Zeller's manual for air-conditioning (4 editions from 1994 to 1999)

Now we write the year 2017 and there are still requests for this book, which has been out of print for many years and is no longer reissued. Meanwhile, electronic distribution channels have largely replaced books, with the advantage that necessary corrections and additions can be implemented much easier. The book was developed from the experiences in the field of climatic engineering when handling humid air since 1970. It did not contain any fundamentally new insights. It was more than one compendium for interested professionals, engineers, lecturers and students. Furthermore it has been used gladly and actively as a supplementary specialist book to our varied multilingual software.

AHH	Mollier diagram and Carrier psychrometric chart with air processes. Range -100/300°C, 0/1000 g/kg, -5000/15000 m, 0.03/16 bar. 150 meteorologcical data, further locations from Meteonorm. 3 different ranges of comfort zone from DIN and ASHRAE. Show your individual measurement points.
MDI	Meteorological data interface: Define the service times. Meteorological data based on Meteonorm.
AHU	Air-handling unit configurator: Element handling per drag and drop. Approximately measurements, weights, pressure drops, prices.
EAC	Economy of AHU's with circuit connected heat recovery systems. Variable air volume flows, amortization time, capital costs.
DEH	Economy of AHU's with different heat recovery systems. Variable air volume flows, amortization time, capital costs.
ESH	Glycol Re-Cooler with axial fans, outside or in AHU's. Dry, adiabatic and hybrid service, container measurements.
HEH	Calculation of fin coils like heater, cooler,condenser, evaporator, heater split, cooler split.
CCS	Calculation of heat recovery circuit systems with fin coils. Different systems with foreign energy in the circuit.
Diverse	GHH, Mollier diagram for different types of gases and steams. Spiral rib and circle rib heat exchangers.

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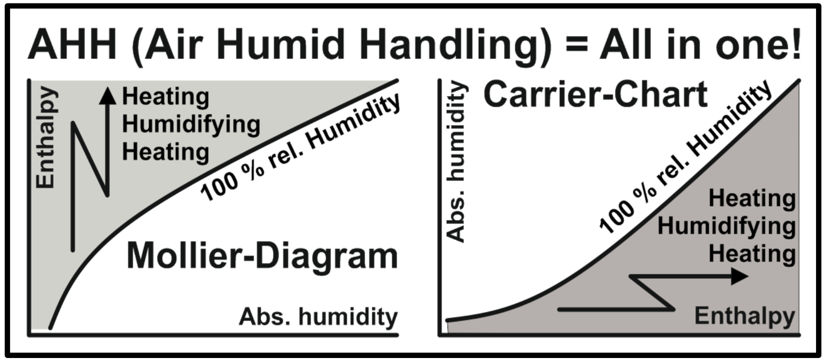

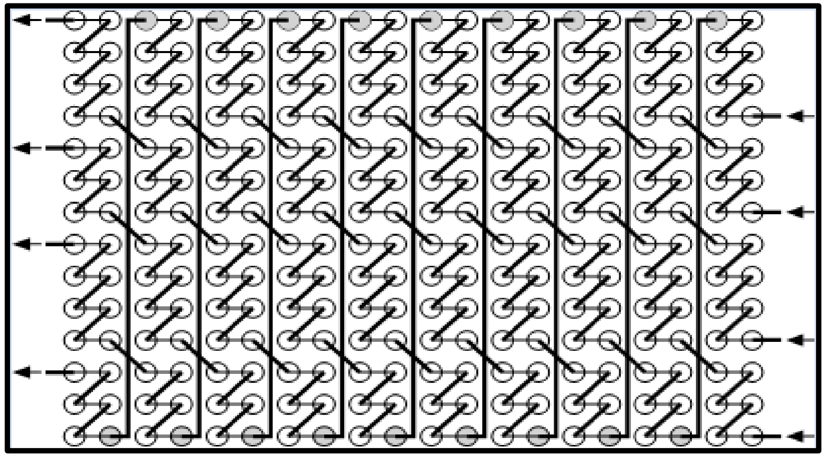

Mollier diagram / Carrier psychrometric chart (AHH) thematics

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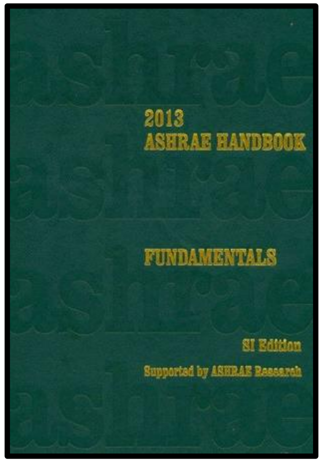

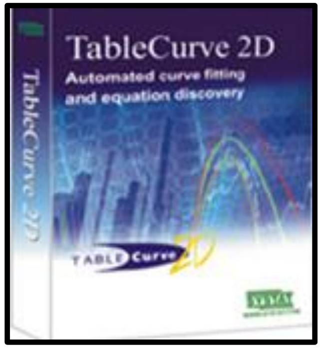

Dr.-Eng. Boris Slipcevic
Sulzer Escher Wyss
Lindau

Laboratory measurements were recorded by empirical equations. There followed numerous publications on heat transfer and pressure loss during condensation and evaporation (convective and bubble boiling) of refrigerants that flowed into the software.

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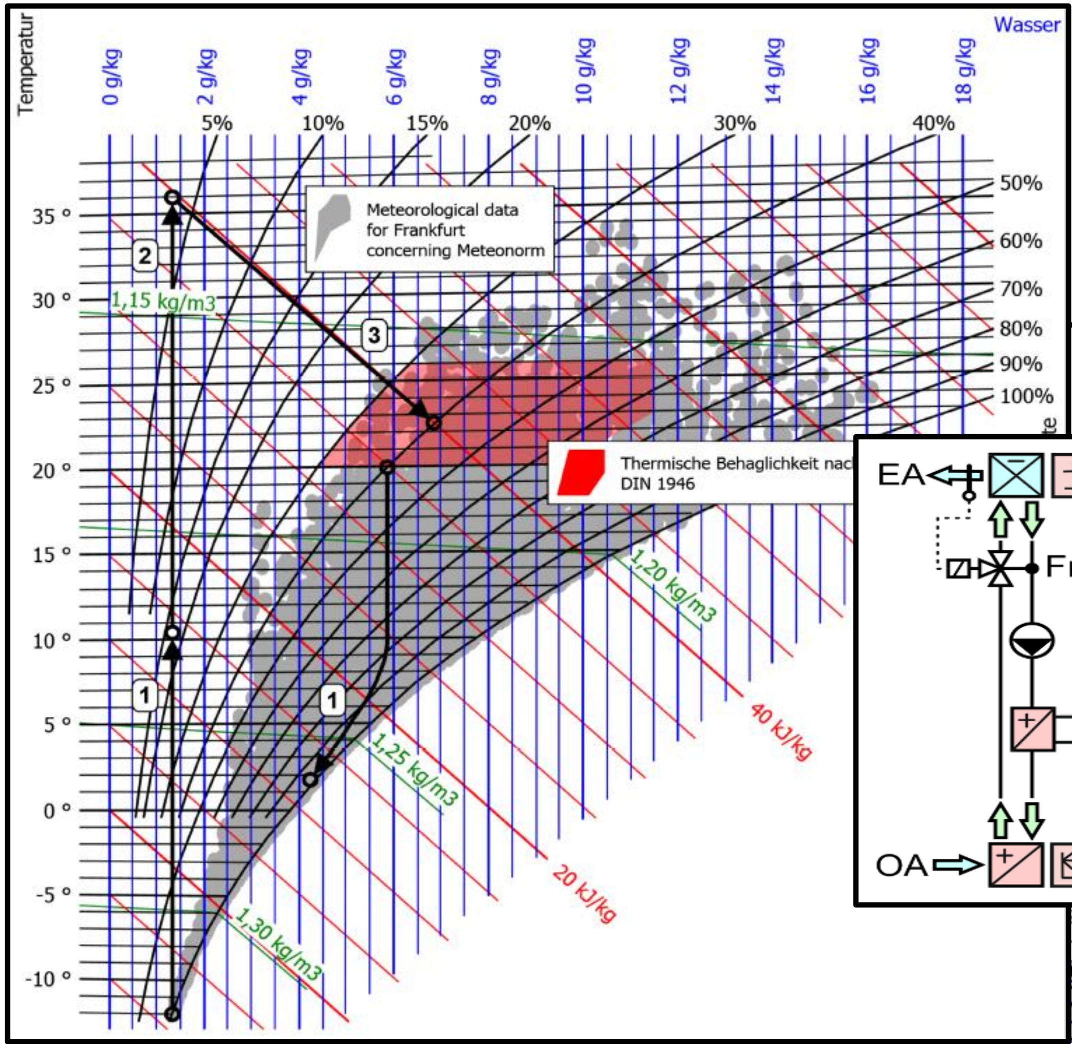
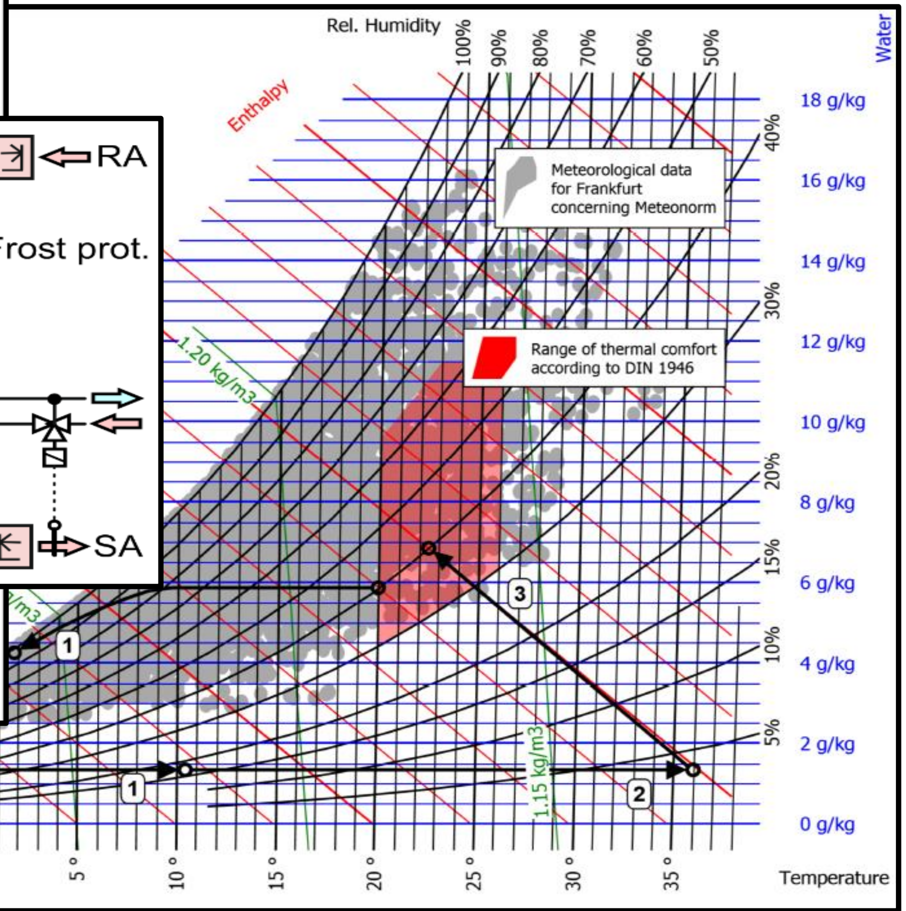
Ittigen

Laboratory measurements were recorded by empirical equations, especially to the lamella impression, to the contact efficiency between tubes and fins, to the cooling process with formation of condensate, the surface temperature and to the pressure drop wet of the moist air.

Representation according to Mollier



Representation according to Carrier



Process data to the moist air

Definition

The Mollier-hx-diagram represents the air water mixture. It is in such a way developed that the 0 °C isotherm is horizontal in the range of unsaturated air. The nebula isotherm of over saturated air by 0 °C go parallel to the enthalpy. On t = 0 °C and x = 0 kg/kg the enthalpy h = 0 J/kg, which leads to ranges with negative enthalpies. By exchange of the centerlines one receives the Carrier-xh-Diagram (Psychrometric chart) within the software AHH to be alternatively worked can.

Area

Usually the Mollier-hx-Diagram is based on a pressure of 1.013 bar according to sea level and exhibits a range, which permits not all applications. The software AHH permits the desired range for each application and supports the good clarity because of each stretching of the axes of coordinates.

Temperature	-100 to 300°C
Absolute humidity	0 to 1000 g/kg
Pressure absolute	0.1 to 16 bar
Height	-5000 to 15000 m

Thermodynamic properties

In specialized books one usually finds the specific temperature-referred thermal capacity. This value points out, how much energy must be spent, in order to warm up the medium with appropriate temperature around 1°C. If one wants to know, which energy is needed, in order the medium of t_1 to t_2 to warm up, the means of the specific temperature-referred thermal capacity must be determined. Below the average values were formed of 0 °C to t °C and combined into tables and approximation polynomials, which make a fast processing possible on computers.

1) Heat recovery - Circuit connect-system

Efficiency temperature	%	70.000
Efficiency hygroscopic	%	0.000
Efficiency humid	%	0.000
Capacity	kW	62.759
Mean temp.diff.	K	11.560
Coefficient	kW/K	5.429

		Cold air In	Cold air Out	Hot air In	Hot air Out
Temperature	°C	-12.000	10.400	20.000	1.771
Rel. Humidity	%	100.000	17.090	40.000	97.451
Abs. Humidity	g/kg	1.346	1.346	5.872	4.255
Density humid	kg/m³	1.330	1.225	1.182	1.261
Enthalpy humid	kJ/kg	-8.737	13.857	35.028	12.435
Volumeflow humid	m³/h	7527.666	8173.325	8511.363	7961.565
Massflow dry	kg/h	10000.000	10000.000	10000.000	10000.000
Condensed water	kg/h		0.000		16.170
Surface temperature	°C				-3.324

Danger of FREEZING!

2) Heating

Capacity	kW	71.780

		Air In	Air Out
Temperature	°C	10.400	36.000
Rel. Humidity	%	17.090	3.641
Abs. Humidity	g/kg	1.346	1.346
Density humid	kg/m³	1.225	1.124
Enthalpy humid	kJ/kg	13.857	39.697
Volumeflow humid	m³/h	8173.325	8911.214
Massflow dry	kg/h	10000.000	10000.000

3) Moistening of air with water

Capacity	kW	0.952
Moistening flow	kg/h	55.020
Moistening temperature	°C	15.000
Moistening enthalpy	kJ/kg	62.302

		Air In	Air Out
Temperature	°C	36.000	22.488
Rel. Humidity	%	3.641	40.000
Abs. Humidity	g/kg	1.346	6.848
Density humid	kg/m³	1.124	1.171
Enthalpy humid	kJ/kg	39.697	40.040
Volumeflow humid	m³/h	8911.151	8596.927
Massflow dry	kg/h	10000.000	10000.000

Mollier diagram / Carrier psychrometric chart

Thermodynamic properties

t	cp_l	cp_d	p_d	h_w	h_d	r
-100	1007.20	1815.40	0.00160			
-90	1006.90	1817.50	0.00933			
-80	1006.63	1819.60	0.05333			
-70	1006.40	1821.70	0.258			
-60	1006.20	1823.80	1.076			
-50	1006.07	1826.00	3.939			
-40	1006.00	1828.10	12.870			
-30	1005.97	1830.30	38.101			
-20	1006.00	1832.50	103.450			
-10	1006.08	1834.70	259.980			
0	1006.18	1836.90	610.480	0	2500500	2500500
10	1006.31	1839.10	1230	42000	2518900	2476900
20	1006.45	1841.40	2340	83900	2537300	2453400
30	1006.60	1843.70	4240	125600	2555500	2429900
40	1006.81	1846.00	7370	167300	2573500	2406200
50	1007.03	1848.30	12300	209100	2591300	2382200
60	1007.30	1850.60	19900	250900	2608800	2357900
70	1007.60	1852.90	31100	292800	2625900	2333100
80	1007.90	1855.30	47300	334700	2642500	2307800
90	1008.30	1857.70	70100	376800	2658700	2281900
100	1008.70	1860.10	101300	418900	2674400	2255500
110	1009.00	1862.50	143300	461100	2689600	2228500
120	1009.50	1864.90	198500	503500	2704200	2200700
130	1009.90	1867.30	270100	546100	2718300	2172200
140	1010.30	1869.80	361400	588900	2731800	2142900
150	1010.80	1872.30	476000	631900	2744500	2112600
160	1011.30	1874.80	618000	675200	2756500	2081300
170	1011.80	1877.30	792000	718800	2767600	2048800
180	1012.40	1879.80	1002700	762700	2777600	2014900
190	1013.00	1882.40	1255200	807000	2786300	1979300
200	1013.60	1884.90	1555100	851800	2793700	1941900
210	1014.20	1887.50	1908000	897100	2799400	1902300
220	1014.80	1890.10	2320100	943000	2803400	1860400
230	1015.50	1892.70	2797900	989600	2805400	1815800
240	1016.20	1895.30	3348000	1036900	2805100	1768200
250	1016.90	1898.00	3978000	1085100	2802500	1717400
260	1017.60	1900.60	4694000	1134300	2797400	1663100
270	1018.40	1903.30	5505000	1184500	2789500	1605000
280	1019.20	1906.00	6419000	1236100	2778700	1542600
290	1020.10	1908.70	7445000	1289300	2764900	1475600
300	1021.00	1911.40	8592000	1344200	2748000	1403800

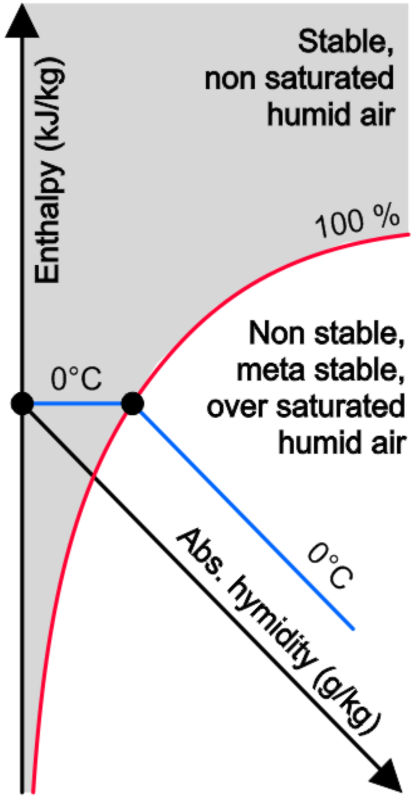
Symbol	Unit	Description
cp_d	J/kgK	Heat capacity from water vapor on Solidus
cp_l	J/kgK	Heat capacity from dry air
h_d	J/kgK	Enthalpy from water vapor on Solidus
h_w	J/kgK	Enthalpy from water vapor on Liquidus
p_d	Pa	Partial pressure from water vapor
r	J/kgK	Evaporation heat from water vapor
t	°C	Temperature

$$cp = \frac{\int_{t_1}^{t_2} cp_t dt}{t_2 - t_1}$$

$$t_1 = 0\ and t_2 = t \rightarrow cp = \frac{\int_0^t cp_t dt}{t}$$

Richard Mollier
1863 - 1935

W. H. Carrier
1876 - 1950



Approximation polynomials (-100 < t < 300°C)

The application TableCurve 2D offers more than 7000 equations, sorted according to the smallest error rate. However, not all equations are suitable to the same extent, but 3 criteria must be observed:

1. The waves within the data should be minimal.
2. Within the data, singularities are to be excluded.
3. The trend outside of the data should be continuous.

```
a =  1.00617203411816E+03
b = -5.14584155927084E-04
c = -5.07744861271335E-01
d = -4.08693984761444E-06
e = -3.94830238325583E-03
f =  3.86998536082132E-10
```

$$cp_l = \frac{a + ct + et^2}{1 + bt + dt^2 + ft^3}$$

```
a =  1.83690225155577E+03
b =  2.96850242760703E-04
c =  7.68576185706328E-01
d = -8.23605125618347E-08
```

$$cp_d = \frac{a + ct}{1 + bt + dt^2}$$

```
a =  6.41424538282508E+00
b =  1.34952974449424E-02
c =  1.68771989526873E-01
d =  3.60425763984253E-05
e =  1.03764255356861E-03
f = -2.54470285416322E-09
```

$$-100 < t \leq 0 \rightarrow ln(p_d) = \frac{a + ct + et^2}{1 + bt + dt^2 + ft^3}$$

```
a =  6.41425292688508E+00
b = -1.17398221741019E-02
c = -2.31391504282494E-03
d =  5.85324578180939E-05
e = -7.97076080224934E-04
f =  3.14129016158240E-07
g =  1.13870924045918E-05
h = -1.05815083120807E-09
i = -2.45705078974294E-08
```

$$0 > t \leq 300 \rightarrow ln(p_d) = \frac{a + ct + et^2 + gt^3 + it^4}{1 + bt + dt^2 + ft^3 + ht^4}$$

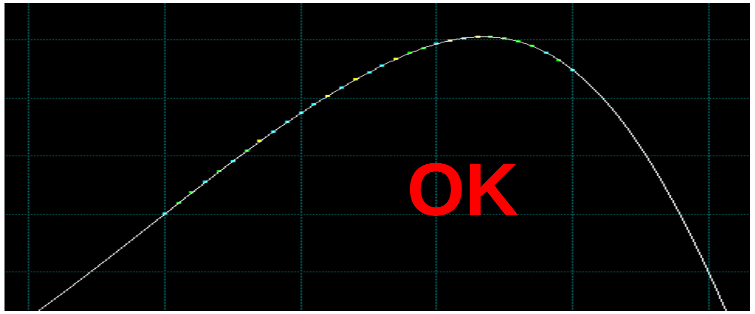
```
a =  3.63051146855678E+00
b = -2.04547872985726E-03
c =  4.19231841629432E+03
d = -8.10031771231269E-07
e = -8.90297066641696E+00
f =  9.66233360384174E-10
```

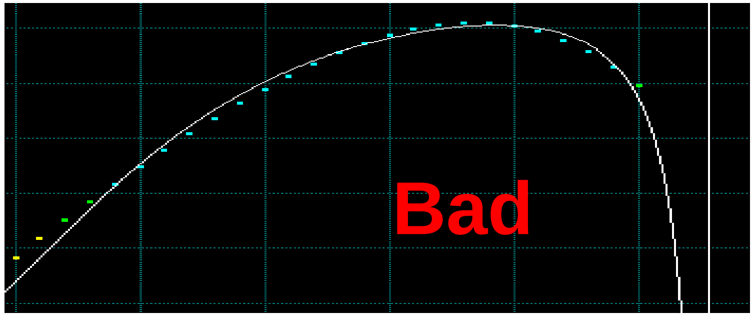
$$h_w = \frac{a + ct + et^2}{1 + bt + dt^2 + ft^3}$$

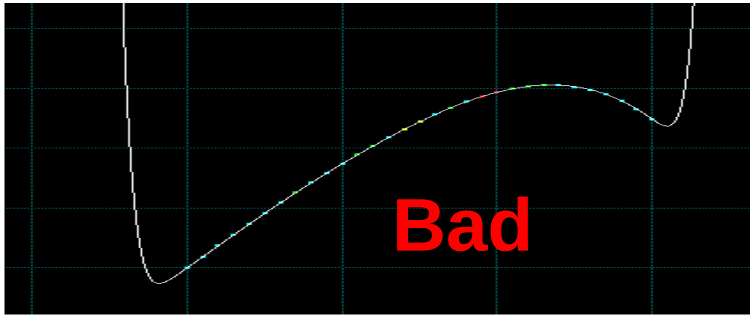
```
a =  2.50049979241906E+06
b = -1.09042949248609E-02
c = -2.54218801181540E+04
d =  6.03477944019292E-05
e =  1.30688801015895E+02
f = -1.24219636812250E-07
g = -2.06485565012501E-01
h =  1.15773454895717E-10
i =  1.21390880909374E-04
```

$$h_d = \frac{a + ct + et^2 + gt^3 + it^4}{1 + bt + dt^2 + ft^3 + ht^4}$$

$$r = h_d - h_w$$







Mollier Diagramm / Carrier Psychrometric Chart

Equations for moist air

The air pressure depends on the height over sea level, the temperature and the humidity. As basis for Mollier diagram and Carrier chart the air pressure is to determine by the height over sea level, the middle yearly temperature (average 10°C for middle Europe) and the middle yearly humidity (average 80% for middle Europe).

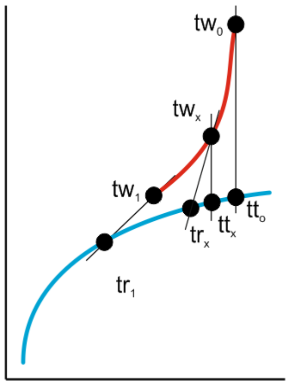
$$T_{lf} = t_{lf} + 273.16 \qquad z = \frac{M_l g H}{R T_{lf}} \frac{1+x}{1+x\frac{M_l}{M_w}} \rightarrow p_{lf} = 1.01325 e^{-z}$$

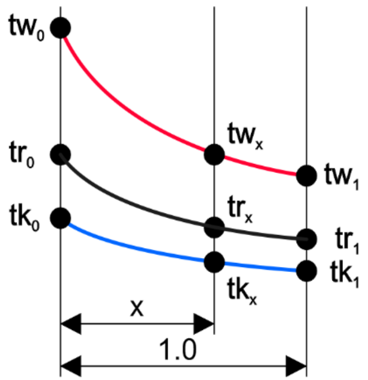
$$\rho_{lf} = \frac{M_l p_{lf}}{R T_{lf}} \frac{1+x}{1+x\frac{M_l}{M_w}} \qquad x_s = \frac{M_w}{M_l} \frac{p_d}{p_{lf} - p_d} \qquad x = \frac{M_w}{M_l} \frac{\varphi_{lf} p_d}{p_{lf} - \varphi_{lf} p_d}$$

$$\varphi_{lf} = \frac{p_{lf} x \frac{M_l}{M_w}}{p_d \left(1 + x\frac{M_l}{M_w}\right)} \qquad h_{lf} = cp_l t + x(r_0 + cp_d t) \qquad \dot{Q}_{lf} = \dot{M}_{lt} \Delta h_{lf}$$

$$t_{lf} = \frac{h_{lf} - x r_0}{cp_l + x cp_d} \qquad \dot{M}_{lt} = \frac{\dot{V}_{lf} \rho_{lf}}{1+x} \qquad \dot{V}_{lf} = \frac{\dot{M}_{lt}(1+x)}{\rho_{lf}}$$

Cooling process





Symbol	Unit	Description
H	m	Height over sea level
h_{lf}	J/kg	Enthalpy from the moist air
M_l	kg/kMol	Molecular weight from air = 28.96 kg/kMol
$\dot{M}_{lt}$	kg/s	Massflow from the dry air
M_w	kg/kMol	Molecular weight from water = 18.02 kg/kMol
p_{lf}	Pa	Pressure from the moist air
φ_{lf}	kg/m3	Relative humidity from the moist air
$\dot{Q}_{lf}$	W	Capacity with moist air
R	J/kMolK	Universal gas constant = 8314.41 J/kMolK
$r_{0(0°C)}$	J/kg	Evaporation heat from water vapor = 2500500 J/kg
ρ_{lf}	kg/m3	Density from the moist air
t_{lf}	°C	Temperature from the moist air
T_{lf}	K	Temperature from the moist air
$\dot{V}_{lf}$	m3/s	Volume flow from the moist air
x	kg/kg	Absolute humidity from the moist air
x_s	kg/kg	Maximally absolute humidity from the moist air

In the software the cooling process in the heat exchanger is divided in 15 cells in air direction.Here, it is assumed that a high degree of cross-counter-flow. The surface temperature trx plays in each cell a crucial role. When this is less than the dew point ttx, condensate forms. For small ttx-trx, the droplets of condensate are small too. These can be separated in a 1st demister only, which produce bigger droplets. A 2nd demister can separate this big droplets. Demisters with less than 100 Pa pressure drop have a bad separation. This play the big rule, when the dehumidification during the cooling process is important.

Humidification (Water, wet steam, saturated steam)

The humidification direction is carried out in the Mollier diagram on paper with the aid of the edge measuring rod. This is not possible in the software AHH because the two axes can be freely selected.

$$h_{b(Wasser)} = h_w \quad h_{b(Nassdampf)} = h_w + a_d r \quad h_{b(Sattdampf)} = h_d$$

$$\Delta h = h_a - h_e = \Delta x h_b = h_b \frac{\dot{M}_b}{\dot{M}_l} \qquad \Delta x = x_a - x_e = \frac{\dot{M}_b}{\dot{M}_l} = \frac{\Delta h}{h_b}$$

$$\dot{M}_b = \Delta x \dot{M}_l \qquad \dot{Q} = \Delta h \dot{M}_l$$

Examples to the Mollier diagram on the right

Humidification with water of 0°C

$$h_b = h_w = 0\,J/kg$$

Humidification with water of 50°C

$$h_b = h_w = 209'100\,J/kg$$

Humidification with wet steam of 110°C, wet part of steam 50 %

$$h_b = h_w + a_d r = 461'100 + 0.5 \cdot 2'228'500 = 1'575'350\,J/kg$$

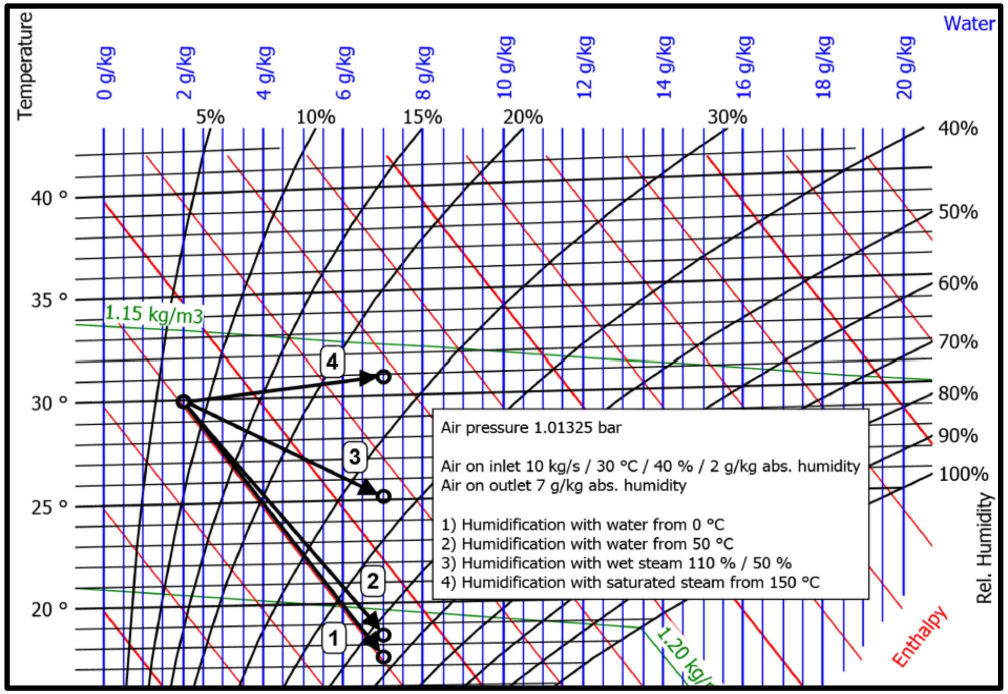
Humidification with saturated vapor of 150°C

$$h_b = h_d = 2'744'500\,J/kg$$

Note that moistening with constant enthalpy can only be achieved with water of 0°C. During humidification with water > 0°C the enthalpy increases, albeit little.

If the desired relative humidity at the outlet is desired as an input value, this must be done by means of iteration, which is the case in the software.

Symbol	Unit	Description
a_d	---	Wet part of steam
Δh	J/kg	Enthalpy difference
Δx	kg/kg	Humidity difference
h_a	J/kg	Enthalpy outlet
h_b	J/kg	Humidification enthalpy
h_e	J/kg	Enthalpy inlet
$\dot{M}_b$	kg/s	Humidification mass flow
x_a	kg/kg	Humidity outlet
x_e	kg/kg	Humidity inlet



Meteorological data and a correct cooler design for sweltering midsummer

The German standard DIN 4710 recorded 87'600 events à 0.1 hours per year as average values of the period from 1961 to 1990 and therefore forms a large area in the Mollier diagram. The software from Meteonorm recorded 8'760 events à 1.0 hours, and therefore forms a smaller area in the Mollier diagram than the German standard DIN 4710 what is a risk in the design of coolers in sweltering midsummer.

A = 32.0°C / 40 % / 63.0 kJ/kg = Usual calculation
B = 31.0°C / 58 % / 73.7 kJ/kg = Meteonorm for 2020
C = 32.0°C / 65 % / 82.9 kJ/kg = DIN 4710, 1961-1990
D = 12.3°C / 100 % / 35.2 kJ/kg = Cooling
E = 18.0°C / 69 % / 41.0 kJ/kg = Reating

A - D: Big risk management

63.0 - 35.2 = 27.8 kJ/kg = 58.28 %

Cooler size loss = 41.72 %

B - D: Middle risk management

73.7 - 35.2 = 38.5 kJ/kg = 80.71 %

Cooler size loss = 19.29 %

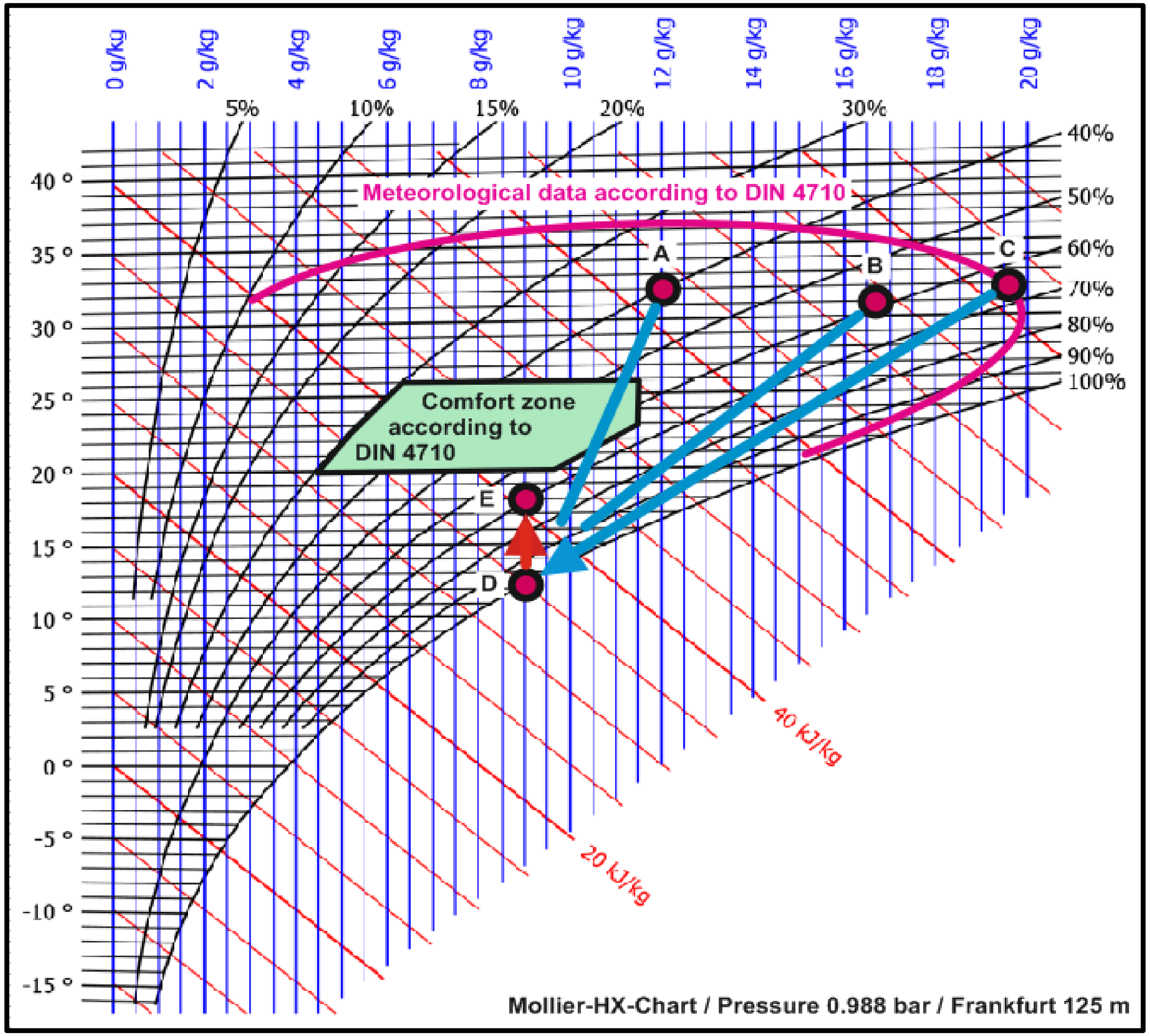
C - D: No risk management

82.9 - 35.2 = 47.7 kJ/kg = 100.00 %

Correct cooler size

Summary

The correct cooler calculation does not depend on the highest summer temperature but on the highest enthalpy in the summer.



AHU: Determine air-conditioning units. Drag and drop the elements

The neutral configurator for air-handling units shows standard values for the weight, the dimensions, the pressure drop and the price of each component of 2 air-handling units.

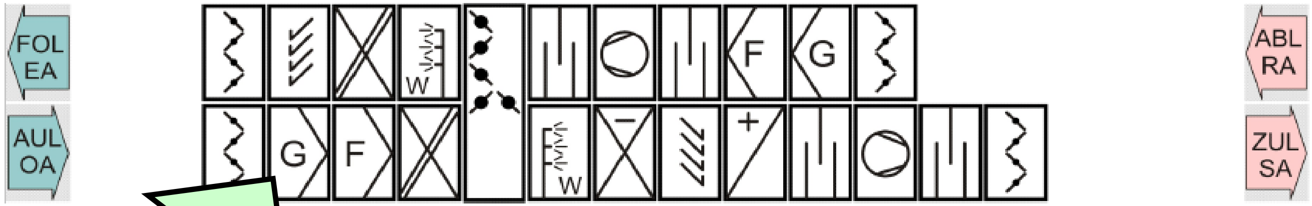
Regarding internal usable width and height, the configurator it based on standard filter dimensions of 610 x 610 mm or divisible units.

The individual components you do not must calculate thermodynamically. The values based on average default values.

After entering the air quantity and the maximum permissible speeds, based on the air filter we offer a variety of dimensions.

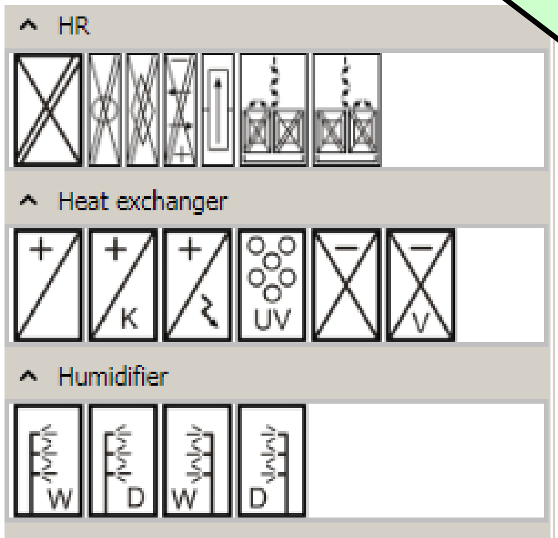
You can select by drag and drop, the individual components and enter the external pressure drops.

As a result, all relevant data is obtain for the two air-handling units and this at a time expenditure of just a few minutes.

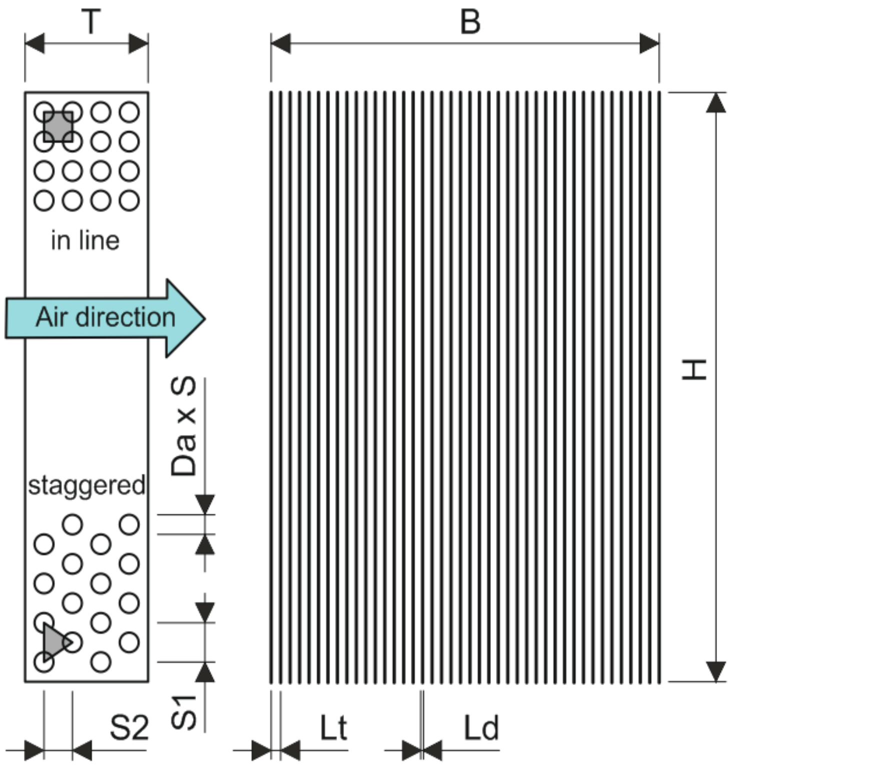


Air-Handling U... = 1960 x 2570 mm) Outside air (2... - Filter 1.87)	Length mm	Weight kg	Pressure drop Pa	Price EUR
Outside ai...			100.00	
Empty ... ith flaps	350.00	100.00	27.00	2180.00
Filter	450.00	140.00	109.00	2290.00
Filte...	650.00	210.00	144.00	3390.00
CC...	650.00	590.00	87.00	9250.00
E... big with flaps	350.00	80.00	18.00	1160.00
... water	1300.00	300.00	88.00	5560.00
	400.00	320.00	63.00	5280.00
...t separator	150.00	70.00	88.00	1500.00
...er	200.00	200.00	23.00	3270.00
...nd absorber	1300.00	300.00	53.00	5560.00
...n - Efficiency 88.00 % - Capacity 12.01 kW	2200.00	780.00	88.00	12900.00
Sound absorber	1300.00	300.00	53.00	5560.00
Empty part little with flaps	350.00	100.00	27.00	2180.00
Supply air			300.00	
Total	9650.00	3490.00	1268.00	60080.00

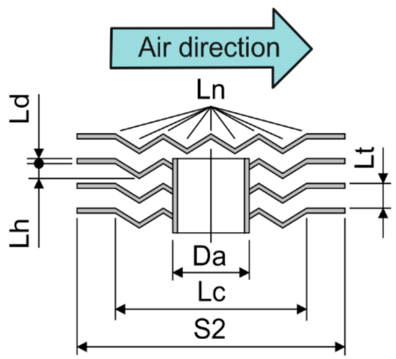
Air-Handling Unit (H x W = 1960 x 2570 mm) Return air (28000 m³/h - Filter 1.74)	Length mm	Weight kg	Pressure drop Pa	Price EUR
Return air			150.00	
Empty part little with flaps	350.00	100.00	23.00	2180.00
Filter G	450.00	140.00	95.00	2290.00
Filter F	650.00	210.00	126.00	3390.00
Sound absorber	1300.00	300.00	46.00	5560.00
Fan - Efficiency 75.00 % - Capacity 9.75 kW	2200.00	780.00	76.00	12900.00
Sound absorber	1300.00	300.00	46.00	5560.00
Empty part big with flaps	350.00	80.00	16.00	1160.00
Humidifier water	1300.00	300.00	76.00	5560.00
CC-System	650.00	590.00	87.00	9250.00
Droplet separator	150.00	70.00	76.00	1500.00
Empty part little with flaps	350.00	100.00	23.00	2180.00
Exhaust air			100.00	
Total	9050.00	2970.00	940.00	51530.00



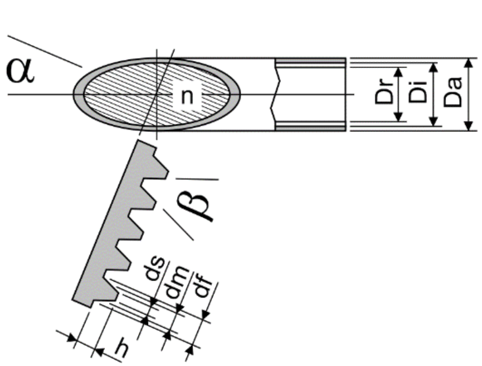
Fin coil heat exchanger



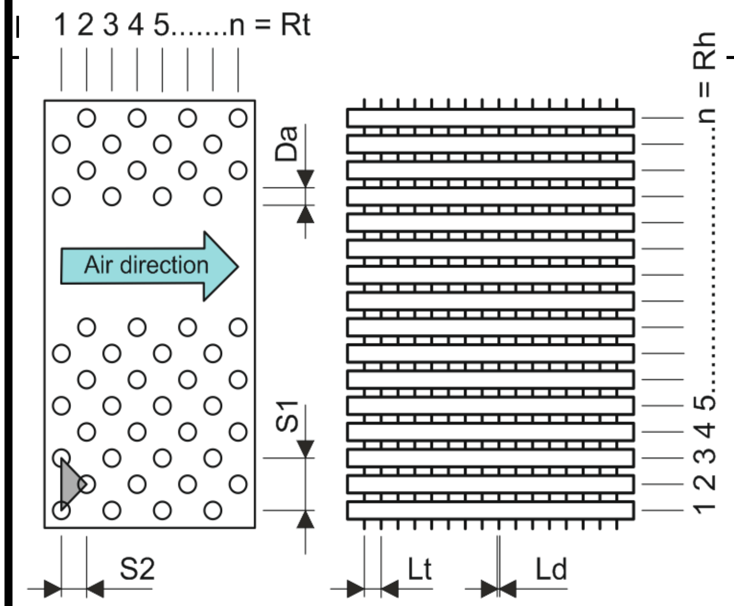
Corrugated fins



Heat exchanger tubes inside grooved



Symbol	Unit	Description
α	°	Inside grooved tubes twist angle
B	m	Fin package width
β	°	Inside grooved tubes flank angle
Da	m	Heat exchanger tubes outside diameter
Di	m	Heat exchanger tubes inside diameter
Dr	m	Heat exchanger tubes grooves diameter
df	m	Grooves thickness at the foot
dm	m	Grooves thickness average
ds	m	Grooves thickness on the head
H	m	Fin package height
h	m	Grooves height
Lc	m	Fin corrugation depth per tube row
Ld	m	Fin thickness
Lh	m	Fin corrugation height
Ln	m	Number of fin creases per tube row
Lt	m	Fin spacing
n	---	Number of grooves
S	m	Heat exchanger tubes thickness
S1	m	Heat exchanger tubes interval on the height
S2	m	Heat exchanger tubes on the depth
T	m	Fin package depth



$\eta_{KRL} = 1.00\ \ for\ fin\ thickness \geq 0.20\ mm$

$\eta_{KRL} = .95\ \ or\ fin\ thickness \geq 0.18\ and < 0.20\ mm$

$\eta_{KRL} = 0.90\ \ for\ fin\ thickness \geq 0.16\ and < 0.18\ mm$

$\eta_{KRL} = 0.85\ \ for\ fin\ thickness \geq 0.14\ and < 0.16\ mm$

$\eta_{KRL} = 0.80\ \ for\ fin\ thickness \geq 0.12\ and < 0.14\ mm$

$\eta_{KRL} = 0.75\ \ for\ fin\ thickness \geq 0.10\ and < 0.12\ mm$

p= $4R[(L_t - L_d)D_a\pi + 2L_dS_1S_2]/(L_tS_1S_2)$ q $= 1 - e^{-p}$

$\eta_{BPL} = q^{0.25}$ r $= L_d\lambda_l/(A_a/A_i)$ s $= 1 - e^{-r}$ $\eta_{WRL} = s^{0.25}$

$\eta_{WTT} = \eta_{KRL}\eta_{BPL}\eta_{WRL}$ $\dot{Q} = \dot{M}_{lt}\Delta h_{lf}$ $\dot{Q} = \dot{M}\Delta h_m$

$\dot{Q} = k_aA_a\Delta t_m$ $\frac{1}{k_a} = \frac{1}{\propto_a} + f_a + \frac{A_a}{A_i}\frac{\delta_w}{\lambda_w} + \frac{A_a}{A_i}\frac{1}{\propto_i} + \frac{A_a}{A_i}f_i$

Symbol	Unit	Description
A_a	m2	Heat exchanger surface outside
A_i	m2	Heat exchanger surface inside
$\propto_a$	W/m2K	Heat transfer coefficient outside
$\propto_i$	W/m2K	Heat transfer coefficient inside
Δh_{lf}	J/kg	Enthalpy difference of the moist air
Δh_m	J/kg	Enthalpy diff. from the heating or cooling agent
Δt_m	K	Average logarithmic temperature difference
δ_w	m	Heat exchanger tubes thickness
η_{BPL}	---	Bypass efficiency between the fins
η_{KRL}	---	Contact efficiency tube / fins
η_{WRL}	---	Heat transfers efficiency tube / fins
η_{WTT}	---	Heat exchanger efficiency total
f_a	m2K/W	Fouling factor outside
f_i	m2K/W	Fouling factor inside
k_a	W/m2K	K-coefficient concerning the outside surface
λ_w	W/mK	Heat exchanger tubes thermal conductivity
$\dot{M}$	kg/s	Mass flow from the heating or cooling agent
$\dot{M}_{lt}$	kg/s	Mass flow from the dry air
$\dot{Q}$	W	Capacity
R_h	Piece	Number of tube rows on the height
R_t	Piece	Number of tube rows on the depth

Fin coil heat exchanger	Symbol	Unit	Description
Average logarithmic temperature difference	t_{11}	°C	Inlet temperature hot agent
	t_{12}	°C	Outlet temperature hot agent
	t_{21}	°C	Inlet temperature cold agent
	t_{22}	°C	Outlet temperature cold agent
	n	Piece	Number of counter flow packages
	i	---	Cell number

Average logarithmic temperature difference

Counter flow

$\Delta t_1 = t_{11} - t_{22}$ $\quad \Delta t_2 = t_{12} - t_{21}$ $\quad \Delta t_1 = \Delta t_2 \rightarrow \Delta t_m = \Delta t_1$

$\Delta t_1 \neq \Delta t_2 \rightarrow \Delta t_m = (\Delta t_1 - \Delta t_2)/ln(\Delta t_1/\Delta t_2)$

Equal flow

$\Delta t_1 = t_{11} - t_{21}$ $\quad \Delta t_2 = t_{12} - t_{22}$ $\quad \Delta t_1 = \Delta t_2 \rightarrow \Delta t_m = \Delta t_1$

$\Delta t_1 \neq \Delta t_2 \rightarrow \Delta t_m = (\Delta t_1 - \Delta t_2)/ln(\Delta t_1/\Delta t_2)$

Cross flow

$\mathrm{p} = (t_{11} - t_{12})/(t_{11} - t_{21})$ $\quad \mathrm{q} = (t_{22} - t_{21})/(t_{11} - t_{21})$

Cross flow 1-time

$\mathrm{r} = 1 - (q/p)ln(1/(1-p))$ $\quad s = q/ln(1/r)$

$\Delta t_m = s(t_{11} - t_{21})$

Cross counter flow 2-times

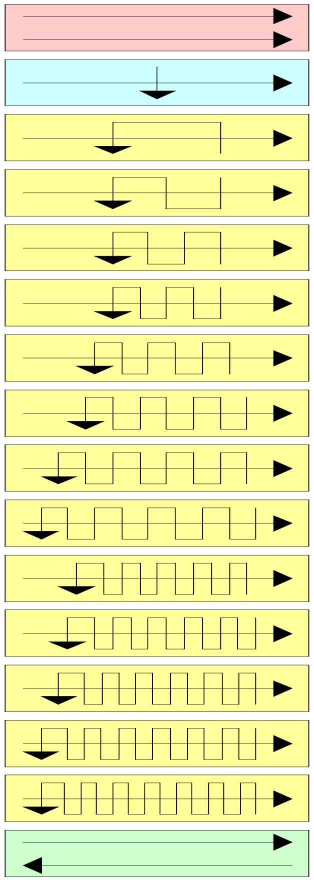
$\mathrm{p} = q \rightarrow x = 1 - p/\big(2(p-1)\big)$

$$\mathrm{p} \neq q \rightarrow x = \frac{\sqrt{(1-q)/(1-p)} - q/p}{1 - q/p}$$

$\mathrm{r} = 1 - (q/p)ln\mathrm{x}$ $\quad s = q/\big(2ln(1/r)\big)$ $\quad \Delta t_m = s(t_{11} - t_{21})$

Cross counter flow n-times

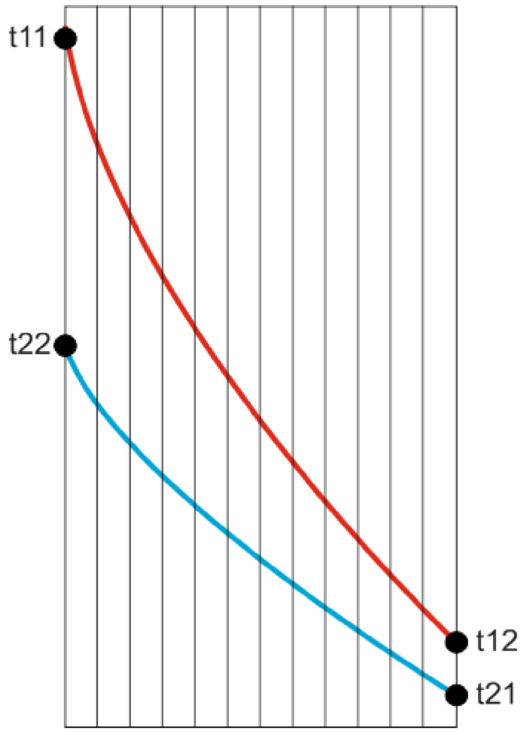
$$\Delta t_m = \dot{Q} \Big/ \sum_{i=1}^{n} k_{a(i)}\, A_{a(i)}$$



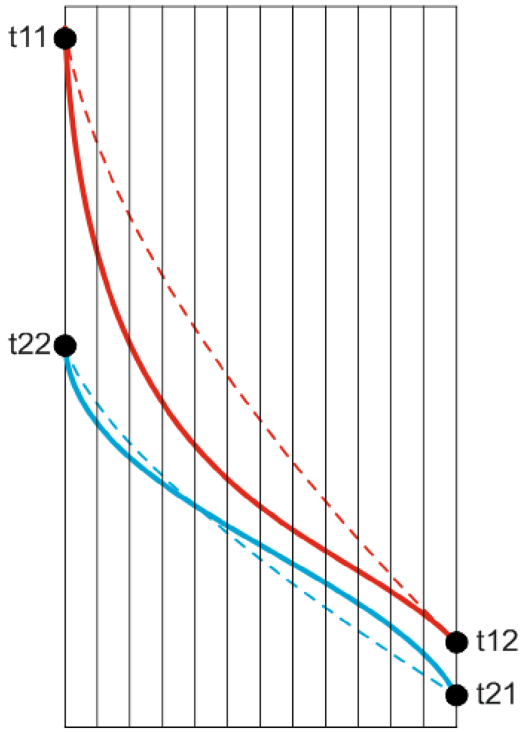
Equal
Cross flow 1-time
Cross counter flow 2-times
Cross counter flow 3-times
Cross counter flow 4-times
Cross counter flow 5-times
Cross counter flow 6-times
Cross counter flow 7-times
Cross counter flow 8-times
Cross counter flow 9-times
Cross counter flow 10-times
Cross counter flow 11-times
Cross counter flow 12-times
Cross counter flow 13-times
Cross counter flow 14-times
Counter flow

Fin coil heat exchanger	Symbol	Unit	Description	Page 14
Cross counter flow n-times	A_l	m2	Heat exchanger fin surface	

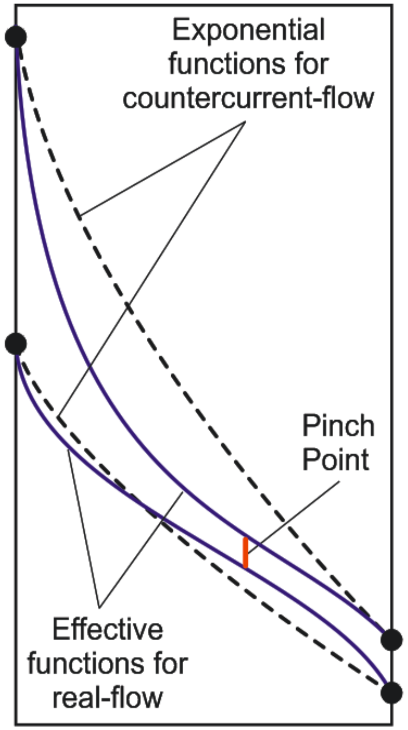
Cooling process sensible only
Temperature profile exponential
Optimal temperature difference
Cells i = 1, 2, 3, 4, 5, ...n

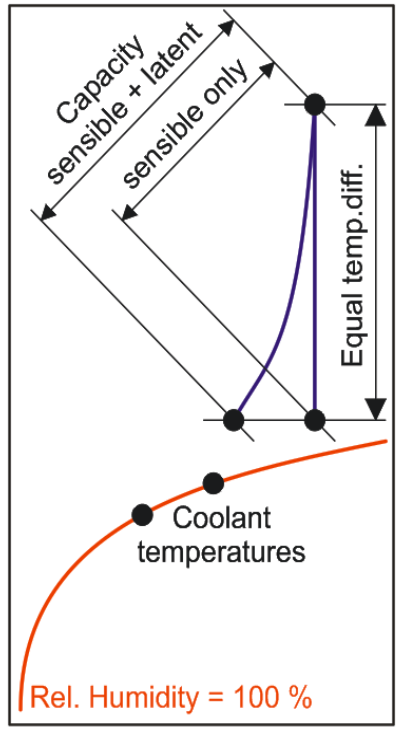


Cooling process sensible & latent
Temperature profile deformed
Reduced temperature difference
Cells i = 1, 2, 3, 4, 5, ...n



Heat exchanger manufacturers, which do not calculate so and specify the Δtm for countercurrent, constantly supply to small air coolers, which is still too often the case, one hopes nevertheless cynically that it will not be measured.





Smooth fins

$$A_l = 2\left(S_1 S_2 - \frac{{D_a}^2 \pi}{4}\right) R_t R_h \left(\frac{B}{L_t} - 1\right)$$

Corrugated fins

Flat or marginal corrugated fins, combined with tubes in line, show the smallest air pressure drops on the air side. Highly corrugated fins, combined with staggered tubes, show the highest air pressure drops on the air side. Manufacturers who offer strongly corrugated fins mostly have too thin fins. They only need heavily corrugated fins to add just a bit of stability. The claim that they are doing it for reasons of increasing performance, one can confidently deny.

$$\xi_a = arctan\left(\frac{L_h(L_n - 1)}{L_c}\right)\frac{180}{\pi} \qquad f_a = \frac{(S_2 + L_c) + \left(\frac{L_c}{cos(\xi_a \pi/180)}\right)}{S_2}$$

$$A_l = 2 f_a \left(S_1 S_2 - \frac{{D_a}^2 \pi}{4}\right) R_t R_h \left(\frac{B}{L_t} - 1\right)$$

Fin coil heat exchanger

Symbol	Unit	Description
A_r	m2	Heat exchanger surface tubes outside
f_h	---	Factor for the heat transfer
f_{dp}	---	Factor for the pressure drop

Heat exchanger surface outside

$$A_r = D_a \pi (L_t - L_d) R_t R_h \left(\frac{B}{L_t} - 1\right) \qquad A_a = A_r + A_l$$

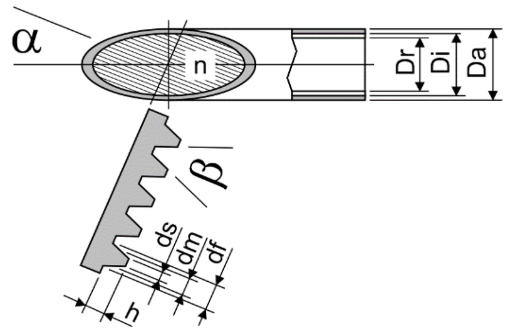
Heat exchanger surface inside with smooth tubes

$$A_i = D_i \pi B R_t R_h$$

Heat exchanger surface inside with grooved tubes

$$d_m = \frac{(D_i + D_r)\pi}{4n} \qquad A_{r1} = n\left[\left(\frac{2h}{cos\left(\frac{\beta\pi}{360}\right)}\right) + \left(d_m - htan\left(\frac{\beta\pi}{360}\right)\right)\right]$$

$$A_{r2} = A_{r1} + \left[n\left(d_m\left(htan\left(\frac{\beta\pi}{360}\right)\right)\right)\right] \qquad A_i = A_{r2} B R_t R_h$$

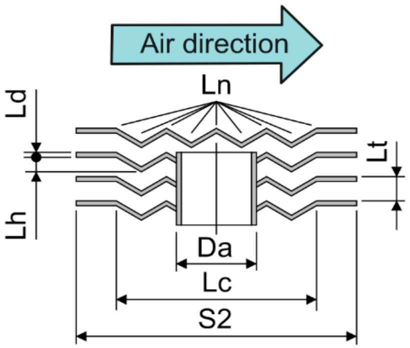


Corrugated fins

Influence on the heat transfer and the pressure drop

$$f_x = 1 + 0.3sin\left(\frac{L_n\pi}{180}\right) + 0.2\left(sin\left(\frac{L_n\pi}{180}\right)\right)^2 \qquad f_y = (L_n + 1)^{0.05}$$

$$f_b = f_x f_y \qquad f_e = \frac{L_h}{L_t - L_d} \qquad f_h = {f_b}^{f_e} \qquad f_{dp} = {f_h}^{L_n}$$



Heat transfer coefficient outside

The equations of the VDI-Atlases of the years 1955 to 1984 supply too high, the equations of the years 1985 to 1997 too low values for the heat transfer coefficient of the moist air. This fact is positive, because it allows calibrating based on laboratory measurements. Thus studies with TUEV showed that the mean value of the two approaches, so a relation of 1:1, resulted in a match with the measurements. VDI-Atlases of recent date are useless because various authors have applied double integrals from zero to infinity and similar impracticable mathematical excesses.

Fin coil heat exchanger	Symbol	Unit	Description Page 16
Heat transfer coefficient outside (VDI 1955 to 1984)	c_l	m/s	Average velocity of the moist air
	cp_{lfm}	J/kgK	Average heat capacity of the moist air
	Δt_{lf}	K	Temperature difference of the moist air
	η_{lfm}	Pas	Average dynamic viskosity of the moist air
	λ_l	W/mK	Thermal conductivity of the fins
	λ_{lfm}	W/mK	Average thermal conductivity of the moist air
	ρ_{lfm}	kg/m3	Average density of the moist air
	$\dot{V}_{lfm}$	m3/s	Average volume flow of the moist air
	x_{lfm}	kg/kg	Average absolute humidity of the most air

$$a = \frac{S_1}{D_a} \quad b = \frac{S_2}{D_a} \quad \psi = \left(1 - \frac{\pi}{4a}\right)\frac{L_t - L_d}{L_t} \quad f_{a(staggered)} = 1 + \frac{2}{3b}$$

$$f_{a(in\ line)} = 1 + \frac{0.7\left(\frac{b}{a} - 0.3\right)}{\psi^{1.5}\left(\frac{b}{a} + 0.7\right)^2} \qquad f_k = \frac{1 + f_a(R_t - 1)}{R_t}$$

$$l_{hyd} = \frac{D_a \pi}{2} \qquad \dot{V}_{lfm} = \frac{\dot{M}_{lt}\left(1 + x_{lfm}\right)}{\rho_{lfm}} \qquad c_l = \frac{\dot{V}_{lfm}}{BH}$$

$$Re_l = \frac{c_l l_{hyd} \rho_{lfm}}{\psi \eta_{lfm}} \qquad cp_{lfm} = \frac{\Delta h_{lf}}{\Delta t_{lf}} \qquad Pr_l = \frac{\eta_{lfm} cp_{lfm}}{\lambda_{lfm}}$$

$$Nu_1 = 0.644 {Re_l}^{0.5} {Pr_l}^{1/3} \qquad Nu_2 = \frac{0.037 {Re_l}^{0.8} Pr_l}{1 + 2.443 {Re_l}^{-0.1}\left({Pr_l}^{2/3} - 1\right)} \qquad Nu_3 = 0.3 + \sqrt{{Nu_1}^2 + {Nu_2}^2} \qquad Nu_l = f_k Nu_3 \qquad \alpha_l = \frac{f_h Nu_l \lambda_{lfm}}{l_{hyd}}$$

$$\rho_{me} = 1.28 \frac{S_2}{D_a}\sqrt{\frac{S_1}{S_2} - 0.2} \qquad h_{me} = \frac{D_a}{2}(\rho_{me} - 1)(1 + 0.35 ln \rho_{me}) \qquad X = h_{me}\sqrt{\frac{2\alpha_l}{\lambda_l L_d}} \qquad \vartheta = \frac{Tanh X}{X}\eta_{KRL} \qquad \eta_l = 1 - \frac{A_l}{A_a}(1 - \vartheta) \qquad \alpha_{a(old)} = \eta_l \alpha_l$$

Heat transfer coefficient outside (VDI 1985 to 1997)

$$a = \frac{S_1}{D_a} \qquad b = \frac{S_2}{D_a} \qquad \psi = \frac{S_1 L_t}{(S_1 - D_a)(L_t - L_d)} \qquad c_l = \frac{\psi \dot{V}_{lfm}}{BH} \qquad Re_l = \frac{c_l D_a \rho_{lfm}}{\eta_{lfm}} \qquad Pr_l = \frac{\eta_{lfm} cp_{lfm}}{\lambda_{lfm}}$$

$$\vartheta = \frac{2\left((S_1 S_2) - \left({D_a}^2 \pi/4\right)\right) + \left(D_a \pi (L_t - L_d)\right)}{D_a \pi L_t} \qquad R_t \leq 3 \rightarrow f_{a(in\ line)} = 0.2 \qquad R_t > 3 \rightarrow f_{a(in\ line)} = 0.22$$

$$R_t \leq 2 \rightarrow f_{a(staggered)} = 0.33 \qquad R_t = 3 \rightarrow f_{a(staggered)} = 0.36 \qquad R_t > 3 \rightarrow f_{a(staggered)} = 0.38 \qquad Nu_l = f_a {Re_l}^{0.6} \vartheta^{-0.15} {Pr_l}^{(1/3)}$$

$$\alpha_l = \frac{f_h Nu_l \lambda_{lfm}}{D_a} \qquad l_r = \sqrt{\left(\frac{S_1}{2}\right)^2 + {S_2}^2} \quad S_2 < \frac{S_1}{2} \rightarrow \rho_{me(staggered)} = 1.27 \frac{2S_2}{D_a}\sqrt{\frac{l_r}{2S_2} - 0.3} \qquad S_2 \geq \frac{S_1}{2} \rightarrow \rho_{me(staggered)} = 1.27 \frac{S_1}{D_a}\sqrt{\frac{l_r}{S_1} - 0.3}$$

Fin coil heat exchanger	Symbol	Unit	Description
	$f_{w(old)}$	---	Factor for the heat transfer coefficient VDI old
	$f_{w(new)}$	---	Factor for the heat transfer coefficient VDI new
	k_g	---	Factor for the surface temperature
	t_l	°C	Temperature of the moist air
	t_m	°C	Temperature of the coolant agent
	t_o	°C	Average surface temperature outside

$$S_1 < S_2 \rightarrow \rho_{me(in\ line)} = 1.28\frac{S_1}{D_a}\sqrt{\frac{S_2}{S_1} - 0.2}$$

$$S_1 \geq S_2 \rightarrow \rho_{me(in\ line)} = 1.28\frac{S_2}{D_a}\sqrt{\frac{S_1}{S_2} - 0.2}$$

$$h_{me} = \frac{D_a}{2}(\rho_{me} - 1)(1 + 0.35 ln\rho_{me}) \qquad X = h_{me}\sqrt{\frac{2\alpha_l}{\lambda_l L_d}}$$

$$\vartheta = \frac{TanhX}{X}\eta_{KRL} \qquad \eta_l = 1 - \frac{A_l}{A_a}(1 - \vartheta) \qquad \alpha_{a(new)} = \eta_l \alpha_l$$

Heat transfer coefficient outside on average

$$\alpha_a = \frac{f_{w(old)} \propto_{a(old)} + f_{w(new)} \propto_{a(new)}}{f_{w(old)} + f_{w(new)}}$$

k-coefficient concerning the outside surface

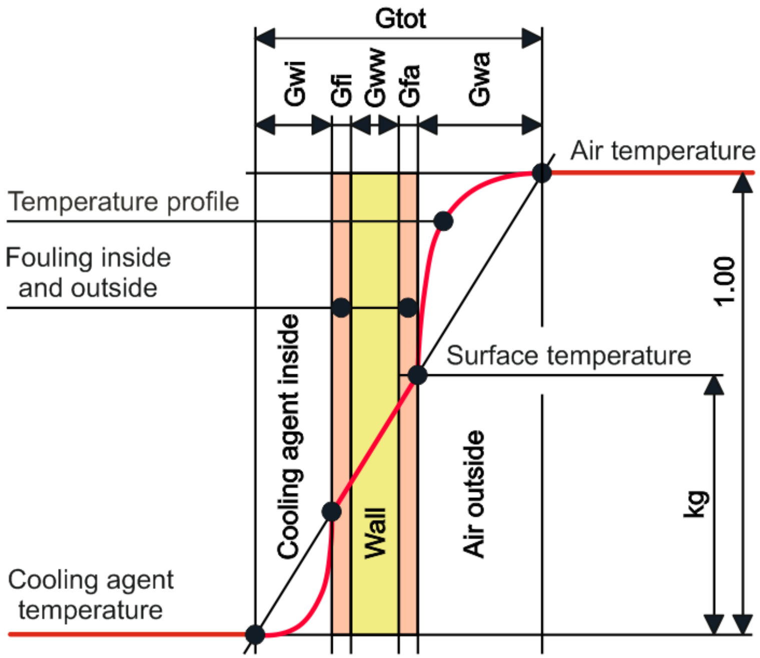
$$G_{wa} = \frac{1}{\propto_a} \quad G_{fa} = f_a \quad G_{ww} = \frac{A_a}{A_i}\frac{\delta_w}{\lambda_w} \quad G_{wi} = \frac{A_a}{A_i}\frac{1}{\alpha_i} \quad G_{fi} = \frac{A_a}{A_i}f_i$$

$$G_{tot} = G_{wa} + G_{fa} + G_{ww} + G_{wi} + G_{fi} \qquad k_a = \frac{1}{G_{tot}}$$

Air cooler surface temperature of the moist air

$$\eta_{WTT} = 1.00 \rightarrow k_g = \frac{G_{tot} - G_{wa}}{G_{tot}} \qquad \eta_{WTT} < 1.00 \rightarrow m = \eta_{WTT}{}^4 \rightarrow k_g = \left(\frac{G_{tot} - G_{wa}}{G_{tot}}\right)^m$$

$$t_o = t_m + k_g(t_l - t_m)$$



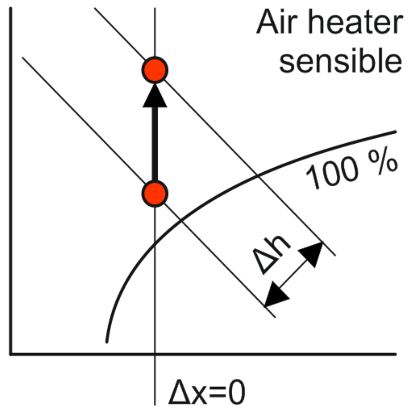
The poorer the total heat exchanger efficiency is the higher is the surface temperature of the air. This is true for much too thin fins with poor contact to the tubes, for a few tubes in the depth and for large fins distribution due to the bypass effect. Such air coolers have too little latent power and can therefore hardly form condensate.

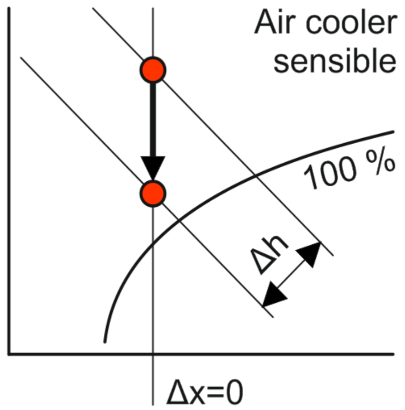
Fin coil heat exchanger

Symbol	Unit	Description
Δp_R	Pa	Pressure drop from the tubes
Δp_L	Pa	Pressure drop from the fins
$\Delta p_{l(dry)}$	Pa	Pressure drop from the air (dry)

Pressure drop outside

$\boldsymbol{\Delta x = 0} \rightarrow$ **Air heater and air cooler sensible**





$$c_l = \frac{\dot{V}_{lfm}}{BH} \qquad a = \frac{S_1}{D_a} \qquad b = \frac{S_2}{D_a} \qquad \Omega = \frac{S_1 - D_a}{S_1}\frac{L_t - L_d}{L_t}$$

$$Re_l = \frac{c_l D_a \rho_{lfm}}{\Omega \eta_{lfm}} \qquad A_1 = \frac{280\pi\left(0.75 + \left(b^{0.5} - 0.6\right)^2\right)}{a^{1.6}(4ab - \pi)} \qquad \xi_1 = \frac{A_1}{Re_l}$$

$$A_{2(staggered)} = 2.5 + \frac{1.2}{(a - 0.85)^{1.08}} + 0.4\left(\frac{b}{a} - 1\right)^3 - 0.01\left(\frac{a}{b} - 1\right)^3 \qquad \xi_{2(staggered)} = \frac{A_2}{Re_l^{0.25}} + \frac{1}{a^2}\left(\frac{1}{R_t} - 0.1\right) \qquad \xi_{R(staggered)} = \xi_1 + \xi_2\left(1 - e^{-\left(\frac{Re_l + 200}{1000}\right)}\right)$$

$$A_{2(in\ line)} = 0.03(a - 1)(b - 1) + \left(0.22 + 1.2\frac{\left(1 - \frac{0.94}{b}\right)^{0.6}}{(a - 0.85)^{1.3}}\right)10^{\left(\frac{b}{a} - 1.5\right)} \qquad \xi_{2(in\ line)} = \frac{A_2}{Re_l^{0.1\left(\frac{b}{a}\right)}} + \frac{1}{a^2}\left(\frac{1}{R_t} - 0.1\right)$$

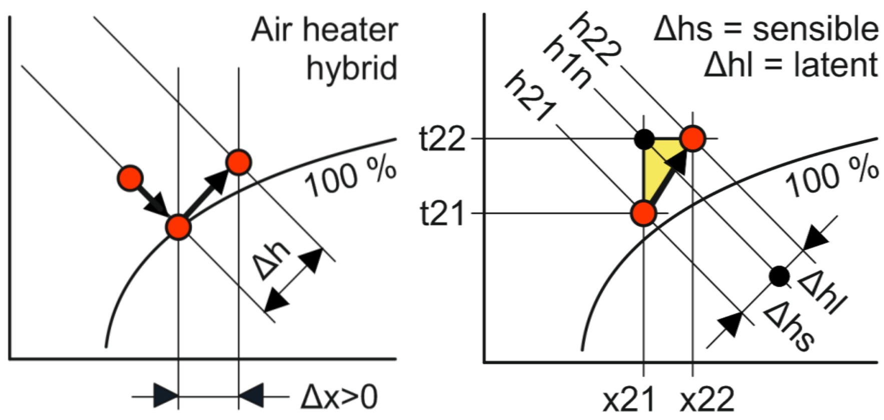
$$\xi_{R(in\ line)} = \xi_1 + \xi_2\left(1 - e^{-\left(\frac{Re_l + 1000}{2000}\right)}\right) \qquad \Delta p_R = \xi_R R_t \rho_l \frac{{c_l}^2}{2\Omega^2} \qquad d_{hyd} = 1.8(L_t - L_d) + 0.1D_a \qquad Re_l = \frac{c_l d_{hyd} \rho_{lfm}}{\Omega \eta_{lfm}} \qquad \delta = \frac{L_t - L_d}{S_1 - D_a}$$

$$\beta = 0.84 + 0.66e^{-\frac{\delta}{0.33}} \qquad r_l = 0.000078 \qquad \xi_1 = \beta\frac{64}{Re_l} \qquad \xi_2 = \left(2log\left(\frac{d_{hyd}}{r_l}\right) + 1.14\right)^{-2} \qquad \xi_3 = \left(-2log\left(\frac{2.51}{Re_l\sqrt{\xi_3}} + \frac{r_l}{3.71d_{hyd}}\right)\right)^{-2}$$

$$\xi_4 = max(\xi_1, \xi_2, \xi_3) \qquad \xi_{L(staggered)} = 0.25 + \frac{2}{\sqrt{3}}\frac{\xi_4 S_2}{d_{hyd}} \qquad \xi_{L(in\ line)} = 0.25 + \frac{\xi_4 S_2}{d_{hyd}} \qquad \Delta p_L = \xi_L R_t \rho_l \frac{{c_l}^2}{2\Omega^2} f_{dp} \qquad \Delta p_{l(dry)} = \Delta p_R + \Delta p_L$$

Symbol	Unit	Description
cp_{km}	J/kgK	Average heat capacity of the condensate
$\Delta h_{lf(sen)}$	J/kg	Enthalpy difference of the moist air (sensible)
$\Delta p_{l(moi)}$	Pa	Pressure drop of the moist air
η_{km}	Pas	Average dynamic viscosity of the condensate
$f_{l(moi)}$	---	Pressure drop factor (moist)
f_n	---	Factor for cells with condensate formation
λ_{km}	W/mK	Average thermal conductivity from the condensate
n	---	First from 15 cells with condensate formation
ρ_{km}	kg/m3	Average density from the condensate

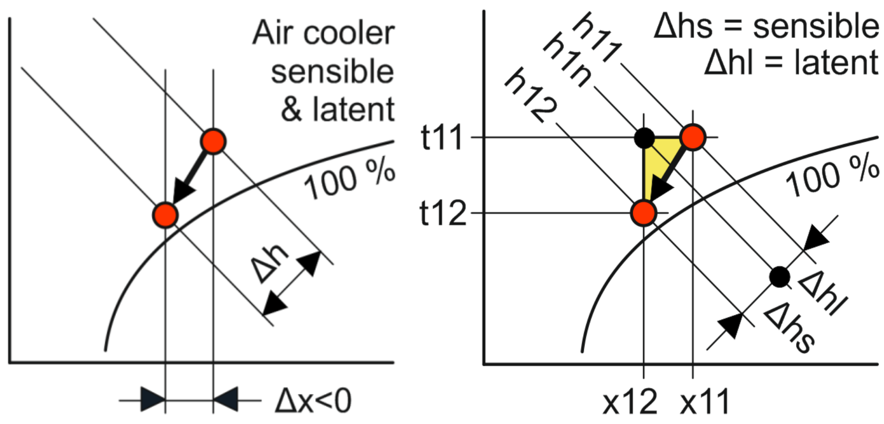
$\Delta x > 0 \rightarrow$ **Air heater hybrid (sensible and latent)**



The equations are identical to those for sensitive air heaters and air coolers to and with the pressure drop of the air dry, see page 18. Now the calculation for the pressure drop with sprayed fins follows.

$$\Delta h_{lf(sen)} = h_{1n} - h_{21} \qquad f_{l(moi)} = \left(\frac{\Delta h_{lf}}{\Delta h_{lf(sen)}}\right)^{0.33} \qquad \Delta p_{l(moi)} = f_{l(moi)} \Delta p_{l(dry)}$$

$\Delta x < 0 \rightarrow$ **Air cooler sensible and latent**



$$f_n = \frac{16 - n}{15}$$

$$f_u = 0.04 f_n \left(1 - e^{-\Delta x / 0.004}\right) \qquad f_v = \frac{\Delta x / \rho_{km}}{\left(\left((1 - \Delta x)/\rho_{lfm}\right) + \left(\Delta x / \rho_{km}\right)\right)}$$

$$\rho_{lfm} = \rho_{lfm}(1 - f_v) + \rho_{km} f_v \qquad cp_{lfm} = cp_{lfm}(1 - \Delta x) + cp_{km} \Delta x$$

$$\lambda_{lfm} = \lambda_{lfm}(1 - f_u) + \lambda_{km} f_u \qquad \eta_{lfm} = \eta_{lfm}(1 - \Delta x) + \eta_{km} \Delta x$$

With the corrected material values (moist air with condensate), the equations for sensitive air heaters and air coolers can be applied to and with the pressure drop of the air dry, see page 18. Now the calculation for the pressure drop with condensate on the fins follows.

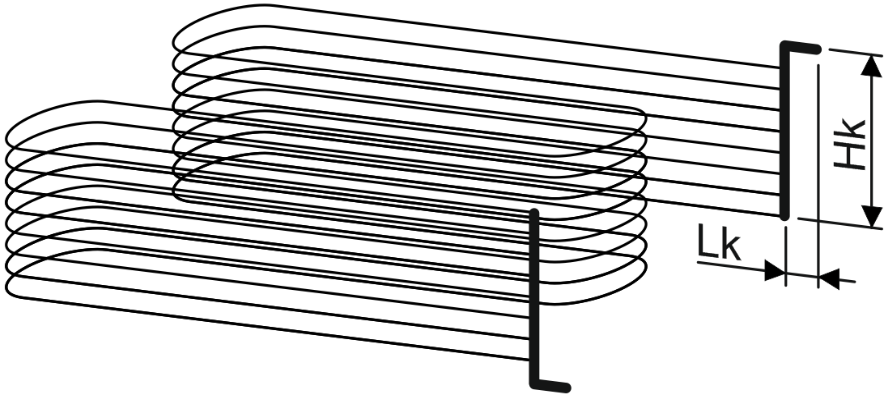
$$\Delta h_{lf(sen)} = h_{1n} - h_{12} \qquad f_{l(moi)} = \left(\frac{\Delta h_{lf}}{\Delta h_{lf(sen)}}\right)^{0.33} \qquad \Delta p_{l(moi)} = f_{l(moi)} \Delta p_{l(dry)}$$

Definition to the tube coupling and to the collectors

Example

Number of circuits (NC) = 8

Number of passages per NC (PA) = 4



Symbol	Unit	Description
c_{km}	m/s	Agent velocity inside of the collector
cp_m	J/kgK	Agent heat capacity
D_{ki}	m	Collector inside diameter
Δp_k	Pa	Collector pressure drop inside
η_m	Pas	Agent dynamic viscosity
λ_m	W/mK	Agent thermal conductivity
NC	---	Number of circuits
PA	---	Number of passages per NC
Q_{ki}	m2	Collector cross section inside
r_k	m	Roughness inside of the collector
ρ_m	kg/m3	Agent density

Pressure drop in the collectors

$$\Delta h_m = cp_m \Delta t_m \qquad \dot{M} = \frac{\dot{Q}}{\Delta h_m} \qquad \dot{V} = \frac{\dot{M}}{\rho_m} \qquad Q_{ki} = \frac{{D_{ki}}^2 \pi}{4} \qquad c_{km} = \frac{\dot{V}}{Q_{ki}} \qquad Re_{km} = \frac{c_{km} D_{ki} \rho_m}{\eta_m}$$

$$r_{k(Copper)} = 0.000002 \qquad r_{k(Stainless\ steel)} = 0.000080 \qquad r_{k(Galvanized\ steel)} = 0.000160 \qquad \xi_1 = \frac{64}{Re_{km}} \qquad \xi_2 = 0.3164 {Re_{km}}^{-0.25}$$

$$\xi_3 = 0.0054 + 0.3964\left({Re_{km}}^{-0.3}\right) \qquad \xi_4 = \left(log\left(Re_{km}\sqrt{\xi_4}\right)\right)^{-2} \qquad \xi_5 = \left(2log\left(\frac{D_{ki}}{r_{km}}\right) + 1.14\right)^{-2} \qquad \xi_6 = \left(-2log\left(\frac{2.51}{Re_{km}\sqrt{\xi_6}} + \frac{r_{km}}{3.71 D_{ki}}\right)\right)^{-2}$$

$$Re_{km} < 100 \rightarrow \xi_7 = \xi_1 \qquad Re_{km} \geq 100 \rightarrow \xi_7 = max(\xi_1, \xi_2, \xi_3, \xi_4, \xi_5, \xi_6) \qquad \xi_{km} = \frac{\xi_7 (H_k + L_k)}{D_{ki}} + 3 \qquad \Delta p_k = \xi_{km} \rho_m \frac{{c_{km}}^2}{2}$$

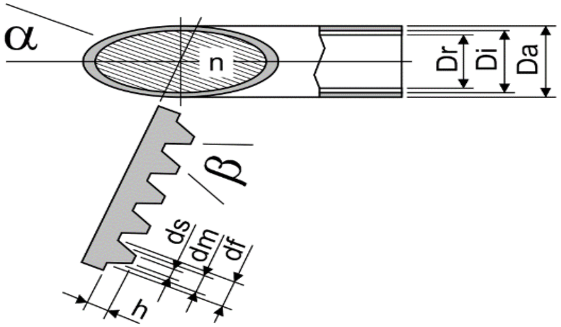
Symbol	Unit	Description
Q_{ri}	m2	Tubes cross section inside total
D_{hyd}	m	Hydraulic diameter from the tubes inside
Δp_r	Pa	Pressure drop in the tubes inside
Δp_{ti}	Pa	Pressure drop total inside

Definition to the smooth tubes

$$Q_{ri} = NC\frac{{D_i}^2\pi}{4} \qquad D_{hyd} = D_i \qquad r_{r(Copper)} = 0.000002$$

$$r_{r(Stainless\ steel)} = 0.000080 \qquad r_{r(Galvanized\ steel)} = 0.000160$$

Definition to the inside grooved tubes



$$Q_{ri} = NC\frac{(D_i + D_r)^2\pi}{16} \qquad D_{hyd} = \frac{4Q_{ri}}{A_{r2}}$$

$$r_{rr(Copper)} = 0.000002$$

$$r_{rr(Stainless\ steel)} = 0.000080$$

$$r_{rr(Galvanized\ steel)} = 0.000160 \qquad r_r = r_{rr} + hsin\left(\frac{\propto \pi}{180}\right)^4$$

Pressure drop in the tubes for agents without aggregate state change

$$c_{rm} = \frac{\dot{V}}{Q_{ri}} \qquad Re_{rm} = \frac{c_{rm}D_{hyd}\rho_m}{\eta_m} \qquad \xi_1 = \frac{64}{Re_{rm}} \qquad \xi_2 = 0.3164{Re_{rm}}^{-0.25} \qquad \xi_3 = 0.0054 + 0.3964\left({Re_{rm}}^{-0.3}\right) \qquad \xi_4 = \left(log\left(Re_{rm}\sqrt{\xi_4}\right)\right)^{-2}$$

$$\xi_5 = \left(2log\left(\frac{D_{hyd}}{r_r}\right) + 1.14\right)^{-2} \qquad \xi_6 = \left(-2log\left(\frac{2.51}{Re_{rm}\sqrt{\xi_6}} + \frac{r_r}{3.71D_{hyd}}\right)\right)^{-2} \qquad \xi_7 = max(\xi_1, \xi_2, \xi_3, \xi_4, \xi_5, \xi_6) \qquad \xi_{rm} = \frac{PA\xi_7B}{D_{hyd}} + (PA - 1)$$

$$\Delta p_r = \xi_{rm}\rho_m\frac{{c_{rm}}^2}{2}$$

Pressure drop inside total from the heat exchanger

$$\Delta p_{ti} = \Delta p_r + \Delta p_{k(Inlet\ collector)} + \Delta p_{k(Outlet\ collector)}$$

Fin coil heat exchanger	Symbol	Unit	Description
Heat transfer coefficient inside	η_{liq}	Pas	Agent viscosity on Liquidus
Agents without aggregate state change	g	m/s2	Fall acceleration (9.81 m/s2)
	λ_{liq}	W/mK	Agent thermal conductivity on Liquidus
	ρ_{liq}	kg/m3	Agent density on Liquidus
	R	J/kg	Agent evaporation heat

This relates to liquids or gases, so also for condensers regarding hot-gas cooling and condensate sub-cooling and for injection evaporators regarding the suction gas superheat.

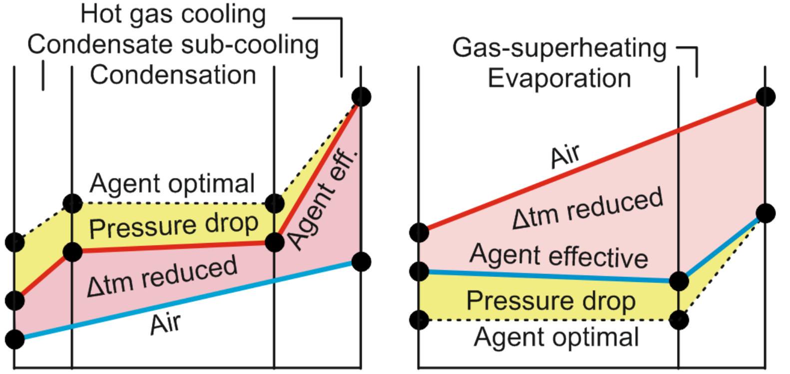
$$Pr_m = \frac{\eta_m cp_m}{\lambda_m} \qquad \xi_m = \left(1.82 log(Re_{rm} - 1.64)\right)^{-2}$$

$$Nu_1 = \frac{\frac{\xi_m}{8}(Re_m - 1000)Pr_m}{1 + 12.7\left(Pr_m^{\frac{2}{3}} - 1\right)\sqrt{\frac{\xi_m}{8}}} \qquad Nu_2 = \sqrt[3]{3.66^3 + 1.61^3 Re_{rm} Pr_{rm} \frac{D_{hyd}}{B}} \qquad Nu_3 = 0.664 Pr_m^{\frac{1}{3}} \sqrt{\frac{Re_m D_{hyd}}{B}} \qquad Nu_m = max(Nu_1, Nu_2, Nu_3)$$

$$\alpha_{i(Smooth\ tubes)} = \frac{Nu_m \lambda_m}{D_{hyd}}$$

Agents with aggregate state change (Boris Slipcevic)

Condensation, evaporation in injected and flooded operation. The equations are closely related. They lead therefore to a result only through iteration. High velocities improve the heat transfer, but lead to higher pressure drops, which reduce the average logarithmic temperature difference. The pressure drop must therefore be converted from Pa to K.



Condensation

$$f_{lam} = 0.943\left(\frac{\lambda_{liq}{}^3 \rho_{liq}{}^2 Rg}{\eta_{liq} B}\right)^{0.25} \qquad gi_{lam} = \frac{A_a}{A_i}\frac{1}{\alpha_{i(lam)}} \qquad gt_{lam} = \frac{1}{k_a} \qquad \Delta t_{lam} = \frac{gi_{lam}}{gt_{lam}} \qquad \alpha_{i(lam)} = f_{lam}\Delta t_{lam}{}^{0.25}$$

$$f_{turb} = 0.003\left(\frac{\lambda_{liq}{}^3 \rho_{liq}{}^2 Bg}{\eta_{liq}{}^3 R}\right)^{0.5} \qquad gi_{turb} = \frac{A_a}{A_i}\frac{1}{\alpha_{i(turb)}} \qquad gt_{turb} = \frac{1}{k_a} \qquad \Delta t_{turb} = \frac{gi_{turb}}{gt_{turb}} \qquad \alpha_{i(turb)} = f_{turb}\Delta t_{turb}{}^{0.5}$$

$$\alpha_{i(Smooth\ tubes)} = max\left(\alpha_{i(lam)}, \alpha_{i(turb)}\right)$$

Fin coil heat exchanger

Symbol	Unit	Description
η_{sol}	Pas	Agent viscosity on Solidus
F_{ge}	---	Agent flashgas part at the inlet point
H_d	J/kg	Agent enthalpy difference
$\dot{m}$	kg/sm2	Agent mass flow density
P_u	---	Agent pump recirculation factor
ρ_{sol}	kg/m3	Agent density on Solidus
S_t	N/m	Agent surface tension
t_o	°C	Evaporation temperature on Solidus

Dry expansion evaporation

$$K_1 = \frac{\lambda_{liq}\rho_{liq}{}^{0.06}}{g^{0.3}\rho_{sol}{}^{0.66}\eta_{liq}{}^{0.575}\eta_{sol}{}^{0.225}} \qquad B_1 = \sqrt{\frac{2S_t}{g(\rho_{liq}-\rho_{sol})}}$$

$$D_1 = 0.511B_1 \qquad F_1 = 0.56\sqrt{\frac{g}{D_1}} \qquad Q_i = \frac{\dot{Q}}{A_i} \qquad H_d = R(1-F_{ge})$$

$$B_2 = \frac{2.059\lambda_{liq}{}^{0.6}H_d{}^{0.133}r_r{}^{0.133}\rho_{sol}{}^{0.133}}{g^{0.2}(t_o+273.16)^{0.4}S_t{}^{0.3}F_1{}^{0.266}D_1{}^{0.399}\rho_{liq}{}^{0.233}}$$

$$\dot{M} = \frac{\dot{Q}}{H_d} \qquad \dot{m} = \frac{\dot{M}}{Q_{ri}} \qquad \alpha_{i(kon)} = \frac{0.9(1-F_{ge})^{0.1}K_1\dot{m}^{1.4}}{D_{hyd}{}^{0.5}}$$

$$\alpha_{i(bub)} = \frac{0.9(1-F_{ge})^{0.1}B_2\dot{m}^{0.1}Q_i{}^{0.7}}{D_{hyd}{}^{0.5}} \qquad \alpha_{i(Smooth\ tubes)} = max(\alpha_{i(kon)},\alpha_{i(bub)})$$

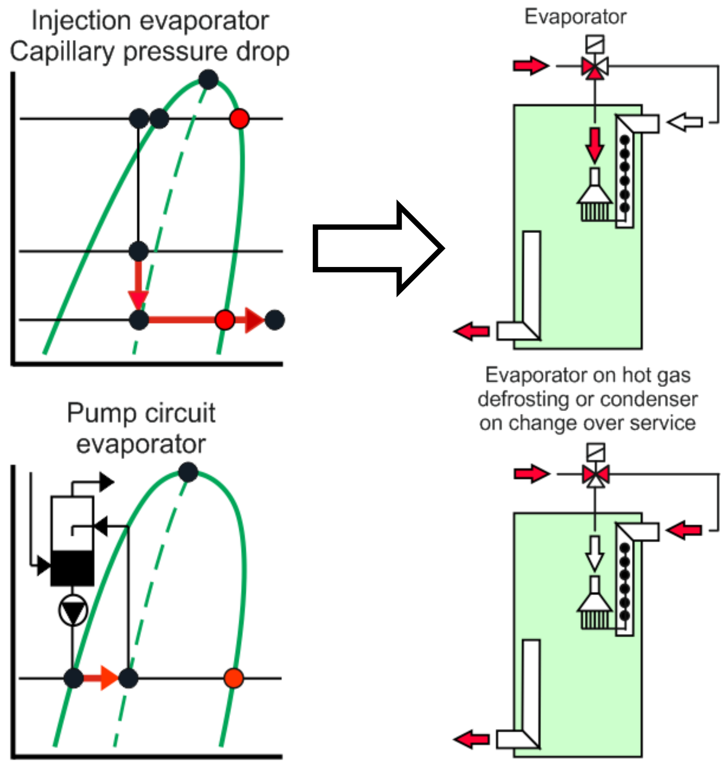
Pump recirculation evaporation

$$K_1 = \frac{\lambda_{liq}\rho_{liq}{}^{0.06}}{g^{0.3}\rho_{sol}{}^{0.66}\eta_{liq}{}^{0.575}\eta_{sol}{}^{0.225}} \qquad B_1 = \sqrt{\frac{2S_t}{g(\rho_{liq}-\rho_{sol})}} \qquad D_1 = 0.511B_1$$

$$F_1 = 0.56\sqrt{\frac{g}{D_1}} \qquad Q_i = \frac{\dot{Q}}{A_i} \qquad H_d = \frac{R}{P_u} \qquad \dot{M} = \frac{\dot{Q}}{H_d} \qquad \dot{m} = \frac{\dot{M}}{Q_{ri}}$$

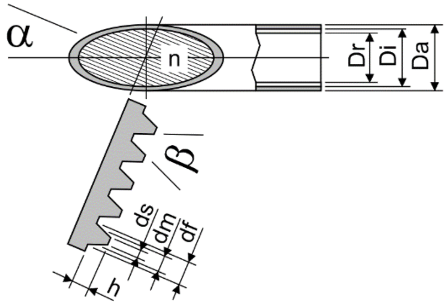
$$B_2 = \frac{2.059\lambda_{liq}{}^{0.6}H_d{}^{0.133}r_r{}^{0.133}\rho_{sol}{}^{0.133}}{g^{0.2}(t_o+273.16)^{0.4}S_t{}^{0.3}F_1{}^{0.266}D_1{}^{0.399}\rho_{liq}{}^{0.233}} \qquad \alpha_{i(kon)} = \frac{K_1\dot{m}^{1.4}}{D_{hyd}{}^{0.5}}$$

$$\alpha_{i(bub)} = \frac{B_2\dot{m}^{0.1}Q_i{}^{0.7}}{D_{hyd}{}^{0.5}} \qquad \alpha_{i(Smooth\ tubes)} = max(\alpha_{i(kon)},\alpha_{i(bub)})$$



Fin coil heat exchanger	Symbol	Unit	Description	Page 24
Inside grooved tubes, heat transfer coefficient inside	$\Delta p_{r(K)}$	K	Pressure drop total inside	
	G_r	Pa/K	Gradient	

$$d_f = d_m + \left(h\tan\left(\frac{\beta\pi}{360}\right)\right) \qquad d_s = d_m - \left(h\tan\left(\frac{\beta\pi}{360}\right)\right)$$



$$d_{ir} = 0.75 d_f + 0.25 d_s \quad X = d_{ir}\sqrt{\frac{2\propto_i}{\lambda_w d_{ir}}} \quad Y = \frac{tanhyp(X)}{X} \quad Z = 1 - \left(\frac{(1-Y)A_{r1}}{A_{r2}}\right)$$

$$\alpha_{i(Grooved\ tubes)} = Z\alpha_{i(Smooth\ tubes)}$$

Pressure drop in tubes for agents with aggregate state change

Pressure drop on condensation

$$f_v = \frac{\eta_{liq}}{\eta_{sol}} \qquad f_d = \frac{\rho_{liq}}{\rho_{sol}} \qquad f_{w1} = \frac{f_v \dot{m}^{0.5}}{62{\eta_{liq}}^{1/6} g^{1/6} {\rho_{liq}}^{1/3} {f_d}^{0.1}}$$

$$f_{w2} = \sum_{n=1}^{10} 0.1\left((0.1n-0.05)^{\frac{14}{19}} + f_{w1}(0.1n-0.05)^{\frac{14}{19}}(1.05-0.1n)^{0.5}\right)^{19/8}$$

$$Re_{rm} = \frac{\dot{m} D_{hyd}}{\eta_{liq}} \qquad \xi_1 = \frac{64}{Re_{rm}} \qquad \xi_2 = 0.3164 {Re_{rm}}^{-0.25}$$

$$\xi_3 = 0.0054 + 0.3964\left({Re_{rm}}^{-0.3}\right) \qquad \xi_4 = \left(2log\left(Re_{rm}\sqrt{\xi_4}\right)\right)^{-2}$$

$$\xi_5 = \left(2log\left(\frac{D_{hyd}}{r_r}\right) + 1.14\right)^{-2} \qquad \xi_6 = \left(-2log\left(\frac{2.51}{Re_{rm}\sqrt{\xi_6}} + \frac{r_r}{3.71 D_{hyd}}\right)\right)^{-2}$$

$$\xi_7 = max(\xi_1, \xi_2, \xi_3, \xi_4, \xi_5, \xi_6) \qquad \xi_{rm} = PA\left(\frac{\xi_7 B}{D_{hyd}} + 1\right) + 5$$

$$\Delta p_r = \left(\frac{\xi_{rm} f_{w2} \dot{m}^2}{2\rho_{sol}}\right) + \left(\dot{m}^2\left(\frac{1}{\rho_{liq}} - \frac{1}{\rho_{sol}}\right)\right) \qquad \Delta p_{r(K)} = \frac{\Delta p_r}{G_r}$$

Gradient, valid for condensation and evaporation

Refrigerant R410A (REFPROP)

Temp. (t) °C	Pressure (p) Pa	Gradient (Gr) Pa/K
30.00	1889338.04	
32.00	1988619.80	50611.13
34.00	2091782.57	52578.39
36.00	2198933.37	54600.09
38.00	2310182.92	56678.16
40.00	2425646.03	58814.84
42.00	2545442.29	61012.67
44.00	2669696.70	63274.56
46.00	2798540.51	65603.88
48.00	2932112.21	68004.53
50.00	3070558.64	

Polynomial approximation (Table Curve 2D)

$$p = a + bt + ct^2 + dt^3$$

a = 777887.5515
b = 27693.40974
c = 234.7441189
d = 2.568982299

Gradient = Differential according to t

$$G_r = b + 2ct + 3dt^2$$

$$t = 40 \rightarrow G_r = 58804.05$$

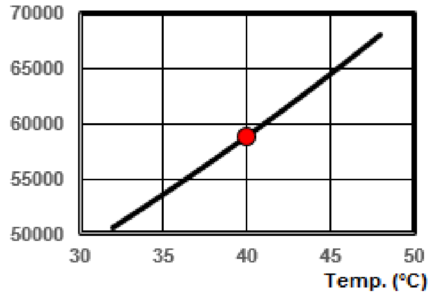
Simplified calculation method
see green cells above

$$G_{r(40)} = \frac{p_{42} - p_{38}}{t_{42} - t_{38}} = 58814.84$$

The error is 0.0183 %
against the exact method

However, one go closer
to the critical point,
the error increases significantly.

Gradient =Pa/K)



Fin coil heat exchanger	Symbol	Unit	Description
Pressure drop on dry expansion evaporation	D_{ak}	m	Capillary outside diameter
	D_{ik}	m	Capillary inside diameter
	Δp_k	Pa	Pressure drop in the capillaries
	S_k	m	Capillary thickness
	F_{gak}	---	Flashgas on capillary outlet
	$h_{(\dots)}$	J/kg	Enthalpie on (…)
	L_k	m	Capillary length

$$f_v = \frac{\eta_{liq}}{\eta_{sol}} \qquad f_d = \frac{\rho_{liq}}{\rho_{sol}} \qquad f_{w1} = \frac{f_v \dot{m}^{0.5}}{62 {\eta_{liq}}^{1/6} g^{1/6} {\rho_{liq}}^{1/3} {f_d}^{0.1}}$$

$$f_{w2} = \sum_{n=1}^{10} 0.1 \left((0.1n - 0.05)^{\frac{14}{19}} + f_{w1} (0.1n - 0.05)^{\frac{14}{19}} (1.05 - 0.1n)^{0.5} \right)^{19/8}$$

$$Re_{rm} = \frac{\dot{m} D_{hyd}}{\eta_{sol}} \qquad \xi_1 = \frac{64}{Re_{rm}} \qquad \xi_2 = 0.3164 {Re_{rm}}^{-0.25}$$

$$\xi_3 = 0.0054 + 0.3964\left({Re_{rm}}^{-0.3}\right) \qquad \xi_4 = \left(2log\left(Re_{rm}\sqrt{\xi_4}\right)\right)^{-2}$$

$$\xi_5 = \left(2log\left(\frac{D_{hyd}}{r_r}\right) + 1.14\right)^{-2} \qquad \xi_6 = \left(-2log\left(\frac{2.51}{Re_{rm}\sqrt{\xi_6}} + \frac{r_r}{3.71 D_{hyd}}\right)\right)^{-2}$$

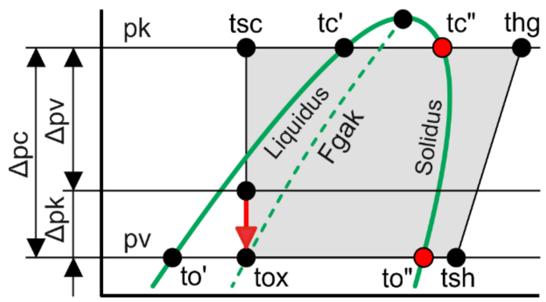
$$\xi_7 = max(\xi_1, \xi_2, \xi_3, \xi_4, \xi_5, \xi_6) \qquad \xi_{rm} = PA\left(\frac{\xi_7 B}{D_{hyd}} + 1\right)+5$$

$$\Delta p_r = \left(\frac{\xi_{rm} f_{w2} \dot{m}^2}{2\rho_{sol}}\right) + \left(\dot{m}^2 \left(\frac{1}{\rho_{sol}} - \frac{1}{\rho_{liq}}\right)\right) \qquad \Delta p_{r(K)} = \frac{\Delta p_r}{G_r}$$

Pressure drop in the capillaries

$$F_{gak} = 1 - \frac{R - h_{(tsc)} + h_{(to")}}{R} \qquad \eta_m = F_{gak}\eta_{sol} + (1 - F_{gak})\eta_{liq} \qquad \rho_m = \frac{1}{\frac{1 - F_{gak}}{\rho_{liq}} + \frac{F_{gak}}{\rho_{sol}}}$$

$$\dot{V}_m = \frac{\dot{M}}{\rho_m} \qquad Q_{ki} = NC\frac{{D_{ik}}^2 \pi}{4} \qquad Re_{km} = \frac{\dot{m} D_{ik}}{\eta_m} \qquad \xi_1 = \frac{64}{Re_{km}} \qquad \xi_2 = 0.3164 {Re_{km}}^{-0.25}$$



$$\xi_3 = 0.0054 + 0.3964\left({Re_{km}}^{-0.3}\right) \qquad \xi_4 = \left(2log\left(Re_{km}\sqrt{\xi_4}\right)\right)^{-2} \qquad \xi_5 = \left(2log\left(\frac{D_{ik}}{r_r}\right) + 1.14\right)^{-2} \qquad \xi_6 = \left(-2log\left(\frac{2.51}{Re_{km}\sqrt{\xi_6}} + \frac{r_r}{3.71 D_{ik}}\right)\right)^{-2}$$

$$\xi_7 = max(\xi_1, \xi_2, \xi_3, \xi_4, \xi_5, \xi_6) \qquad \xi_{km} = PA\left(\frac{\xi_7 L_k}{D_{ik}}\right)+2.5 \qquad \Delta p_k = \frac{\xi_{km}\dot{m}^2}{\rho_m}$$

Fin coil heat exchanger	Symbol	Unit	Description Page 26
Pressure drop on pump recirculation evaporation	F_{ga}	---	Flashgas part on the outlet point

$$f_v = \frac{\eta_{liq}}{\eta_{sol}} \qquad f_d = \frac{\rho_{liq}}{\rho_{sol}} \qquad f_{w1} = \frac{f_v \dot{m}^{0.5}}{62\eta_{liq}{}^{1/6} g^{1/6} \rho_{liq}{}^{1/3} f_d{}^{0.1}} \qquad F_{ga} = \frac{1}{P_u} \qquad q = \frac{F_{ga}}{20}$$

$$f_{w2} = \sum_{n=1}^{10} 0.1\left(\left(q(2n-1)\right)^{\frac{14}{19}} + f_{w1}\left(q(2n-1)\right)^{\frac{14}{19}}\left(F_{ga} - q(2n-1)\right)^{0.5}\right)^{19/8} \qquad Re_{rm} = \frac{\dot{m}D_{hyd}}{\eta_{liq}} \qquad \xi_1 = \frac{64}{Re_{rm}} \qquad \xi_2 = 0.3164Re_{rm}{}^{-0.25}$$

$$\xi_3 = 0.0054 + 0.3964\left(Re_{rm}{}^{-0.3}\right) \qquad \xi_4 = \left(2log\left(Re_{rm}\sqrt{\xi_4}\right)\right)^{-2} \qquad \xi_5 = \left(2log\left(\frac{D_{hyd}}{r_r}\right) + 1.14\right)^{-2} \qquad \xi_5 = \left(2log\left(\frac{D_{hyd}}{r_r}\right) + 1.14\right)^{-2}$$

$$\xi_6 = \left(-2log\left(\frac{2.51}{Re_{rm}\sqrt{\xi_6}} + \frac{r_r}{3.71D_{hyd}}\right)\right) \qquad \xi_7 = max(\xi_1, \xi_2, \xi_3, \xi_4, \xi_5, \xi_6) \qquad \xi_{rm} = PA\left(\frac{\xi_7 B}{D_{hyd}} + 1\right) + 5$$

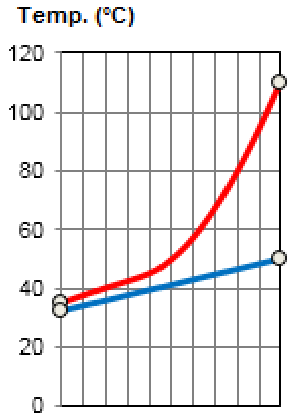
$$\Delta p_r = \left(\frac{\xi_{rm} f_{w2} \dot{m}^2}{2\rho_{sol}}\right) + \left(\dot{m}^2\left(\frac{1}{\rho_{sol}} - \frac{1}{\rho_{liq}}\right)\right) \qquad \Delta p_{r(K)} = \frac{\Delta p_r}{G_r}$$

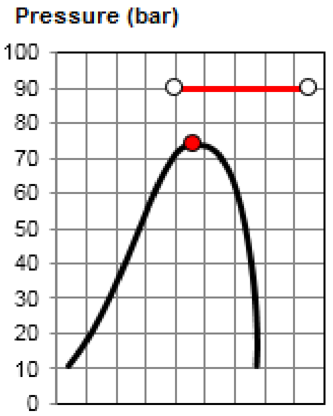
CO2-Cooler in the supercritical area

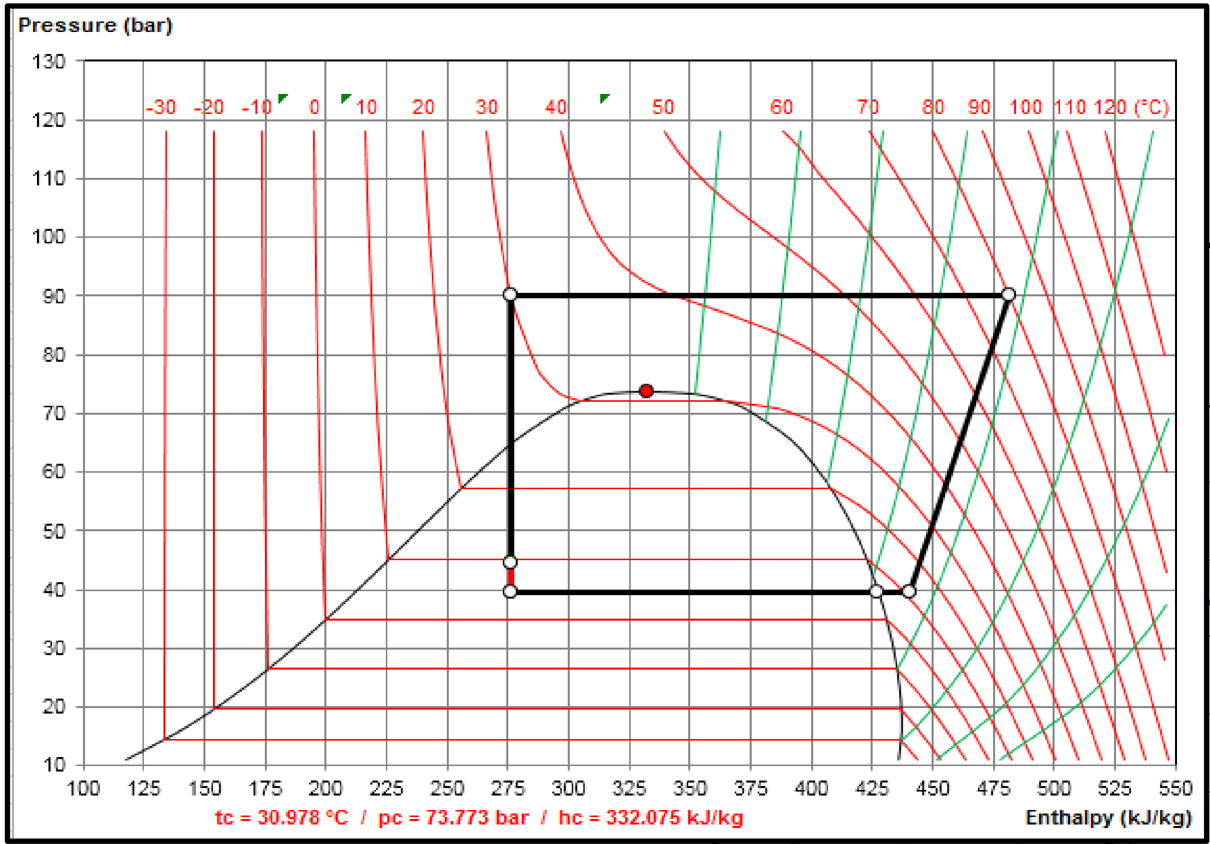
Because the thermodynamic values for cooling of CO2 in the supercritical range extremely change, the cooling process must be divided into 15 cells. The heat transfer coefficients, the mean logarithmic temperature difference and thus the need for a heat exchange surface change in the cells.

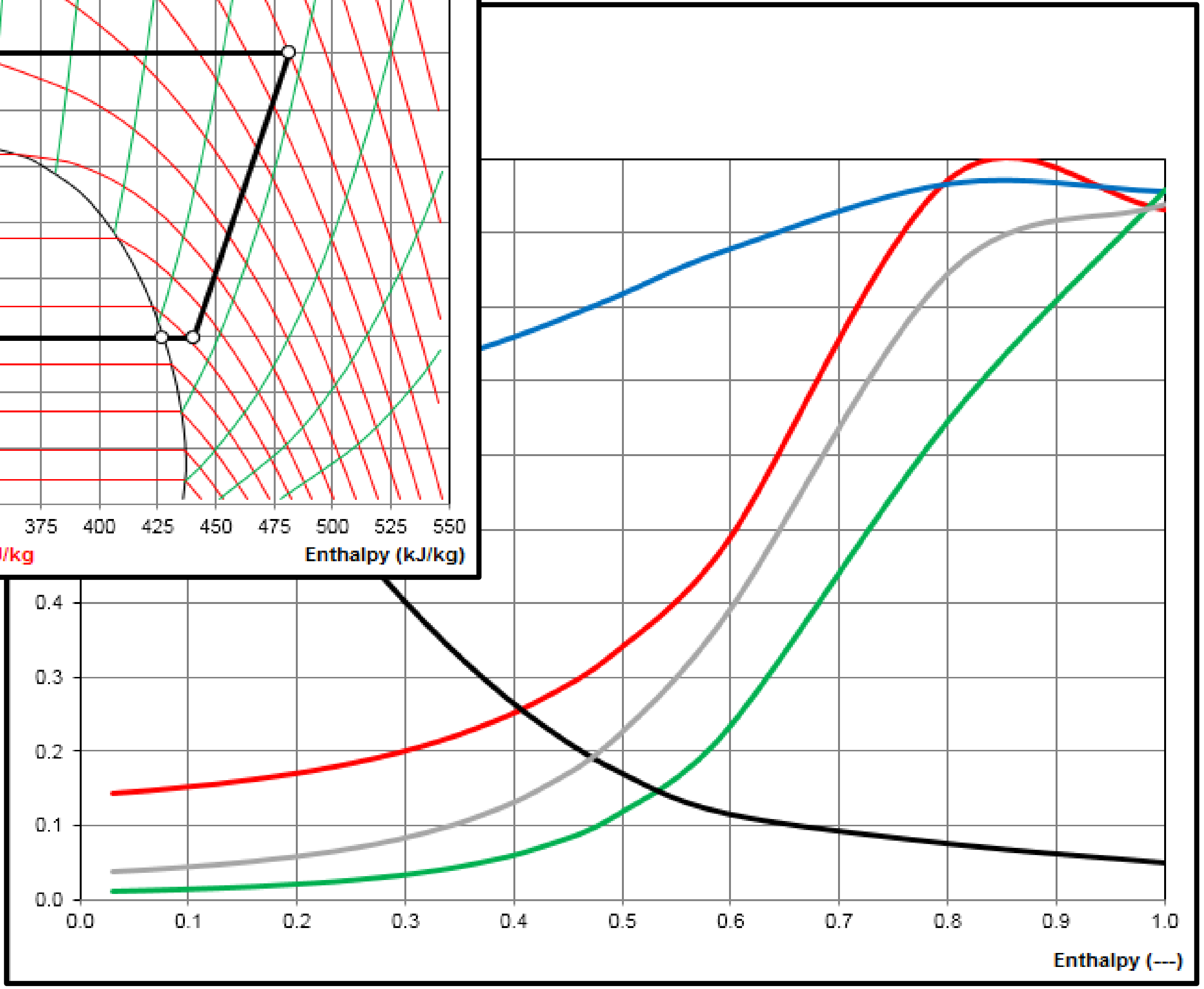
The calculation for the heat transfer coefficient and pressure drop occurs in the individual cells, as in the case of the media without a change in the state of the aggregate.

Carbon dioxide R744 (CO2) supercritical		
Pressure	bar	90.000
Inlet	°C	110.000
Outlet	°C	35.000
Mass flow	kg/h	3214.497
Density in	kg/m3	154.930
Density out	kg/m3	662.130
Volume flow in	m3/h	20.748
Volume flow out	m3/h	4.855
Velocity in	m/s	3.467
Velocity out	m/s	0.811
Pressure drop	kPa	42.955









The precise determination of the injection point (flash gas) and the calculation of the capillaries are also a problem. These are carried out in the application HEH-DX-Evaporator.

There is very good software, such as REFPROP from NIST, which allows the calculation of the thermodynamic values of non-condensable mixed gases. A partial condensation of water is not taken into account. Software that would allow such calculations is not offered at a reasonable price. With the applications GHH and HEH-SR-G we offer the possibility to carry out such calculations at a reasonable price. By means of REFPROP from NIST, the mixed gas is first determined without condensable water vapor. Subsequently, the cooling process with condensable water vapor is calculated in the application GHH. GHH is therefore a Mollier diagram for non-condensable mixed gases with the possibility of calculating a partial condensation of water without determining the exact size of the heat exchanger. Alternatively, HEH-SR-G applications can be directly used when determining the size of the heat exchanger accurately.

Mixture Information

Mixture name: nitrogen/oxygen/carbon dioxide/carbon monoxide

Molar mass: 30.638 kg/kmol

Saturation fixed points

	Critical Point	Cricondenther (Max Temp.)
Temperature (°C)	Unknown	-67.404
Pressure (Pa)	Unknown	7091200.0
Density (kg/m³)	Unknown	160.34

Components and composition

	Mass Fraction
nitrogen	0.55
oxygen	0.25
carbon dioxide	0.15
carbon monoxide	0.05

REFPROP - NIST Reference Fluid Properties (DLL version 9.1)

File Edit Options Substance Calculate Plot Window Help Cautions

1: nitrogen/oxygen/carbon dioxide/carbon monoxide: p = 100000.0 Pa (55/25/15/5)

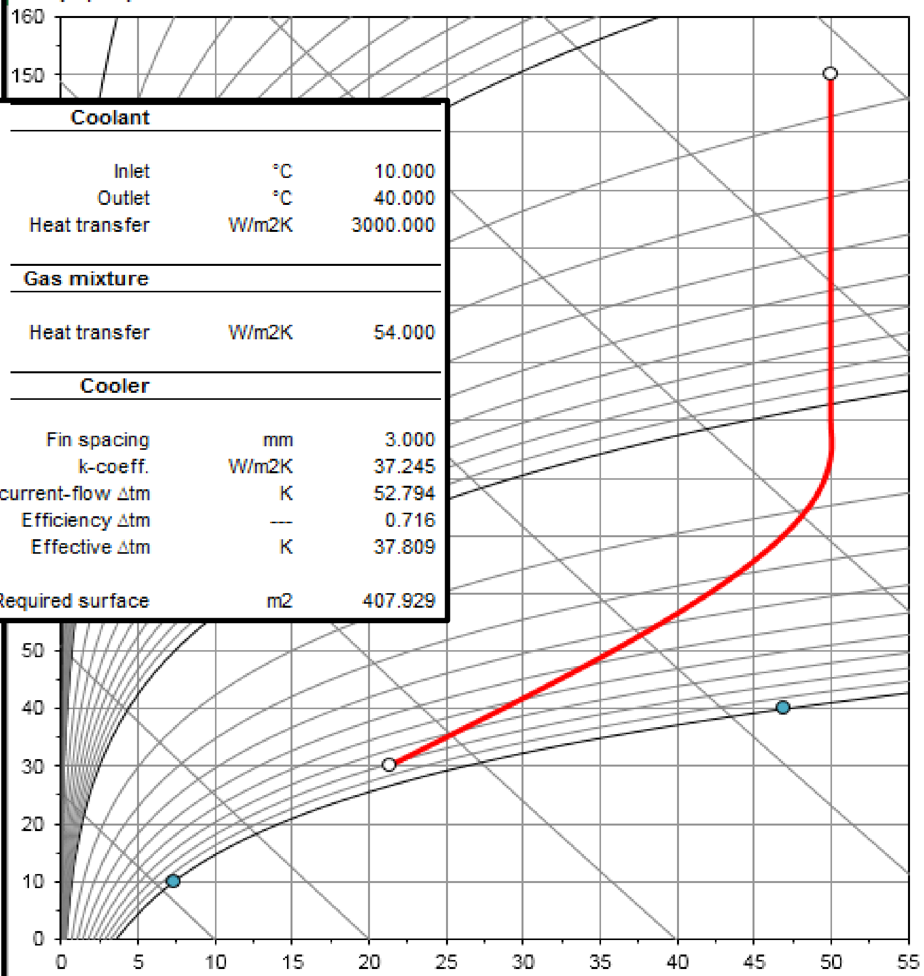
	Temperature (°C)	Pressure (Pa)	Density (kg/m³)	Cp (J/kg-K)	Therm. Cond. (W/m-K)	Viscosity (Pa-s)	Prandtl
1	-100.00	100000.	2.1395	965.59	0.015077	0.000011493	0.73607
2	-75.000	100000.	1.8660	966.75	0.017183	0.000012951	0.72864
3	-50.000	100000.	1.6550	969.52	0.019194	0.000014351	0.72492
4	-25.000	100000.	1.4872	973.15	0.021158	0.000015697	0.72200
5	0.00000	100000.	1.3504	977.31	0.023085	0.000016994	0.71944
6	25.000	100000.	1.2367	981.86	0.024974	0.000018244	0.71727
7	50.000	100000.	1.1407	986.74	0.026826	0.000019453	0.71555
8	75.000	100000.	1.0586	991.93	0.028641	0.000020624	0.71430
9	100.00	100000.	0.98756	997.43	0.030419	0.000021760	0.71352
10	125.00	100000.	0.92545	1003.2	0.032162	0.000022864	0.71320
11	150.00	100000.	0.87071	1009.3	0.033873	0.000023939	0.71331
12	175.00	100000.	0.82209	1015.7	0.035553	0.000024986	0.71379
13	200.00	100000.	0.77861	1022.2	0.037204	0.000026008	0.71461
14	225.00	100000.	0.73951	1029.0	0.038828	0.000027006	0.71571
15	250.00	100000.	0.70416	1036.0	0.040427	0.000027981	0.71704
16	275.00	100000.	0.67203	1043.0	0.042002	0.000028936	0.71856
17	300.00	100000.	0.64271	1050.2	0.043555	0.000029871	0.72023

Example of a mixed gas of 4 non-condensable gases, thus without condensable water vapor, calculated with REFPROP from NIST. For our applications GHH and HEH-SR-G the values of -100 to 300°C are required in steps of 25 K even if, for example, only a cooling process from 150 to 30°C is to be calculated. Furthermore, the proportion of partially condensable water vapor has to be entered into these applications. In the following example, the cooling occurs with water from 10 to 40°C.

A water vapor quantity of 50 g/kg at the inlet was assumed. At the outlet it is still 20 g/kg. Accordingly, 30 g/kg water vapor condenses from. The size of the heat exchanger was determined only rudimentarily. A more precise calculation can be made with the 3 applications HEH-SR-G:

1. Cooling with liquid media
2. Cooling by injection evaporation
3. Cooling by pump-circulating evaporation

Mollier TX diagram for		Gas	Steam
	Name	Gas mixture	Water
	Formula	N2+O2+CO2+CO	H2O
	CAS	---	7732-18-5
Molecular weight	kg/kMol	30.638	18.015
Triple point temperature	°C	0.000	0.010
Evaporation-Enthalpy (0.000 °C)	J/kg		2500900.000



Gas mixture / Water		Inlet	Outlet			
				Coolant		
Pressure	barabs	1.000		Inlet	°C	10.000
Temp.	°C	150.000	30.000	Outlet	°C	40.000
Rel. humidity	%	1.646	82.377	Heat transfer	W/m2K	3000.000
Abs. humidity	g/kg	50.000	21.317			
Density humid	kg/m3	0.843	1.199	**Gas mixture**		
Enthalpy humid	kJ/kg	290.790	83.989			
				Heat transfer	W/m2K	54.000
Wet bulb temperature	°C	51.887	27.439			
Dew point temperature	°C	41.119	26.662	**Cooler**		
Volume flow humid	m3/h	12461.503	8520.704			
Mass flow dry	kg/h	10000.000	10000.000	Fin spacing	mm	3.000
Condensate flow	kg/h		286.833	k-coeff.	W/m2K	37.245
				Countercurrent-flow Δtm	K	52.794
Capacity sensible	kW		352.322	Efficiency Δtm	---	0.716
Capacity latent	kW		222.127	Effective Δtm	K	37.809
Capacity frost	kW		0.000			
Capacity total	kW		574.449	Required surface	m2	407.929

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software by www.zcs.ch
created for the range
0.5 / 20 bar
-100 / 300 °C
0 / 1000 g/kg

Reduce the pollutants by cooling, condensation and optimal separation

Cooler: 42/36/20-12R-30T-1800A-3.0PA-30C-AISI 316/AISI 316/AISI 316

Capacity	kW	570.980	------ sensible:	352.635
Surface reserve	%	7.321	latent:	218.345
Present surface	m2	434.640	frost:	0.000
Required surface	m2	404.989		
k-coeff.	W/m2K	37.254	------------- ffi:	5.000E-05
Mean temp. diff.	K	37.845	ffa:	5.000E-05

Gas mixture / Water		Inlet	Outlet	Medium
Pressure	bar	1.000		
Temp.	°C	150.000	30.000	90.000
Rel. humidity	%	1.646	84.197	10.504
Abs. humidity	g/kg	50.000	21.805	46.797
Density humid	kg/m3	0.843	1.198	0.984
Enthalpy humid	kJ/kg	290.790	85.237	214.544
Volume flow humid	m3/h	12461.503	8527.533	10639.337
Mass flow dry	kg/h	10000.000	10000.000	10000.000
Condensate flow	kg/h		281.950	
Surface temperature	°C	77.812	16.875	
Velocity	m/s	1.526	1.044	
Pressure drop (dry 119 Pa)	Pa		139.544	

Water		Inlet	Outlet	Medium
Temp.	°C	10.000	40.000	25.000
Density	kg/m3			997.209
Spec. heat	kJ/kgK			4.180
Heat cond.	W/mK			0.611
Viscosity	Pas			8.901E-04
Volume flow	m3/h			16.438
Velocity	m/s			0.598
Pressure drop	kPa			15.066

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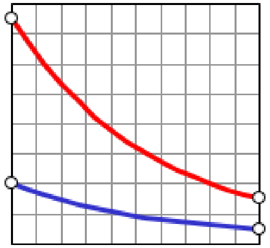
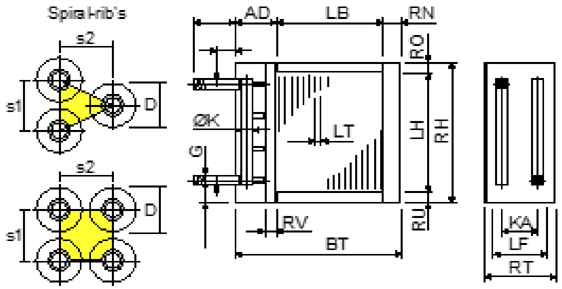


Application	GHH	HEH-SR-G
Capacity sensible	352 kW	353 kW
Capacity latent	222 kW	218 kW
Capacity total	574 kW	571 kW
Required surface	407 m2	405 m2

		Piece	360	Tubes:	flat	AISI 316
		Piece	0		staggered	
		Piece	0	Collectors:	1.23 m/s	AISI 316
		Piece	0	Connections:	1.23 m/s	AISI 316
		Piece	12	Fins:	ribbed	AISI 316
		Piece	30	Frame:	2.00 mm	AISI 316
		Piece	12	Circulations:	1	Default
		Piece	30	Protection:		without
		l	191			---
		kg	2024	El. heat rods:		---
		---	2 ½"	Air flow direction:		horizontal
		mm	1340	Special:		Bottom plate perforated
		mm	2070			for perfect condensate drain
		mm	510			
		mm	1260			
Finned width	LB	mm	1800			
Finned depth	LF	mm	436			
Frame on top	RO	mm	40			
Frame on bottom	RU	mm	40			
Frame in front	RV	mm	30			
Frame on back (~84mm)	RN	mm	84			
Collector-Diameter	K	mm	76			
Collector covering	AD	mm	186			
Collector distance	KA	mm	401			
Fin spacing	LT	mm	3.000			
Fin thickness	LD	mm	1.000	Delivery:		5-6 weeks
Tube diameter	d / D	mm	20.000 / 40.000	Validity:		12 weeks
Tube thickness	S	mm	1.000	Condit.:		net, prepaid address
Tube interval on the height	S1	mm	42.000	Payment:		30 days net
Tube interval on the depth	S2	mm	36.373	**Non el. rods:**	**EUR**	**38802.00**



Who prefers to let lay out the whole, can contact us for what we need of course all data. We will gladly provide you with an offer on a fee base.

It should be noted, that we make such calculations only against prepayment.

Cert.-Eng. Marin Zeller TU, VDI