}_{m, fwd} = \dot{q}_{m, ret} + \dot{q}_{m, feedUp} \quad (4.1.4)$$

$$\dot{Q}_{USH, pipe} = \dot{Q}_{UPH, proc} + \dot{Q}_{WH, pipe} + \dot{Q}_{loss} \quad (4.1.5)$$

where

$$\dot{Q}_{USH, pipe} = \dot{q}_{m, fwd} h_{fwd, i} - \dot{q}_{m, ret} h_{ret, o} - \dot{q}_{m, feedUp} h_{feedUp} \quad (4.1.6)$$

$$\dot{Q}_{UPH, proc} = \dot{q}_{m, fwd} (h_{fwd, o} - h_{ret, i}) \quad (4.1.7)$$

$$\dot{Q}_{WH, pipe} = \dot{q}_{m, feedUp} (h_{ret, i} - h_{feedUp}) \quad (4.1.8)$$

$$\dot{Q}_{loss} = \dot{q}_{m, fwd} (h_{fwd, i} - h_{fwd, o}) + \dot{q}_{m, ret} (h_{ret, i} - h_{ret, o}) \quad (4.1.9)$$

Q_{loss} furthermore can be determined from the heat loss coefficients of the piping (approximation valid for $UA \ll q_{mCp}$):

$$\dot{Q}_{loss} = UA_{fwd} (T_{fwd, i} - T_{env}) + UA_{ret} (T_{ret, i} - T_{env}) \quad (4.1.10)$$

EINSTEIN assumes $UA_{fwd} = UA_{ret} = UA/2$ (same length and degree of insulation in forward and return pipe) for closed and partially closed pipes, and $UA_{fwd} = UA$; $UA_{ret} = 0$ for open pipes (no return).

If pipe length, pipe diameters and insulation thickness are known, EINSTEIN calculates the UA value as

$$UA = \frac{2\pi k}{\ln(d_{ext}/d_{pipe})} l \quad (4.1.11)$$

with $d_{ext} = d_{pipe} + 2\delta_{ins}$

where:

k: thermal conductivity of the insulation material
 d_{pipe} : outer pipe diameter
 d_{pipe} : external diameter (pipe + insulation)
 δ_{ins} : thickness of insulation material
l: pipe length
UA: heat loss coefficient

For k, a constant, temperature independent default value of 0,05 W/mK is supposed.

4.1.4 Data estimates

| Parameter | Value [min , max] | Description |
|--|--|--|
| Pipe length | 100 m [30, 300] | Typical pipe length (one way) used as estimate if no real pipe length is given |
| Central supply temperature $T_{fwd,i}$ | 100 °C [50, 200] | Heating, working fluids without phase change |
| | 0°C [-10, +10] | Cooling, working fluids without phase change |
| | $T_c + 5K$ [$T_c+0,1$, T_c+50] | Heating, working fluids with phase change |
| | $T_c - 5K$ [T_c-50 , $T_c-0,1$] | Cooling, working fluids with phase change |
| Central supply (return temperature) $T_{ret,o}$ | $\min(T_{fwd,i} - 20K, T_c-5)$ [$\min(\sim-50, \sim-10)$, $\min(\sim-5, \sim-0,1)$] | Heating |
| | $\max(T_{fwd,i} + 20K, T_c+5)$ [$\max(\sim+50, \sim+10)$, $\max(\sim+5, \sim+0,1)$] | Cooling |

4.2 EINSTEIN – Heat recovery (HR) Module

4.2.1 Module parameters

Inputs (simulation)

| Parameter | Symbol | Description |
|----------------------|--------------------|---|
| Available waste heat | $Q_{WH}(T,t)$ | Available waste heat from all objects that may act as waste heat sources: equipments, pipes, waste heat from heating processes and cooling demands from cooling processes, waste heat from electrical equipment (WHEE). |
| Required waste heat | $Q_{HX_Req}(T,t)$ | Required waste heat (from all objects that may act as sink for waste heat : equipments, pipes, heating demands from heating processes. |

Outputs (simulation)

| Parameter | Symbol | Description |
|----------------------|---------------------|---|
| Waste heat used | $Q_{WH_Used}(T,t)$ | Waste heat used (from all objects that may act as waste heat sources: equipments, pipes, waste heat from heating processes and cooling demands from cooling processes, waste heat from electrical equipment (WHEE). |
| Waste heat delivered | $Q_{HX}(T,t)$ | Waste heat delivered to all objects that may act as sink for waste heat : equipments, pipes, heating demands from heating processes. |
| Residual waste heat | $Q_{WH_Res}(T,t)$ | Residual waste heat after heat exchanger |

Parameters

The heat exchanger network is defined by:

- (a) the characteristic parameters of each heat exchangers
- (b) the connections of heat exchangers with sources / sinks

Table 1. Characteristic parameters of heat exchangers

| Parameter | Description | Default unit (SI-kWh) |
|---|---|-----------------------|
| UA | Equivalent heat transfer coefficient of a counter-flow heat exchanger. Considered as a constant property of the heat exchanger. Possible variations of the heat transfer coefficient with flow rate, temperature or fluid state (single phase vs. condensation/evaporation) are not taken into account. | kW/K |
| $(mc_p)_s$ | Effective thermal mass of associated heat storage | kWh/K |
| Type | Heat exchanger type | |
| Fluid_h, Fluid_c | Working fluid at hot and cold side | |
| <i>Parameters defining nominal operating conditions:</i> | | |
| P_{nom} | Nominal power of a heat exchanger (at nominal operating conditions (mass flow rates, inlet temperatures)) | kW |
| $\Delta T_{eff, nom}$ | Effective average temperature difference at nominal conditions. $P_{nom} = UA * \Delta T_{eff, nom}$ | K |
| $T_{hi,nom}, T_{ci,nom}$ | Inlet temperatures at hot and cold side defining the nominal working conditions | °C |
| $q_{m,h,nom}, q_{m,c,nom}$ | Nominal mass flow rate at hot and cold side | kg/h |
| <i>Parameters defining average operating conditions in the actual system:</i> | | |
| $Q_{HX,h}$ | Energy flow over the heat exchanger: heat extracted from source (hot side) | kW |
| $Q_{HX,c}$ | Energy flow over the heat exchanger: heat delivered to sink (cold side) | kW |
| ΔQ_s | Variation of heat stored in the associated storage. $Q_{HX,h} = Q_{HX,c} + \Delta Q_s$ | kWh |
| P_{act} | Actual power of a heat exchanger | kW |
| $\Delta T_{eff, act}$ | Effective average temperature difference at actual working conditions. $P_{act} = UA * \Delta T_{eff, act}$ | K |
| $q_{m,h,act}, q_{m,c,act}$ | Nominal mass flow rate at hot and cold side | kg/h |

b) connections

A heat exchanger can have several sources and several sinks.

Limitation V2.0 ff: all sources / all sinks have to be components of the same object

Sources / sinks can be components (e.g. one or several incoming / outgoing streams in the case of processes, ...) of the waste heat / heat demand of different objects: equipments, pipes, processes, electrical equipment.

Data files and data bases

Currently information on heat exchangers in EINSTEIN is hardcoded and cannot be modified by the user.

4.2.2 General issues

The heat recovery module consists of the following functional blocks:

a) calculation of heat exchanger networks (as part of system simulation algorithm)

Two modes that can be selected by the user are available for calculation:

- estimative mode: does **not** consider the actually existing heat exchangers in the system, but a certain fraction of the theoretical maximum of heat recovery
- actual network calculation – Version 2 (ActualHXNet V2)

b) consistency checking of heat exchanger network data

c) design of new / optimised heat exchanger network

4.2.3 Definition of heat exchanger networks

Heat exchanger networks are defined by the connections of heat exchangers to streams and possible “nodes”: splitters or unions (mixing) of streams (see Figure below for the heat source – HS – side; analogous for heat sinks).

Hot streams (heat sources) can be waste heat of processes, pipes, thermal or electrical equipments, and components of cooling demands.

Cold streams can be components of heat demands and waste cooling of the above mentioned objects.

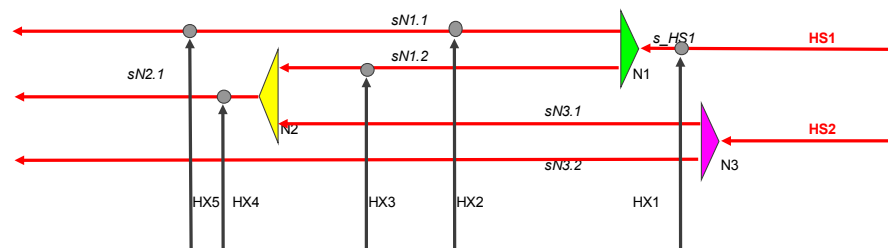


Figure 4. Possible connections of heat exchangers (source side) to hot streams. The cold or sink-side is analogous.

The connections of a heat exchanger (both source and sink side) are defined by:

- origin(s) of the stream at the inlet at source and sink side
- mass fraction of the stream at the origin, that is passed through the heat exchanger (default = 100 %)

All connection possibilities as shown in Figure 4 can be built with the following 3 basic interconnection types:

4.2.3.1 Serial HX connection

Serial connections can be defined on the sink side and/or on the source side.

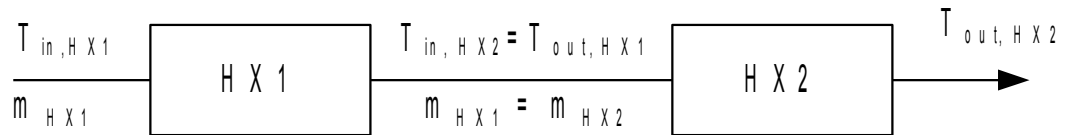


Figure 5. Example of serial connected heat exchangers on sink side: $HX2, inlet = HX1, outlet$

4.2.3.2 Paralell HX connection

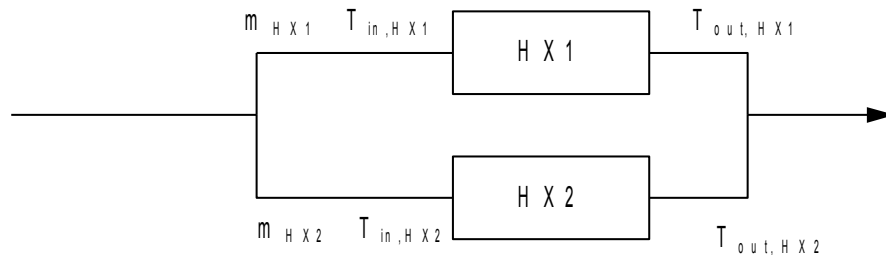


Figure 6. Example of serial connected heat exchangers on sink side: $HX2, inlet = HX1, outlet$

Parallel connections are defined by specifying the same source or sink to the heat exchangers connected in parallel, and specifying the mass flow ratio for each of them.

The mass flow ratios are considered as a constant and do not vary throughout operation (e.g. as a function of some system regulation strategy).

EINSTEIN checks that the sum of all mass flow ratios over parallel heat exchangers is $\leq 100\%$.

4.2.3.3 Mixing of streams

Streams can be mixed by specifying more than one inlet stream for a heat exchanger.

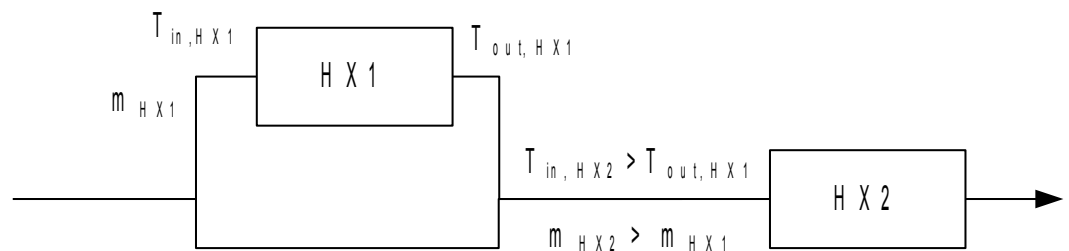


Figure 7. Example of mixing: serial connected heat exchangers with bypass at HX1.

4.2.4 Mathematical description

4.2.4.1 Basic heat exchanger equations

The heat exchanger model used in EINSTEIN are based on the following simplifying assumptions:

- heat exchangers are modeled as counter-flow heat exchangers. no explicit modeling of cross-flow or other heat exchanger geometries is foreseen.
- no thermal losses of the heat exchanger (it is supposed, that insulation of heat exchangers and related piping is good enough so that thermal losses can be neglected)
- no inertia (thermal mass of heat exchangers is considered to be neglectable)
- no explicit consideration of fouling (fouling factors can be considered implicitly by correcting heat exchanger UA values appropriately).
- constant heat transfer coefficient that does neither vary with variation of flow rates nor with variations of temperature

The basic equations describing the heat exchangers are:

a) energy balance at hot and cold side

$$\dot{Q}_{HX} = q_{m,h}(h_{hi} - h_{ho}) = q_{m,c}(h_{co} - h_{ci}) \quad (4.2.1)$$

b) heat transfer equation

$$\dot{Q}_{HX} = UA \Delta T_{eff} \quad (4.2.2)$$

where for situations without phase change (constant c_p at both hot and cold side) ΔT_{eff} is equal to the logarithmic mean temperature difference

$$\Delta T_{eff} = \Delta T_{LM} = \frac{\Delta T_{hi-co} - \Delta T_{ho-ci}}{\ln\left(\frac{\Delta T_{hi-co}}{\Delta T_{ho-ci}}\right)} \quad (4.2.3)$$

for heat exchangers with phase change in one of the streams (hot or cold side) equations (4.2.2) and (4.2.3) subdividing the heat exchanger into three zones (above condensation, condensation, below condensation), with the additional conditions that the outlet temperature of one zone is the inlet temperature of the successive zone, and that the sum of the areas (sum of heat transfer coefficients) of the three zones has to be equal to the total values of the heat exchanger.

$$UA_{bc} + UA_c + UA_{ac} = UA \quad (4.2.4)$$

c) regulation: constraints to heat exchange imposed by maximum / minimum temperatures of sink / source.

Both source and sink heat flows in EINSTEIN may be associated to a constraint with respect to outlet temperatures. This may lead to situations, where heat exchange in unconstrained situation would be higher than the maximum allowed one:

$$\dot{Q}_{HX,max} = \min(q_{m,h}[h_{hi} - h_{ho,min}], q_{m,c}[h_{co,max} - h_{ci}]) \quad (4.2.5)$$

In these cases, the actual heat exchange is constrained to the maximum allowed one and outlet temperatures at both hot and cold side are calculated correspondingly from equation (4.2.1). This in practice can be obtained by a partial bypass either at the hot or at the cold side.

c) definition of average values of parameters for real operation

In real operation heat exchangers may operate under varying conditions (varying flow rates or varying inlet temperatures at either hot or cold side or even both).

Average values in EINSTEIN are defined so that equations (4.2.1) and (4.2.2) hold also for the average values:

- for mass flows, powers and effective temperature difference the simple average is used:

$$\bar{\dot{Q}}_{HX} = \frac{1}{t_{op}} \int_{\dot{Q}_{HX} > 0} \dot{Q}_{HX}(t) dt \quad (4.2.6)$$

$$\bar{q}_m = \frac{1}{t_{op}} \int_{\dot{Q}_{HX} > 0} q_m(t) dt \quad (4.2.7)$$

$$\bar{q}_m = \frac{1}{t_{op}} \int_{\dot{Q}_{HX} > 0} q_m(t) dt \quad (4.2.7)$$

- for temperatures weighted averages by mass flow rate at cold and hot side are used:

$$\bar{T}_{c/h,oli} = \frac{1}{t_{op}} \int_{\dot{Q}_{HX} > 0} T_{c/h,oli}(t) q_{m,c/h}(t) dt \quad (4.2.9)$$

where t_{op} is the operation time .

4.2.4.2 Heat exchangers with associated storage

In the model, storage is always considered at the cold side for simplicity. Nevertheless, this model can also represent real systems with storage at hot side, if a small correction in the storage heat loss coefficient is added considering the higher temperatures in the storage that leads to higher thermal losses.

A 5-node fully stratified model is used for storage modeling, where mixing and/or heat transfer between nodes takes place only by charge/discharge but not due to internal conduction or convection heat transfer.

Regulation of this sub-system is considered by 5 operation modes:

- **charge:** the storage is charged by the hot stream. no heat is transferred to the cold stream.
- **discharge:** the storage is discharged, transferring heat to the cold stream. no heat is extracted from the hot stream.
- **direct:** heat is transferred directly from hot to cold stream. no charge or discharge of storage takes place.

- **direct and charge:** heat is transferred directly from hot to cold stream. excess heat is stored.
- **direct and discharge:** heat is transferred directly from hot to cold stream. excess demand is covered by discharge of the storage.

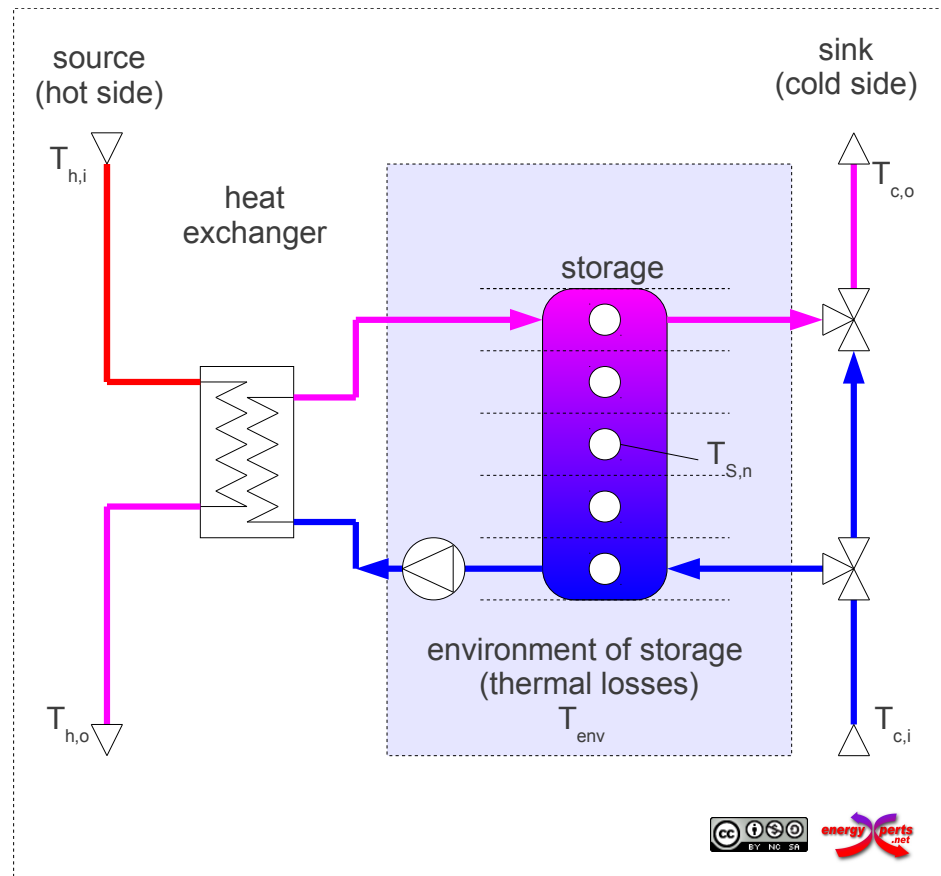


Figure 8. Model of heat exchanger and associated storage with 5-node model for storage.

Charge and discharge modes are always activated when only one of the streams (either hot or cold) is available and temperature levels are sufficient for adding / extracting heat to the storage.

In case that there is simultaneous availability of source and sink streams direct-and-discharge or direct-and-charge modes are activated depending on whether there is excess availability or excess demand. Direct mode is used in this situation when either no storage is available or temperature levels in the storage are so that no heat can be added or extracted from the storage.

For systems with storage at the cold side, equation (4.2.1) splits up into two equations.

For the hot side the heat extracted from the hot stream is identical to the heat transferred across the heat exchanger:

$$\dot{Q}_h =: q_{m,h}(h_{hi} - h_{ho}) = \dot{Q}_{HX} \quad (4.2.10)$$

For the cold side the heat delivered to the cold stream can be calculated as:

$$\dot{Q}_c =: q_{m,c}(h_{co} - h_{ci}) \quad (4.2.11)$$

The energy balance of the storage then can be written as

$$\dot{Q}_{storage} = \dot{Q}_h - \dot{Q}_c - \dot{Q}_{loss} \quad (4.2.12)$$

where:

$\dot{Q}_{storage}$: variation of energy stored; (+) for net charge, (-) for net discharge.
 \dot{Q}_{loss} : thermal losses of storage

4.2.4.3 *Mixing of streams*

(work in progress. not yet active)

Several sinks or several sources can be connected to one heat exchanger by mixing of the streams. In this case the mixing enthalpy and the combined mass flow of the streams are obtained as

$$q_{m,mix} = \sum_i q_{m,i} \quad (4.2.13)$$

$$h_{mix} = \frac{\sum_i q_{m,i} h_i}{q_{m,mix}} \quad (4.2.14)$$

4.2.5 **Consistency checking of heat exchangers**

Consistency checking of heat exchangers carries out the following checks:

- Equation (4.2.2) for both nominal and average actual conditions
- Calculation of average power by yearly energy transferred

$$Q_{HX} = \dot{Q}_{HX} t_{op} \quad (4.2.15)$$

- Yearly energy balance (integral of equation (4.2.12))
- Second law constraints:

$$T_{hi} > T_{co} \quad (4.2.16a)$$

$$T_{ho} > T_{ci} \quad (4.2.16b)$$

$$T_{hi} > T_{ho} \quad (4.2.16c)$$

$$T_{ci} > T_{co} \quad (4.2.16d)$$

Currently (Version 2.2) no consistency checking with nominal flow rates and temperature levels of connected of connected streams is carried out at the stage of consistency check. Possible inconsistencies will be detected only by calibration of system simulation.

4.2.6 Design assistant for heat exchanger networks

The goal of auto-design is to automatically generate a close to optimum heat exchanger network based on pinch analysis.

The heat exchanger network design assistant disposes of three modes:

- **Flow rate matching:** this mode is based on the algorithm used in previous versions of EINSTEIN (since V2.2), resulting in most cases in a simple heat exchanger network, but with a result further away from the optimum of energy recovery.
- **Exergy optimisation (EINSTEINplus):** in this mode, a heat exchanger network is proposed which maximises energy recovery but at the same time minimising the requirement of exergy for residual heating / cooling.

Within the constraint of maximising the total amount of heat recovery as specified by the result of pinch analysis, waste heat is taken at the minimum possible temperature level in order to provide heat at the maximum possible temperature level.

- **Minimum size (EINSTEINplus):** in this mode energy recovery is maximised while using a heat exchanger with minimum size (and, therefore, cost).

Within the constraint of maximising the total amount of heat recovery as specified by the result of pinch analysis, waste heat is taken at the maximum possible temperature level in order to provide heat at the minimum possible temperature level.

The following general criteria for HX network design are hereby taken into account:

- Maximisation of total heat recovered
- Use of heat that has to be actively cooled down leads to double savings both in external cooling in external heating energy

Other aspects have to be manually checked by the user, such as:

- Heat integration within the same process should be given priority - direct use of waste heat
- Distance between the heat source (hot stream) and heat sink (cold stream)
- Practical issues, such as fouling factors, necessity of indirect heat exchange via heat transfer media, temperature and pressure aspects etc.
- Investment costs and economic performance

4.2.6.1 Flow rate matching

The EINSTEIN auto-design algorithm mode "*FLOWRATE*" is derived from an algorithm of Kemp [2007] described in Figure 9. Hereby there is the option to maintain already existing heat exchangers, or – alternatively – to design the full network from scratch.

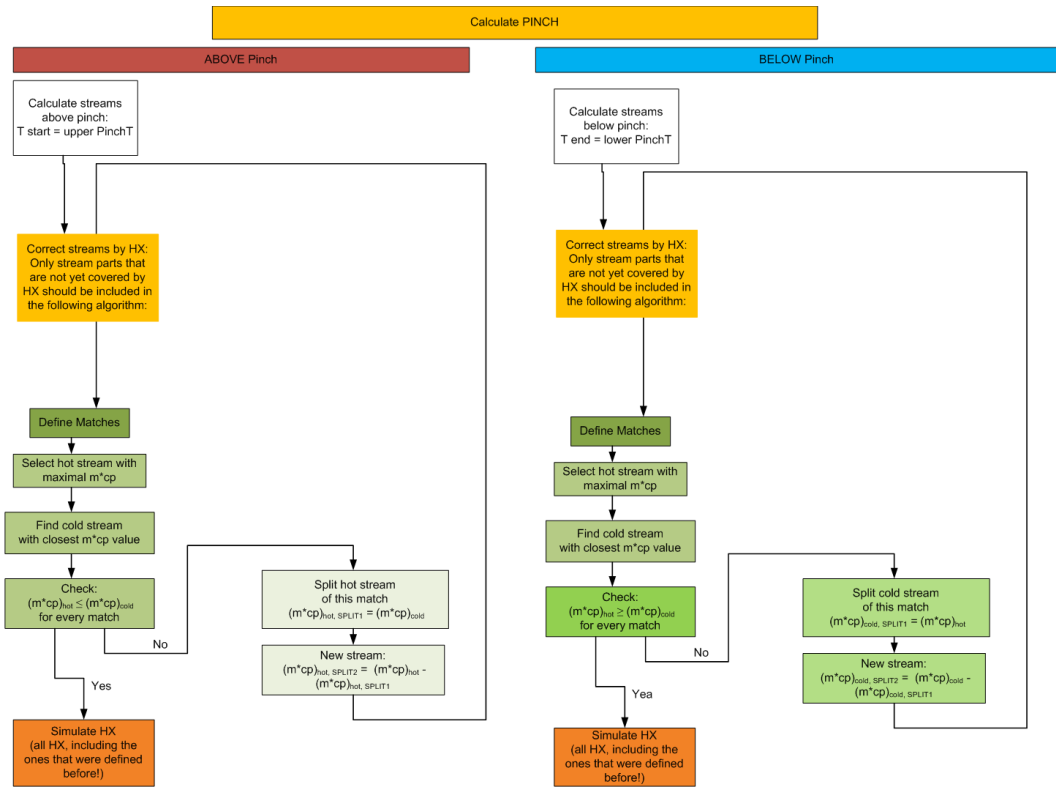


Figure 9. Scheme of the algorithm for heat exchanger network auto-design.

References:

Kemp, I.C. [2007]: *Pinch Analysis and Process Integration*. Elsevier, Amsterdam, 2007.

4.2.6.2 Exergy optimisation and minimum size (EINSTEIN plus)

The EINSTEIN auto-design algorithm modes “EXERGY” and “MINSIZE” are based on the following algorithm:

- first pinch analysis is carried out and the composite curves (cold: CCC, hot: HCC and grand: GCC) are determined together with the pinch temperature and the maximum possible heat exchange $Q_{hx, max}$
- then a fraction of both the cold and the hot streams is selected building new composite curves, representing the *used* part of the cold streams, CCCu, and of the hot streams, HCCu, so that the total amounts of energy are identical and equal to $Q_{HX, max}$,

$$Q_{HCCu}(\infty) = Q_{CCCu}(\infty) = Q_{hx, max} \quad (4.2.17)$$

and under the constraint that maximum heat exchange should remain possible:

$$T_{HCCu}(Q) \geq T_{CCCu}(Q) + \Delta T_{min} \quad \forall Q \text{ where } 0 \leq Q \leq Q_{hx, max} \quad (4.2.18)$$

The selection of the appropriate fraction is different for the two modes, following the criteria of either minimum exergy loss (EXERGY mode) or maximum

temperature difference (equivalent to minimum heat exchanger area, MINSIZE mode):

- i. **EXERGY mode – minimum exergy loss.** For heat exchange above pinch temperature heat is supplied to the heat demands at maximum possible temperature. For heat exchange below pinch, waste heat is used from streams with lowest possible temperature.
- ii. **MINSIZE mode – minimum heat exchanger area.** For heat exchange above pinch temperature heat is supplied to the heat demands at minimum possible temperature. For heat exchange below pinch, waste heat is used from streams with highest possible temperature.

The procedure for the two modes and the resulting used parts of the composite curves, CCCu and HCCu, are illustrated in Figure 10.

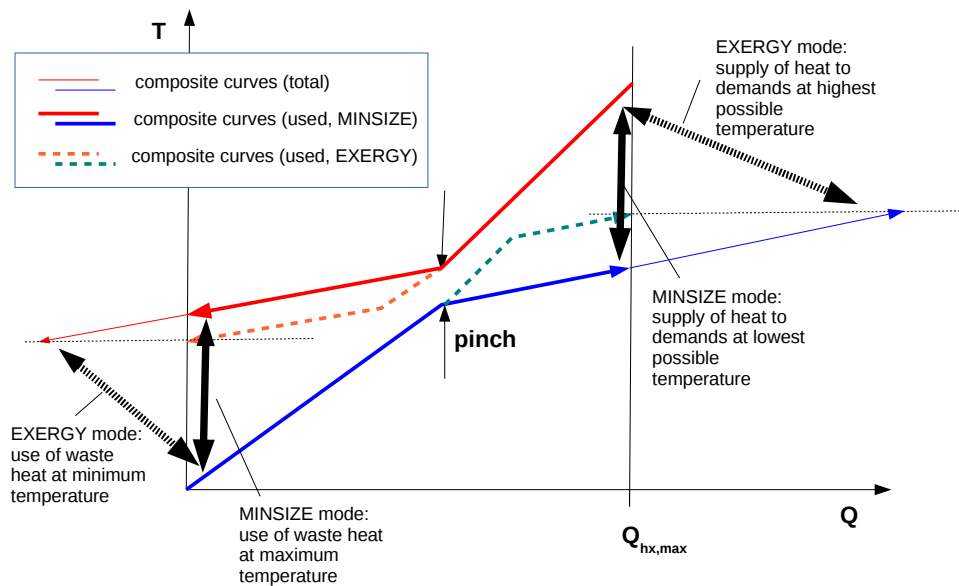


Figure 10. Scheme illustrating the determination of the used part of the hot and cold composite curves: EXERGY and MINSIZE modes.

- once defined the curves CCCu and HCCu, a subset of cold streams above pinch and for hot streams below pinch is selected (splitting or dividing streams where necessary) that in its sum builds up exactly the respective parts of these curves. In order to minimize the number of streams, priority is given to streams with high heat transport coefficients mc_p .

The hot streams above pinch and the cold streams below pinch are fully used, so that no subset has to be selected in these sections of the curves.

- then pairs of streams are formed trying to match streams with equal or similar amounts of energy, in order to minimise the need for splitting and dividing streams.
- finally those pairs of streams are merged into one pair, where both on hot and cold side there is a direct serial connection.
- for each of the resulting pairs of streams an appropriate size of the heat exchanger (UA-value and, where necessary, size of heat storage) is defined.

Note: in practice, in some cases the resulting heat transfer of the so defined heat exchanger network is smaller as the ideal value $Q_{hx,max}$ given by pinch analysis. This may be due to some of the following facts:

- **non-simultaneity of streams and storage losses.** In the above algorithm possible shifts in time between different streams are taken into account only ex-post, i.e. when defining an appropriate size of heat storage to each heat exchanger.
- streams or splits of **streams with very low amounts of energy are neglected** in order to limit the number of resulting heat exchanger.

4.3 EINSTEIN – CHP (cogeneration) Module

4.3.1 Module parameters

Inputs (system simulation)

| Parameter | Symbol | Description |
|----------------------|---------------|--|
| Required heat demand | USHj,Req(T,t) | Time and temperature dependent heat demand |

Outputs (system simulation)

| Parameter | Symbol | Description |
|-----------------------|--------------|---|
| Generated useful heat | USHj (T,t) | Useful heat delivered to connected pipes (including temperature distribution) |
| Generated electricity | FETel,j(t) | Useful electricity generated |
| Generated waste heat | QWHj(T, t) | Useful waste heat generated (including temperature distribution) |
| Fuel consumption | FETfuel,j(t) | Fuel consumption (power equivalent based on LCV) |

Parameters

| Parameter (user interface) | Symbol | Description |
|---|---------------------------|--|
| Nominal power (heating, output) | Pnom | Nominal thermal power output of the CHP equipment corresponding to full load (in current version considered as constant independent of working temperatures) |
| Nominal power (electrical) output | Pel | Nominal electrical power output of the CHP equipment corresponding to full load (in current version considered as constant independent of working temperatures) |
| Nominal fuel consumption | Pfuel | Power consumed (calculated based on LCV of fuel used) |
| Temperature at which heating is supplied | T_supply | Average temperature (during operating hours) in the real system at which heating is supplied (outlet of equipment). |
| Maximum temperature at which heat can be supplied | TmaxSupply, (TmaxSupply2) | Maximum temperature at which heat can be supplied (for engines, two temperature levels are given for waste water (TmaxSupply) and for exhaust gas (TmaxSupply2). |
| Thermal efficiency | η_{th} | Ratio between thermal output and fuel consumption (Pnom/Pfuel) |
| Electrical efficiency | η_{el} | Ratio between generated electricity and fuel consumption (Pel/fuel) |
| CHP Type | CHPTYPE | The following types are currently supported: engine gas turbine steam turbine fuel cell |

Data files and data bases

The EINSTEIN release contains a default database with CHP equipment that can be extended by the user. Data have been elaborated based on a data set of market available equipment [COGEN Europe, 2008].

4.3.2 Mathematical description

4.3.2.1 Definitions of electrical and thermal efficiency

The nominal thermal and electrical power of the equipment are given by:

$$\dot{Q}(t) = \eta_{th} \dot{E}_{fuel} \quad (4.1.1)$$

$$\dot{E}_{el}(t) = \eta_{el} \dot{E}_{fuel} \quad (4.1.2)$$

where:

- $\dot{Q}(t)$: the instantaneous thermal power
- $\dot{E}_{el}(t)$: electrical power
- \dot{E}_{fuel} : fuel input (power, based on LCV)
- η_{th} , η_{el} : thermal and electrical efficiency

Both thermal and electrical efficiency in current version (V2.2) are considered as constant, independent of part load ratio.

4.3.2.2 Temperature level of available heat

It is supposed that heat from CHP is available in one or two circuits of heat carriers at different temperature levels:

$$\dot{Q}^{nom}(t) = f_1 \dot{q}_{m,1} c_{p,1} (T_{fwd,1} - T_{ret,1}) + f_2 \dot{q}_{m,2} c_{p,2} (T_{fwd,2} - T_{ret,2}) \quad (4.1.3)$$

The default parameters for forward and return temperatures and the distribution of total thermal power on the first and second circuit are listed in Table 2.

Table 2 Temperature levels of generated heat for different types of CHP equipment

| CHP Type | fraction of total generated heat | fluid Tfwd/Tret [°C] | fluid / description |
|---------------|----------------------------------|--|---------------------|
| engine | 70% | TmaxSupply/TmaxSupply - 20 (default: 95 / 75) | exhaust gas |
| | 30% | TmaxSupply / 0 (*) (default: 200 / 0) | cooling water |
| gas turbine | 100% | TmaxSupply / 0 (*) (default: 250 / 0) | exhaust gas |
| steam turbine | 100% | TmaxSupply/TmaxSupply (default: 140 / 140) | expanded steam |
| fuel cell | 100% | TmaxSupply / 0 (*) (default: 250 / 0) | cooling air |

(*) reference temperature for exhaust gas: 0 °C

4.3.2.3 Control strategy: Matching of available heat and heat demand

A continuous regulation of the CHP equipment between partload ratios of 20% and 100% is supposed in order to obtain an optimum matching between the available heat and the (temperature dependent) heat demand. By matching heat availability for a given part load ratio f_p with heat demand, the utilisation ratio for the available heat is obtained:

$$\dot{Q}_u(t) = f_u f_p \dot{Q}^{nom}(t) \quad (4.1.4)$$

where:

- $\dot{Q}_u(t)$: useful heat supplied to demand after optimum matching
- f_p : part load ratio
- f_u : fraction of total waste heat generated that is used for covering heat demand

Matching of temperature levels of availability and demand is illustrated for an example in Figure xxx: Waste heat of a CHP engine is available mainly in the temperature range of cooling water (about 75 – 95 °C) and only in minor degree at higher temperatures in the exhaust gas; this may be in contrast with the heat demand (in the example air preheating from 15 to 150 °C) so that due to the different temperature distribution not all of the demand can be covered, and not all of the generated heat can be used, although the totals of availability are identical (in the example: 100 kW). Similar to pinch analysis, matching is obtained by shifting the supply curve down along the power-axis until for any power level the supply curve is always at the right (higher temperature level) than the demand curve. In this case, the maximum power that can be transferred to the demand is 75 kW or 75 % of the total heat generated, leading to $f_u = 0,75$.

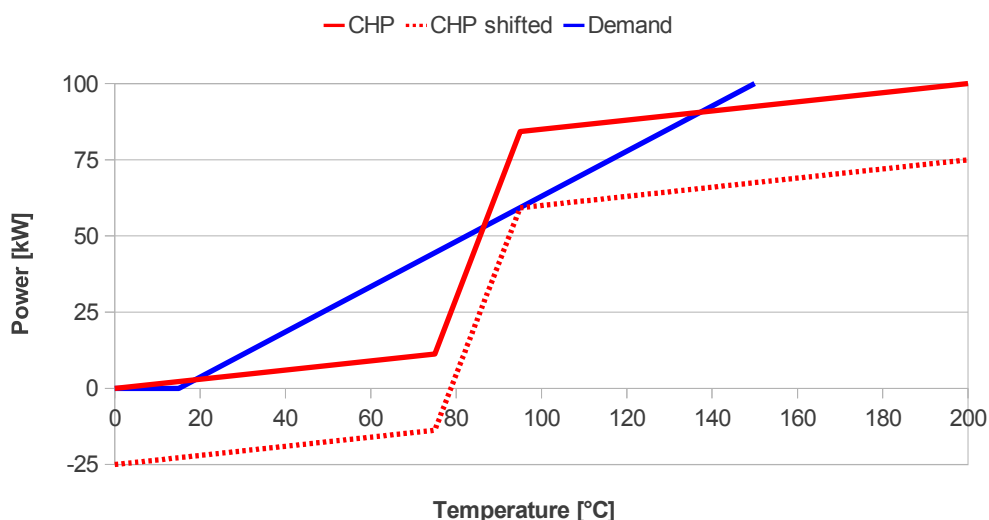


Figure 11. Example for matching of temperature levels of generated heat and heat demand

Based on f_u the effective electrical efficiency (see Audit Guide, section 3.7) is calculated:

$$\eta_{el,eff} = \frac{\dot{E}_{el}(t)}{\dot{E}_{fuel}(t) - \frac{\dot{Q}_u(t)}{\eta_{ref}}} = \frac{\dot{E}_{el}^{nom}(t)}{\dot{E}_{fuel}^{nom}(t) - f_u \frac{\dot{Q}^{nom}(t)}{\eta_{ref}}} \quad (4.1.5)$$

where

η_{ref} : reference efficiency for generation of heat (default: 0,9 as typical value for conventional boilers)

The part load ratio is adjusted so that a maximum electricity generation is obtained under the constraint that a minimum effective electrical efficiency is obtained.

$$\eta_{el,eff} = \eta_{el,eff}^{min} \quad (4.1.6)$$

If under the given demand this is not possible for a part load ratio > 20%, the equipment is switched off.

The unused part of the heat generated in the equipment is considered as waste heat, that is available for a potential external use in the heat recovery system:

$$\dot{Q}_{WH}(t) = (1 - f_u) f_p \dot{Q}^{nom}(t) \quad (4.1.7)$$

4.3.2.4 Heat storage

For all CHP equipments, implicitly a fully stratified ideal storage (zero heat losses) with a maximum storage capacity corresponding to the heat generated at full load operation and storage period τ is supposed (default: $\tau = 4$ hours).

The variation of the energy stored is given by the difference of the useful heat generated by the CHP equipment and the heat delivered to the demand (USH).

$$\dot{Q}_{storage}(t) = \dot{Q}_u(t) - \dot{Q}_{USH}(t) \quad (4.1.8)$$

4.3.3 CHP auto-design

The design assistant for CHP systems carries out a series of design choices in order to propose a CHP system:

- selection of CHP type
- dimensioning of CHP system and selection of a specific equipment from the equipment data base

The design choices are based on a modified temperature and time dependent demand curve, that takes into account the possibility of heat storage.

4.3.3.1 Smoothed time- and temperature dependent demand curve

In order to take into account the possibility of heat storage, not instantaneous values of (temperature dependent) thermal power requirements are used as a base for design, but a smoothed version of this curve after applying a low-pass filter with a time constant τ of a typical storage (default value used: 2 hours).

$$\dot{Q}_{smooth}(T, t) = \frac{1}{\tau} \int_0^t \dot{Q}_{USH}(T, t') e^{-\frac{t-t'}{\tau}} dt' \quad (4.1.9)$$

After smoothing, a monotonously descending curve $\dot{Q}_{smooth}^{mono}(T, t)$ is constructed ordering the different time intervals so that the intervals with the highest power requirement come first

4.3.3.2 Selection of system type

The algorithm of CHP type selection consists in the following steps:

a) for each of the available CHP types, a design temperature (indicating the maximum temperature typically to be supplied) is given as default.

Table 3 Default design temperature for different types of CHP equipment

| CHP Type | Tdesign [°C] |
|---------------|--------------|
| engine | 95 |
| gas turbine | 250 |
| steam turbine | 140 |
| fuel cell | 250 |

As a function of the minimum number of operating hours, a maximum power can be determined as

$$\dot{Q}_{max} = \dot{Q}_{smooth}^{mono}(T_{design}, t_{op, min}) \quad (4.1.10)$$

The maximum power for each CHP type is then corrected by a priority ratio (default value for each CHP type) that – at equal possible maximum power – gives priority to certain types of CHP equipment in order to take into account not only maximum possible installed thermal power, but also aspects of efficiency and cost. From this a CHP type classification parameter Z is obtained for each CHP type:

$$Z = f_{priority} \dot{Q}_{max} \quad (4.1.11)$$

And finally, the type of equipment with the highest Z-value is selected.

The user can preconfigure the design assistant by manually pre-selecting the CHP type, or by selecting some specific fuel type, which implicitly may exclude certain types of equipment.

4.3.3.3 Selection of system dimensions and specific equipment model

The design value for thermal system power is the maximum possible power as \dot{Q}_{max} of the selected CHP type.

Based on this parameter, EINSTEIN carries out a look-up in the CHP database for appropriate equipment in the power range [$0,7\dot{Q}_{max} : 1,4\dot{Q}_{max}$]

Then, the CHP model with the maximum value for the classification parameter z is selected which takes into account:

- the global (primary energy) energetic efficiency $\eta_{global} = 2.5 * \eta_{el} + \eta_{th}$ ⁵
- the absolute deviation from the required power level
- the signed deviation from the required power level (priority for smaller equipment, which usually means better economic performance)

$$z_i = \eta_{global} - 0,25 \left| \frac{\dot{Q}_i^{nom} - \dot{Q}_{max}}{\dot{Q}_{max}} \right| - 0,125 \frac{\dot{Q}_i^{nom} - \dot{Q}_{max}}{\dot{Q}_{max}} \quad (4.1.12)$$

5 The factor 2,5 takes into account a typical primary energy conversion factor for electricity

where:

i : index of the equipment model

\dot{Q}_i^{nom} : nominal thermal power of the specific equipment model I

References:

COGEN Europe [2008]: Database of market available CHP equipment, delivered by COGEN Europe in the Framework of the IEE project EINSTEIN-I.

4.4 EINSTEIN – HTP (heat to power) Module

4.4.1 Module parameters

Inputs (system simulation)

| Parameter | Symbol | Description |
|-------------------------------------|-------------|--|
| Waste heat supplied as driving heat | QHXj (T, t) | |
| Master control | f_ctrl(t) | Control parameter with $0 \leq f \leq 1$ |
| Ambient temperature (dry bulb) | T_amb,db(t) | Dry bulb temperature for dissipation of waste heat |

Outputs (system simulation)

| Parameter | Symbol | Description |
|-----------------------|-----------------|--|
| Generated electricity | FETel,j(t) | Useful electricity generated |
| Generated waste heat | QWHj(T, t) | Useful waste heat generated (including temperature distribution) |
| Heat consumption | FETheat,j(T, t) | Heat consumption (driving heat) |
| Required waste heat | QHXj, req(T, t) | Required waste heat for full load operation |

Parameters

| Parameter (user interface) | Symbol | Description |
|---|-------------|---|
| Nominal power (electrical) output | Pel, nom | Nominal electrical power output of the HTP equipment corresponding to full load (in current version considered as constant independent of working temperatures) |
| Nominal driving heat consumption | Q_drive,nom | Power consumed |
| Nominal power (heating, output) | Q_nom | Nominal thermal power output (heat dissipation) of the HTP equipment corresponding to full load |
| Temperature at which driving heat is supplied | T_drive | Inlet temperature of heat supply medium (constant) |
| Nominal air temperature | T_amb, nom | Temperature of ambient air defining nominal conditions (temperature at which the nominal efficiency is defined) |
| Electrical efficiency | η_{el} | Ratio between generated electricity and heat input (Pel/Pheat) |
| HTP Type | HTPTYPE | The following types are currently supported: rankine cycle (RC) trilateral flash cycle (TFC) |

Data files and data bases

The EINSTEIN release does not provide a default database with HTP equipment.

4.4.2 Mathematical description

For simulations within EINSTEIN HTP equipment is treated as a *black box* with an input for the driving hot stream (steam, hot water, other) and the stream for dissipation of the residual heat (cooling water or air), and the generated electricity as an output.

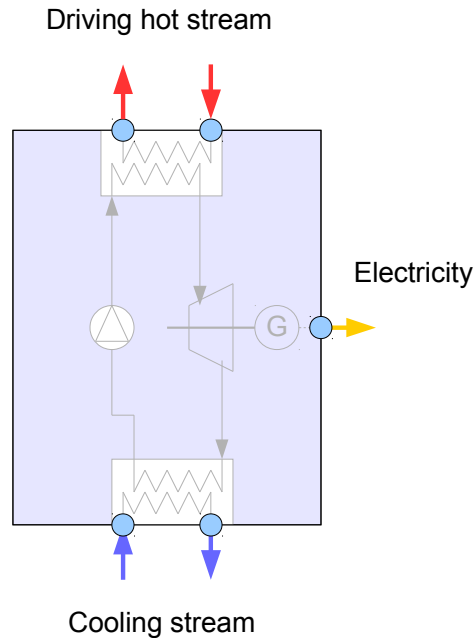


Figure 1. Black box model for waste heat to power generating units (Rankine cycle in the inner of the box only for illustrative purposes; can be substituted by any other cycle).

A specific equipment then is defined by:

- nominal operation conditions
- performance at nominal operation conditions
- part load behaviour
- performance at off-nominal operation temperatures

4.4.2.1 Nominal operation conditions

The nominal operation conditions are defined by the medium (e.g. hot air, hot exhaust gas, hot water, steam, ...), flow rate and inlet temperature of both hot driving stream and cooling stream.

Hot side (driving heat source):

The inlet temperature of the heat source is considered as constant (user input) and the temperature drop is assumed depending on the type of HTP equipment:

a) Rankine cycle

For Rankine cycle most of the heat is delivered as latent heat, and therefore a small temperature drop in the medium delivering the driving heat is supposed.

The temperature drop is assumed as fixed: $\Delta T = 5 \text{ K}$

b) Trilateral flash cycle (TFC)

Trilateral flash cycle is characterised by a high temperature drop in the driving medium. In this case the return temperature of the driving medium is assumed to be constant:

$$T_{ret} = 25 \text{ }^\circ\text{C}$$

Cold side (heat dissipation)

It is assumed that heat is dissipated to ambient air. The return temperature of the medium within the power cycle is assumed to be above ambient by a fixed value of $\Delta T = 10 \text{ K}$. Therefore, the dissipation temperature varies with time following the ambient (dry bulb) temperature:

$$T_{ret} = T_{amb,db} + \Delta T \quad (4.1.1)$$

4.4.2.2 Performance at nominal operation conditions

The performance at nominal operation conditions and steady state is defined by the (net) electrical power generated (output), the power input (heat), and the nominal heat output (dissipated and/or delivered to some low temperature heat demand).

The thermal and electrical efficiency of the equipment are defined by:

$$\dot{Q}(t) = \eta_{th} \dot{Q}_{drive} \quad (4.1.1)$$

$$\dot{E}_{el}(t) = \eta_{el} \dot{Q}_{drive} \quad (4.1.2)$$

where:

$\dot{Q}(t)$: the instantaneous thermal power (output at low temperature)

$\dot{E}_{el}(t)$: (net) electrical power generated

\dot{Q}_{drive} : power of driving heat (input at high temperature)

η_{th} , η_{el} : thermal and electrical efficiency

Thermal and electrical efficiency are function of both part load ratio and temperature levels of driving heat and heat dissipation (ambient).

For simplicity losses are neglected, so that the following relationship holds:

$$\eta_{th} = 1 - \eta_{el} \quad (4.1.3)$$

4.4.2.3 Part load correction

The part load behaviour is described by the conversion efficiency as a function of the part load ratio

$$\eta_{el} = \eta_{el}(f_{partload}) \quad (4.1.1)$$

where the part load ratio is defined by the ratio of actual and nominal electric power

$$\dot{E}_{el} = f_{partload} \dot{E}_{el, nom} \quad (4.1.1)$$

Specific function for part load correction:

The following specific function for part load correction has been implemented in the EINSTEIN HTP module, as it describes well the behaviour of some ORC turbines [ENOGIA 2016]: the part load behaviour is defined implicitly by the linear equation

$$f_{th} =: \frac{\dot{Q}_{in}}{\dot{Q}_{in, full load}} = c + (1-c) f_{partload} \quad (4.1.1)$$

which leads to the following function for the part load correction

$$\eta_{el}(f_{partload}) = \eta_{el, fullload} \frac{f_{partload}}{c + (1-c) f_{partload}} \quad (4.1.1)$$

The coefficient c is a user input. If missing the default value $c = 0.2195$ is assumed.

4.4.2.4 Temperature levels of driving heat and heat dissipation

In the current version (V2.4) the temperature level of the driving heat (both forward and return) is considered as constant (fixed parameter), whereas the temperature of dissipation may vary with the ambient temperature.

Extrapolation from nominal to real working conditions (compression chillers)

For simplicity, the power at full load of the driving heat (input) is considered as constant, whereas the electrical power output is variable depending on the variation in η_{el} with temperature.

The (time dependent) electrical efficiency at real operation conditions and at full power is obtained as:

$$\eta_{el, real, fullLoad}(t) = \eta_{ex, nom} \eta_{carnot}(T_{c, real}(t), T_{drive, real}(t))$$

where:

$\eta_{el, real, fullLoad}(t)$: real energy efficiency at full load considering the real temperatures for driving heat and heat rejection.

$\eta_{ex, nom}$: nominal exergy efficiency

$\eta_{carnot}(T_{c, real}(t), T_{drive, real}(t))$: carnot energy efficiency ratio of the power cycle calculated using the inlet temperature of the driving heat and the heat rejection temperature.

It is assumed that the exergetic efficiency of the equipment remains constant for different working conditions:

$$\eta_{ex, nom} = \eta_{ex, real}$$

The exergetic efficiency is obtained from the electric efficiency at nominal conditions (user input):

$$\eta_{ex,nom} = \frac{\eta_{el,nom}}{\eta_{carnot}(T_{c,nom}, T_{drive,nom})}$$

where

$\eta_{el,nom}$: energy efficiency at full load at nominal conditions.

The electricity consumed for fans or cooling towers is considered included in the (net) electrical efficiency.

The carnot efficiency η_{carnot} is given by:

$$\eta_{carnot}(T_c, T_{drive}) = \frac{(T_{drive} - T_c)}{T_{drive} [K]}$$

where T_{drive} [K] is the absolute temperature in K (T_{drive} in [°C] + 273.15).

If the exergetic efficiency η_{ex} cannot be calculated from user specified data, $\eta_{ex} = 0,5$ is assumed.

4.4.2.5 Control strategies

The desired heat supply is calculated from the nominal power of the driving heat, the thermal part load ratio f_{th} , and the temperature levels:

$$\dot{Q}_{heat, req}(T) = f_{th} \dot{Q}_{heat}^{nom} \left(1 + \frac{T - T_{drive}}{\Delta T_{drive}}\right) \text{ for } T_{drive} - \Delta T_{drive} \leq T \leq T_{drive} \quad (4.1.4)$$

where:

- $\dot{Q}_{heat, req}(T)$: power requirement of the driving heat as a function of temperature
- T_{drive} : required temperature level (inlet temperature of driving heat supply medium)
- ΔT_{drive} : (maximum) temperature drop of driving heat supply medium

a) master control

If the equipment is connected to a heat supply equipment via a distribution pipe, then the equipment is fully regulated by the master control:

$$f_{th}(t) = f_{ctrl}(t)$$

b) control by availability of waste heat

For HTP equipment driven exclusively by heat recovery and not connected to any additional active driving heat supply equipment / distribution pipe, the equipment is fully controlled by the amount of available waste heat.

The part load ratio is adjusted in iterative calculations so that:

$$\dot{Q}_{heat, req}(T_{drive}) = \dot{Q}_{HX}$$

where \dot{Q}_{HX} is the amount of heat exchanged via the heat exchanger.

The required temperature levels are set as fixed parameters without adaptation of the desired level of driving temperature to the waste heat source.

References:

ENOGIA [2016]: Data of ORC turbines from ENOGIA (private communication)

4.5 EINSTEIN – Solar Thermal Module

4.5.1 Module parameters

Inputs (system simulation)

| Parameter | Symbol | Description |
|----------------------|----------------------------|--|
| Required heat demand | USH _{j,Req} (T,t) | Time and temperature dependent heat demand |

Outputs (system simulation)

| Parameter | Symbol | Description |
|-------------------------|------------------------|---|
| Generated useful heat | USH _j (T,t) | Useful heat delivered to connected pipes (including temperature distribution) |
| Electricity consumption | FETel,j(t) | Electricity consumption for auxiliary devices (circulation pumps in solar loop, etc.) |

Parameters

| Parameter (user interface) | Symbol | Description |
|---------------------------------|----------------------|---|
| Nominal power (heating, output) | P _{nom} | Nominal thermal power The default convention $P_{nom} = 0,7 A_{ap}$ (effective - usually aperture – area) is used. |
| Nominal power (electrical) | P _{el} | Nominal electrical power consumption |
| System efficiency | h _{sys} | System efficiency defined as ratio between useful heat delivered to load and useful solar heat at collector output. Typically close to 1. |
| Storage volume | V _{storage} | Maximum temperature at which heat can be supplied (for engines, two temperature levels are given for waste water (T _{maxSupply}) and for exhaust gas (T _{maxSupply2}). |
| Inclination | η _{th} | Ratio between thermal output and fuel consumption (P _{nom} /P _{fuel}) |
| Azimuth | η _{el} | Ratio between generated electricity and fuel consumption (P _{el} /fuel) |

Data files and data bases

The EINSTEIN release contains a default database with CHP equipment that can be extended by the user.

For data files / data input on solar radiation data an ambient temperature see the section on weather data.

4.5.2 Mathematical description

4.5.2.1 General overview

For solar fields in EINSTEIN a quasi-steady state approach is used, this means that the output of the solar field is fully determined by the instantaneous values of solar radiation, working and ambient temperature.

Inertia is exclusively considered in the modeling of the solar buffer storage tank forming part of the solar thermal subsystem.

The modeling includes explicitly only the solar collectors and the storage tank (including thermal losses). Losses of solar field piping are considered implicitly correcting the useful solar heat gains by a fix user-defined coefficient, the solar system efficiency.

Solar radiation is considered as composition of direct or beam radiation and homogeneously distributed diffuse radiation.

4.5.2.2 Collector efficiency equation

The output of the solar collectors is calculated based on the collector efficiency equation:

$$\eta = c_0 - (c_1 + c_2 \Delta T) * \frac{\Delta T}{G_T} \quad (4.5.1)$$

where c_0 is the optical efficiency, c_1 , c_2 are the linear and quadratic heat loss coefficients (c_1 [W/K m²]; c_2 [W/K²m²]), ΔT [K] is the difference between the average temperature of the solar heat carrier and the atmospheric temperature and G_T [W/m²] is the amount of incident total solar radiation on the solar collector.

For concentrating collectors, G_T in equation (4.5.1) is replaced by the incident beam radiation G_{bT} .

The temperature difference ΔT is obtained as the difference between the ambient temperature, the current temperature of the solar storage, and the temperature difference $\Delta T_{primary}$ due to (a) the heat exchanger of the solar primary circuit and (b) half the temperature lift between collector inlet and outlet temperature ($\Delta T_{primary}$ is assumed as constant with a fix default value of 7 K):

$$\Delta T = T_{storage} + \Delta T_{primary} - T_{amb} \quad (4.5.2)$$

4.5.2.3 Incidence angle modifier

For non-normal incidence angles of solar radiation on the collector, the optical efficiency c_0 has to be corrected by the incidence angle modifier, leading – for the special case of biaxial incidence angle modifiers - to the equation for the optical efficiency:

$$\eta_o = c_0 K(\theta_L, \theta_T) = c_0 K^L(\theta_L) K^T(\theta_T) \quad (4.5.3)$$

The incidence angle modifier functions K_L and K_T in EINSTEIN are approximated by the following function, which is defined by a single parameter K_{50} giving the value of the function for an incidence angle of 50°:

$$K(\theta) = e^{-b(\tan \theta)^2} \quad (4.5.4a)$$

with $b = -\ln(K_{50}) / (\tan 50^\circ)^2$

For the special case $K_{50} > 1$ ($b < 0$), for angles $t > 50^\circ$ equation (4.5.4) is replaced by

$$K(\theta) = K_{50} e^{1 - \frac{\tan \theta}{\tan 50^\circ}} \quad (4.5.4b)$$

For Fresnel collectors, for the transverse incidence angle modifier function K_T instead of equations (4.5.4a) and (4.5.4b) the following correlation is used which has been obtained approximating the measured incidence angle modifier of a commercially available Fresnel collector:

$$K_{Fresnel}^T(\theta_T) = 1 + b_1 \theta_T + b_2 \theta_T^2 \quad (4.5.4c)$$

where $b_1 = \frac{1}{2} \left(\frac{K_{50}^T - 1}{50^\circ} \right)$ and $b_2 = \frac{1}{2} \left(\frac{K_{50}^T - 1}{50^\circ} \right)^2$.

4.5.2.4 Energy balances, storage charge and discharge

The maximum possible solar field output at a given time t is given by

$$\dot{Q}_{u, solar}^{max} = \eta_{sys} \eta_c G_T A_c \quad (4.5.5)$$

where:

- η_c : collector efficiency as given by equation (4.5.1)
- η_{sys} : solar system efficiency (user defined coefficient that accounts for the thermal losses in the solar field piping; indicates the average ratio of useful solar heat delivered to the storage and solar heat at collector output)
- G_T : total solar radiation incident on collector plane (to be substituted by beam radiation G_{bT} for concentrating collectors)
- A_c : effective (usually aperture⁶) area of solar collectors

The solar buffer storage is modeled for simplicity as a fully mixed storage (i.e. all the storage volume is considered at one temperature).

Control strategy:

a) potential solar system output > demand + storage losses

In this case in first place demand and storage losses are covered by the generated solar heat. The surplus is stored in the solar buffer storage (under the constraint of a maximum allowed temperature).

b) potential solar system output < demand + storage losses

In this case in first place part of the demand and storage losses are covered by the useful solar heat gains. The uncovered part of the demand is covered by storage discharge (under the constraint of a minimum temperature in the storage, that cannot be lower than the minimum temperature of heat demand).

Mathematically, this control strategy is realised as described below. Total demand as seen by the solar field is given by:

⁶ *Technical data for solar collectors may be given based on gross area, absorber area or aperture area. The solar collector area used in this equation has to be consistent with the one used as basis for the collector efficiency parameters*

- the actual heat demand $\dot{Q}_{USH}(T_{storage})$; it is considered that only the demand up to $T_{storage}$ can be covered
- the remaining capacity for storage charge $\Delta Q_{storage}^{max} = (m c_p)_{storage} (T_{max} - T_{storage})$ given by the thermal mass of the storage, the actual and the maximum storage temperatures
- thermal losses of the storage to be compensated $\dot{Q}_{storage}^{loss} = UA(T_{storage} - T_{env})$

The actual solar field output then is calculated as

$$\dot{Q}_{u,solar} = \min(\dot{Q}_{u,solar}^{max}, \dot{Q}_{USH,Req}(T_{storage}) + \frac{\Delta Q_{storage}^{max}}{\Delta t} + \dot{Q}_{storage}^{loss}) \quad (4.5.6)$$

The maximum amount of heat available for supplying demand is given by:

- the net solar field output (= field output – thermal losses of storage)
- the remaining capacity for storage discharge $\Delta Q_{storage}^{discharge}(T_{min}) = (m c_p)_{storage} (T_{storage} - T_{min})$ given by the thermal mass of the storage, the actual and the minimum storage temperatures, where the minimum storage temperature is given by the minimum temperature of demand plus a certain $\Delta T_{stratification}$ accounting for stratification (considered constant at default value 5K)

$$\dot{Q}_{USH} = \min\left[\left(\dot{Q}_{u,solar}^{max} - \dot{Q}_{storage}^{loss} + \frac{\Delta Q_{storage}^{discharge}(T_{min})}{\Delta t}\right), \dot{Q}_{USH,Req}(T_{storage})\right] \quad (4.5.7)$$

4.5.2.5 Inclination and azimuth of solar collectors

In order to minimise the need for user input, EINSTEIN uses a default algorithm for defining the inclination and azimuth of the solar collectors from the available surface areas.

a) surfaces undefined or more than one surface available

In this general case EINSTEIN auto-selects the inclination and orientation for the solar collectors considered optimum.

For collector types parabolic trough, Fresnel and ... 0° inclination is considered optimum (most commercially available collectors are not designed for inclined mounting)

For flat plate collectors and evacuated tube optimum inclination = |latitude| – 10°.

Optimum orientation is to south in the northern hemisphere and to north in the southern hemisphere.

b) exactly one surface available

Also in this case, the optimum inclination and orientation described above are used as default value, but in certain special cases different orientations are used:

- for surfaces oriented to south, south-east or south-west (in the northern hemisphere) or to north, north-east or north-west (in the southern hemisphere) with a sufficient inclination in-roof-mounting is considered as the best option.

- the orientation of the solar collectors is identical to that of the available surface (if deviation from the south or north is larger than 45° and surface inclination is small, then the solar collectors are rotated by 90 (supposing that surface areas are rectangular shaped).

4.5.3 Design assistant

4.5.3.1 General overview

The auto-design of solar thermal systems is carried out in the following steps

- a) check of available surface area
- b) pre-dimensioning of solar system
- c) selection of solar collector type
- d) final dimensioning of proposed solar system (adjustment of solar fraction and solar yield per unit of installed power)

4.5.3.2 Check of available surface area

a) Ratio of gross surface area to installed nominal power

The required surface area for installing a given size of solar field is given by the following data:

- ratio of gross to net (effective) collector area of the specific collector type
- ratio of surface area to gross collector area depending on inclination of collectors and minimum required row separation in order to avoid shading
- minimum fraction of additional area required for piping, access for maintenance, separation between collectors within one collector row, etc.

The defaults as given in Table 4 are used depending on collector type.

Table 4. Area ratios for different collector types

| Collector type | Ratio roof area per unit of nominal power | | | | |
|-------------------------------------|---|-------------------------------------|---|---------------------------------------|--|
| | ratio gross to net collector area (*) | optimum inclination at 40° latitude | ratio roof area to gross collector area | ratio roof area to net collector area | ratio roof area to nominal power (***) |
| | - | ° | - | - | m ² /kW |
| Flat plate collector (FPC) | 1,152 | 30 | 2,12 | 2,44 | 3,49 |
| Evacuated tube collector (ETC) (**) | 1,355 | 0 | 1,10 | 1,49 | 2,13 |
| Parabolic trough collector (PTC) | 1,063 | 0 | 3,30 | 3,51 | 5,01 |
| Fresnel collector (FRC) | 1,364 | 0 | 1,10 | 1,50 | 2,14 |

(*) for typical collectors of the given type

(**) in-roof mounting, turning tubes into optimum direction

(***) using standard conversion factor of 0,7 kW/m² independent of collector type

b) Check of appropriateness of surfaces

In order to check the appropriateness of available surfaces, the following checks are carried out:

- Absolute size of surface is enough to install a solar system of at least minimum allowed size (default: 14 kW).
- Appropriate inclination and orientation of the surface:

- inclined surface facing to N, NE, NW (on northern hemisphere) or to S, SE, SW (on southern hemisphere) are excluded
- Surfaces with severe shading problems are excluded
- Surfaces with insufficient static load capacity are excluded (minimum 25 kg/m²) or at too large distance

4.5.3.3 Predimensioning

Predimensioning is obtained analysing the demand at a supposed default maximum temperature of 200 °C.

Using a default solar fraction of $f_{sol} = 50\%$ and a default desired solar yield of 700 kWh/kW and year an estimate for the required solar system power is obtained by

$$P_{nom}^{predimensioning} = f_{sol} \frac{Q_{USH, Req}(200\text{ }^{\circ}\text{C})}{q_{u, solar}} \quad (4.5.8)$$

From the heat demand by temperature curve the estimated average working temperature in the collector is calculated, that then in the following step will be used for selecting an appropriate collector type.

4.5.3.4 Selection of solar collector type

Automatic selection of the solar collector type and model is done by searching the database of solar collectors.

The collector with maximum efficiency at the estimated average working temperature (from predimensioning) and an incident radiation of 800 W/m² (normal incidence) is selected (efficiency of concentrating collectors is furthermore corrected by a factor of 0,8, in order to take account for the fact that those collectors use only direct radiation).

4.5.3.5 Final dimensioning

By an iterative algorithm that solar system size is selected that results in the desired (user specified) solar fraction.

If for this system size, the solar yield per unit system size is smaller than the desired (user specified) value, system size is reduced until this lower limit is reached.

For the case that the resulting solar system size is smaller than the minimum system size (14 kW), no solar system is proposed.

References:

-

4.6 EINSTEIN – Cooling Module

4.6.1 Module parameters

Inputs

(...)

Outputs

(...)

Parameters

| Parameter (user interface) | Symbol | Description |
|--|-----------------------|--|
| Nominal power (cooling, output) | Pnom | Power output of the chiller corresponding to full load (in current version considered as constant independent of working temperatures) For fresh or ground water as cooling supply the nominal power is defined as the cooling power delivered by the given (maximum) water flow rate and a nominal temperature drop of 25K between inlet and outlet. |
| Electrical power input | Pel | Electrical power input for the chiller, including cooling towers and auxiliary equipment (circulation pumps, etc.). $P_{el} = P_{nom} / EER_{sys_fullLoad_real}$ |
| Chiller efficiency | EER_ch | Ratio of useful cooling supplied to driving energy (electricity, fuels, heat). In the case of air cooled compression chillers, electricity for fans is INCLUDED into the chiller EER ! |
| Thermal efficiency | EER_sys_fullLoad_real | Energy Efficiency Ratio (EER) of the system (chiller + cooling tower) at full load under real operation conditions (as given by T_supply and T_reject). Includes electricity consumption of cooling tower |
| Temperature at which cooling is supplied | T_supply | Average temperature (during operating hours) in the real system at which cooling is supplied (outlet of chiller). |
| Heat rejection temperature | T_reject | Average temperature (during operating hours) in the heat rejection circuit of the chiller (cooling water or air). Inlet temperature to the chiller. |
| Exergetic efficiency | η_{ex} | Ratio between real and ideal (Carnot) efficiency of chiller |
| Chiller type | CHTYPE | The following types are currently supported: compression chiller (air cooled) compression chiller (water cooled) cooling tower (dry) cooling tower (wet) fresh or ground water |
| Minimum cooling supply temperature | T_supply_min | Minimum temperature at which cooling can be supplied (chiller outlet) given by technical constraints of the chiller or the working fluids used. If nothing is specified, -60 °C (the lower limit of the temperature scale) is used. |

Data files and data bases

Chillers and heat pumps are listed in the same data base. For a description of this database see the chapter on heat pumps.

4.6.2 *Mathematical description*

a) *Energy calculations*

Yearly energy balances

For the yearly energy balance of the chiller the following equation holds:

$$FET_j = \frac{USC_j}{EER_{sys, partLoad, real}}$$

where:

USC_j: the yearly amount of cooling supplied by the chiller

FET_j: the yearly amount of driving energy (electricity, fuel, heat) consumed by the chiller

$EER_{sys, partLoad, real}$: energy efficiency considering temperature levels and part-load efficiency losses (if applicable).

Instantaneous efficiency

In EINSTEIN system simulations, the instantaneous efficiency of the system is calculated:

$$FET_j(t) = \frac{USC_j(t)}{EER_{sys}(t)}$$

with

$$\frac{1}{EER_{sys}(t)} = \frac{1}{EER_{ch}(T_{ch, outlet}(t), T_{reject}(t)) * PLC(t)} + f_{reject} \frac{QWH_j(t)}{USC_j(t)}$$

where:

USC_j(t): cooling power supplied by the chiller

FET_j(t): power consumption of driving energy (electricity, fuel, heat) consumed by the chiller

PLC(t): part load correction factor for chillers (see section on part load correction)

f_{reject}: electrical power consumption ratio of a cooling tower (water cooled compression chillers only, 0 for other types of cooling equipment). Currently a default value of f_{reject} = 1/ 33.333 is assumed when wet cooling towers are selected, and 1/22,222 for dry cooling towers (value derived from [EIPPCB, 2001]).

QWH(t) is the waste heat to be dissipated in the cooling tower (heat rejection)

The energy efficiency ratio EER_{ch} is assumed to be constant for cooling towers (wet and dry) and fresh and ground water.

For cooling towers (wet and dry), default values for EER (considers circulation pump, cooling tower pump and cooling tower fans) are available that are used whenever no appropriate user input data are given:

- dry cooling tower: EER = 22.222
- wet cooling tower: EER = 33.333

For fresh or ground water a default of EER = 100 is supposed, accounting for electricity consumption of circulation pumps (value derived from [EIPPCB, 2001]).

For compression chillers, EER_{ch} is calculated from the exergetic efficiency depending on instantaneous working temperatures (see following sections).

Extrapolation from nominal to real working conditions (compression chillers)

For simplicity, the full load power (cooling, output) is considered as constant, whereas the power input (electricity or other driving source) is variable depending on the variation in the EER.

The (time dependent) chiller efficiency at real operation conditions and at full power is obtained as:

$$EER_{ch, real, fullLoad}(t) = \eta_{ex, nom} * EER_{carnot}(T_{c, real}(t), T_{h, real}(t))$$

where:

$EER_{ch, real, fullLoad}(t)$: real energy efficiency at full load considering the real temperatures for chilled water and heat rejection.

$\eta_{ex, nom}$: nominal exergy efficiency

$EER_{carnot}(T_{c, real}(t), T_{h, real}(t))$: carnot energy efficiency ratio of the chiller calculated using chilled water and heat rejection temperatures.

For compression chillers it is assumed that the exergetic efficiency of the chiller remains constant for different working conditions:

$$\eta_{ex, nom} = \eta_{ex, real}$$

The working conditions of the chiller are defined by:

- the outlet temperature of the chilled water / air / fluid circuit (cooling supply)
- the inlet temperature of the heat rejection circuit (cooling water or air)

From those temperatures, the evaporation (T_c) and condensation (T_h) temperatures are calculated using the correlations specified below (see chapter on default values and operating conditions).

The exergetic efficiency at nominal conditions is obtained from the nominal EER by:

$$\eta_{ex, nom} = \frac{EER_{ch, nom}}{EER_{carnot}(T_{c, nom}, T_{h, nom})}$$

where

$EER_{ch, nom}$: energy efficiency at full load at nominal conditions.

EER_{carnot} , ($T_{\text{c,nom}}$, $T_{\text{h,nom}}$): carnot energy efficiency ratio of the chiller calculated using chilled water and heat rejection nominal temperatures.

For air cooled compression chillers the electricity consumed for fans is considered included in the chiller efficiency:

$$EER_{ch, \text{nom}} = EER_{\text{sys}, \text{fullLoad}, \text{real}}$$

whereas for water cooled compression chillers the electricity consumption of the cooling tower is considered not to be included in the chiller data:

$$\frac{1}{EER_{ch, \text{nom}}} = \frac{1}{EER_{\text{sys}, \text{fullLoad}, \text{nom}}} - f_{\text{reject}} \frac{QWH_{j, \text{nom}}}{USC_{j, \text{nom}}}$$

The carnot energy efficiency ratio EER_{carnot} is given by:

$$EER_{\text{carnot}}(T_c, T_h) = \frac{T_c [K]}{(T_h - T_c)}$$

where T_c [K] is the absolute temperature in K (T_c in [$^{\circ}$ C] + 273.15).

T_c : evaporation temperature of the refrigerant

T_h : condensation temperature of the refrigerant

If the exergetic efficiency η_{ex} cannot be calculated from user specified data (nominal EER or nominal temperature level unknown), the following default values are used:

- compression chiller (air cooled): 0,4983
- compression chiller (water cooled): 0,5194

These default values for exergetic efficiencies η_{ex} were determined based on data from the manufacturer York: 33 water and 94 air cooled chillers using either R134a or R407C were assessed as to their exergetic efficiency by considering chilled and cooling water temperatures of:

1. 7/12 $^{\circ}$ C and 35/30 $^{\circ}$ C,
2. 6/13 $^{\circ}$ C and 30/25 $^{\circ}$ C, and
3. 8/14 $^{\circ}$ C and 40/35 $^{\circ}$ C

In case the EER provided by the user is not at 7/12 $^{\circ}$ C and 35/30 $^{\circ}$ C (water cooled) or 35 $^{\circ}$ C (air cooled), EINSTEIN recalculates this value at these conditions (Eurovent conditions), or at 2,7/12,2 and 29,4/35 $^{\circ}$ C (ASHRAE conditions). The user can then compare his EER value with those of actual equipments on the market.

Part load correction (compression chillers)

The real instantaneous chiller efficiency is furthermore corrected for part load behaviour by the part load correction factor PLC:

Source: Wellig et al. [2006]

$$EER_{ch, \text{real}} = EER_{ch, \text{real}, \text{fullLoad}} * PLC \quad (2.1)$$

with

$$PLC = \frac{PLR}{(C * PLR) + (1 - C)} \quad (2.2)$$

and

$$PLR = Q_{Load} / Q_{Nom} \quad (2.3)$$

$EER_{ch,real}$: Energy Efficiency Ratio at part load

$EER_{ch,real,fullLoad}$: Energy Efficiency Ratio at full load = Q_{nom}/P_{nom}

Q_{Nom} – Cooling power at full load – value either as data input or calculated by EINSTEIN

P_{Nom} – Electrical power at full load – value obtained from data acquisition

PLR: Part Load Ratio

Q_{Load} : Cooling power at considered load

C: Degradation factor - value: 0,9 (source: prEN 14825 standard, mentioned in [Cecchinato 2010])

The calculation of the average yearly part load correction factor requires a time dependent simulation of the system throughout the year, which in the current version is not carried out at the stage of data consistency checking. Therefore, during consistency checking of user input data EINSTEIN applies a simplified procedure, considering the following:

- there is a lower limit for the part load correction factor given by

$$PLC_{min} = PLC(PLR_{average})$$

valid for the limit case in which the system operates constantly during all operating hours at $PLR = PLR_{average} = const.$

- there is an upper limit for the part load correction factor given by

$$PLC_{max} = 1$$

valid for the limit case in which the system operates on/off (neglecting energy losses due to cycling)

As real systems behave somewhere in between, currently in Version 2.0 an estimative conservative yearly-average part load correction function is used:

$$PLC = 0.9 + 0.1 * PLR_{average}$$

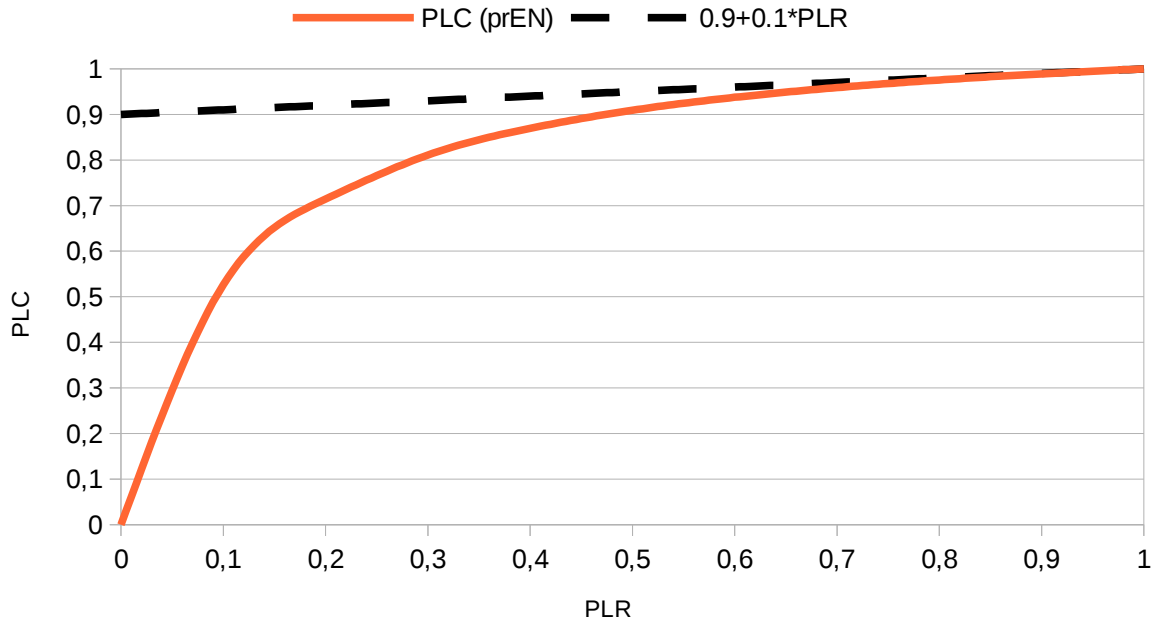


Figure 12. Part load correction coefficient (PLC) as a function of the part load ratio (PLR).

Waste heat

For reasons of simplicity it is supposed that energy input to the system is dissipated in the heat rejection circuit, and there is no dissipation / losses elsewhere.

Then, the following equation is valid:

$$Q_{WHj} = USC_j + FET_j$$

The temperature level of the waste heat is supposed to be that of the actual heat rejection circuit.

b) Default values and operating conditions

In order to limit the volume of data to be entered by the user, some of the system parameters that are of secondary importance are currently (Version 2.0) fixed to constant default values:

i) Temperature difference between ambient temperature and the inlet temperature of the heat rejection circuit (cooling water) to the chiller (= outlet temperature from cooling tower).

The temperature drop is currently fixed to constant: 7 K.

$$T_{reject_in} = T_{wb} + 7 \text{ K}$$

Jacquard et al [2004], mentions a range of 3-7K for water cooled chillers using open wet cooling towers, 7-15K when using closed wet cooling towers and 7-12K when using dry cooling towers.

If no data file is specified with explicit values for dry and wet bulb temperatures (simplified weather data input) then the following estimate is used:

$$T_{wb} = T_{db} - 5K$$

For air cooled chillers the following equation holds

$$T_{reject_in} = T_{db}$$

whereas for dry cooling towers the cooling water outlet temperature is given by

$$T_{ch,out} = T_{db} + 9,5$$

The value of 9,5 is the average of the range 7 – 12 K as specified by Jacquard et al. [2004].

A minimum condensation temperature can be fixed and is currently set to a constant value:

$$T_{h,min} = 20 \text{ °C} = \text{const.}$$

Suggestion for working with cooling module (compression chillers)

The user is requested to enter the (average) temperature for heat rejection (inlet of cooling water / cooling air circuit to the chiller). In system simulation this parameter is a function of the ambient temperature which is time dependent. For systems working 24 hours a day and 365 days a year the average ambient temperature during system operation matches with the yearly average value.

For systems operating mainly during the day, about 3K should be added to the average value of T_{amb} ⁷.

For systems operating mainly during summer time, about 5 - 10 K (depending on the amplitude of temperature variations between summer and winter; see section on weather data for default correlations used by EINSTEIN) should be added to the average value of T_{amb} .

Therefore the average heat rejection temperature can be estimated to

$$T_{reject_in_avg} = T_{db/wb_avg} + \Delta T_{dayoperation} + \Delta T_{summeroperation} + \Delta T_{CT}$$

where ΔT_{CT} is the difference between dry or wet bulb temperature (depending on chiller type, see above) and the inlet temperature to the chiller heat rejection circuit.

ii) Temperature difference between forward and return temperature in the heat rejection circuit.

Currently fixed to constant 5 K.

$$T_{reject_out} = T_{reject_in} + 5K$$

⁷ In simplified weather data a sine-function profile with an amplitude of +/- 5 K and a maximum at 14:00 h is supposed for the daily variation. See section on → weather data.

Jacquard et al [2004], mentions a range of 4-5K for plate heat exchangers, 5K for shell and tube heat exchangers in combination with air cooled chillers, water cooled chillers using wet cooling towers and 2-5K for shell and tube heat exchangers in combination with water cooled chillers using dry cooling towers.

iii) Temperature difference between forward and return temperature in the chilled water / air / fluid circuit.

Currently fixed to constant 5 K.

$$T_{ch_in} = T_{ch_out} + 5K$$

iv) Temperature difference between cooling water / air inlet temperature and condensing temperature T_h

Currently (EINSTEIN V2.0) fixed to constant 10 K (water-cooled chillers) and 13,5 K (air-cooled chillers) [Jacquard et al., 2004].

According to Jacquard et al. the temperature difference between cooling water outlet (to heat rejection circuit) and condensation is 5K. With the water temperature difference between inlet and outlet of 5K (see above) this results in a ΔT of 10K.

For the temperature difference between air entry and condensing temperature Jacquard et al. [2004] mention a range between 12 and 15K.

Currently (EINSTEIN V2.0) ΔT is fixed to constant 7 K.

$$T_h = T_{reject_in} + 10K = T_{reject_out} + 5K \text{ (water-cooled)}$$

$$T_h = T_{reject_in} + 13,5K = T_{reject_out} + 8,5K \text{ (air-cooled)}$$

Example: for a cooling water circuit with 30/35 °C a condensing temperature of 30 (inlet to chiller) + 10 = 40 °C is supposed.

v) Temperature difference between chilled water / air / fluid inlet temperature and evaporation temperature T_c

Currently (EINSTEIN V2.0) fixed to constant 9,25K.

According to Jacquard et al. [2004] the temperature difference between chilled water outlet (to chilled water distribution circuit) and evaporation is between 4-6K for shell and tube heat exchangers coupled to air cooled chillers and water cooled chillers equipped with wet cooling towers, and 2-5K for chillers equipped with plate heat exchanger. An average of 4,25K was calculated to use one single value. For water cooled chiller equipped with a dry cooling tower, a difference of 6-10K between water entry and evaporation temperature is given. With the water temperature difference between inlet and outlet of 5K (see above) this results in a ΔT of 9,25K.

$$T_c = T_{ch_in} - 9,25K = T_{ch_out} - 4,25K$$

Example: for a chilled water circuit with +7/+12 °C an evaporation temperature of 12 (inlet to chiller) – 9,25 = +2,75 °C is supposed

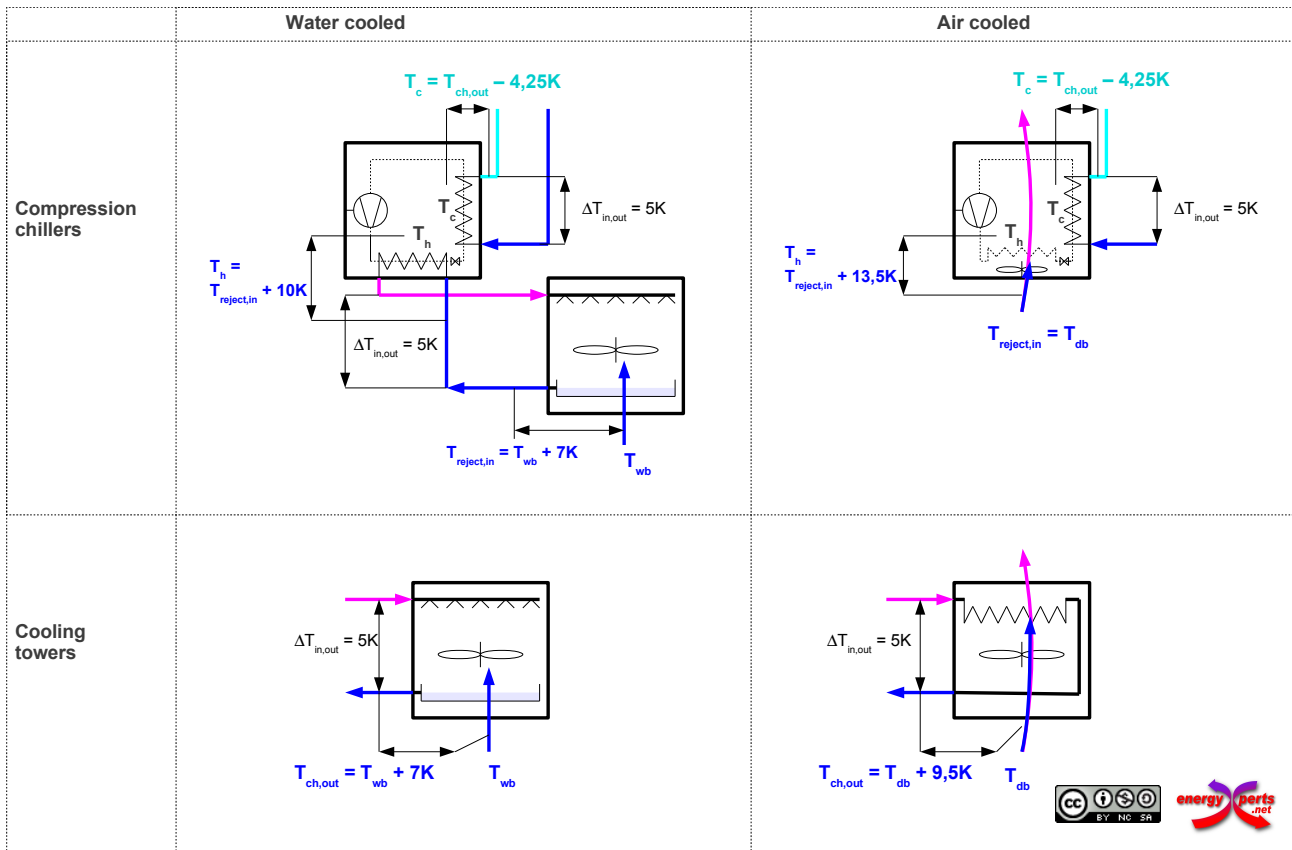


Figure 13. Overview of default correlations for temperature differences in compression chillers and cooling towers

c) Matching of flow rates and temperature levels in load and chiller

In real systems, flow rates and temperature levels in the chiller and in the load can be made independent by using a four way mixing valve.

In this case, and whenever the flow rate in the load is lower or equal than the chiller flow rate, the required outlet temperature of the chilled water / air / fluid is given by the minimum temperature of the cooling load.

For reasons of simplicity this assumption is applied in EINSTEIN (V2.0) as a general rule in order to obtain the chiller working point at each time t:

$$T_{ch_out} = T_{load_min}$$

d) Water consumption

The water consumption of fresh and ground water systems is considered to be proportional to the cooling supply USC:

$$m_{H2O} = \frac{USC}{(c_p \Delta T)}$$

where c_p is the specific heat of water and ΔT is the temperature difference between water inlet and outlet. In EINSTEIN calculations currently a fix ΔT of 25 K is assumed. For systems with different ΔT the water quantities have to be corrected manually.

The water consumption of wet cooling towers is considered to be proportional to the cooling supply USC (dissipated heat):

$$m_{H2O} = USC r_{H2O}^{CT}$$

A constant value of 0,002 m³/kWh is assumed for the water demand ratio r_{H2O}^{CT} [EIPPCB, 2001].

For the water consumption of wet cooling towers, it is assumed that the cooling tower is an open wet cooling tower. Water consumption of closed wet cooling towers is indicated as “variable” in reference [EIPPCB, 2001].

The water consumption of the water cooled compression chiller is calculated using the same model than for the wet cooling tower, but considering that the dissipated heat is equal to the waste heat generated by the chiller:

$$m_{H2O} = USC r_{H2O}^{CT}$$

References:

Cecchinato L., Chiarello M., Corradi M. [2010]: A simplified method to evaluate the seasonal energy performance of water chillers, International Journal of thermal Sciences 49, 1776-1786

European Integrated Pollution and Prevention Control Bureau (EIPPCB), [2001], Best reference document on the application of best available technologies to industrial cooling systems, EIPPCB, <http://eippcb.jrc.ec.europa.eu/>

Jacquard P., Sandre S. [2004]: La pratique du froid, PYC édition.

4.7 EINSTEIN – Heat Pump Module

4.7.1 Module parameters

Inputs

a) Temperature and time dependent heat demand and availability curves

| Parameter | Unit | Min/Max | Description |
|------------|------|---------|---|
| DBHeatPump | - | - | Heat pump database (table in einsteinDB) |
| QDa(T) | MWh | | Annual remaining heat demand after heat recovery (HRM output) |
| QAa(T) | MWh | | Annual remaining waste heat availability after heat recovery (HRM output) |
| QDh(T,t) | MWh | | Hourly remaining heat demand after heat recovery (HRM output) |
| QAh(T,t) | MWh | | Hourly remaining waste heat availability after heat recovery (HRM output) |

Outputs

a) Dimensioning of heat pump

As a result the Heat Pump Module selects the smallest heat pump from the database in order to match the hourly heat demand and availability curves and to fulfil the specified user-defined conditions (see above: user-defined specifications).

If a heat pump application is not appropriate to the specific process characteristics, a warning message is shown.

| Parameter | Unit | Min/Max | Description |
|-----------------|------|---------|--|
| QGenerationHC | | | Table in einsteinDB where the data for the HC Generation equipment is stored. A modified list of HC Generation equipment will be created, including the new heat pump. |
| HPApplicability | | | Internal table in einsteinDB where flag that indicates the applicability of a heat pump is stored (execution state): (0): everything ok, heat pump has been designed (1): not applicable due to XY (2): not applicable due to ZZ (3): not applied because user didn't want ... |

b) Energy-flows in the heat pump

Other results

- Hourly and annual energy Q_h, Q_c, Q_w , [MWh]
- Hourly T_h, T_c, COP_{real}
- Hourly part load factor $F_{pl}(i)$
- Annual working hours F_{pla}
- Annual averaged heat pump energy flows $\dot{Q}_h, \dot{Q}_c, \dot{Q}_w$, [kW]
- Annual averaged T_h, T_c, COP_{real}

c) Remaining heat demand curve

| Parameter | Unit | Min/Max | Description |
|-------------|------|---------|---|
| QDhp,a(T) | MWh | | Annual remaining heat demand after heat pump application |
| QAhp,a(T) | MWh | | Annual remaining available heat after heat pump application |
| QDhp,h(T,t) | MWh | | Hourly remaining heat demand after heat pump application |
| QAhp,h(T,t) | MWh | | Hourly remaining available heat after heat pump application |

Parameters

The parameters are the user defined specifications which are adopted in the calculation and selection process. These are stored in the einsteinDB, internal table UHeatPump.

For the Heat Pump Module the following parameters are required:

| Parameter | Unit | Min/Max | Description |
|------------|------|---|--|
| UHPTtype | - | - | Heat pump type |
| UHPPminHop | h | 1 - 8760 | Minimum desired annual operation hours |
| UHPPDTMax | °C | 0 – 100 | Maximum desired temperature lift |
| UHPTgenIn | °C | depending on heat pump type (single or double effect) | Only for absorption heat pumps: Inlet temperature of heating fluid in generator. |
| UHPPmaxT | °C | depending on heat pump type in DB, not greater than limit temperature | Maximum desired condensing temperature |
| UHPPminT | °C | depending on heat pump type in DB, not smaller than limit temperature | Minimum desired evaporating temperature |

Depending on the level of user interaction selected from the main menu (Level1: Interactive, Level2: Semi-automatic, Level3: Automatic), these parameters will be either required from the user through an interactive menu, or default values will be used, provided by the set-up menu (see chapter 3 of this document).

Data files and data bases

Data bases used by the heat pump module:

| Data Base | Description solo lista de bases de datos |
|------------------|---|
| DBHeatPumps | Data base on available heat pumps |
| UHeatPump | User-defined parameters from GUI, table in einsteinDB |
| DBFluid | Fluid properties ... (??? do we need them here ???) |
| DB XY | ??? other data bases that might be needed / accessed directly by the module |
| ...to define... | Tables where to store the selected equipment for the alternative proposals |

4.7.2 Heat Pump Database

a) Heat pump database (DBHeatPump)

Database comprising vapour-compression and absorption (thermal) heat pumps

| Parameter | Name in einsteinDB | Unit | Code name | Description |
|--|--------------------|------|-----------|---|
| Manufacturer data | HPManData | - | | Manufacturer's data for heating or cooling mode, or both? Options: heating only (H); cooling only (C); heating and cooling (HC) |
| Manufacturer | HPManufacturer | - | | Manufacturer name-EINSTEIN standard |
| Model | HPModel | - | | Model of the equipment |
| Type | HPTType | - | | COMP: compression or ABS: absorption |
| Sub-type | HPSubType | - | | Heat source - heat sink indication (e.g. water - water) |
| Effects (absorption) | HPAbsEffects | - | | Only for absorption type: Number of effects: 1, 2, or 3 |
| Heating medium (absorption) | HPAbsHeatMed | - | | Only for absorption type: Heating medium in the generator. Options: hot water, vapour, exhaust gas, direct fired |
| Working fluid | HPWorkFluid | - | | Refrigerant, absorbent-refrigerant pair |
| Cooling capacity | HPCoolCap | kW | Qdot,c | Nominal cooling capacity (manufact. catalogue) |
| Nominal cooling COP | HPCoolCOP | - | COPc | Nominal COP for cooling mode (manufact. catalogue) |
| Exergetic cooling COP | HPExCoolCOP | - | COPc,ex | Calculated from the nominal and theoretical COP at the manufact. catalogue nominal conditions and applied as a constant in extrapolation for other working conditions (see next point). |
| Theoretical cooling COP | HPThCoolCOP | - | COPc,t | Carnot COP for cooling mode at nominal conditions ⁸ (see next point). |
| Temperature range of constant exergetic COP application, cooling | HPConstExCoolCOP | °C | DTcop,c | Range of variation of the temperatures (Tab,Tco,Tev,Tge) around the nominal, where the constant exergetic COP calculated above can be applied (e.g. +/-20 K) |
| Absorber inlet temperature, cooling mode | HPAbsTinC | °C | | Only for abs. HP: Absorber secondary fluid's inlet temperature at nominal conditions, cooling mode (manufact. catalogue) ² |

⁸ For the calculation of the theoretical COP the temperature difference between the secondary fluid's inlet and the primary fluid (in absorber, condenser, evaporator and generator) is assumed to be 7 K

| Parameter | Name in einsteinDB | Unit | Code name | Description |
|--|--------------------|------|-----------|---|
| Condenser inlet temperature, cooling mode | HPCondTinC | °C | | Condenser secondary fluid's inlet temperature at nominal conditions, cooling mode (manufact. catalogue) ² |
| Generator inlet temperature, cooling mode | HPGenTinC | °C | | Only for abs. HP: Generator secondary fluid's inlet temperature at nominal conditions, cooling mode (manufact. catalogue) ² |
| Evaporator inlet temperature, cooling mode | HPEvapTinC | °C | | Evaporator secondary fluid's inlet temperature at nominal conditions, cooling mode (manufact. catalogue) ² |
| Heating capacity | HPHeatCap | kW | Qdot,h | Nominal heating capacity (manufact. catalogue) |
| Nominal heating COP | HPHeatCOP | - | COPh | Nominal COP for heating mode (manufact. catalogue) |
| Exergetic heating COP | HPExHeatCOP | - | COPh,ex | Calculated from the nominal and theoretical COP at the manufact. catalogue nominal conditions and applied as a constant in extrapolation for other working conditions (see next point). |
| Theoretical heating COP | HPThHeatCOP | - | COPh,t | Carnot COP for heating mode at nominal conditions ¹ (see next point). |
| Temperature range of constant exergetic COP application, heating | HPConstExHeatCOP | °C | DTcop,h | Range of variation of the temperatures (Tab, Tco, Tcv, Tge) around the nominal, where the constant exergetic COP calculated above can be applied (e.g. +/-20 K) |
| Absorber inlet temperature, heating mode | HPAbsTinH | °C | Tab,in | Only for abs. HP: Absorber secondary fluid's inlet temperature at nominal conditions (manufact. catalogue). ⁹ |
| Condenser inlet temperature, heating mode | HPCondTinH | °C | Tco,in | Condenser secondary fluid's inlet temperature at nominal conditions (manufact. catalogue). ² |
| Generator inlet temperature, heating mode | HPGenTinH | °C | Tge,in | Only for abs. HP: Generator secondary fluid's inlet temperature at nominal conditions (manufact. catalogue). ² |
| Evaporator inlet temperature, heating mode | HPEvapTinH | °C | Tcv,in | Evaporator secondary fluid's inlet temperature at nominal conditions (manufact. catalogue). ² |
| Limit temperature difference | HPLimDT | °C | DTlim | Maximum acceptable temperature difference between evaporator and condenser temperatures (primary fluid: Tco - Tcv) - working limit |

⁹ the abs. temperature difference inlet-outlet in the secondary fluid is considered to be 5 K

| Parameter | Name in einsteinDB | Unit | Code name | Description |
|---------------------------------|--------------------|------------|-----------|--|
| Minimum generator temperature | HPGenTmin | °C | Tge,min | Only for abs. HP: Minimum generator temperature (primary fluid) - working limit |
| Maximum condensing temperature | HPCondTmax | °C | Tco,max | Maximum condensing (and absorption) temperature (primary fluid) - working limit |
| Minimum evaporating temperature | HPEvapTmin | °C | Tev,min | Minimum evaporating temperature (primary fluid) - working limit |
| Electricity consumption | HPElectConsum | kW | | Only for abs. HP: Electricity consumption of the equipment (pumps, ...) |
| Equipment factory price | HPPrice | € | | Equipment price at factory applied installer's discount |
| Turn-key price | HPTurnKeyPrice | € | | Price of installed equipment (including work, additional accessories, pumps, regulation, etc) |
| Ratio for O&M costs (fixed) | HPOandMfix | €/kW year | | Annual operational and maintenance fixed costs (approximate average per kW heating) |
| Ratio for O&M costs (variable) | HPOandMvar | €/MWh year | | Annual operational and maintenance variable costs dependant on usage (approximate average per MWh heating) |
| Year of last data update | HPYearUpdate | year | | Year of last update of the heat pump equipment and the economic data |

b) Extrapolation outside the nominal conditions of operation

Absorption cooling and heating equipment

When the application working conditions are different from the nominal, the following procedure is used to approximate the real COP of absorption equipment

Cooling mode

The theoretical Carnot COP is calculated from the temperatures specified in the manufacturer's catalogue as nominal.

A correction is applied to the temperatures in order to approximate the primary fluid temperatures (7 K correction, corresponding to logarithmic mean difference of 4 K):

$$T_{ev} = T_{ev}^{inlet} - 7 K; T_{ab} = T_{ab}^{inlet} + 7 K; T_{co} = T_{co}^{inlet} + 7 K; T_{ge} = T_{ge}^{inlet} - 7 K$$

The temperatures are converted to absolute temperatures in [K]

$$T_{ev}^K = T_{ev} + 273.15; T_{ab}^K = T_{ab} + 273.15; T_{co}^K = T_{co} + 273.15; T_{ge}^K = T_{ge} + 273.15$$

Then the theoretical Carnot COP is calculated:

$$COP_c^{th} = \frac{T_{ev}^K (T_{ge} - T_{ab})}{T_{ge}^K (T_{co} - T_{ev})}$$

The exergetic COP_c(COP_{c,ex}) is calculated at the nominal conditions, assuming that the real COP can be expressed as a product of the theoretical and the exergetic COPs:

$$COP_c^{real} = COP_c^{ex} \cdot COP_c^{th}$$

For extrapolation of the real COP_c outside the nominal conditions (in the defined limit) the previously calculated COP_{c,ex} is assumed as a constant, and the real COP_c is calculated from:

$$COP_c^{real} = COP_c^{ex} \cdot COP_c^{th, real}$$

where COP_{c,th,real} is the theoretical Carnot COP for the real temperature working conditions.

Heating mode

In similar way the calculations are done for the COP in heating (heat pump) mode. A correction is applied to the temperatures in order to approximate the primary fluid temperatures (7 K correction, corresponding to logarithmic mean difference of 4 K), and then the temperatures are converted to absolute temperature in [K]:

$$T_{ev} = T_{ev}^{inlet} - 7 K ; T_{ab} = T_{ab}^{inlet} + 7 K ; T_{co} = T_{co}^{inlet} + 7 K ; T_{ge} = T_{ge}^{inlet} - 7 K$$

$$T_{ev}^K = T_{ev} + 273.15 ; T_{ab}^K = T_{ab} + 273.15 ; T_{co}^K = T_{co} + 273.15 ; T_{ge}^K = T_{ge} + 273.15$$

The theoretical COP of the absorption heat pump is then calculated:

$$COP_h^{th} = \frac{T_{ev}^K (T_{ge} - T_{ab})}{T_{ge}^K (T_{co} - T_{ev})} + 1$$

The exergetic COP_h for heat pump is calculated at the nominal conditions from:

$$COP_h = COP_h^{ex} \cdot COP_h^{th}$$

The real COP_h of the heat pump at different than the nominal conditions is calculated, considering the exergetic COP_h constant, from:

$$COP_h^{real} = COP_h^{ex} \cdot COP_h^{th, real}$$

Vapour-compression refrigeration and heat pump equipment

When the application working conditions are different from the nominal, the following procedure is used to approximate the real COP of the equipment

Cooling mode

The theoretical Carnot COP is calculated from the temperatures specified in the manufacturer's catalogue as nominal.

A correction is applied to the temperatures in order to approximate the primary fluid temperatures (7 K correction, corresponding to logarithmic mean difference of 4 K):

$$T_{ev} = T_{ev}^{inlet} - 7 K ; T_{co} = T_{co}^{inlet} + 7 K$$

The temperatures are converted to absolute temperatures in [K]

$$T_{ev}^K = T_{ev} + 273.15 ; T_{co}^K = T_{co} + 273.15$$

Then the theoretical Carnot COP is calculated:

$$COP_c^{th} = \frac{T_{ev}^K}{(T_{co}^K - T_{ev}^K)}$$

The exergetic COP (COP_{c,ex}) is calculated at the nominal conditions, assuming that the real COP can be expressed as a product of the theoretical and the exergetic COPs:

$$COP_c = COP_c^{ex} \cdot COP_c^{th}$$

For extrapolation of the real COP_c outside the nominal conditions (in the defined limit) the previously calculated COP_{c,ex} is assumed as a constant, and the real COP_c is calculated from:

$$COP_c^{real} = COP_c^{ex} \cdot COP_c^{th,real}$$

where COP_{c,t,real} is the theoretical Carnot COP for the real temperature working conditions.

Heating mode

In similar way the calculations are done for the COP in heating (heat pump) mode.

A correction is applied to the temperatures in order to approximate the primary fluid temperatures (7 K correction, corresponding to logarithmic mean difference of 4 K), and then the temperatures are converted to absolute temperature in [K]:

$$T_{ev} = T_{ev}^{inlet} - 7 K ; T_{co} = T_{co}^{inlet} + 7 K$$

$$T_{ev}^K = T_{ev} + 273.15 ; T_{co}^K = T_{co} + 273.15$$

The theoretical COP of the absorption heat pump is then calculated:

$$COP_h^{th} = \frac{T_{ev}^K}{(T_{co}^K - T_{ev}^K)} + 1$$

The exergetic COP_h for heat pump is calculated at the nominal conditions from:

$$COP_h = COP_h^{ex} \cdot COP_h^{th}$$

The real COP_h of the heat pump at different than the nominal conditions is calculated, considering the exergetic COP_h constant, from:

$$COP_h^{real} = COP_h^{ex} \cdot COP_h^{th, real}$$

4.7.3 Heat Pump Calculation Module

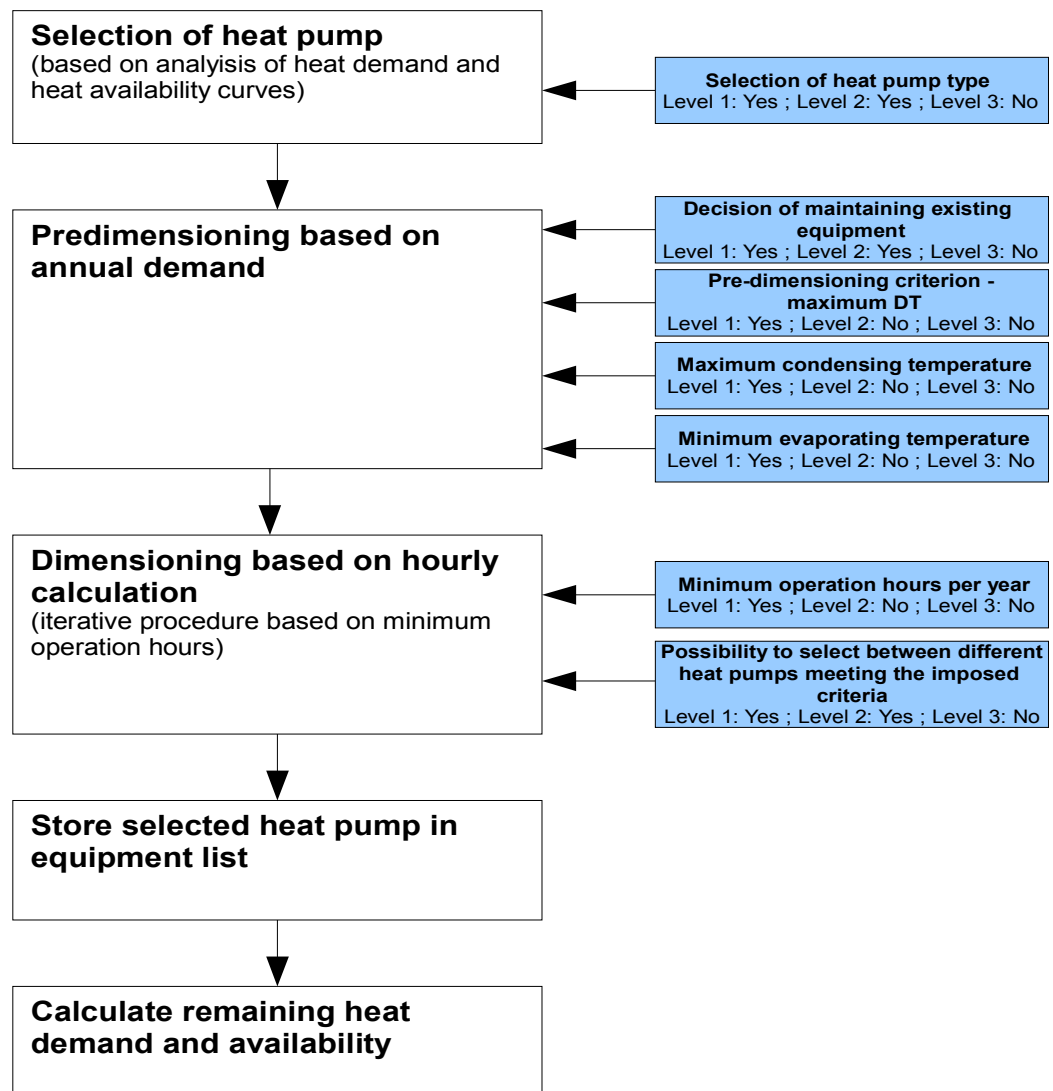
a) Information flow and user interactions

The module can operate with three levels of user interaction, that can be chosen from the main menu:

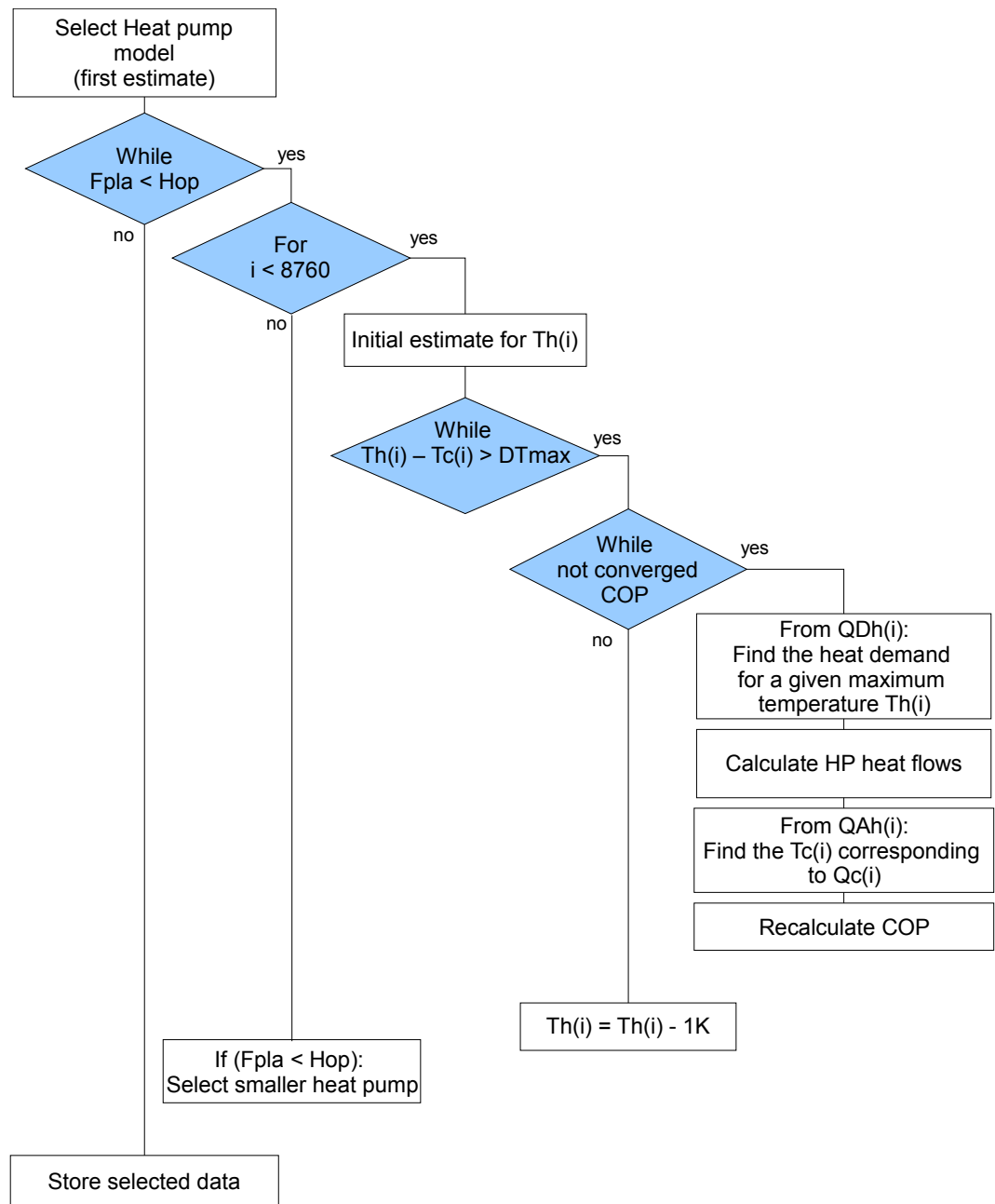
- Level 1 - Interactive mode requiring maximum user interactions for a personalized analysis.
- Level 2 - Semi-automatic mode requiring some user interactions, taking the rest of values as defaults from the set-up menu (stored in table e.g. SetUpHPDefaultValues).
- Level 3 - Automatic mode. In this mode the module runs without requiring user interaction, taking all the values as defaults from the set-up menu.

In the figure below is indicated with “yes” or “no” if user interaction is required for the corresponding level. When user decision is required, this is supported with help menus.

user interactions



b) Flowchart of calculation algorithm



c) Internal parameters description (Python code)

| | |
|-------------------|---|
| QD(h/a) | Hourly and annual temperature dependent heat demand curves [MWh] |
| QA(h/a) | Hourly and annual temperature dependent heat availability curves [MWh] |
| HPtype | Heat pump type: vapour-compression (COMP) or thermal-absorption (THERM) |
| Tmind | Minimum temperature of annual demand curve $QDa(T)$ (where demand = zero) |
| Tmaxa | Maximum temperature of annual availability curve $QAa(T)$ (availability = zero) |
| DTlim | Maximum temperature lift for the heat pump type (from database) (working limit for the condensing - evaporating temperature difference) |
| Qh° | Initial estimation of annual heat delivered by the heat pump [kWh]or[MWh] |
| dotQh° | Initial estimation of heating capacity ($\text{dot}Qh^\circ = Qh^\circ / (8760 \text{ or } Hop)$) [kW] |
| dotQh(i) | Calculated heating capacity for the i-th hour [kW] |
| dotQc(i) | Calculated cooling capacity for the i-th hour [kW] |
| dotQw(i) | Calculated power consumption of the heat pump for the i-th hour [kW] |
| j | Index for the position of the selected heat pump in the heat pump list from DBHeatPump (in ascending order by heat capacity) |
| dotQ(j,DB) | Nominal heating capacity for the j-th heat pump in the heat pump list [kW] |
| Fpla | Annual part-load factor (equal to annual operation hours for the heat pump) |
| Fpl(i) | Part-load factor for the i-th hour of the year |
| Th(i) | Condenser secondary fluid's inlet temperature (heating) |
| Th°(i) | Condenser secondary fluid's inlet temperature (heating) in the previous iteration |
| Tc(i) | Evaporator secondary fluid's inlet temperature (cooling) |
| Tc°(i) | Evaporator secondary fluid's inlet temperature (cooling) in the previous iteration |
| COPh(i) | Calculated real heating mode COP for the i-th hour |
| COPh°(i) | Calculated real heating mode COP for the i-th hour in the previous iteration |

4.8 *EINSTEIN – Boiler and Burner (BB) Module*

To be completed

4.9 EINSTEIN – Building Module

4.9.1 Mathematical description

4.9.1.1 Estimation of temporal distribution of heating, cooling and hot water demand from building parameters

The building energy demands (heating, cooling, hot water) are calculated based on the following parameters:

- yearly energy demand
- building occupation schedule
- heating and cooling periods specified

$$\dot{Q}(t) = \dot{Q}_{max} f_{daily}(t) f_{holidays}^{general}(t) f_{holidays}^{building}(t) f_{seasonal}(t)$$

where

- $\dot{Q}(t)$: instantaneous power requirement at time t
- $f_{daily}(t)$: daily variation, normalized so that the maximum value = 1
- $f_{holidays}^{general}(t)$: general holidays of the production site [binary 0/1]
- $f_{holidays}^{building}(t)$: building specific holidays where the building is unused [binary 0/1]
- $f_{seasonal}(t)$: seasonal variation, normalized so that maximum value = 1

The daily profile is used (a) for creating a realistic short term variation of the demand and (b) for normalizing the yearly integral of demand to the user input Q_{yearly} .

- hot water demand is considered constant throughout the hours of occupancy of the building (symmetric distribution around noon like for simple default schedules in processes). Peak power for hot water demand is not given as user input, so that no additional free parameter is needed for normalizing.
- for heating demand in current version a 24 h continuous operation is considered
- for cooling demands it is considered that the cooling system operates continuously at constant load during the building operation hours

The seasonal profile for heating and cooling is calculated using a sine function so that for $d_{start} \leq d \leq d_{stop}$:

$$f_{seasonal}(d) = \sin\left(\pi \frac{d - (d_{start} - 1)}{(d_{stop} + 1) - (d_{start} - 1)}\right)$$

and 0 elsewhere.

The seasonal profile for hot water demand is considered as constant.

4.9.1.2 Default process types and parameters used

The following process types are used for building demands:

- heating: maintenance and air renovation (circulation)

- cooling: maintenance and air renovation (circulation)
- hot water: continuous process with circulation heat (water) only at 50 / 10 °C

4.9.1.3 Air handling units / air renovation

A nominal air flow rate can be specified as input parameter. In this case the yearly maintenance heat demand of the building is obtained by:

$$Q_m = Q - Q_c = Q - q_m^{nom} c_p^{air} (T_{indoor} - T_{i,rec}) t_{op,eff}$$

Important note: due to current limitations of the EINSTEIN process model (constant temperature levels) in simulations the real behaviour of building air renovation (usually constant mass flow rate and variable (outdoor) temperature is approximated by an energetically equivalent stream with variable flow rate and constant (average) outdoor temperature. It is considered that the mass flow rate varies proportional to the total heat (or cooling demand).

4.10 Weather Data

4.10.1 Weather data files

Where available, the user can read hourly weather data from weather data files in TMY2 (typical meteorological year) format (for a description of this format see <http://rredc.nrel.gov/solar/pubs/tmy2/tables.html>)

4.10.2 Calculation from yearly averages

Where detailed hourly data are not available, EINSTEIN can generate artificial hourly distributions of solar radiation and ambient temperature based on yearly averages.

The correlations used are:

$$T_{amb}(t) = \bar{T}_{amb} - \hat{T}_{yearly} \cos\left(2\pi \frac{t - 45 * 24h}{8760}\right) - \hat{T}_{daily} \cos\left(2\pi \frac{t - 2h}{24}\right) \quad (4.10.1)$$

where:

\hat{T}_{yearly} : amplitude (half difference of minimum and maximum) of yearly variations in temperature. Default correlation: $\hat{T}_{yearly} = 10 K \frac{\theta}{40^\circ}$

\hat{T}_{daily} : amplitude (half difference of minimum and maximum) of daily variations in temperature. Default value: 5 K

θ : geographic latitude in degrees

The default correlations use a shift of the yearly maximum of temperature of 45 days with respect to the middle of the calendar year, and a shift of the daily maximum of 2 hours with respect to solar noon.

4.10.3 Default correlations

If no data on the average yearly ambient temperature are available, for fast assessment EINSTEIN can use the following default correlations based on the geographic latitude, which have been obtained from interpolation of data for some southern and central European cities:

$$\bar{T}_{amb} = 20.5 + 0.4668\theta - 0.0143\theta^2 \quad (4.10.2)$$

$$\bar{G} = -0.0002\theta^4 + 0.2547|\theta^3| - 31.055\theta^2 + 1245.4*|\theta| - 14419 \quad (4.10.3)$$

for :

θ : geographic latitude in degrees

5 Economic Analysis

To be completed